Compressed Air
Systems Specialists

N_2O_2 SYSTEM
AUDIT REPORT

PREPARED BY

FROM
05/12/2026

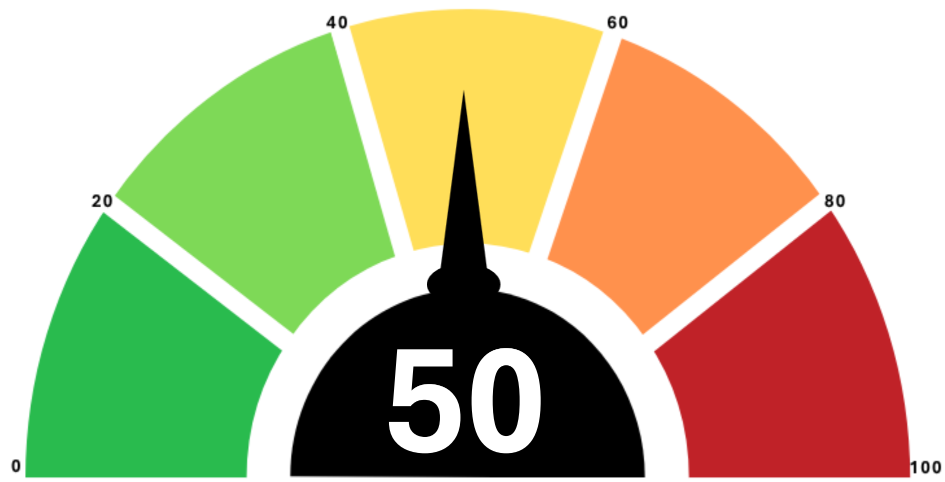
PREPARED FOR
[Company Name]
[Location]

[phone number]
www.N2O2.com
[email]

N2O2 Compressed Air Systems Specialists

COMPRESSED AIR SYSTEM SUMMARY

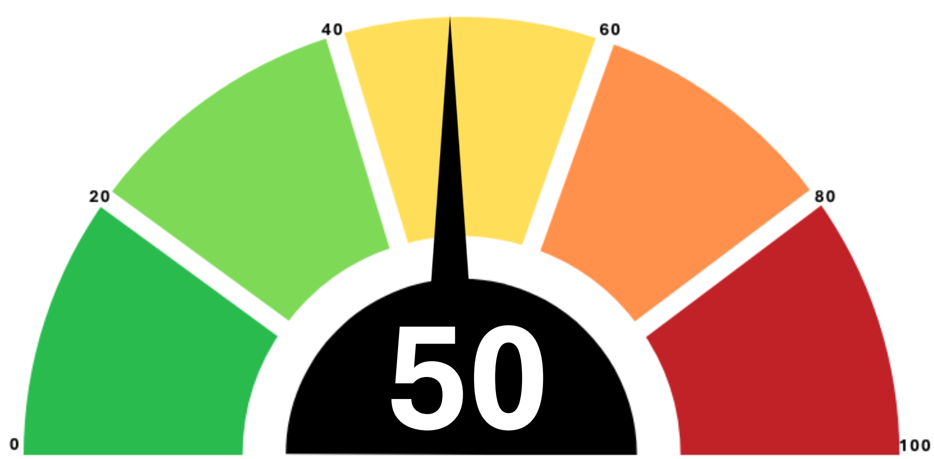
Supply



Volatility Score

Supply side volatility reflective of compressed air production, dewpoint, storage, reliability and sustainability.

Demand



Volatility Score

Demand side volatility reflects system air quality and pressure stability

RECOMMENDATION SUMMARY

Priority I: Implement as soon as possible

1. Retrofit all condensate drains with air-free drains.

2. Remove the 1" reducer as described in the report below. This will eliminate the current 5 psig delta P currently seen after this reducer.

3. Replace LS-20S-125L with a 200 hp 2-stage with variable frequency drive. Compressor capable of a full load pressure of 125 psig.

4. Insert a flow meter at the rail unload to determine the demand. This should be done so that a remedy can be implemented to increase the dewpoint from -30F to a minimum -50F.

5. Check with the dryer manufacturer to see if there is an energy management system that can be retrofitted to the dryers.

6. The air lines feeding the FV3687 and Process Comp Area are regulating the air but are not preventing pressure fluctuations. Priority II will resolve the issue.

7. The secondary compressor should be turned off. If that is not possible at this time than it should be after Priority II is implemented.

Priority II: Implement 6 – 12 months

1. Increase storage by 2000 gallons.
2. Add flow control valve.
3. Turn secondary compressor off.

Priority III: Implement 12 – 18 months

1. Replace 20/16 compressor with identical unit mentioned in priority I.
2. Have a site wide leak audit performed.
3. Replace heated/purge dryer with a heated/blower purge dryer.

Estimated Costs

Priority I (U.S. $)

Air -Free Drains..$ 1,200.00 each
150Hp Compressor 2 – stage with VFD..................................$188,000.00 each
Rail Unloading Flow Meter..$ 1,000.00 each

Priority II (U.S. $)

2000 Gallon Air Receiver w/relief valve and gauge................$ 18,000.00
1500 cfm Flow Control Valve..$ 18,500.00

Priority III (U.S. $)

150Hp Compressor with VFD...$188,000.00 each
Site Wide Leak Audit..$ 5,000.00

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EXECUTIVE SUMMARY

An onsite compressed air quality audit was performed at the [Company Name] facility in [Location] April 21st through April 28th, 2026. The findings, analysis and recommendations are detailed in this report.

Compressed Air Audit Objectives:

The objectives of the Compressed Air Audit were as follows:

1. Analyze the current supply side compressed air system components. This includes compressors, dryers, air receivers and condensate drains.

2. Measure dewpoint in various areas of the plant.

3. Take pressure readings in various parts of the plant to determine the pressure profile of the facility.

4. Determine the minimum pressure required for the Plant.

5. Determine the demand (cfm) of the facility.

6. Make recommendations to upgrade or replace current equipment to improve efficiency, reliability and sustainability.

Summary of findings

The audit results determined there are several correctable inefficiencies that will improve reliability, sustainability and reduce operational costs.

1. Analysis of the plant air demand has determined that the LS20S-125L compressor 17.84 $m^{(3)}$/min (620 cfm) and the 20/16L-125 (17.84 $m^{(3)}$/min, 690 cfm) were both on-line. The demand range was as follows:

Overall Average Demand = 636 cfm /18.0 $m^{(3)}$)/min

Peak Demand = 1000 cfm / 28.3 $m^{(3)}$)/min

Min Demand = 620 cfm / 17.56 $m^{(3)}$)/min

2. There is no useful storage in the system. Useful storage is achieved by storing air in the receiver at a higher pressure than is consumed in the plant. The useful storage in turn keeps pressure constant within 2 psig of the set point through the facility in conjunction with implementation of a flow control valve. Industry standard calls for 4 gallons of storage per cfm

3. Artificial demand, which is the amount of pressure produced versus amount Required. Currently up to 115 psig is produced with pressure requirement of 600 kPa-650 kPa (as high as 700 Kpa) (87 psig - 95 psig) in the plant resulting in 137.9 kPa (20 psig) in artificial demand.

4. It is estimated that air leaks are consuming approximately 4% of the compressed Air produced or 38 cfm (1.08m^3/min).

5. There is a 2” air-line, (40-R-039 FRM 8) that has a 1”reducer than returns to a 2” line. Based on the recorded data this appears to induce a drop of 3 psig (20.68 kPa) - 5 psig (39.5 kPa) psig.

6. Float and timed type drains are used to remove condensate from coolers and filters. Each drain when engaged consumes 100 cfm. There are approximately 6 drains. They do not fire at the same time. Based on the length of time and frequency,they are open, it is estimated this equates to a demand of 15 cfm on average or 3 bhp (2.238 kW) or $1,490 per year in lost energy.

7. The 20S-125L is on-line producing no air. Its on-line in case the pressure falls to quickly or the primary compressor trips. However this cost over $25,000. Per year to run in this fashion.

8. Each of the dryers requires up to 120 cfm of compressed air and a 12kW heater to assist in the regeneration of the saturated desiccant bed.

Heated dryers with an 8-hour cycle breakdown like this:

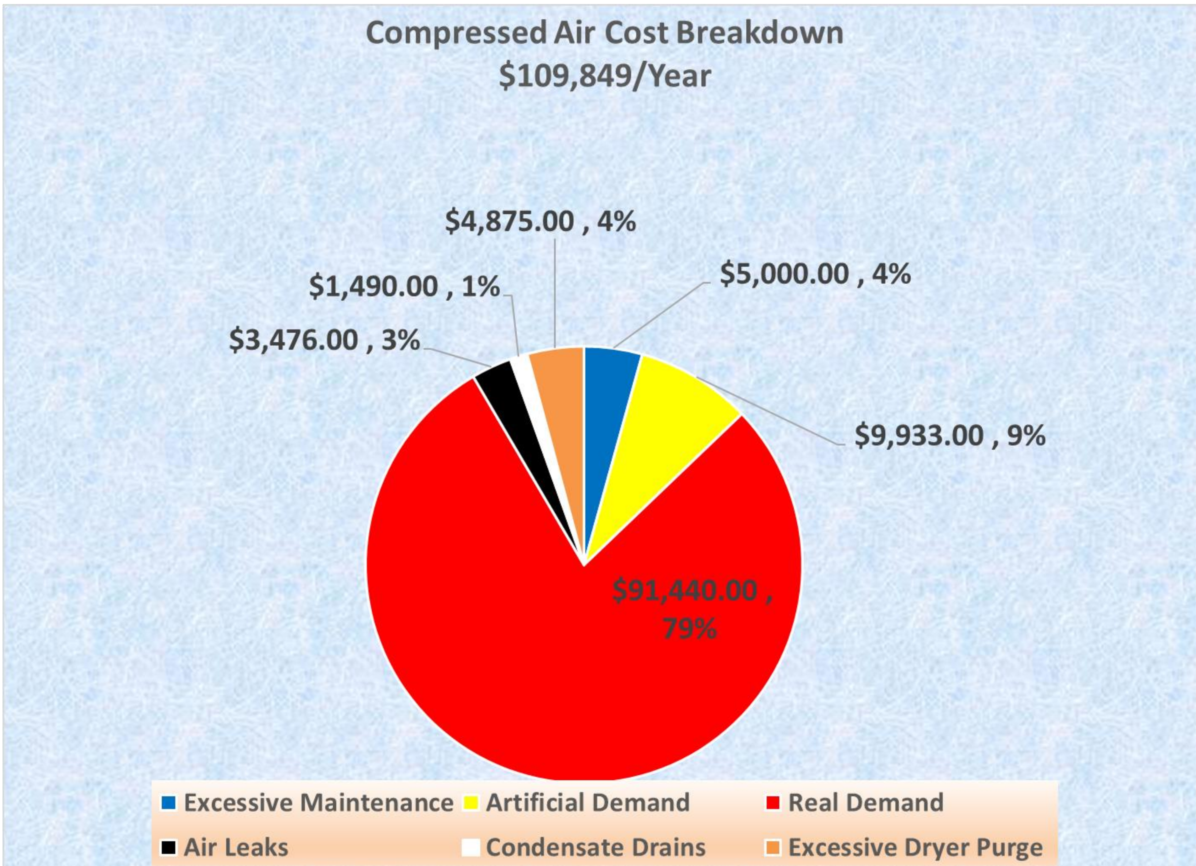
Per tower (one full cycle = 8 hrs total):

Phase	Duration (typical)	What’s happening
Drying (online)	**4 hours**	Tower drying compressed air
Heating / Regen	**2–3 hours**	Heated purge air drives moisture out

Cooling	**1–2 hours**

The purge air is on for approximately 4 hours and then stops. This coincides with the flow fluctuations exhibited. Thus, the heater on average consumes 3 kW/hr and the purge air consumes 60 cfm continuously or 8.952 kW/hr for a total of 11.952 kW/hr. This equates to **$7,957 per year.**

Compressed Air Cost Breakdown



N2O2 Compressed Air Systems Specialists

ENGINEERING REPORT

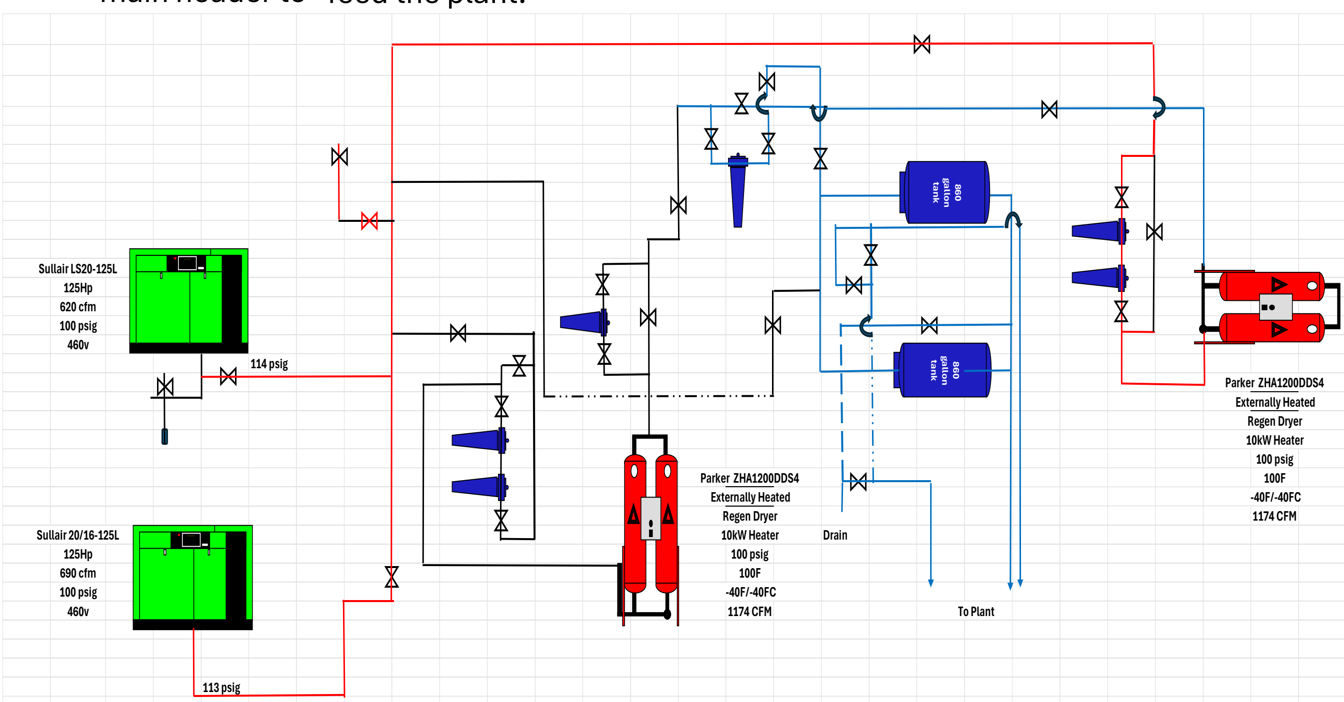
Engineering Report:

Compressor Area

[Company Name] has one main compressor area located in the Water Treatment Building.

Water Treatment Area

This area contains two air compressors. The first is a Sullair model 20/16 – 125L oil-flooded rotary screw compressor. The unit is an air-cooled, oil-flooded, rotary screw compressor designed to produce approximately 690 cfm at 100 psig, utilizing a 125 Hp, 460-volt motor. The second unit is a Sullair LS20S – 125L oil-flooded rotary screw compressor. The unit is an air-cooled, oil-flooded, rotary screw compressor designed to produce approximately 620 cfm at 100 psig, utilizing a 125 Hp, 460-volt motor. During the audit both compressors were on-line. From the compressors air is sent into one of two identical dryers. Each is a Parker model ZHA1200DDS4 externally heated desiccant dryer. It is designed to dry 1174 cfm of air at 100F and 100 psig to a -40F dewpoint utilizing a 10kw heater and 7% - 10% purge air (84 cfm – 120 cfm) for desiccant regeneration. From the dryer the air feeds into two 860 gallon "dry" air receivers before discharging into the 3" main header to feed the plant.



Note:

The sketch above reflects the air circuit with the corresponding drier that was on-line.

Blue lines represent dried air.

Red lines represent wet air

Black lines represent valved off

[Company Name] Compressor & Dryer List

[Company Name] Compressor List

Comp#	Area	Type	Air/Water Cooled	Lube/Oil-Free	Manufacturer	Model	Published HP	Published CFM	Rated Pressure	Actual Avg Pressure	Actual Avg BHP	Actual Avg CFM	Voltage	Status
1	Water Treatment	Rotary Screw	Air	Oil-Flooded	Sullair	LS20-125L	125	620	100	105	52.3	0	460	On-Line
2	Water Treatment	Rotary Screw	Air	Oil-Flooded	Sullair	20/16-125L	125	690	100	105	125	636	460	On-Line

[Company Name] Dryer List

Dryer #	Area	Type	Manufacturer	Model	Published CFM	Published Dewpoint	Actual Dewpoint	Heater kW	Purge Cfm	Voltage	Status
1	Water Treatment	Regenerative	Parker	ZHA1200DDS4	1174	-40F	**-88.25F**	10	120	460	Off-Line
2	Water Treatment	Regenerative	Parker	ZHA1200DDS4	1174	-40F	**-88.10F**	10	120	460	On-Line

Plant Air Pressure, Bhp & Air Demand

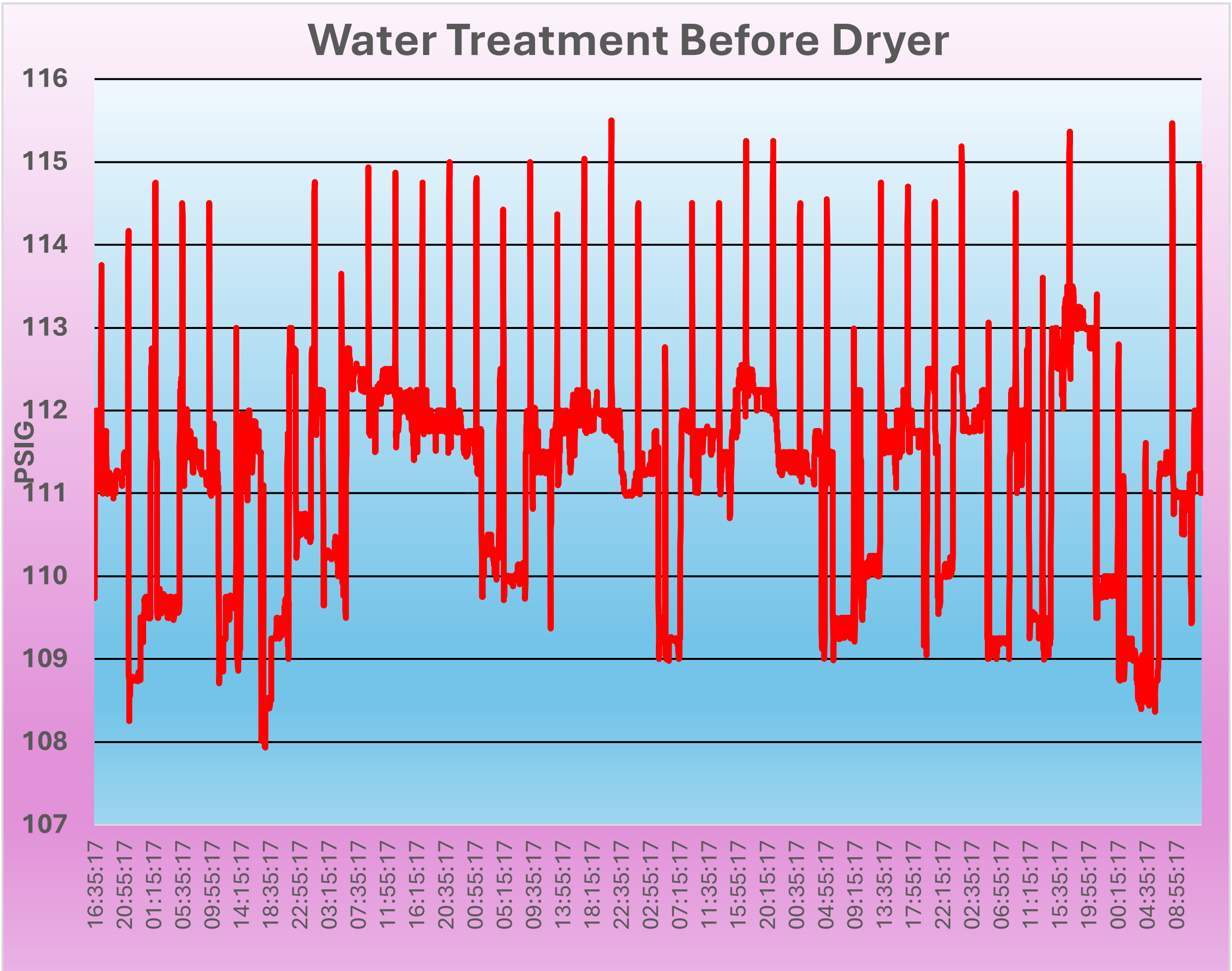
The following graphs are the recorded data from 04-21-26 thru 04-28-26. The location of each can be seen on the following drawing with red dots representing the locations:

Air Pressure

1. Compressor Discharge Before Dryer

At this point the pressure fluctuated between 116 psig and 108 psig with an average pressure of 111.1 psig. Thus, the pressure fluctuates 8 psig.

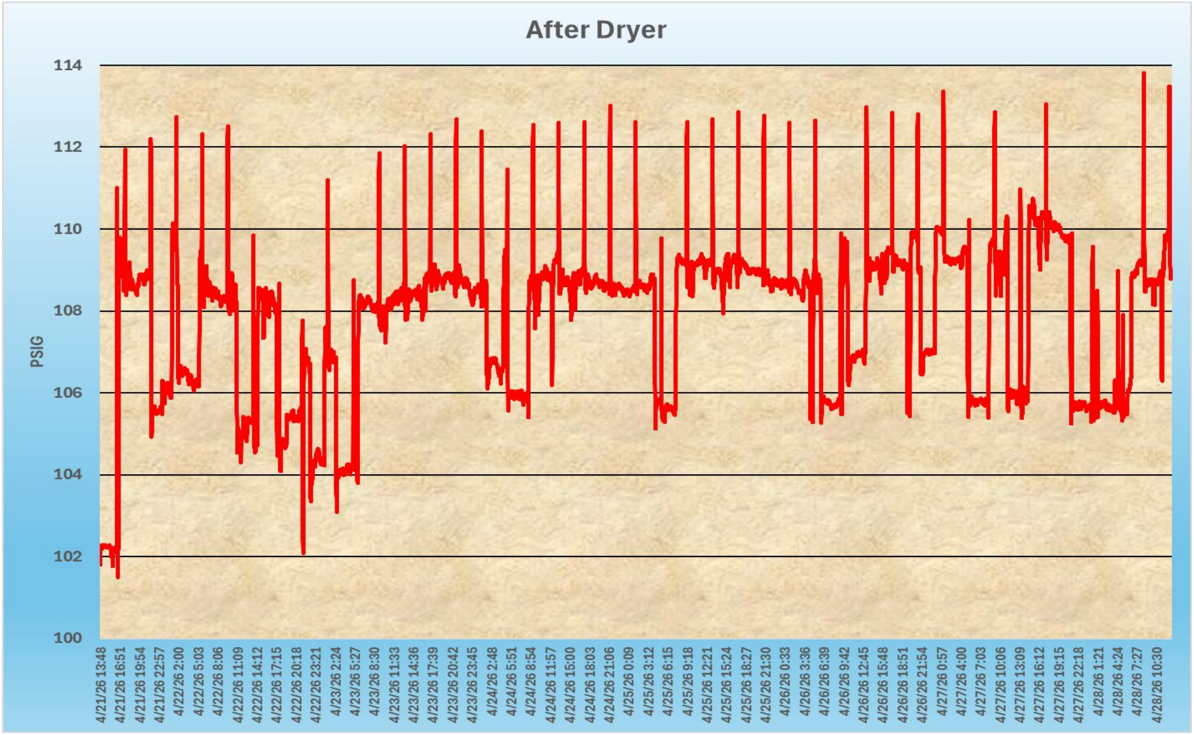
Graph #1:



2. Water Treatment Area After Dryer

At this location the pressure fluctuated between 113.4 psig and 101.5 psig with an average pressure of 107.8 psig. Pressure fluctuates up to 11.9 psig.

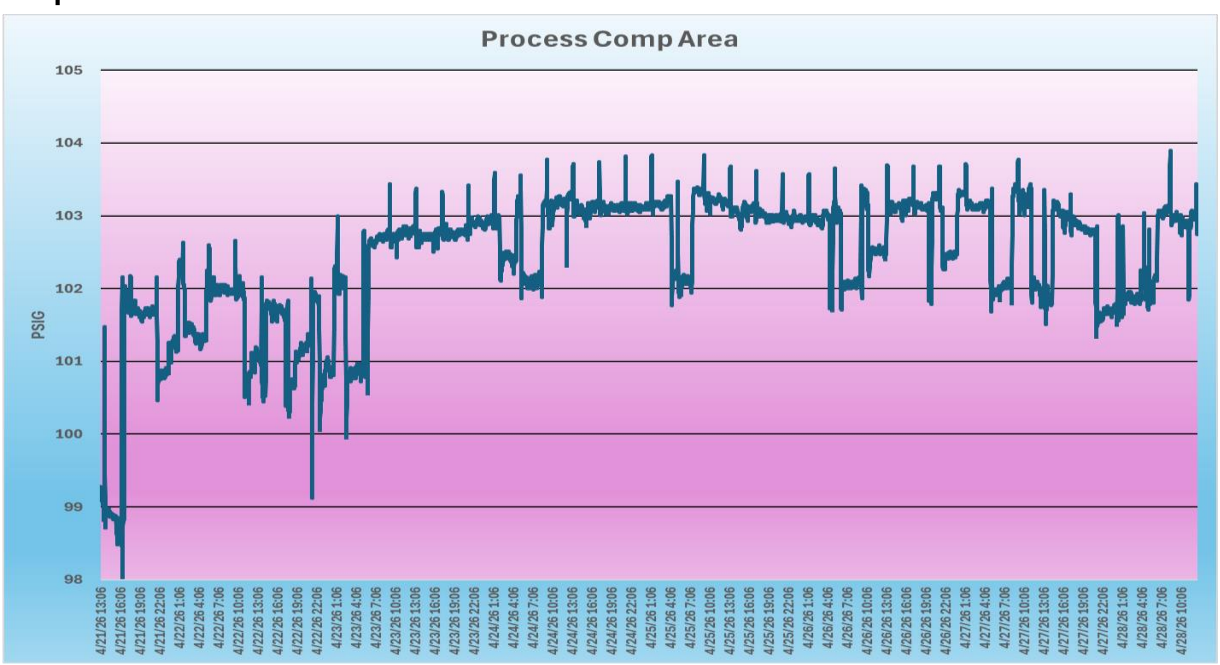
Graph #2:



3. Process Compressor Area

Pressure in this area fluctuated between 103.8 psig and 98 psig (5.8 psig delta P) with an average pressure of 102.4 psig.

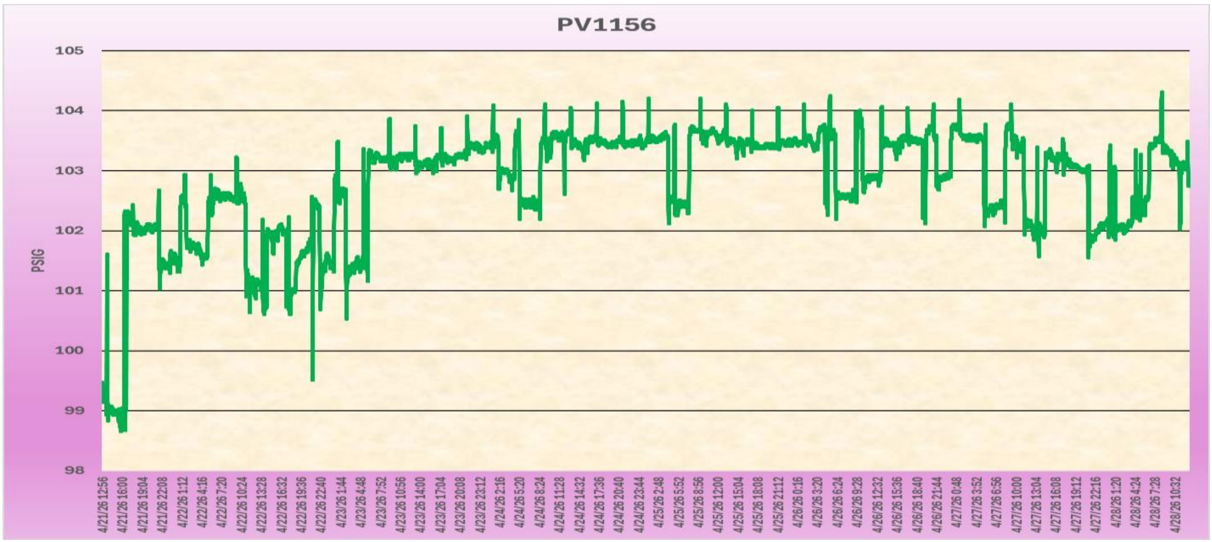
Graph #3:



4. PV 1156/1157

Pressure in this area fluctuated between 104.3 psig and 98.7 psig (5.6 psig delta P) with an average pressure of 102.8 psig.

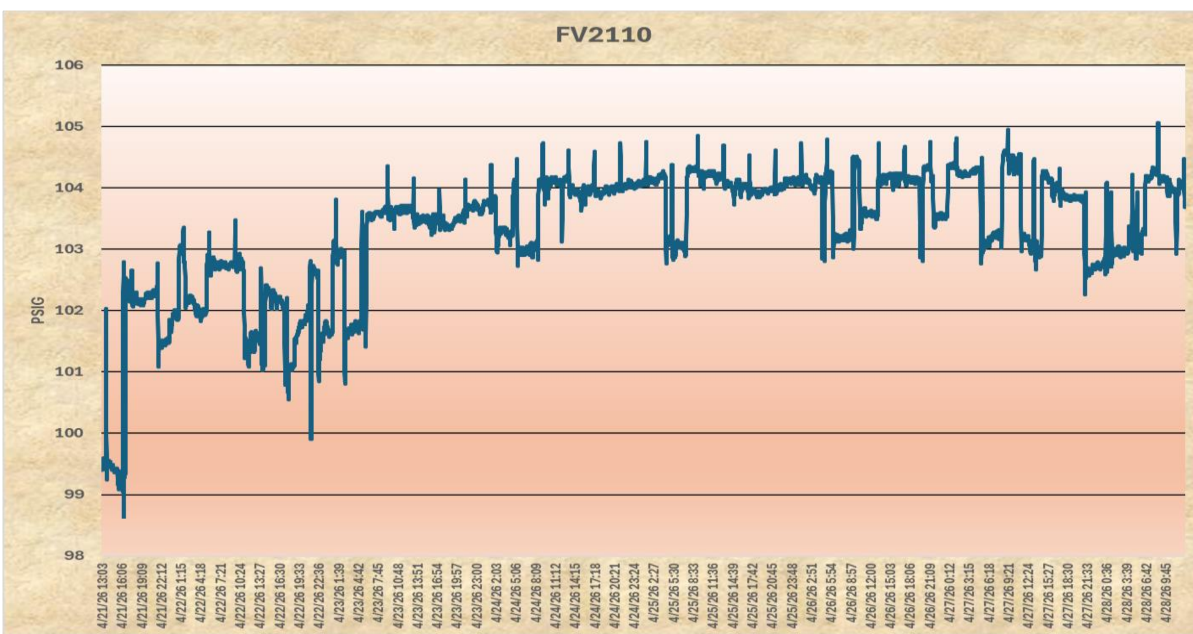
Graph #4:



5. FV2110

Pressure in this area fluctuated between 88.7 psig and 103.5 psig (14.8 psig delta P) with an average pressure of 98.6 psig.

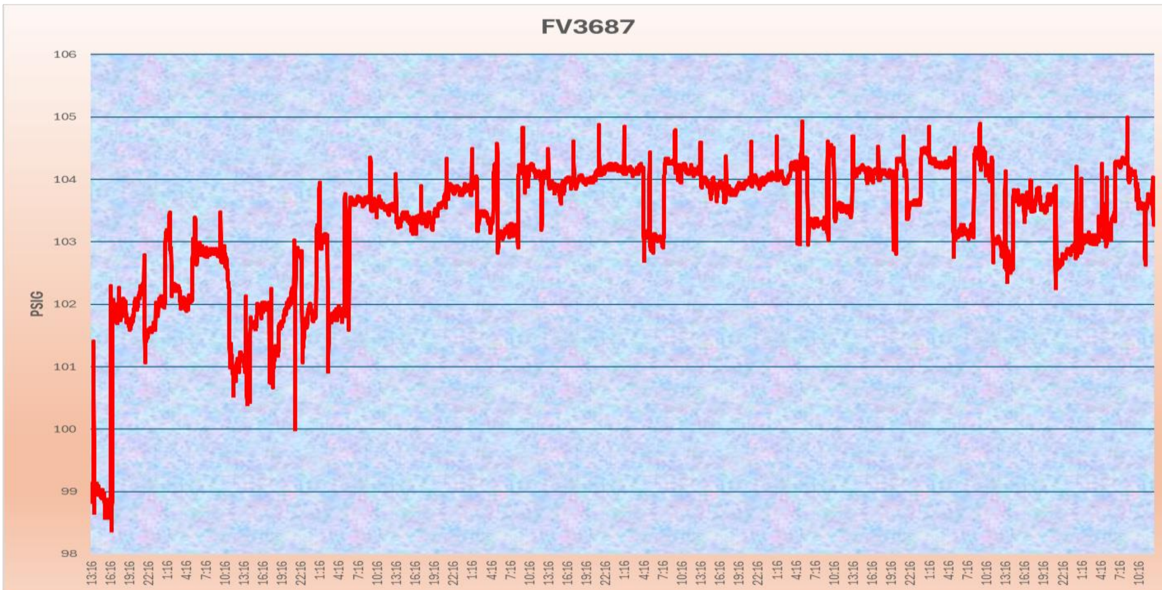
Graph #5



6. FV3687

Pressure in this area fluctuated between 104.9 psig and 98.4 psig (6.5 psig delta P) with an average pressure of 103.3 psig.

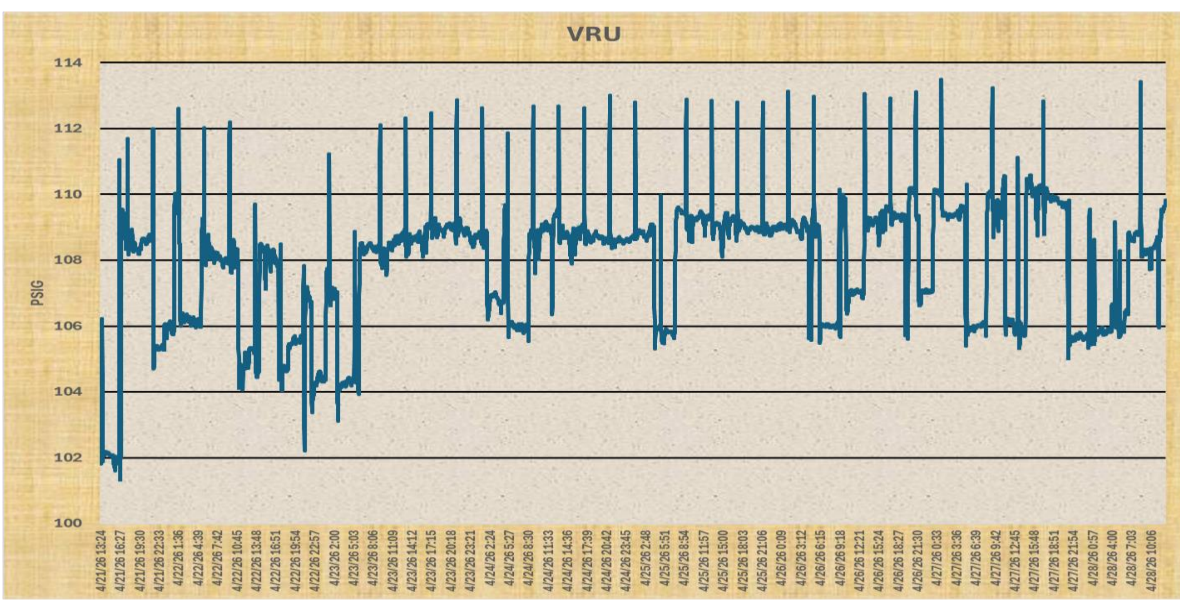
Graph #6



7.0 VRU

Pressure in this area fluctuated between 113.5 psig and 101.3 psig (13.2 psig delta P) with an average pressure of 107.8 psig.

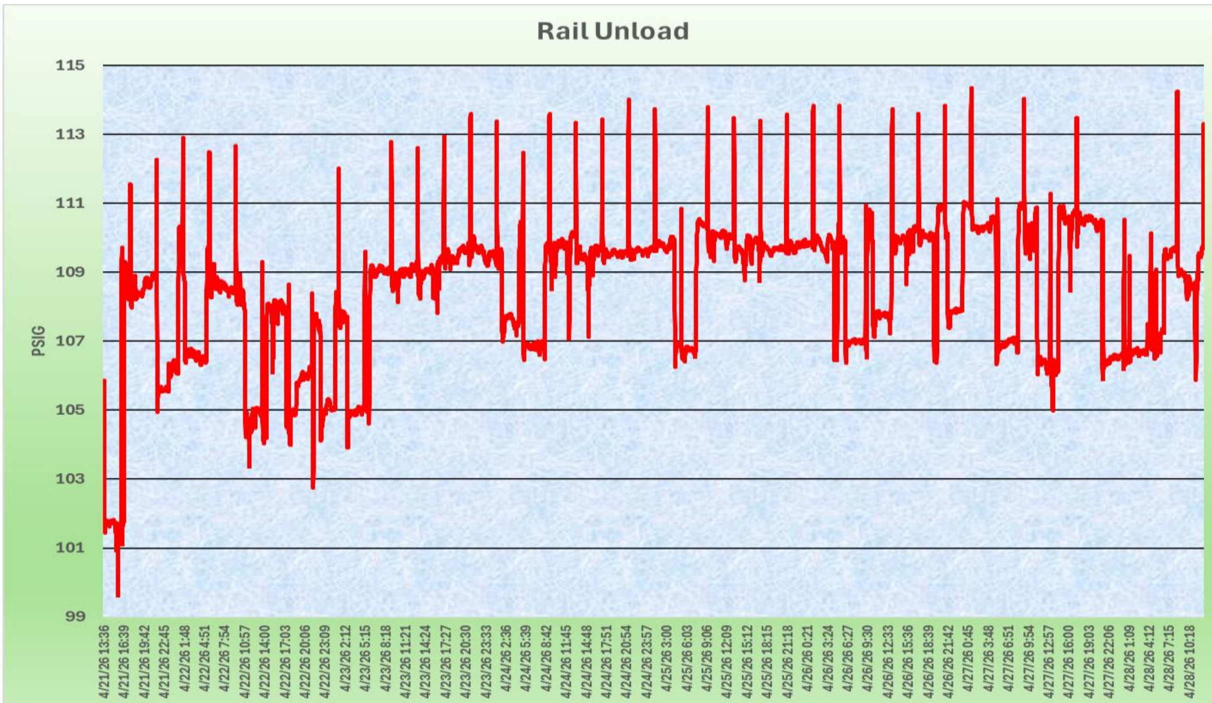
Graph #7



8.0 Rail Unload

Pressure in this area fluctuated between 114.4 psig and 99.6 psig (14.8 psig delta P) with an average pressure of 108.5 psig.

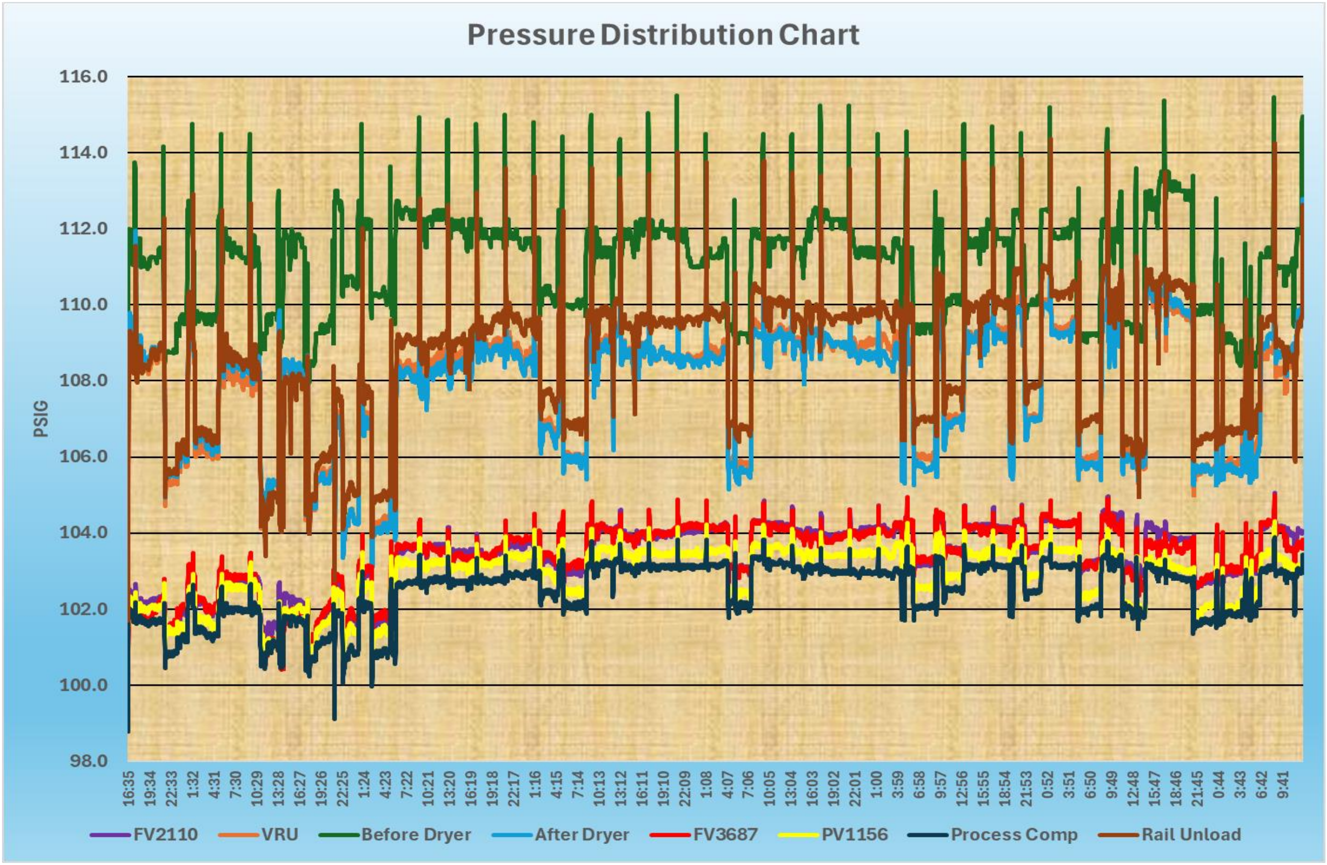
Graph #8



Overall Pressure Distribution Profile

The graph below depicts the pressure profile of the plant. It is evident that an event in any part of the plant affects all areas of the plant. What is of interest is that the FV2110, Process Comp Area, PV1156, and FV3687 operate approximately 10 psig below the rest of the recorded areas. These areas are fed from the air-lines that have a pressure regulator.

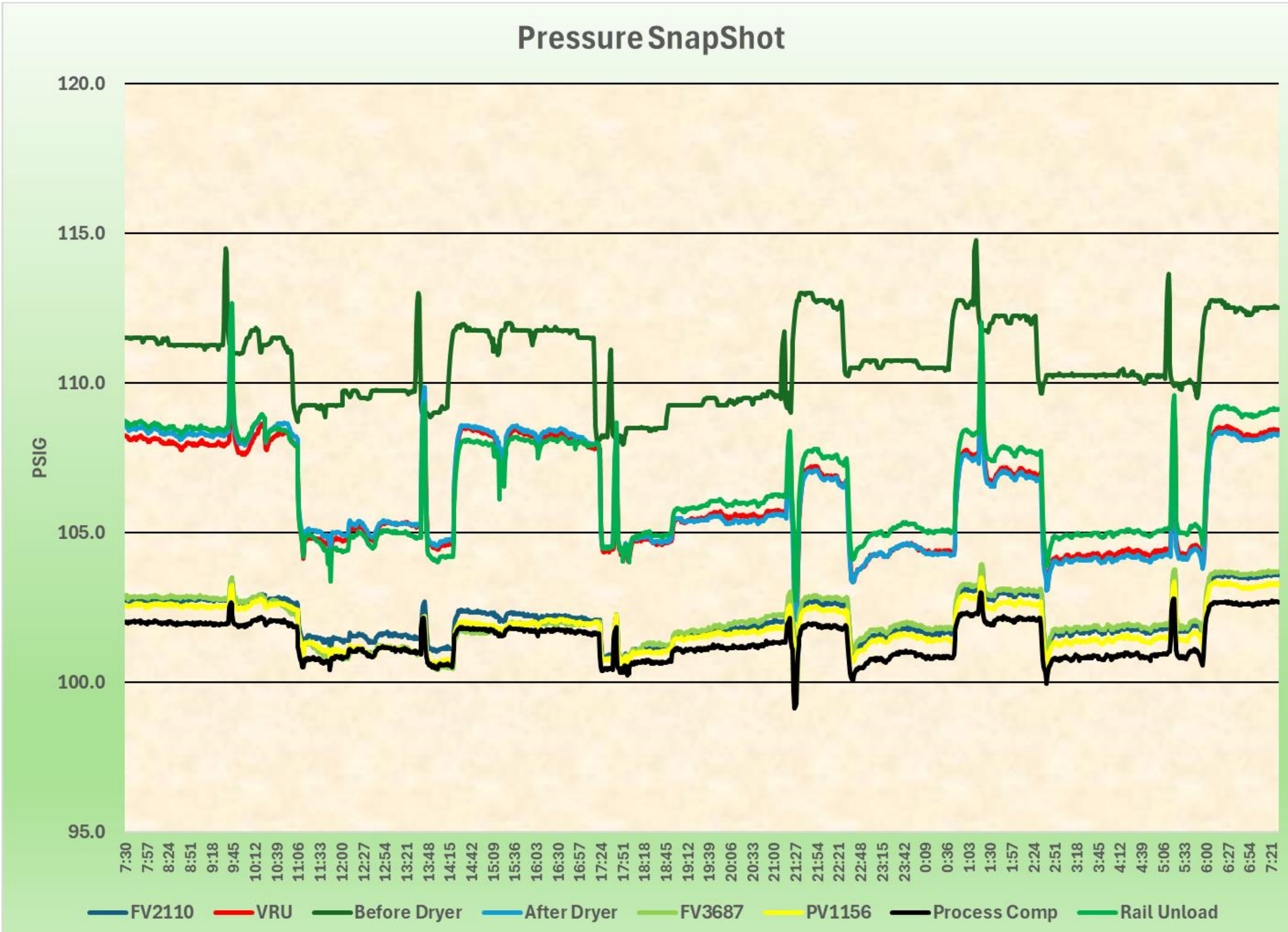
Graph #10:



Pressure Snapshot

The snapshot below of the pressure profile, shows that all parts of the plant are connected. Whenever and wherever an event occurs, it affects all parts of the plant. The graphs also show that the pressure is not consistent and fluctuations occur continuously due to a lack of useful storage.

Graph #11:



BHP & Demand

The Bhp was obtained by attaching amp meters to the compressors online. The resultant Bhp of the system was ascertained by converting the amps to Bhp as follows:

BHP = (Amps x 1.73 x 460v x .9 PF) /0.746 / 1000

kW = (Amps x 1.73 x 460v x 0.9PF)/1000

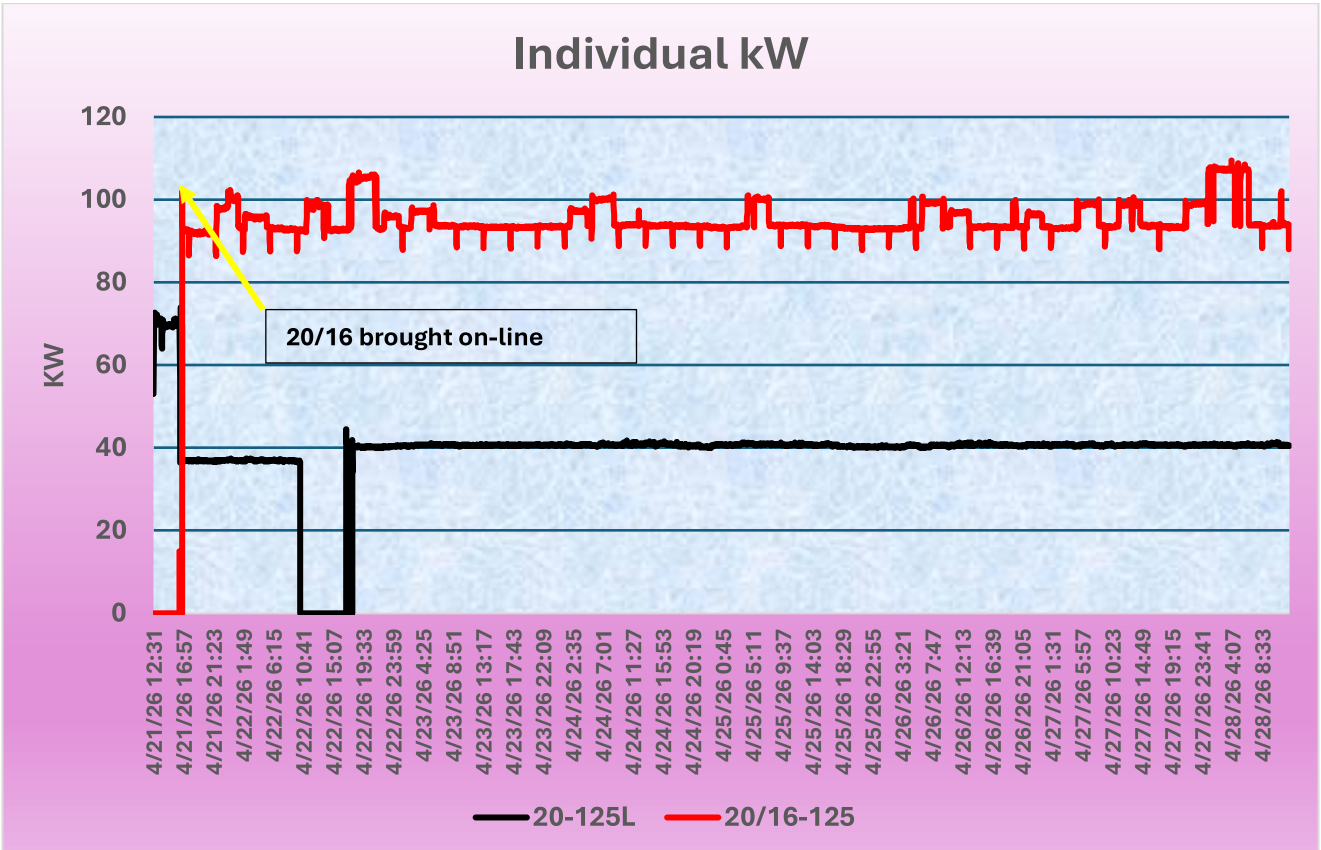
Power Factor (PF) was estimated at 0.9

Flow was ascertained using the recorded bhp and CAGI published CFM data for the compressors. Computer modeling converts amp draw to BHP and CFM.

Compressor kW

The graph below depicts the kW of each compressor over the course of the entire audit. Note the 20/16-125 was off-line for service during the start of the audit. The LS-20-125L and rental were on-line. The rental was not recorded but remained off-line after the 20/16-125 came back on-line after servicing was completed.

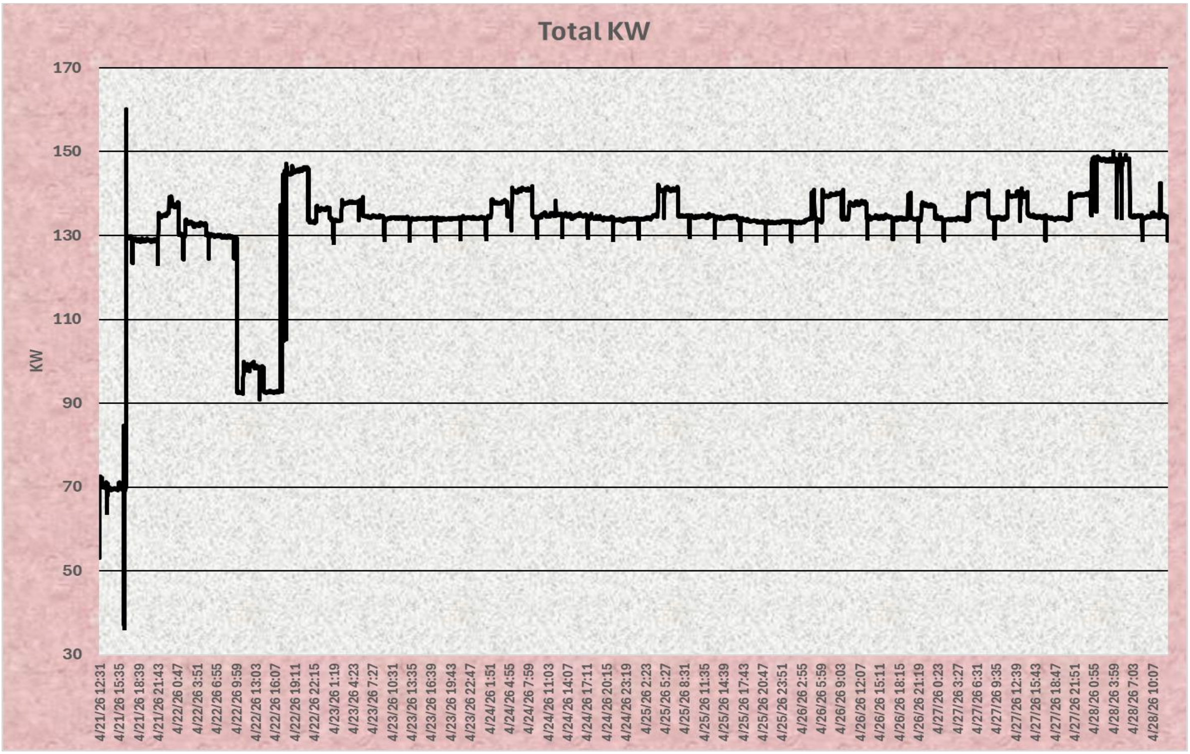
Graph #12:



BHP/kW Table

20/16-125	LS20-125L
Avg - 125 bhp/ 93 kW	Avg - 52.3 bhp/ 39 kW
Min - 115.3 bhp/ 86 kW	Min – 0 bhp/ 0 kW
Max – 146.1 bhp/ 109 kW	Max – 109.5 bhp/ 72 kW

Graph #10:



Total Average BHP = 177 bhp   kW = 132 kW

Minimum BHP = 122 bhp   kW = 91 kW

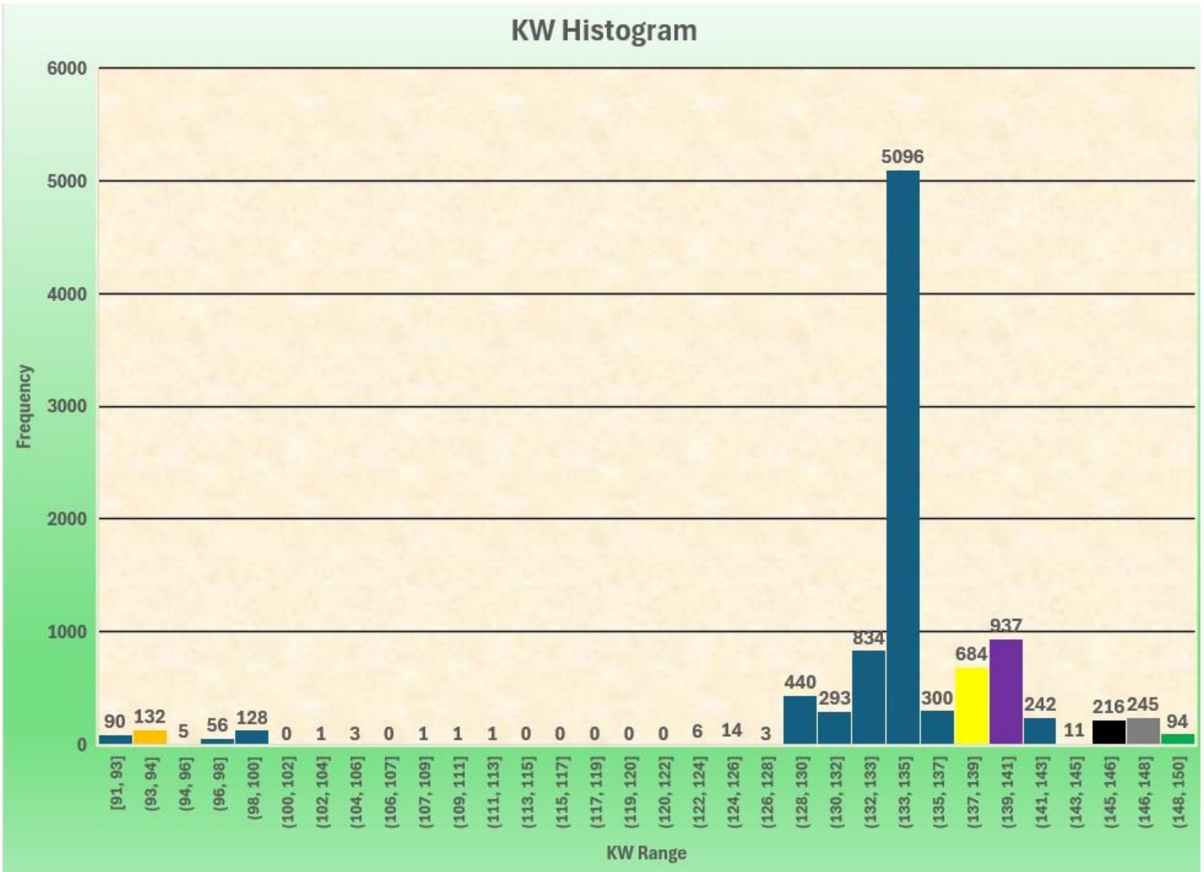
Maximum BHP = 201 bhp   kW = 150 kW

Note: The minimum and max values listed above exclude the start-up of the 20/16-125 unit.

kW Histogram

The kW histogram is designed to show the number of times (frequency) a particular kW range was on-line during production. The following chart depicts the frequency breakdown for the plant. The range of 132kW to 135 kW is the most predominant.

Graph #11:



Plant Air Demand

Individual Compressors CFM

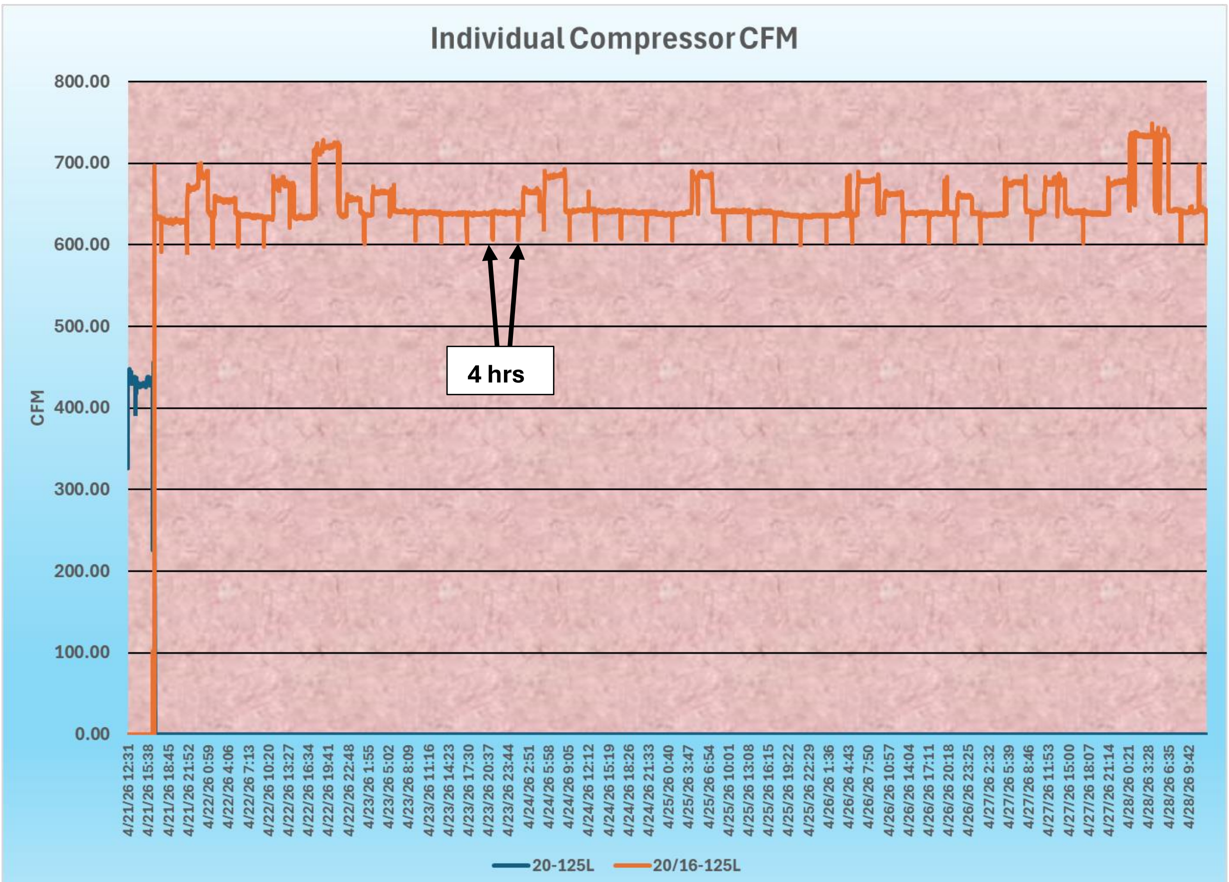
The graph below depicts the CFM produced by each compressor over the course of the entire audit.

CFM / M(3)/Min

20/16-125	LS20-125L
Avg - 636 cfm / 18 m(3)/min	**Avg - 10.62 cfm / 0.283m(3)/min**
Min - 591cfm / 16.73 m(3)/min	**Min – 0 cfm / 0 m(3)/min**
Max – 748 cfm / 21.18 m(3)/min	**Max – 444 cfm / 12.57 m(3)/min**

It should be noted that the small dips in demand on the following chart may be due to the dryer purge cycle. Each occurs approximately every 4 hours. When the dryer stops purging the demand drops. This cycle repeated throughout the audit. When this cycle occurs, there is also a corresponding rise in pressure of 4 psig for the same duration.

Graph #12:



CFM Table

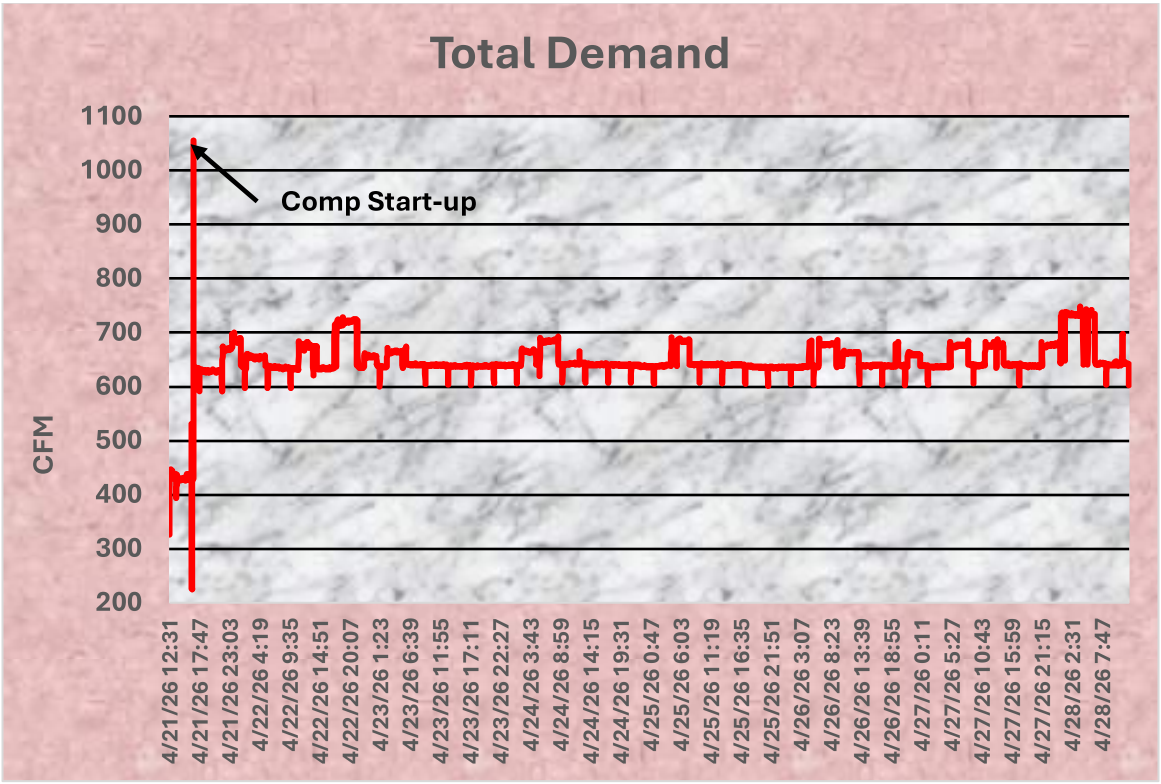
The following chart depicts the total demand of the plant over the course of the audit. Excludes when rental was in service.

Overall Average Demand = 646 cfm /18.29 m$^{(3)}$/min

Peak Demand = 741 cfm / 20.98 m$^{(3)}$/min

Min Demand = 591 cfm / 16.74 m$^{(3)}$/min

Graph #13:

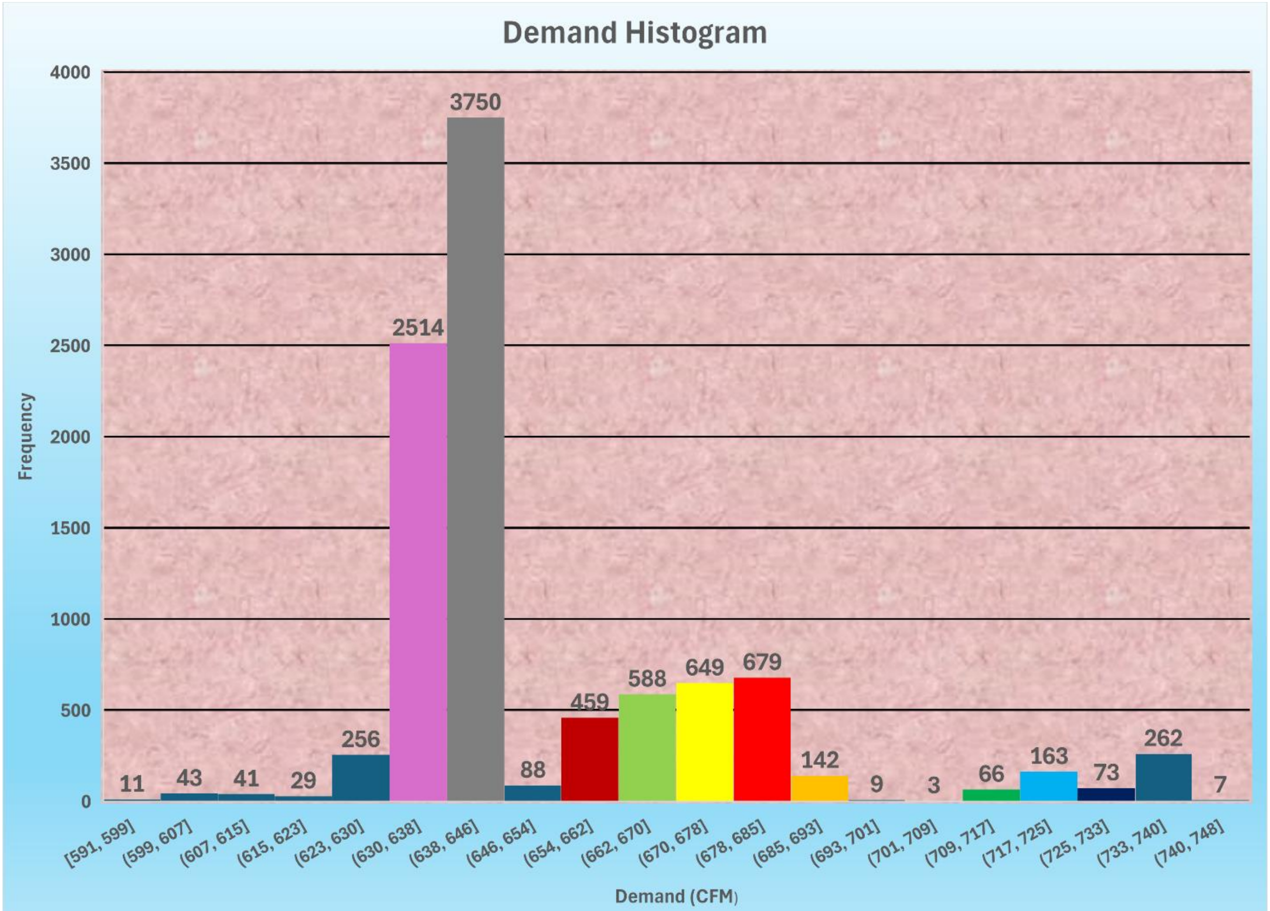


Demand Histogram

The Demand histogram is designed to show the number of times (frequency) a particular range of cfm was on-line during production. The following chart depicts the frequency breakdown.

The histogram shows that the predominant flow range to be 638 cfm – 646 cfm (18.0 m^3/min – 18.29 m^3/min).

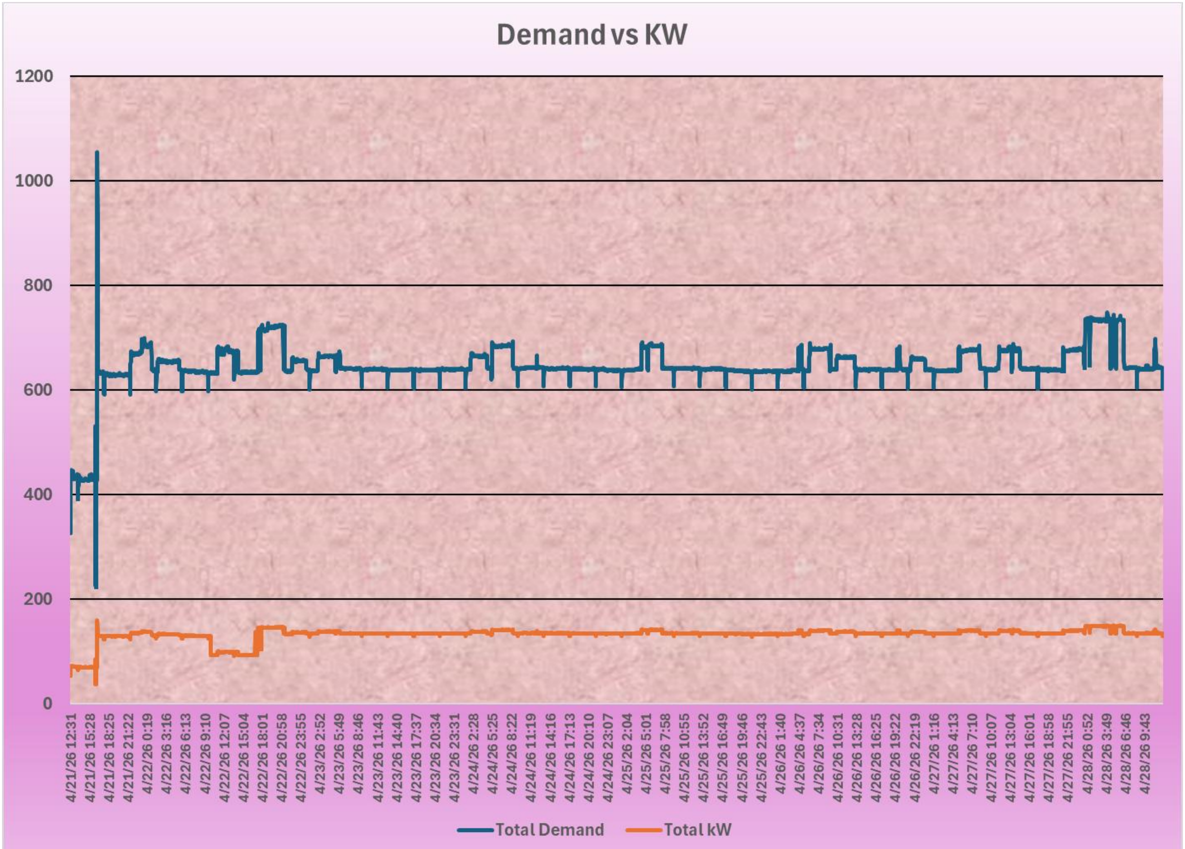
Graph #14:



Demand vs BHP

The following graph shows the relationship between demand and power consumption. There is a corresponding drop in kW to match the reduction in demand that occurs every 4 hours and a pressure increase of 4 psig. In addition there is very little fluctuation in demand and horsepower but there is a 10 – 12 psig fluctuation in pressure.

Graph #15:



Friction Loss

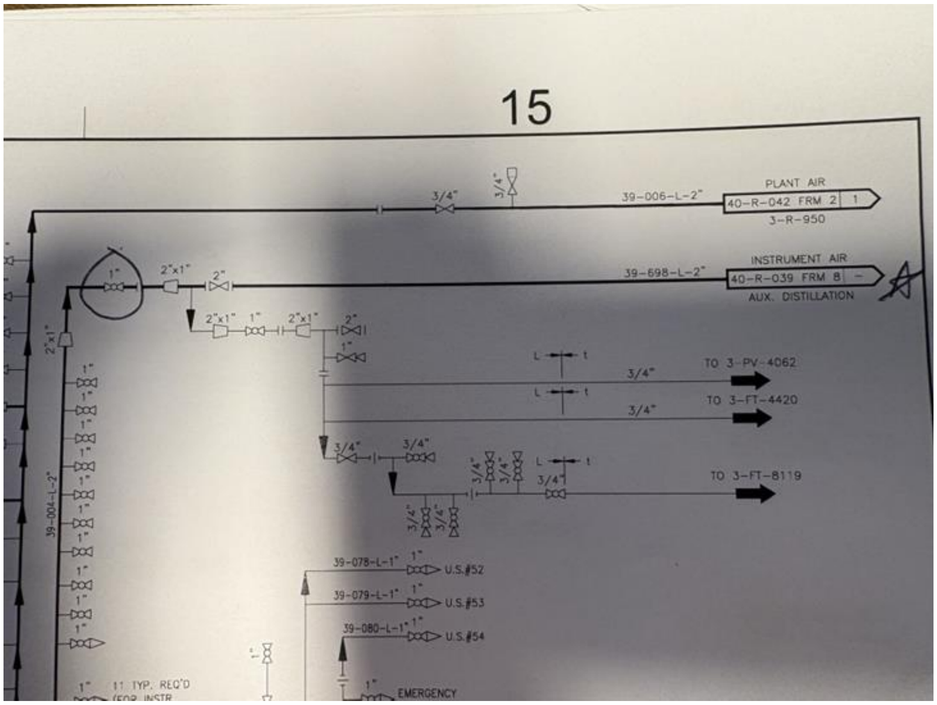
Based on the data obtained, friction loss is minimal. Analysis of the pressure charts above clearly shows very little difference from the water treatment area (compressors) to the rest of the plant. The exceptions are FV2110, PV1156, FV3687 and Process Comp Area which shows an average pressure approximately 5 psig (34.47 kPa) below the rest of the plant. This would indicate the lines are regulated.

Reducer Valve 40-R-039 FRM 8 (See drawing below)

There is a reducer valve which reduces the 2" header to 1" and expands back to 2". This would affect the following lines:

1. 40-R-039 FRM 8
2. 3-PV-4062
3. 3-FT-4420
4. 3-FT-8119

The amount of pressure drop across this reducer is based on the flow passing through it. Since the flow rate was not available, a table (below) was produced that shows the effect of flow in relation to delta P.



Pressure Drop

- **Fluid:** Compressed air
- **Line pressure: 100 psig** (114.7 psia)
- **Temperature:** ~68 °F (20 °C)
- **Pipe:** Schedule 40 steel
  - 2" ID = **2.067 in**
  - 1" ID = **1.049 in**
- **Geometry:**
  2" → reducer → **1 ft of 1"** → expander → 2"
- **Losses included:**
  - Friction in 1 ft of 1" pipe
  - Reducer (K ≈ 0.5)
  - Sudden expander (K ≈ 1.0)

1. Flow Conversion (SCFM ↔ ACFM)

When the flow is given as **SCFM**, it must be converted to **actual volumetric flow in the pipe (ACFM)** at line conditions.

$$\boxed{ACFM = SCFM \times \frac{P_{std}}{P_{line}} \times \frac{T_{line}}{T_{std}}}$$

For your case (same temperature):

$$ACFM = SCFM \times \frac{14.7}{P_{psig} + 14.7}$$

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2. Air Density (Ideal Gas Law)

Air density at **line pressure** is calculated using:

$$\boxed{\rho = \frac{P_{abs}}{R\,T}}$$

Where:

- P_{abs}= absolute pressure (Pa)
- R= 287 J/(kg·K)
- T= absolute temperature (K)

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3. Velocity in the Pipe

$$\boxed{v = \frac{Q}{A}}$$

With:

- Q= volumetric flow rate (m^3/s)
- $A = \frac{\pi D^2}{4}$

Used separately for:

- 2" pipe velocity
- 1" pipe velocity

4. Reynolds Number (Check Flow Regime)

$$\boxed{Re = \frac{\rho v D}{\mu}}$$

Where:

- μ= dynamic viscosity of air (~1.85 × 10^{-5} Pa·s)

All cases are **fully turbulent**.

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5. Darcy Friction Factor (Turbulent, Smooth Pipe Approximation)

Blasius correlation (valid for turbulent flow):

$$\boxed{f = \frac{0.3164}{Re^{0.25}}}$$

This is acceptable for short steel piping sections like this.

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6. Friction Loss (Darcy–Weisbach)

Used for the **1 ft length of 1" pipe**:

$$\boxed{\Delta P_{friction} = f\frac{L}{D} \cdot \frac{\rho v^2}{2}}$$

Where:

- L= pipe length
- D= pipe inside diameter

---

7. Minor Losses (Reducer + Expander)

$$\boxed{\Delta P_{minor} = \sum K \cdot \frac{\rho v^2}{2}}$$

For your geometry:

- Sudden reducer: $K \approx 0.5$
- Sudden expander: $K \approx 1.0$

$$\sum K = 1.5$$

Velocity used: upstream (2") velocity — conservative and standard practice.

8. Total Pressure Drop

$$\boxed{\Delta P_{total} = \Delta P_{friction} + \Delta P_{minor}}$$

Converted from pascals to psi:

$$1 \text{ psi} = 6894.76 \text{ Pa}$$

Summary (All in One Line)

$$\boxed{\Delta P = \left[f\,\frac{L}{D_{1"}} + (K_{red} + K_{exp})\right]\frac{\rho v^2}{2}}$$

Pressure drop across the reducer is dependent on the flow through it. The table below list a flow rate and associated pressure drop.

Pressure Drop vs Calculated Flow (SI Units) Through the Reducer

(at 100 psig air)

Actual Flow (m^3/s)	Velocity in 1" Pipe (m/s)	Pressure Drop (kPa)
0.024	42.3	1.95
0.047	84.6	7.08
0.071	127.0	15.1
0.094	169.3	25.9
0.118	211.6	39.5
0.142	253.9	55.7
0.165	296.2	74.4
0.189	338.6	95.8
0.212	380.9	119.7
0.236	423.2	146.1

For the reducer valve to induce a 3 psig to 5 psig drop infers 175 cfm (4.96 m^3/min) to 250 (7.08 m^3/min) cfm would need to be delivered to the reducer.

Compressed Air System Efficiency (CFM Delivered Per BHP Input)

[Company Name] compressed air system produces a certain amount of compressed air or CFM for every 1 BHP of input energy. This is commonly called the CFM/BHP ratio. The compressor's type and its design discharge pressure, will determine the cfm/ bhp ratio as listed below
(electric motors):

1. Single Stage Oil Flooded Rotary Screw @ 100 psig: 4.0 - 4.6 cfm/bhp (3.8 - 4.3 cfm/Bhp @ 125 Psig).

2. Two Stage Oil-Free Rotary Screw @ 100 psig: 4.6 – 5.2 cfm/bhp (4.2 - 4.7 cfm/Bhp @ 125 Psig).

For rotary screw and reciprocating compressors (oil free & oil flooded), the cfm/bhp ratio is regardless of the size of the compressors or ambient conditions. The above is based on electric drive compressors.

The following calculations are derived using the manufacturers published data. The overall published cfm/bhp will be compared to that of the actual data recorded at Caros Chemical over the course of the audit. This includes the cfm and bhp.

Individual Compressor Efficiency

Sullair 20/16 IR-125L Compressor

According to the compressor manufacturer's published data, the cfm produced from this unit is 690 cfm at 100 psig, at a full load Bhp of 135 Bhp (includes fan motors). Thus the manufacturers stated cfm/bhp ratio is 690 cfm / 135 bhp / = 5.11 Cfm/Bhp.

The actual measured data of the compressor, on average, produced 636 cfm and consumed 128 bhp at 100 psig discharge pressure. The resultant Scfm/Bhp ratio is calculated as follows:

636 Cfm / 128Bhp = 4.97 Cfm/Bhp

The compressor operates in accordance with manufacturers specification.

Sullair LS20L-125L Compressor

According to the compressor manufacturer's published data, the cfm produced from this unit is 620 cfm at 100 psig, at a full load Bhp of 135 Bhp. Thus the manufacturers stated cfm/bhp ratio is 620 cfm / 135 bhp / = 4.59 Cfm/Bhp.

The actual measured data of the compressor, on average, produced 0.0 cfm and consumed 52.66 bhp at 100 psig discharge pressure. The resultant Scfm/Bhp ratio is calculated as follows:

0.0 Cfm / 52.3 Bhp = 0 Cfm/Bhp

This unit is within manufacturers specification.

Overall System Efficiency

According to the compressor manufacturer's published data, the total cfm available is 1310 cfm at 100 psig, at a full load Bhp of 27 Bhp (includes fan motors). Thus the manufacturers stated cfm/bhp ratio is 1310 cfm / 270 bhp / = 4.85 Cfm/Bhp.

The actual measured data of the compressors, on average, produced 646 cfm and consumed 174.8 bhp at 100 psig discharge pressure. The resultant Scfm/Bhp ratio is calculated as follows:

646 Cfm / 180.3 Bhp = 3.58 Cfm/Bhp

The system is operating 26% below manufacturers specifications.

Plant Air System Estimated Pressure & Flow Requirements

Plant Air Pressure Requirement

As the pressure data logs depict, at various times throughout the day, the air system pressure fluctuated between 114 psig to 102 psig at the compressor area and in the plant. An average overall delivered pressure of approximately 102.7 psig. The air system could possibly operate properly at 90 psig (620 kPa)or lower. The lower the pressure to the Plant, the more energy savings can be attained. It will also create more useful storage in the primary air receivers. Note, for energy savings calculations, 90 psig will be used as the minimum pressure requirement.

Lowering the pressure to the Plant, will also reduce the amount of cfm consumed. An example would be to view the Plant as an open orifice. Thus, the total amount of air consumed at 115 psig, can be compared to that consumed at 90 psig as follows:

14.485 x D^2 x (psig + 14.7) = “X” SCFM

Diagram 1: 115 psig

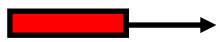
1” diameter nozzle with — Yields

Inlet pressure  1878 cfm output

of 115 psig

Diagram 2: 90 psig

1” diameter nozzle with — Yields

Inlet pressure  1516 cfm output

of 90 psig

Whereas diagram 1 shows a flow of 1878 cfm at 115 psig, diagram 2 shows a flow of 1516 cfm at 90 psig using the same size nozzle. When the pressure is reduced from 115 psig to 90 psig the overall cfm consumption drops by 20%. That is every piece of unregulated equipment in the facility would consume 20% less compressed air. As a rule of thumb, *for every 10 psig drop in consumption pressure equates to a 5% savings in energy to produce compressed air.*

The above examples shows that by lowering pressure to the Plant, air consumption will also decrease. All unregulated users of compressed air will consume less cfm including air leaks. This will affect the operation/performance of the air compressors.

Dealing with Mass Flow, a computation can be performed to calculate the proposed demand of the system. Based on increasing storage and replacing a the 20-125L compressor we have the following information:

Calculation:

Weight of Air @ 125 psig = 0.7125 lb/cu.ft. @ 70F (compressor rated discharge pressure)

Weight of Air @ 90 psig = 0.585 lb/cu.ft. @ 70F (proposed system pressure)

CFM @ 125 psig = “X” cfm

CFM @ 90 psig = 878 cfm

Thus: 646 cfm x 0.585 lb/cu.ft. = “X” cfm x .7125 lb/cu. ft

Solving for "X" = 646 cfm at 90 psig = 530 cfm @ 125 psig

This would occur with proper useful storage and pipe size.

Calculation of Useful Storage Based on 3720 Gallon Storage

Currently there is no useful storage in the system. However, implementing the recommendations listed below, the useful storage would be calculated as follows:

P(in) = 125 psig (selected compressor discharge pressure)

P(out) = 90 psig

Storage = 3720 gal = 497.3 cu.ft.

Useful storage = (125 psig – 90 psig) * 497.3 / 14.7 = 1184 cu.ft.

In conjunction with a flow or pressure control valve, the Plant would have 1184 cu.ft. of useful stored air. Thus, the Plant would not lose 1 psig of pressure until the useful storage was depleted. Based on 646 cfm demand there would be approximately 1.8 minutes of useful storage before the plant would see any change in pressure. Ample time for the compressors to respond to demand changes in the storage.

Compressed Air Dewpoint

Dewpoints were taken at various locations around the plant. The table below shows the locations and dewpoints recorded. All dewpoints recorded were well above the required -50F/-45.6C except for the Rail Loading. This area is the furthest point from the air dryers. It does not have a continuous demand and therefore it is susceptible to stagnation dewpoint as described below:

With **low or intermittent flow**, air can:

- Sit stagnant in the pipe
- Cool to ambient temperature
- Pick up moisture from:
    - Residual condensate
    - Internal pipe walls
    - Dead-end volume downstream of regulators or valves

When flow finally resumes, that moisture is carried downstream and shows up as a **much higher dewpoint reading.**

Area Dewpoints

DewPoint Table			
Area	Dewpoint	DewPoint	Min Required Dewpoint
	F	C	F/C
After Dryer 1	-56.2	-49	-50F/-45.6C
After Dryer 2	-88.2	-66.8	-50F/-45.6C
FV2110	-84.1	-64.5	-50F/-45.6C
FV3687	-70.62	-57	-50F/-45.6C
Lube Oil Process Compressor	-80.9	62.7	-50F/-45.6C
PV1156/1157	-88.25	-66.8	-50F/-45.6C
Rail Loading	-30	-34.4	-50F/-45.6C
VRU	-95.9	-71.1	-50F/-45.6C

Note: After Dryer 2 reading was taken at the same spot as After Dryer 1. The line was blown down to clear it of moisture

ISO Standards

Following the guidelines as set forth by ISO 8573.1 (see table below), for moisture, oil and particles in the air, [Company Name's] air quality is a class 1 except for Rail Loading which is a class 3. The amount of solid particles and vapor carryover downstream is provided in the table below. Should the compressed air be used for instrumentation, valves, etc. purposes the proper level for this region is class 1. This is to protect the valve from contamination and to prevent it from failing due to moisture/oil content and prolong the life of the component.

ISO 8573.1 Quality Classes

AIR QUALITY STANDARDS
ISO 8573-1 CLASSES

Class	**A:** Solid Particle - Maximum number of particles per m^3			**B:** Pressure Dew Point F° (C°)	**C:** Oil (incl. vapor) mg/m^3
	0.1-0.5 micron	0.5-1.0 micron	1.0-5.0 micron		
0	As specified by the end-user or manufacturer, and more stringent that Class 1				
1	≤ 20,000	≤ 400	≤ 10	≤ -94° (-70°)	0.01
2	≤ 400,000	≤ 6,000	≤ 100	≤ -40° (-40°)	0.10
3	—	≤ 90,000	≤ 1,000	≤ -4° (-20°)	1.00
4	—	—	≤ 10,000	≤ 37.4° (3°)	5.00
5	—	—	≤ 100,00	≤ 44.6° (7°)	—
6	—	—	—	≤ 50° (10°)	—

Compressed Air System Inefficiencies

1. There is no “Useful Storage” located within the compressed air system. Without useful storage the local compressor controls cannot react properly to demand changes in the Plant. Thus, pressure swings can and do occur. The result is that various parts of the facility are susceptible to low pressure. The current air receivers located in the plant act as transport mechanisms. After the receiver is filled, the inlet pressure equals the discharge pressure, thus there is no useful storage. Basically, the air receiver acts as a pipe, allowing the same amount of air and pressure to pass through, as enters it. To act as a true storage vessel, air must be stored at higher pressure and released to the Plant at a lower, controlled, pressure. This will create “Useful Storage”. Creation of useful storage will in effect allow compressors to run at optimal levels. Event demands will be satisfied from stored air in the receiver vessels and dampen the level of pressure fluctuations throughout the Plant. The result being elimination of low-pressure problems and savings of valuable operating dollars.

2. The LS-20S-125L is online consuming 52.3 bhp (39.0 kW) to produce no compressed air. **This costs [Company Name] $25,975 per year.**

3. The artificial demand of the system is the difference between the highest pressure, 114 and the minimum pressure required by the plant. In this case the minimum pressure required is 90 psig. The 24-psig difference equates to approximately 20 Bhp (14.92 kW) or in energy or **$9,933.00 per year**.

4. There is a 2” air-line, (40-R-039 FRM 8) that has a 1”reducer than returns to a 2” line. This may possibly induce an excess pressure drop in the line up to 5 psig.

5. Float and timed drains are used to remove condensate from coolers and filters. Each drain when engaged consumes 100 cfm. There are approximately 6 drains. They do not fire at the same time. Based on the length of time they are open it is estimated that this equates to a demand of 15 cfm on average or 3 bhp (2.238 kW) or **$1,490 per year** in lost energy.

6. Each of the dryers requires up to 120 cfm of compressed air and a 12kW heater to assist in the regeneration of the saturated desiccant bed.

Most heated dryers with an 8-hour cycle breakdown like this:

Per tower (one full cycle = 8 hrs total):

Phase	Duration (typical)	What's happening
Drying (online)	**4 hours**	Tower drying compressed air
Heating / Regen	**2–3 hours**	Heated purge air drives moisture out
Cooling	**1–2 hours**	

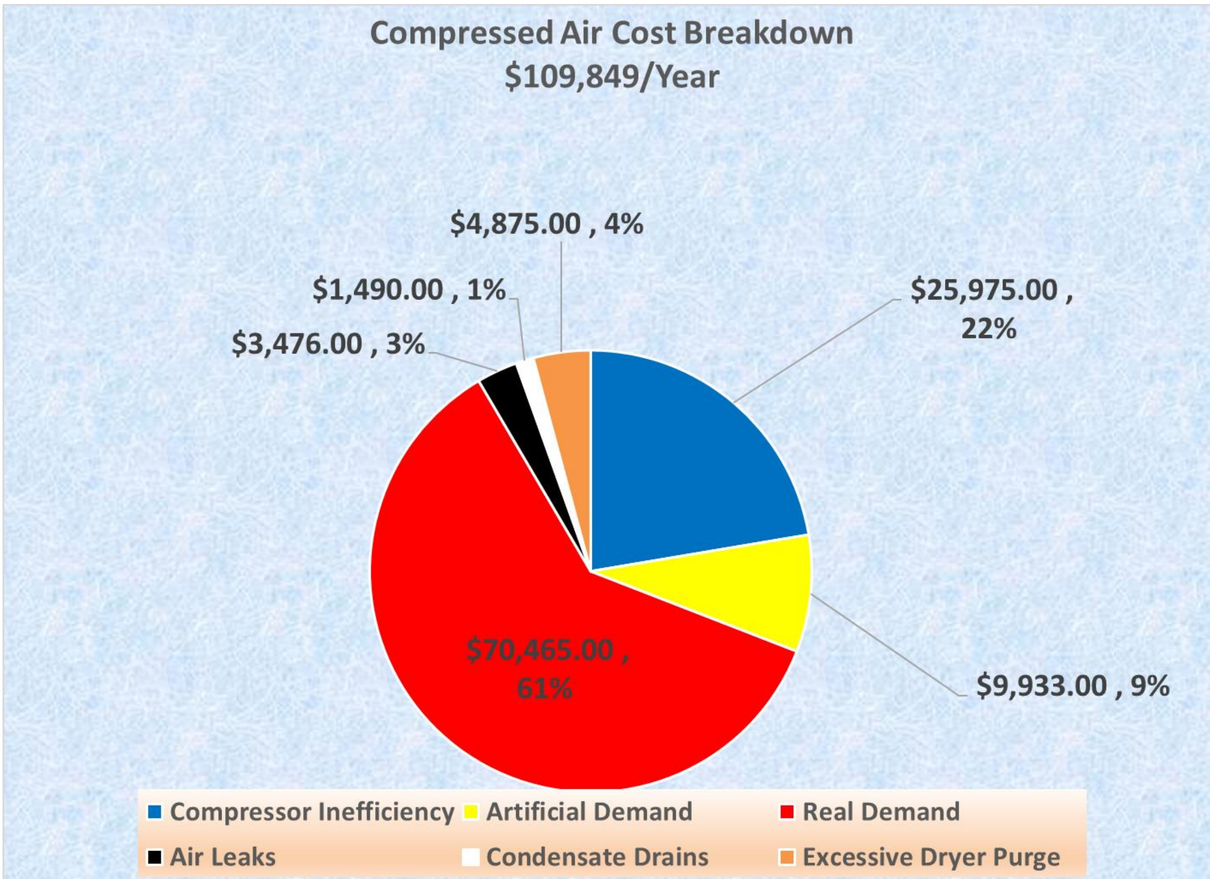
The purge air is on for approximately 4 hours and then stops. This coincides with the flow fluctuations exhibited above. Thus, the heater on average consumes 3 kW/hr and the purge air consumes 60 cfm continuously or 8.952 kW/hr for a total of 11.952k W/hr. This equates to **$7,957 per year.**

7. It is estimated that air leaks amount to approximately 4% of the total cfm consumed. This equates to 35 cfm or 7 hp costing [Company Name] **$3,476.00** per year.

8. Rail Unloading dewpoint is below acceptable levels as discussed above. Based on the low demand there is the potential of freezing in the line.

9. Pressure drop in FV2110 and PV1156 lines can be explained by faulty valves. There is a pressure drop in the FV3687 and Process Comp Area lines which is also inducing a 5 psig pressure drop induced by regulators but is not preventing pressure fluctuations from occurring.

Compressed Air Cost Breakdown



Recommendations

Priority I: Implement as soon as possible

1. Retrofit all condensate drains with air-free drains.

2. Remove the 1" reducers as described above. This will eliminate the current 5 psig delta P currently seen after this reducer.

3. Replace LS-20S-125L with a 200 hp compressor with variable frequency drive. Compressor should be capable of a full load pressure of 125 psig. The new unit will be capable of running the plant on its own. Currently both compressor are running.

4. Insert a flow meter at the rail unload to determine the demand consumed. This is to be done so that a remedy can be implemented to increase the dewpoint from -30F to a minimum -50F.

5. Check with the dryer manufacturer to see if there is an energy management system that can be retrofitted to the dryers. Since the dryers are rated for 1174 cfm and only 878 cfm is passing through it, an energy management system will ensure the system only switches towers, when necessary, not on a fixed cycle (8hrs). This will save on the current purge used by each dryer and cut energy consumption by more than half.

6. Implementation of flow control valve (see below) will resolve the pressure fluctuations in the FV3687 and Process Comp Area.

7. Turn the LS-20S-125L off or do it in conjunction with Priority II.

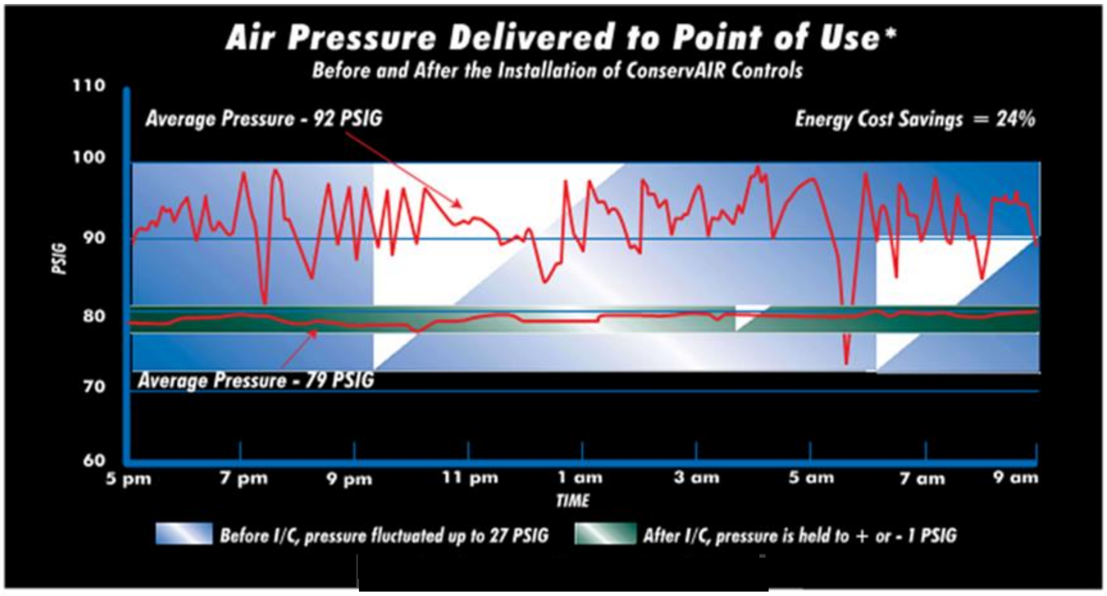
Priority II: Implement 6 – 12 months

8. Increase storage by 2000 gallons. This will give [Company Name] approximately 3720 gallons of storage. The storage should be set at 125 psig (new compressor pressure rating).

9. Add a flow control valve. Discharge pressure to the plant should be set at 90 psig (600Mpa). To create useful storage as described previously explained in a prior section above. It is recommended to install a Demand Flow Stabilizer after the 2 air receivers (3720-gallons total). The Demand Flow Stabilizer (DFS) will regulate the air to the systems. The valve works in conjunction with the air receiver. The air will be produced by the air compressors at approximately 125 psig, pass through the dryers and then be stored in the air receiver. The DFS will be installed on the downstream side of the 3720 gallons of storage. The tanks will be hold air after the air dryers. The DFS valve should be set at 90 psig for the system. This will ensure proper flow distribution to the entire facility at the proper pressure. It will also ensure air pressure remains constant within 1– 2 psig instead of the current 14 psig. Event demands will be satisfied by the tank. Storing at 125 psig and releasing at 90 psig will create useful storage as described above. Note the control valve should be sized for 2000 cfm maximum and 500 cfm minimum.

10. The LS-20S-125L should be turned off. With additional storage the cut in pressure can be used to start the compressor versus cascading. With over 1.8 minutes of useful storage will be ample time for the compressor to start up.

Pressure Control Valve Function



Priority III: Implement 12 – 18 months

11. Replace 20/16 compressor with identical unit mentioned in priority I.

12. Have a site wide leak audit performed. We recommend to have a plant wide leak audit performed. With the completion of the air leak audit, Caros Chemical should institute a leak repair program. The following information may be useful in the future in determining leak rates:

    * Hear and feel a leak: it's over 5 scfm.
    * Felt but not heard: Its leaking 2 - 5 scfm.
    * Can't be detected by human senses: It still can leak 1 - 2 scfm.

    Leaks represent a real demand that must be satisfied, or production will suffer. Large leaks tend to get fixed, but small leaks do not. Short of implementing a full program, leak losses can be mitigated by feeding them at the lowest possible pressure. The air consumption savings are approximately 1% per psig. A 10-psig reduction in the supply pressure results in 10% savings in the air consumed by a leak. Addressing point of use issues by applying pressure regulation will also, lower leak losses since most of the small insidious leaks are associated with the work stations located downstream of the main piping header.

13. Replace heated/purge dryer with a heated/blower purge dryer if a purge management controller is not available.

Estimated Costs

Priority I

Air -Free Drains...$ 1,200.00 each
150Hp Compressor with VFD 125 psig..................................$188,000.00 each
Rail Unloading Flow Meter..$ 1,000.00 each

Priority II

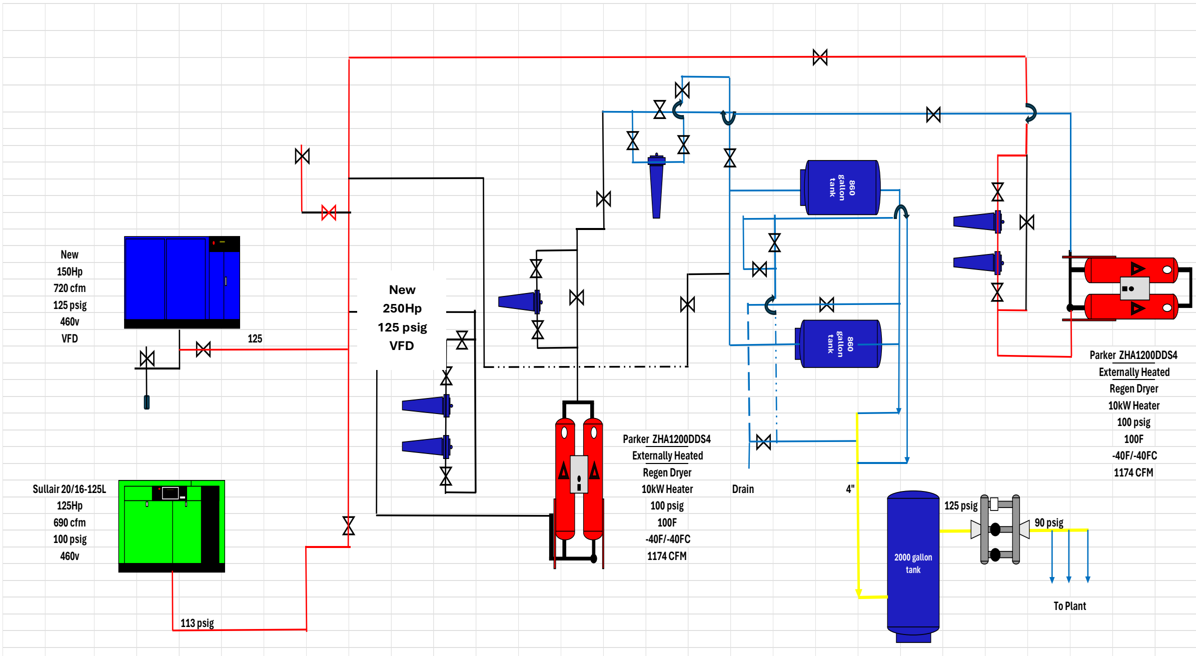
2000 Gallon Air Receiver w/relief valve and gauge................$ 18,000.00
1500 cfm Flow Control Valve...$ 18,500.00

Priority III

150 Hp Compressor with VFD 125 psig.................................$188,000.00 each
Site Wide Leak Audit...$ 5,000.00

Total estimated cost savings...$24,775/yr

Upgraded Compressor Layout Sketch



Final Summary

System Usage Table

	Average	Estimated Average
CFM	646	567
BHP	177	113
PSIG	114	90
KW Usage	1,156,687	741,273
KWH Savings		415,414

Implementation of the recommendations in this report will have several benefits for [Company Name] as follows:

1. Elimination of false demands.
2. Elimination of pressure fluctuations.
3. Creation of back-up compressor.
4. Elimination of rental compressors.
5. Reduction of on-line Hp

6. Elimination of artificial demand.
7. Savings of 414,415 KWh (based on 8760 hrs).
8. Increase in reliability, sustainability and efficiency of the compressed air system.

The success of the audit is due to the people at [Company Name] who assisted in the process. Their help was essential in understanding and resolving the many complex issues. Please extend our thanks to everyone involved. Any questions concerning the findings or subsequent recommendations should be addressed to [N2O2 Auditor].

Sincerely,

[N2O2 Auditor]

Director of Engineering Services