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Compressed Air System Survey at Army Industrial Facilities

Mike C.J. Lin , Robert T. Lorand, Doug Presny, John F. Westerman, Stephen W. Aylor, John C. Skelton, and Hank Van Ormer

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Foreword

This study was conducted for the Headquarters, U.S. Army Corps of Engineers (HQUSACE), under Project 40162784AT45, "Facility Infrastructure Technology"; Work Unit X051 "Compressed Air System Modernization." The technical monitor was Robert Reeves, AMXIS-C, AMC I&SA.

The work was performed by the Energy Branch (CF-E) of the Facility Division (CF), U.S. Army Construction Engineering Research Laboratory (CERL). The CERL Principal Investigator was Dr. Mike C.J. Lin. Surveys conducted at Picatinny Arsenal, Watervliet Arsenal, Corpus Christi Army Depot, Lone Star Army Ammunition Plant, Pine Bluff Arsenal, and the Combat Equipment Group - Afloat were done under contract No. DACA88-98-0007-0002 by Technology and Management Services, Inc. (TMSI) with Xenergy Inc. as subcontractor. Stephen W. Aylor is associated with TMSI; and John Skelton and Hank Van Ormer are associated with Xenergy. Surveys conducted at Aberdeen Proving Ground, Lake City Army Ammunition Plant, Redstone Arsenal, Rock Island Arsenal, and Sierra Army Depot were done by Science Applications International Corporation (SAIC) under contract No. DACA88-98-003-0010. Robert T. Lorand, Doug Presny, and John F. Westerman are associated with SAIC. Appreciation is expressed to Base Energy Coordinators for their efforts in support of this survey. Dr. Thomas Hartranft is Chief, CEERD-CF-E, and L. Michael Golish is Chief, CEERD-CF. The associated Technical Director was Gary W. Schanche. The technical editor was William J. Wolfe, Information Technology Laboratory. The Director of CERL is Dr. Alan W. Moore.

CERL is an element of the U.S. Army Engineer Research and Development Center (ERDC), U.S. Army Corps of Engineers. The Commander and Executive Director of ERDC is COL John Morris III, EN and the Director of ERDC is Dr. James R. Houston.

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Background

Compressed air (CA) is an indispensable commodity used in manufacturing and maintenance facilities. As in the private sector, Department of Defense (DOD) facilities make widespread use of air compressors. In fact, the variety of tools and machinery that operate on compressed air is increasing. Although CA is a convenient power source, CA systems are not cheap to operate. Air leakage as high as 30 to 50 percent of the total compressor output is not uncommon, and is often the largest waste of energy associated with CA usage. An analysis of the cost breakdown of a CA system shows that as little as 10 percent of the input power supplied to the compressor is delivered as CA to the system. Nearly all industrial plants can realize 25 to 40 percent savings on the power costs for the CA system without additional capital expenditures.*

The electricity used by these air compressors is a major contributor to annual energy operating costs. The use of natural gas engine driven air compressors (NGEDACs) in place of conventional electric motor driven air compressors offers an opportunity to reduce these costs. The operating cost savings associated with the NGEDAC are a result of the lower price of natural gas relative to electricity, the efficient operation of the NGEDAC at partial loads, and the greater opportunities for heat recovery compared to electric motor driven air compressors. These savings can more than offset the additional capital costs and maintenance costs of NGEDAC installation. NGEDACs can also be configured to eliminate the need for external sources of electric power, enabling operation even when there are electric power supply disruptions.

A CA system survey is an essential maintenance step to keep CA systems in efficient working condition, and a necessary preliminary step before comparing the energy use of electrical motor-driven CA systems with NGEDACs. This survey

* Henry L. Kemp, Jr., Strategic Air Concepts, http://www.strait-air.com/index.html (Qualified instructor for the Compressed Air Challenge program).

was undertaken to evaluate CA systems at Army industrial facilities to identify opportunities for energy savings—by improving system efficiencies and by identifying suitable candidates for installation of NGEDACs.

Objectives

The overall objective of this work was to conduct a CA system survey at Army industrial facilities that can greatly benefit from CA performance improvements. The specific purpose of the survey was:

1. To identify opportunities for reducing energy operating costs associated with the existing compressed air system.
2. To evaluate sites for their suitability as candidates for the installation of NGEDACs.

Approach

The site survey involved the following steps:

1. Site surveys ("Level I" audits) were conducted on the performance of the major CA systems at 11 Army installations.
   a. Aberdeen Proving Ground (APG), Aberdeen, MD
      http://www.apg.army.mil/default.htm
   b. Combat Equipment Group—Afloat (CEGA), Charleston, SC.
      http://www.globalsecurity.org/military/agency/army/ceg-afloat.htm
   c. Corpus Christi Army Depot (CCAD), Corpus Christi, TX
      http://www.ccad.army.mil/
   d. Lake City Army Ammunition Plant (LCAAP), Independence, MO
      http://www.terc-itcorp.com/kcterc/to03/index.htm
   e. Lone Star Army Ammunition Plant (LSAAP), Texarkana, TX;
      http://www.globalsecurity.org/military/facility/aap-lonestar.htm
   f. Picatinny Arsenal (PICA), Dover, NJ
      http://www.pica.army.mil/Public/
   g. Pine Bluff Arsenal (PBA), Pine Bluff, AK
      http://www.pba.army.mil/
   h. Redstone Arsenal (RSA), Huntsville, AL
      http://www.redstone.army.mil/

i. Rock Island Arsenal (RIA), Rock Island, IL
   http://www.ria.army.mil/

j. Sierra Army Depot (SIAD), Herlong, CA.
   http://www.sierra.army.mil/index.html

k. Watervliet Arsenal (WVA), Watervliet, NY
   http://www.wva.army.mil/

2. Detailed background information was gathered on the CA systems, including, manufacturers, sizes and vintage of compressors, accessories, and air distribution systems, and the various uses of the compressed air.
3. The cost of operating the CA systems was estimated based on the performance data, utility rate data, and maintenance expenses.
4. Specific measures to reduce CA system energy use were identified, including the magnitude of the possible operating cost savings and associated implementation costs.
5. The economics of an NGEDAC implementation were determined, assuming the application was technically suitable (i.e., loads and hours of operation were sufficiently large, natural gas supply was accessible, etc.).
6. Best candidates for NGEDAC demonstration were selected from the surveyed sites.

Scope

Survey results were documented for 11 Army industrial facilities. While this work focused on specific sites, the results of these surveys may be broadly applicable to other similar DOD manufacturing facilities.

Mode of Technology Transfer

The information presented in this report will be furnished directly to the sponsoring organization and the 11 Army bases studied, and to the U.S. Army, Installation Management Agency Headquarters and Regional Offices:

- Installation Management Agency
  http://www.ima.army.mil/index.asp
- Installation Management Agency, Northwest Region
  http://www.ima.army.mil/northwest/index.asp
- Installation Management Agency, Northeast Region
  http://www.ima.army.mil/northeast/index.asp

- Installation Management Agency, Southwest Region
  http://www.ima.army.mil/southwest/index.asp
- Installation Management Agency, Southeast Region
  http://www.ima.army.mil/southeast/index.asp
- Installation Management Agency, Pacific Region
  http://www.ima.army.mil/pacific/index.asp
- Installation Management Agency, European Region
  http://www.ima.army.mil/europe/index.asp
- Installation Management Agency, Korea Region
  http://www.ima.army.mil/korea/index.asp

It will also be published via the World Wide Web (WWW) at URL:

http://www.cecer.army.mil/

Technical papers documenting the survey results will be prepared and presented at the industrial energy technical conferences. All industrial facilities with extensive compressed air usage should have a periodic system survey performed and an air pressure leak management program established. Follow-up meetings with survey sites will be held to see if they have implemented any of the survey recommendations, and if so, estimate/document the benefits.

Units of Weight and Measure

U.S. standard units of measure are used throughout this report. A table of conversion factors for Standard International (SI) units is provided below.

SI conversion factors					
1 in.	=	2.54 cm	1 cu ft	=	0.028 m^3
1 ft	=	0.305 m	1 cu yd	=	0.764 m^3
1 yd	=	0.9144 m	1 gal	=	3.78 L
1 sq in.	=	6.452 cm^2	1 lb	=	0.453 kg
1 sq ft	=	0.093 m^2	1 kip	=	453 kg
1 sq yd	=	0.836 m^2	1 psi	=	6.89 kPa
1 cu in.	=	16.39 cm^3	°F	=	(°C x 1.8) + 32

2 Summary of the Six-Site Survey Conducted by TMSI/Xenergy

Technology and Management Services, Inc. (TMSI) and Xenergy, Inc. (Xenergy) (referred to collectively as the "Project Team") conducted six site surveys (Level I Audits) to identify opportunities to reduce compressed air system operating expenses and to determine the suitability for installing the NGEDAC. The six sites (listed here and discussed in this Chapter in alphabetical order) were:

1. Combat Equipment Group–Afloat (CEGA), SC
2. Corpus Christi Army Depot (CCAD), TX
3. Lone Star Army Ammunition Plant (LSAAP), TX
4. Picatinny Arsenal (PICA), NJ
5. Pine Bluff Arsenal (PBA), AK
6. Watervliet Arsenal (WVA), NY.

Each site was visited by a team consisting of TMSI and Xenergy personnel along with representatives from the U.S. Army Engineer Research and Development Center, Construction Engineering Research Laboratory (ERDC/CERL). The Level I Audit provided an overview of each facility's compressed air system, outlined potential areas for reducing system demand, evaluated system economics, and identified potential benefits and problems of a NGEDAC system at the site.

Following a Level I Audit kick-off meeting held at each site, the team surveyed all central system air compressors at the facility and some of the standalone air compressors. Data were collected on the individual air compressor units and on the compressed air system as a whole. Where possible, a data logger was used to record compressed air system operation for a 24-hour period. Data were then analyzed and the preliminary results presented at a wrap-up meeting held at each site.

Following the Level I Audit, a report was prepared for each site. Copies of the reports were provided to the facility. Based on the recommendations contained in these reports, two sites (Picatinny Arsenal and Watervliet Arsenal) were selected as the two NGEDAC demonstration sites.

The following sections summarize the Level I Audits for each of the six candidate sites. Appendixes A-K contain the Level I Audit reports. Appendix L includes the scope of work of the CAS survey including a description of “Level I” and “Level II” audits.

Corpus Christi Army Depot

Introduction

The Level I Survey at Corpus Christi Army Depot (CCAD) was conducted on 18-19 January 2001. The survey team consisted of Stephen Aylor from TMSI; John Skelton, Hank VanOrmer, and Don VanOrmer from Xenergy: and Mike Lin from CERL.

The evaluation of the Corpus Christi Army Depot site reflects several issues that complicate the analysis, and thereby, the overall recommendation. First, the Corpus Christi site is in the process of evaluating a large-scale project to decentralize both the compressed air and steam systems at the depot. This means the assessment of the NGEDAC demonstration project, as well as the planning of its implementation, need to be made in the context of a hypothetical system that is expected to be operational some time after the termination date of the NGEDAC demonstration project.

Second, with the implementation of the Compressed Air Decentralization Plan, the NGEDAC unit could provide needed redundancy and fuel choice flexibility within the proposed decentralized system. In assessments of other sites where the NGEDAC unit was replacing an existing electric unit, the accumulative operational savings of the NGEDAC unit would defray the entire capital cost of the NGEDAC unit within a reasonable time. In the case of CCAD, a more appropriate approach might entail having the accumulative operational savings defray the incremental costs of the NGEDAC unit relative to an electric-based approach of providing needed redundancy. Third, the evaluation at CCAD needs to reflect the existence of two distinct sets of energy price signals: those provided by the Corpus Christi Naval Air Station to Corpus Christi Army Depot and those provided by the energy utilities to the Corpus Christi Naval Air Station. The electric rate structure governing the cost of electricity paid by the Corpus Christi Naval Air Station reflects an unusually high percentage of the electric bill made up by a large demand charge. The rate structure is based on the peak demand during regular business hours during four summer months and reduced for the remaining 8 months at a 90 percent level. Such rate structures are ideal candidates for summer peak-load shaving, for which the NGEDAC unit is well suited. However, the rate structure for the Corpus Christi Army Depot includes no demand charge element and hence no savings associated with a peak-load shavings strategy—a lost benefit of the NGEDAC system on the order of $22,000 to $28,000, annually.

Existing Compressed Air System

The compressed air system at CCAD is a centralized system with three Ingersoll-Rand three-stage centrifugal compressors as the primary air supply: 4,000 cfm, 2,500 cfm, and 1,200 cfm class. CCAD is evaluating a proposal to decentralize this system and place dedicated compressors in many of the buildings now served by the central system. This will create several areas of savings.

The largest production area, Building 8, will set up its air system with two of the central centrifugals (2,500 and 1,200 class), along with potentially a 750 cfm and a 450 cfm Quincy rotary screw. These four units will be controlled by a networking capacity unloading control system. The objectives will be to assure that:

1. No centrifugal is in blow-off
2. Only one unit operates at part load, all others are at full load or off
3. Create energy savings by replacing current noncycling and other inefficient dryers with effectively sized cycling-type refrigerated dryers.

Most of the CCAD staff believes this Decentralization Plan will be implemented; thus, the economic benefits of a natural gas engine driven unit are being evaluated in the context of the proposed decentralized system rather than the current centralized system.

In Building 8, the Quincy 450 rotary screw 100-hp compressor will be moved next to the Quincy 750 rotary screw (both air cooled) under the outside shed in the northwest corner of Building 8 as part of the decentralized system. Both units will become part of Building 8's air system. Buildings 30, 252, 259, 1209, 1219, and Wheel Tower would be left as part of Building 8's air system and be fed by underground lines. Building 8 would be the likely site if the NGEDAC unit were installed at CCAD. The measured air flow in Building 8 during CCAD's September 2000 investigation was for weekday production (2,080 hours), a peak of 2,400 cfm, and an average of 2,200 cfm; for nonproduction (6,240 hours), an average flow of 900 cfm; and for weekend production (440 hours), an average flow of 1,450 cfm. The estimated leaks, which will be repaired, were 400–450 cfm. The projected average demands with leaks repaired are weekday production at 1,750 cfm, nonproduction at 450 cfm, and weekend production at 1,000 cfm. Numerous other buildings and hangars would have new or existing compressors installed as part of the Decentralization Plan.

Results of Level I Audit

Current air flow of the system associated with Building 8, the main manufacturing location, and various other buildings nearby averages 2,200 cfm during production hours, based on measurements taken in the decentralization study. Current electric costs paid by the Corpus Christi Army Depot for this portion of the current "Main System" total $126,000. As part of the leak reduction associated with implementing the decentralization plan, it is expected that the average airflow in the Building 8 system will decrease by at least 450 cfm for a savings of $42,000. The annual electric cost for operating the Building 8 system with the leak savings included will total $85,000.

The NGEDAC program was evaluated with two different engine system alternatives. The first alternative includes a 1,400 cfm class NGEDAC unit that could serve three-quarters of the anticipated airflow during weekday production in the Building 8 system under the Decentralization Plan. The second alternative adds both a 1,400 cfm class and an 800 cfm class NGEDAC unit that could serve the entire level of air flow requirements.

Using the basic energy charges paid by CCAD of $6/10^6 Btu and $65/megawatt hour (MWh), the NGEDAC system program is expected to reduce energy costs by an additional $5,000. The incrementally higher maintenance costs associated with NGEDAC of $8,000 to $17,000 are almost equally balanced by the benefits of $9,000 to $20,000 accruing to the NGEDAC system through the incorporation of heat recovery linked with the new boilers installed as part of the decentralization plan.

The savings accruing to the NGEDAC system as part of its summer peak-shaving capability will not likely be reflected in lower costs to CCAD, but will be reflected in lower electric costs for the Corpus Christi Naval Air Station in its bill from the utility. These savings are the equivalent of the avoided demand ratchet during the non-summer months, or an amount ranging from $22,000 to $28,000. Future energy costs are highly uncertain. One of the advantages provided by the NGEDAC unit is fuel choice flexibility. Although the analysis is based on the most recent energy costs ($6/10^6 Btu and $65/MWh), it is expected that the energy prices charged by the Corpus Christi Naval Air Station could increase. If the electric price were to increase from $65/MWh to $85/MWh (an increase of 30 percent) and the price of gas were to increase from $6/10^6 Btu to $7.20/10^6 Btu (an increase of 20 percent), the NGEDAC unit would accrue an additional $10,000 to $14,000 in savings.

Recommendations

The CCAD site has a number of positive aspects that help make it a potential demonstration site candidate. Gas, electric, and water supplies are readily accessible. The CCAD site provides for a fairly straightforward technology application and a supportive staff. It affords DoD an excellent opportunity to test operating a "gas/electric hybrid system," developing a heat recovery system, and being integrated within an overall modernization and decentralization plan.

However, logistically, the site presents many difficulties for installing a NGEDAC unit for three primary reasons. First, the design phase for the Decentralization Plan needs to be moved forward enough for the NGEDAC project to be designed in the same time frame. Also, the Decentralization Plan needs to pick up certain costs associated with the NGEDAC project. For example, these costs could include the cooling tower and other items that would only be constructed during implementation of the Decentralization Plan that will be needed after the NGEDAC project is complete, e.g., the heat recovery system with the NGEDAC unit. Second, The NGEDAC unit needs to be assigned to play a critical role in the Decentralization Plan and, as such, its accumulative savings do not need to be measured against the entire capital cost of the NGEDAC unit. Third, the NGEDAC needs to be able to claim the full theoretical savings of its peak shaving capability, which will be reflected in the electric bill paid by the Corpus Christi Naval Air Station, but not necessarily reflected in the electric bill paid by the Corpus Christi Army Depot to the Corpus Christi Naval Air Station.

As a result of the Level I Audit, the following demand-side recommendations were made:

- Check flow regulators. Some flow regulators are probably set at higher than necessary feed pressure to the process, with some wide open to full header pressure. In this type of operation, it is very important that the actual inlet pressure to the process be known and that the lowest effective pressure be held steady for the proper product quality. CCAD may need to install storage bottles downstream of the regulator to "close up" the pressure readings at rest and in operation. Production and maintenance personnel must achieve and adhere to the minimum effective pressure in operation, established at the unit, for each product run.
- Install electric-operated automatic ball valves that can be placed in the main feed line to a piece of equipment that are wired to open and close when the pneumatic device is powered up or turned off, respectively. This will eliminate off-production leaks and valves left open.
- Set up a continuing leak inspection by maintenance personnel in the short term, so that each primary sector of the plant is inspected once a quarter or

at a minimum once every 6 months, to identify and repair leaks. A record should be kept of these findings and overall results.

- Consider setting up cost-reduction programs in the long term where the production people (particularly the operators and their supervisors) could receive incentives for identifying and repairing these leaks.
- Review compressed air condensate handling system to ensure compliance with environmental regulations.
- Evaluate cabinet coolers. There may be cabinet coolers in use in the facility. Some cabinet coolers use refrigeration (1,500 Btu), some cabinet coolers use compressed air-driven vortex coolers, and some merely use compressed air for cooling. These cabinet coolers could be replaced with “heat tube” cabinet coolers with a potential savings of 3.5 to 4 kW each.
- Evaluate blow-off design. The facility may have ¼-in. lines running as blow-off on units at 80 psig. These will use approximately 32 cfm each. An alternate is an air amplifier that takes less compressed air and through Venturi action amplifies the usable air by pulling in significant amounts of ambient air and mixing it directly into the air stream. These have amplification ratios up to 25:1. Using 10 cfm of compressed air would generate a savings of 22 cfm compressed air per ¼-in. blow-off and will supply 250 cfm blow air to the process.
- Review any vacuum generators. Vacuum generators are very convenient, very responsive, and very inefficient compared with positive displacement pumps, i.e., rotary screw, reciprocating.
- Examine air-operated diaphragm pumps. There are several steps that can be taken to generate significant air savings: (1) use an electric motor driven diaphragm pump, which is significantly more power efficient than an air-operated diaphragm pump; (2) consider the installation of electronic or ultrasonic controls to shut the pumps off automatically when they are not needed; and (3) arrange controls to ensure that the lowest possible pressure is used as appropriate for the operation, which may generate significant savings.
- Review the compressed air system and take measurements to identify if there is any potential energy savings in using an alternate source of low-pressure air in the production area because using high-pressure air for very low-pressure applications is not an efficient use of energy.

The full Level I Audit report for CCAD is contained in Appendix A

Combat Equipment Group—Afloat

Introduction

The Level I survey at the Combat Equipment Group—Afloat (CEGA) in Charleston, SC was conducted on 19 January 2001. The survey team consisted of George Powers from TMSI; John Skelton, Hank VanOrmer, and Henry Kemp from Xenergy: and Mike Lin from CERL.

Existing Compressed Air System

The compressed air system at CEGA consists of many buildings served by large underground fiberglass lines delivering 125-psig air from three separate compressed air supplies:

1. A 100-hp Ingersoll-Rand rotary screw EP100 supplies 446 cfm (110 bhp) with an air-cooled aftercooler
2. A 75-hp Ingersoll-Rand rotary screw EP75 supplies 320 cfm (82.5 bhp)
3. A 40-hp Ingersoll-Rand rotary screw EP40SE supplies 157 cfm (47.2 bhp).

Current airflow of the system averages 80 cfm. When working, CEGA's 40-hp compressor can supply the entire system on a part-load basis. The other existing units serve as backups or as additional air supply resources should air demands at CEGA increase with future work. The underground fiberglass lines form a very effective storage for the entire system, approaching the equivalent of 10,000 gal.

The facility has good control over its system leak level. The maintenance personnel already have a far-reaching leak identification and repair program, not only on the demand side, but also for all the underground lines. As leaks were identified underground (in rain storms air bubbles can be seen on the ground), the pipe is then exposed and the pipe is repaired or replaced. Because the pipe is fiberglass and not "black iron," the leaks are usually caused by ground movement and not by overall deterioration. This program has led to a very efficient compressed air system that generally runs all the time on a part loaded 40-hp compressor.

There is no apparent need for compressed air dryers for this work. Since the underground storage is also at the underground temperature, the air is cooled to about 50 °F or less. The lines are sloped and have automatic condensate drains to remove condensed water to control pits. At the compressor supply areas, all the receivers, aftercoolers, and risers also have mechanical, level-actuated automatic condensate drains.

Results of Level I Audit

Annual electric costs for operating the existing system total almost $15,000, based on an electric rate of $0.075/kWh. Projected annual gas costs for a NGEDAC unit also total almost $15,000, based on a gas rate of $8/million Btu. Thus, the NGEDAC unit does not have an operating cost advantage over the existing electric system when the incremental costs of about $7,000 annually for maintaining the gas unit are included.

Unlike some of the other sites, CEGA does not gain a significant tactical or strategic benefit by operating or having a gas unit, due in large part to the soundness and magnitude of the current system relative to current air requirements and due to the significantly higher gas costs paid by CEGA. The relative price differential between gas and electric would have to change by some combination of gas rates decreasing or electric rates increasing by at least 50 percent, before the economics of the NGEDAC unit begin to look attractive.

As part of its overall system review, the survey team evaluated the viability of installing one or two 20-hp electric units and operating one to satisfy the typical 80 cfm load. While the proposed system could reduce electric costs by $2,000 annually, estimated costs for installing the system ranged from $10,000 to $16,000, resulting in a payback period that is probably too long for this type of investment by CEGA.

Recommendations

The preliminary assessment concludes that CEGA is not a good candidate for additional consideration at this time as a NGEDAC demonstration site. The current system already consists of well-applied electric compressors and a well-managed distribution system. The distribution system has recently undergone a systematic leak repair program and is monitored effectively by a Johnson Controls energy system. In addition, the gas rates currently charged to CEGA are disproportionately high relative to the electric rates being charged.

As a result of the Level I Audit, the following demand-side recommendations were made:

- Consider reconfiguring the existing system to add one or two 20-hp units, and operate one as base load to supply CEGA's 80 cfm air requirements. There is an approximate $1,900 annual savings in electric power by running a 20-hp rotary screw in lieu of the current 40-hp at an average demand of 80 cfm. One or two new 20-hp units will involve a turnkey cost of about $10,000 to $16,000.

- Consider modifying the compressed air control system software to read every "5-minutes averaged" or something similar to get a more representative load profile. There are six different Fox heated-wire anemometer thermal mass flowmeters installed with the Johnson Controls central air management system. These meters currently read the instantaneous "rate of flow" every 15 minutes, which provides limited data on load profile.
- Check with the Ingersoll-Rand service provider to see if there has been a modification or change to the drive system that may help prevent the 40-hp Ingersoll-Rand rotary screw EP40SE compressor from "throwing belts." Changing these belts should normally be no more than annual or an every-other-year event.
- Review compressed air condensate handling system to ensure compliance with environmental regulations.
- Replace all timer drains with level-activated drains. Separately connect each drain point (aftercooler, pre-filter, dryer, after-filter, receivers, and all risers) to individual level-activated electric or pneumatic drains to collect and direct the condensate to a proper handling point, such as in a large plastic vented line (4 or 6 in.). Be sure maintenance personnel can effectively and visually monitor the drain's action.
- Review the compressed air system and take measurements to identify if there is any potential energy savings in using an alternate source of low-pressure air in the production area, because using high-pressure air for very low-pressure applications is not an efficient use of energy.

The full Level I Audit report for CEGA is contained in Appendix B.

Lone Star Army Ammunition Plant

Introduction

The Level I Survey at Lone Star Army Ammunition Plant (LSAAP) was conducted on 13-14 March 2001. The survey team consisted of Stephen Aylor from TMSI; John Skelton, Henry Kemp, and Dave Beary from Xenergy; and Mike Lin from CERL.

Existing Compressed Air System

Relatively efficient air compressors that are capable of delivering the 100-psig full load pressures in a continuous manner produce the primary compressed air supply at LSAAP. The units are well applied. They appear to be in good operating order and well maintained. The compressors serving the plant are generally

300-hp class Ingersoll-Rand XLE units. The units are equipped with 5-step control, which allows each unit to operate efficiently at partial loadings. LSAAP spends about $100,000 annually to operate the compressed air systems in Areas B, G, P, and Q.

The four compressor units (Areas B, G, P, and Q) that were being used at the time of the visit were operating correctly. A fifth compressor in the machine shop was briefly reviewed and was operating correctly, but the requirements for compressed air are too small in the machine shop to be considered for an NGEDAC unit application.

The compressors currently in operation at the main sites are the same brand and size. They were all built in the early 1970s, but have been recently serviced along with a general updating of the entire drying system. Each of the compressor units is equipped with a 300-horsepower synchronous motor, five-step control, water-cooled aftercooler, and twin tower desiccant compressed air dryers. A 1,660-gal receiver supports each compressor. In these compressors, the air flows from the compressor through the aftercooler to the receiver. The route is then from the receiver to the pre-filter, to the twin tower desiccant dryer, to the aftercooler, and then out to the load. This is particularly good because the dryers should only see dry air after the aftercooler, separator, and receiver.

Only Area G was pinpointed as a potential for the NGEDAC system due to the potentially long distance between the installation site of the NGEDAC unit and the production areas to be served in Areas B, P, and Q. Area G also had the economic advantage of being able to incorporate a heat recovery system for the boiler operation nearby, and it is close to a natural gas supply.

Results of Level I Audit

Assuming an Ingersoll-Rand PCD200-NG platform and a Caterpillar G3306TA (780L) engine with heat recovery as the NGEDAC equipment for this site, the estimated cost to operate the NGEDAC unit, including a heat recovery benefit, is 20 percent higher than the current electric unit—$12,600 versus $10,300. The operating cost comparison is based on an electric rate of $0.045/kWh and a gas rate of $5/10^6 Btu. The incremental maintenance costs of $4,500 associated with the NGEDAC unit are negated by the $4,400 credit given to the NGEDAC unit from the heat recovery application.

The total capital cost estimate for the NGEDAC unit (780 cfm class) is $170,000. The capital cost includes the catalytic converter for the gas engine. The cost also includes an estimate for all installation and freight costs and a budget estimate to erect a compressed air line to link the NGEDAC unit in Area G with the production facilities in Area G.

Recommendations

The compressed air system at Lone Star Army Ammunition Plant is a very efficient and correctly applied system. The compressors have been recently serviced and the entire drying system upgraded. The system is well maintained. The plant enjoys very low electric rates, which average $0.045/kWh—about half of some of the other sites that have been considered.

High system efficiency and low electric rates combine to give Lone Star one of the lowest cost structures among all the compressed air systems that have been reviewed in the program. While this low cost structure is a major benefit to Lone Star in controlling its operating costs, the low cost structure reduces the opportunity for a cost-effective application for the proposed NGEDAC unit. It is, therefore, the conclusion of the preliminary assessment that the Lone Star site not be considered as a NGEDAC demo site, unless prevailing operating and economic conditions change.

As a result of the Level I Audit, the following demand-side recommendations were made:

- Check the aftercooler in P75 for fouling since there was only a 5 °F heat gain across the aftercooler, rather than the more normal 15 to 20 °F gain. This shortfall has the effect of reducing dryer capacity downstream of the aftercooler by 30 to 40 percent.
- Review compressed air condensate handling system to ensure compliance with environmental regulations.
- Replace all timer electronic drains with level-actuated electronic or air-operated drains for air conservation and enhanced performance. Timer-activated drains or dual-timer drains may not be able to handle heavy loads of condensate unless continuously monitored during the summer conditions. LSAAP should verify that auto drains are set to work effectively. The drains should not be tied together to a common header. Lone Star should ensure that all drains could be checked easily for operation. All drains must be properly vented.
- Review the compressed air system and take measurements to identify if there is any potential energy savings in using an alternate source of low-pressure air in the production area because using high-pressure air for very low-pressure applications is not an efficient use of energy.
- Set up a continuing leak inspection by maintenance personnel in the short term, so that each primary sector of the plant is inspected once a quarter or

at a minimum, once every 6 months to identify and repair leaks. A record should be kept of these findings and overall results.

- Consider setting up cost-reduction programs in the long term where the production people (particularly the operators and their supervisors) could receive incentives for identifying and repairing these leaks.

The full Level I Audit report for LSAAP is contained in Appendix C.

Picatinny Arsenal

Introduction

The Level I Survey at Picatinny Arsenal (PICA) was conducted on 28–31 August 2000. The survey team consisted of Stephen Aylor and George Powers from TMSI; John Skelton, Hank VanOrmer, and Don VanOrmer from Xenergy; and Martin Savoie and Mike Lin from CERL.

Existing Compressed Air System

The compressed air system at PICA encompasses an extensive geographical area. There are almost 27 miles of compressed air piping that join over a dozen areas of production buildings. Air usage levels are significantly less than those required during the height of production at Picatinny. Numerous opportunities exist to improve system energy efficiencies and to further reduce system operating costs by using a gas engine driven compressed air system.

An air compressor plant located in the Main Power House—Building 506—supplies the main compressed air system. Running either compressor Unit 1 or 2 provides the basic air supply for the facility. Unit 1 is currently not operational. Both units are 18 ½-in. and 11 x 8 ½-in. stroke, double-acting reciprocating Ingersoll-Rand 200 bhp (1,130 actual cu ft/min [acfm] at 100-110 lb/sq in. gage [psig]) compressors with 5-step unloading. The air delivered from the Main Power House is dried only with a water-cooled aftercooler. Average flow for the main system is 925 acfm at 80 psig with the system operating 8,760 hr/yr. These units are the most power efficient units at Picatinny and have a capacity control system, which effectively translates lower air demand into lower input energy. The Project Team observed Unit 2 running and, except for slightly excessive oil from the oiler, it appeared to be in very good shape. When the site visit took place on a 79 °F ambient day, the compressed air system was delivering 80 °F saturated air at 80 psig to the system.

These units are well applied. There are no more power efficient units available in this size class. The units are still "state of the art." In addition to the main compressed air system, numerous standalone systems are installed and operating at the facility.

Results of Level I Audit

The economics for a NGEDAC installation at Picatinny were favorable. The main air compressor system was originally designed to operate at 100 psig, but it has been lowered by plant personnel to 80 psig, which was estimated to have lowered the annual operating cost from $125,000 to $100,000. Demand-side improvements could reduce demand by 500 acfm and lower the annual costs to $58,000/yr. The estimated energy costs for a NGEDAC system to meet the same demands after demand-side improvements were estimated at $26,000/yr—a $32,000/yr savings in energy costs based on an interruptible natural gas price of $3.41/10^6 British thermal units (Btus). The gas driven engine system would incur $11,000 in incremental annual maintenance costs based on a 2-year maintenance contract for the gas system. The resulting net operating cost of the gas system was estimated at $21,000 less than the current electric system based on an electric rate of 8.8 cents/kWh.

The preliminary estimate of the installed system capital cost for the gas technology was approximately $160,000. System environmental emission levels were based on limits of 0.70 grams/brake horsepower/hr (gm/bhp/hr) for NOx and 0.48 gm/bhp/hr for CO. The total estimated project cost does not include any potential electrical demand reduction rebates for which this project may qualify.

The Picatinny site presented a number of other positive aspects that helped make it a good demonstration site candidate. Gas supply was readily accessible. Physical space was available and plant modifications would be minimal. Experience and confidence gained by Picatinny staff and contractors in developing and operating the 2.2 megawatts-electric (MWe) natural gas-fueled cogeneration system were a significant plus and could help reduce cost estimates for maintenance contracts for the gas engine compressed air system.

On the demand side, for a campus facility of this type, an effective leak control program could save in the average range of 300 to 400 cfm, which could be $30,000 to $40,000 in potential annual power cost savings. The estimated recoverable value is $25,000/yr.

To effectively control and manage leaks in such an extensive operation as Picatinny Arsenal, a continuing cost-reduction program must be in place. Generally speaking, the most effective programs are those that involve the production supervisors and operators working with the maintenance personnel in a coordinated manner.

Recommendations

The Picatinny site provided for a fairly straightforward technology application and demonstration with a very manageable system size. The location affords Government an excellent opportunity to test operating a "gas/electric hybrid system." In addition, Picatinny has an energy savings performance contract (ESPC) vehicle for implementing the recommended system optimization improvements that are outside the scope of this NGEDAC project, but are essential in properly sizing the project's gas engine system and reducing the overall operating costs.

As a result of the Level I Audit, the following demand-side recommendations were made:

- Repair a significant air leak that was identified in an above-ground, rusted-out distribution line under enclosed walkway between Building 807 and Building 810.
- Disconnect and cap off the piping and disconnect and remove the Sullivan WN4 600-hp horizontal, double-acting, reciprocating, water-cooled unit at the wind tunnel in Building 266 to stop constant leaks from the discharge valve.
- Modify the control systems on the two Sullair two-stage lubricated rotary screw compressors because they are experiencing productivity problems during supersonic operations in the wind tunnel caused by too slow top end refill that is a result of the type of modulation control system in use.
- Remove the (out of service) Ingersoll-Rand 40-hp ESV NL in Building 3150.
- Remove the (out of service)7½-hp Brunner tank-mounted unit on mezzanine in Building 3150.
- Evaluate extending main compressed air lines to both Buildings 3150 and 3028. If the IMC single-phase, 5-hp SP tank-mounted units were used for backups and compressed air were supplied from the Main Power House, there would be an approximate $13,000/yr electrical energy savings.
- Evaluate a microprocessor-driven, centralized, full networking electronic control system. This would automatically place the most efficient machine on line and assure no more than one partial loaded unit at a time.
- Consider replacing drains in the compressed air system. Automatic drain traps are a much better idea than manual drains for Picatinny's circumstances. For air conservation and enhanced performance, level-actuated electronic or air-operated drains should replace all dual-timer electronic drains and manual drains. Timer-activated drains and dual-timer drains may not

be able to handle heavy loads of condensate unless continuously monitored during the summer conditions.

- Ensure that auto drains are set up to work effectively, for example:
  - drains should not be tied together to a common header
  - ensure all drains can be checked easily for operation
  - ensure all drains are properly vented.
- Set up a continuing leak inspection by maintenance personnel in the short term, so that each primary sector of the plant is inspected once a quarter or, at a minimum, once every 6 months to identify and repair leaks. A record should be kept of these findings and overall results.
- Consider setting up cost-reduction programs in the long term where the production people (particularly the operators and their supervisors) could receive incentives in identifying and repairing these leaks.
- Review compressed air condensate handling system to ensure compliance with environmental regulations.
- Install electric-operated automatic ball valves in each piece of equipment's main feed line, and wire the valves to open and close when the pneumatic device is powered up or turned off, respectively. This will eliminate off-production leaks and valves left open.
- Check flow regulators. Some flow regulators are probably set at higher than necessary feed pressure relative to the process, with some wide open to full header pressure. In this type of operation, it is very important that the actual inlet pressure to the process be known and that the lowest effective pressure be held steady for the proper product quality. Picatinny may need to install storage bottles downstream of the regulator to "close up" the pressure readings at rest and in operation. Production and maintenance personnel must achieve and adhere to the minimum effective pressure in operation, established at the unit, for each product run.
- Evaluate cabinet coolers. There may be cabinet coolers in use in the facility. Some cabinet coolers use refrigeration (1,500 Btu), some cabinet coolers use compressed air-driven vortex coolers, and some merely have compressed air for cooling. These cabinet coolers could be replaced with "heat tube" cabinet coolers with a potential savings of 3.5 to 4 kW each.
- Evaluate blow-offs. Picatinny may have 1/8-in. and 1/4-in. lines running as blow-off on units at 80 psig. These will use 8 to 35 cfm each. An alternate is an air amplifier, which uses less compressed air, and through Venturi action amplifies the usable air by pulling in significant amounts of ambient air and mixing it directly into the air stream. These devices have amplification ratios up to 25:1. Using 10 cfm of compressed air would generate a savings of 25 cfm compressed air/¼-in. blow-off and flow 250 cfm total air at the process.
- Examine air-operated diaphragm pumps. These devices are generally used because they tolerate hostile conditions relatively well and run without

catastrophic damage even if the pump is dry. Efficiency is not usually considered. However, several steps can be taken to generate significant air savings:
- use an electric motor driven diaphragm pump, which is significantly more power efficient than an air-operated diaphragm pump
- consider the installation of electronic or ultrasonic controls to shut the pumps off automatically when they are not needed
- arrange controls to ensure that the lowest possible pressure is used as appropriate for the operation, which may generate significant savings.

- Eliminate the use of high-pressure air for very low-pressure applications. A close review of the compressed air system should be made and measurements taken to identify if there is any potential energy savings in using an alternate source of low-pressure air in the production area.

The full Level I Audit report for PICA is contained in Appendix D.

Pine Bluff Arsenal

Introduction

The Level I survey at Pine Bluff Arsenal (PBA) was conducted on 15 November 2000. The survey team consisted of Stephen Aylor from TMSI; John Skelton, Hank VanOrmer, and Don VanOrmer from Xenergy; and Mike Lin from CERL.

Existing Compressed Air System

The air system at PBA was recently upgraded. According to plant personnel, the air system has operated satisfactorily since the system upgrade.

The main compressed air system at PBA serves six different production areas—Area 3 (Sections 1, 2, 3, and 4) and Area 4 (Sections 2 and 4). There are six Ingersoll-Rand remanufactured two-stage, double-acting, water-cooled air compressors (16-in. and 10 x 7-in. stroke). Three are 150-hp units at 585 rpm. Three are 200-hp units at 705 rpm. These units are less than 1 year old and are very power efficient and very responsive to demand changes.

There is a pair of 150-hp and 200-hp units in Buildings 32-060, 33-060, and 34-140. These units appear to be well installed and maintained. However, the cooling water system seems to be acting a bit unstable and perhaps should be reviewed. The air from these units goes through water-cooled aftercoolers and then to air receivers (1,000 gal) and to a heatless-type regenerative desiccant dryer. These also were recently purchased. There are also numerous dedicated air systems at PBA.

Results of Level I Audit

Current airflow of the main system is approximately 2,000–3,200 acfm at a supply pressure of 110–115 psig during production times. Estimated annual electric cost to operate the three main satellite compressed air systems is $186,000. From a current annual level of $186,000, PBA should consider a number of modifications to the system that could reduce usage levels by almost 1,000 cfm or $70,000 annually.

The current 150-hp unit costs about $69,400 annually in electric expenses based on an electric rate of 5.7 cents/kWh. A comprehensive maintenance contract would add another $13,000 for a total energy and maintenance cost of $82,400, annually. The proposed NGEDAC unit would cost $53,100 annually in energy costs based on a gas rate of $4.00/10^6 Btu or a savings of $16,000 in energy costs. However, a comprehensive maintenance contract would add $26,300 for a total energy and maintenance cost of $79,400, leaving a net gain of only $3,000 over the existing system.

There are two key reasons why the Pine Bluff site is apparently less cost-effective than other sites. First, the existing compressors at Pine Bluff are relatively new and very energy efficient. Second, the current electric costs at Pine Bluff are on the order of 5 to 6 cents/kWh or 40 percent less than electric rates at some of the other sites. A 40 percent increase in electric costs with gas costs holding at $4.00/million Btu increases the net savings of the NGEDAC demonstration to $31,000.

The preliminary estimate of the installed system cost for the NGEDAC is approximately $228,000. This estimate could vary up or down depending on specific installation conditions and desired equipment features. System environmental emission levels are based on limits of 2.00 gm/bhp/hr for NOx and 2.00 gm/bhp/hr for CO.

Recommendations

PBA has a number of positive aspects that help make it a potential demonstration site candidate. Gas supply is readily accessible. The Pine Bluff site provides for a fairly straightforward technology application and a supportive staff. It affords the Department of Defense an excellent opportunity to test operating a gas/electric hybrid system. However, Pine Bluff demonstrates only marginally favorable economic conditions for a gas engine driven system to replace one of the existing 150-hp XLE units.

As a result of the Level I Audit, the following demand-side recommendations were made:

- Review the cooling water system on the 150-hp and 200-hp units in Buildings 32-060, 33-060, and 34-140 because it appears to be somewhat unstable.
- Consider installing a 20-hp unit in Building 44-120. The old Worthington HBs have been replaced with four Ingersoll-Rand EP75 lubricant-cooled rotary screw compressors; however, during the site visit, only one of four units was on and it was loaded about 7–8 percent (31 cfm). It was mostly at idle with an average 38 kW and annualized electric cost of $18,000.
- Evaluate a microprocessor-driven, centralized, full networking electronic control system because this would automatically place the most efficient machine on line and assure no more than one partial loaded unit at a time.
- Add a dew point demand purge controller on the primary dryers, which will reduce total purge by 50 percent. The primary dryers are twin tower, heatless, regenerative, desiccant dryers capable of delivering a consistent -40 pressure dew point (PDP), which is a measure of the “degree of dryness.” For three dryers from 858 to 429 cfm average, the estimated cost of three controllers is $45,000 and the estimated electric energy savings is $30,000/yr.
- Eliminate the water (condensate) and oil carryover problems in the air system. This problem is significant and can be expected to increase in magnitude during the summer. The correct way to eliminate water and oil in the air system is to clean and dry the air immediately after it is produced in the compressor room. Clean dry air then can be stored in a separate air receiver and flow to the system as required. Some guidelines for controlling oil and water carryover include:
  - generally, it is best to eliminate the water and oil right at the air source before it enters the air system
  - every 20 °F increase in temperature doubles the moisture load the compressed air will hold
  - compressed air dryers are usually capacity rated with 100 °F, 100 psig inlet air conditions (at 120 °F, 100 psig, the dryer’s capacity rating is reduced 50 percent)
  - putting “dry/or oil free” air into system 90 percent of the time and then allowing wet/oily air in sporadically 10 percent of the time will, in reality, make the system wet or oily all the time (the liquid water and/or oil will fall out in the piping system continuing to re-entrain and contaminate and/or collect in the low spots of the system, causing recontamination as it is pulled into the flowing compressed air system). A wet/oily system

may well take many months of continued flowing of clean dry air to clean up.

- Review compressed air condensate handling system to ensure compliance with environmental regulations.
- Replace all timer drains with level-activated drains. Separately connect each drain point (aftercooler, pre-filter, dryer, after-filter, receivers, and all risers) to individual level-activated electric or pneumatic drains to collect and direct the condensate to a proper handling point, such as in a large plastic vented line (4 or 6 in.). Be sure maintenance personnel can effectively and visually monitor the drain's action.
- Check flow regulators. Some flow regulators may be set at higher than necessary feed pressure relative to the process, with some wide open to full header pressure. In this type of operation, it is very important that the actual inlet pressure to the process be known and that the lowest effective pressure be held steady for the proper product quality. PBA may need to install storage bottles downstream of the regulator to "close up" the pressure readings at rest and in operation. Production and maintenance personnel must achieve and adhere to the minimum effective pressure in operation, established at the unit, for each product run.
- Implement a continuing leak identification and repair program with ultrasonic locators. With a plant of this type, an effective leak control program could save 1,200 cfm or the equivalent of repairing 300 leaks averaging 4 cfm each. On a percentage basis, this leak level is about the same as leak levels in other plants. Leaks totaling 1,200 cfm translate into an annual loss of $102,000 in electric cost. A comprehensive leak management program could reduce such levels by 70 percent or $71,400 annually.
- Set up a continuing leak inspection by maintenance personnel in the short term, so that each primary sector of the plant is inspected once a quarter or at a minimum once every 6 months, to identify and repair leaks. A record should be kept of these findings and overall results.
- Consider setting up cost-reduction programs in the long term where the production people (particularly the operators and their supervisors) could receive incentives for identifying and repairing these leaks.
- Evaluate cabinet coolers. There may be cabinet coolers in use in the facility. Some cabinet coolers use refrigeration (1,500 Btu), some cabinet coolers use compressed air-driven vortex coolers, and some merely have compressed air for cooling. These cabinet coolers could be replaced with "heat tube" cabinet coolers with a potential savings of 3.5 to 4 kW each.
- Review any vacuum generators. Vacuum generators are very convenient, very responsive, and very inefficient compared with positive displacement pumps, i.e., rotary screw, reciprocating.

- Examine air-operated diaphragm pumps. These devices are generally used because they tolerate aggressive conditions relatively well and run without catastrophic damage even if the pump is dry. Efficiency is not usually considered. However, there are several steps that can be taken to generate significant air savings:
  - use an electric motor driven diaphragm pump, which is significantly more power efficient than an air-operated diaphragm pump
  - consider the installation of electronic or ultrasonic controls to shut the pumps off automatically when they are not needed
  - arrange controls to ensure that the lowest possible pressure is used as appropriate for the operation, which may generate significant savings.
- Review the compressed air system and take measurements to identify if there are any potential energy savings in using an alternate source of low-pressure air in the production area because using high-pressure air for very low-pressure applications is not an efficient use of energy.

The full Level I Audit report for PBA is contained in Appendix E.

Watervliet Arsenal

Introduction

The Level I Survey at Watervliet Arsenal (WVA) was conducted on 30 October—1 November 2000. The survey team consisted of Stephen Aylor and George Powers from TMSI; John Skelton, Paul Wenner, Hank VanOrmer, and Don VanOrmer from Xenergy; and Mike Lin from CERL.

Existing Compressed Air System

The Watervliet Arsenal has a very extensive compressed air system linking many separate buildings spread over a large geographical area. The air system reaches most production sectors and runs building to building, eventually completing a full loop system. Generally, the compressed air supply comes from Building 110 with a large 2,000-acfm (450-hp) class Joy centrifugal and two 125-hp Ingersoll-Rand XLEs (650 cfm/machine). There are six other major compressors tied in to the main air system in surrounding buildings. There are a number of smaller air-cooled reciprocating units throughout the Arsenal either as part of the separate “controls air system” or dedicated air to a particular process. The air drying is handled by both desiccant and refrigeration units and appears to be operating well. Most of the compressors are water-cooled, but some have their own air-cooled, radiator-type, closed-cooling systems, which also appear to operate well.

The complete air system appears to be very well laid out, well maintained and controlled, and consistent with the type of controls on the units. However, within the demand side of the system, a number of areas should be reviewed in the future in more detail, as there appear to be opportunities for significant reduction in demand.

Results of Level I Audit

The conceptual design for the gas engine driven system is based on providing about two-thirds of the requirements of the main system at Watervliet. The system will be configured as a hybrid system in conjunction with the existing electric system. In this way, the existing electric system can serve as a backup to the gas engine system, if the gas system has a planned or unplanned shutdown or if the air requirements of the base are suddenly increased. Using this approach, the Department of Defense (DoD) can gain experience with not only operating a gas engine driven system, but also integrating it with electric systems to improve overall compressed air system reliability and reduce operating costs. This flexibility is especially important given the increasing uncertainty associated with the price and supply reliability of most energy sources.

Environmental issues are expected to be minimal in this application given the key areas to be addressed in any major project of this nature, particularly on the East Coast. The assessment is based on emitting 2.60 gm/bhp/hr for NOx and 1.75 gm/bhp/hr for CO.

Annual costs are $306,000 for the existing electric system and $210,000 for the NGEDAC system, a savings of $96,000 annually, based on the cost of gas at $5/10^6 Btu and the cost of electricity of 9 cents/kWh. Adding or reducing the gas cost by $1/10^6 Btu would change the savings level by about $25,000 annually. A 2-year comprehensive maintenance contract is about $15,000 higher for the proposed system when compared with the existing system. Quoted maintenance contract levels range from $3.15 to $3.85/hr. This estimate is based on $75.00/hr and 17,000 hours for a 2-year operation. The price includes all parts, fluids, and scheduled and unscheduled maintenance. The net annual savings for the proposed system incorporating both the lower energy costs and higher maintenance costs is $81,000. Without consideration to potential cost reductions resulting from negotiating or utility rebates, the capital costs for the natural gas systems are on the order of $350,000 to $400,000.

If Watervliet is selected as a demonstration site, two specific demand-reduction strategies should be explored. First, potential reductions in air leaks on the order of 300 cfm could save $25,000 annually. Second, the use of low-pressure air or blowers for agitator applications could save even more.

Recommendations

The Watervliet site demonstrates favorable economic conditions for use of a gas engine driven system. Such a system is estimated to cost $210,000 annually in fuel expense and save $96,000. Maintenance costs for the NGEDAC technology are $15,000 higher annually based on a 2-year maintenance contract. The resulting net operating cost of the gas system is $81,000 less than the current electric system, when the centrifugal compressor is operating.

As a result of the Level I Audit, the following demand-side recommendations were made:

- Use low-pressure air where possible. There are some agitation applications that could perhaps be powered by low-pressure air compressors or blowers rather than costly high pressure air.
- Consider an automatically controlled high-performance secondary inline cooler between the radiator discharge and the compressor water inlet for the 450-hp Joy centrifugal in Building 110. The compressor has a closed-radiator-type system, and according to plant personnel, it works well except for several hours a day during extremely hot weather (greater than 90 °F). To alleviate this problem, there is a manually operated spray line set up to supercool when necessary. Centrifugal and rotary screws are more sensitive to cooling conditions in both life and performance than industrial reciprocating units.
- Consider removing or using as backups the Worthington M-Line, single-acting, air-cooled reciprocating units in Buildings 133 and 40 that are not operating under continuous duty. This type of unit is not well adapted for use in industrial production applications. It is rated very low in power efficiency. One of these units is damaged and out of service. These units should be kept only for emergency back-up air, if at all.
- Consider eliminating smaller, less efficient compressors in Building 15, unless higher pressures are needed for particular equipment. There are at least nine 25-hp air-cooled Ingersoll-Rand compressors in Building 15, one 15-hp air-cooled Wayne compressor in Building 120, and one 25-hp Champion (Speedair) compressor in Building 120. These types of units are well applied at or near the point-of-use production area (particularly where pressure higher than the 85 psig systems pressure is needed) to feed an intermittent demand. They are not continuous duty and should be applied on about a 50

percent duty cycle. They are not particularly power efficient and should not be run in place of general system units unless higher pressure is required.

- Consider eliminating smaller less efficient compressors in other buildings unless higher pressures are needed for particular equipment. Well over 20, 5-hp and smaller air-cooled reciprocating compressors are set up on appropriately sized horizontal air receivers and refrigerated air dryers throughout the buildings. Most of these are part of the control system and separate from the main system air. Where a 5-hp or fractional-hp unit is run instead of the general air system, the use should be reviewed unless it is for higher air pressure. These units are not nearly as power efficient when compared to the main air system units.
- Evaluate a new control system if operational performance deteriorates. For Watervliet's centrifugal compressors, there are modern electronic control systems that can be applied that will effectively close off the inlet and will blow the unit down to idle and significantly reduce the kilowatt demand. The existing Quad II control system is somewhat limited, but the new Quad 2000 by Cooper (Joy) would improve operation with some system storage and piping modification. However, there is no reason to pursue this option as long as the unit stays in base load and does not go into continuing blow-off.
- Evaluate a microprocessor-driven, centralized, full networking electronic control system because this would automatically place the most efficient machine on line and assure no more than one partial loaded unit at a time.
- Check flow regulators. Some flow regulators are probably set at higher than necessary feed pressure to the process, with some wide open to full header pressure. In this type of operation, it is very important that the actual inlet pressure to the process be known and that the lowest effective pressure be held steady for the proper product quality. Picatinny may need to install storage bottles downstream of the regulator to "close up" the pressure readings at rest and in operation. Production and maintenance personnel must achieve and adhere to the minimum effective pressure in operation, established at the unit, for each product run.
- Review compressed air condensate handling system to ensure compliance with environmental regulations.
- Set up a continuing leak inspection by maintenance personnel in the short term, so that each primary sector of the plant is inspected once a quarter or at a minimum once every 6 months, to identify and repair leaks. A record should be kept of these findings and overall results.
- Consider setting up cost reduction programs in the long term where the production people (particularly, the operators and their supervisors) could receive incentives (monetary reward, time-off, etc.) for identifying and repairing these leaks.

- Shut off the air supply to machinery when not in use. There are usually some very economical and easy methods to automatically shut off air supply when not in use.
- Evaluate cabinet coolers. There may be cabinet coolers in use in the facility. Some cabinet coolers use refrigeration (1,500 Btu), some cabinet coolers use compressed air-driven vortex coolers, and some merely use compressed air for cooling. These cabinet coolers could be replaced with “heat tube” cabinet coolers with a potential savings of 3.5 to 4 kW each.
- Review the compressed air system and take measurements to identify if there are potential energy savings in using alternate sources of low-pressure air in lieu of high-pressure air in the production area.

The full Level I Audit report for WVA is contained in Appendix F.

3 Summary of the Five-Site Survey Conducted by SAIC

Between 2-24 April 2001, SAIC conducted site surveys and evaluations at the following five Army industrial installations (Figure 1):

- Aberdeen Proving Ground (APG) —Aberdeen, MD
- Lake City Army Ammunition Plant—Lake City, MO
- Redstone Arsenal (RA)—Huntsville, AL
- Rock Island Arsenal—Rock Island, IL
- Sierra Army Depot—Herlong, CA

The results of the surveys are documented in individual reports for each base, which can be found in Appendices G through K. A summary of the survey and findings is provided below.

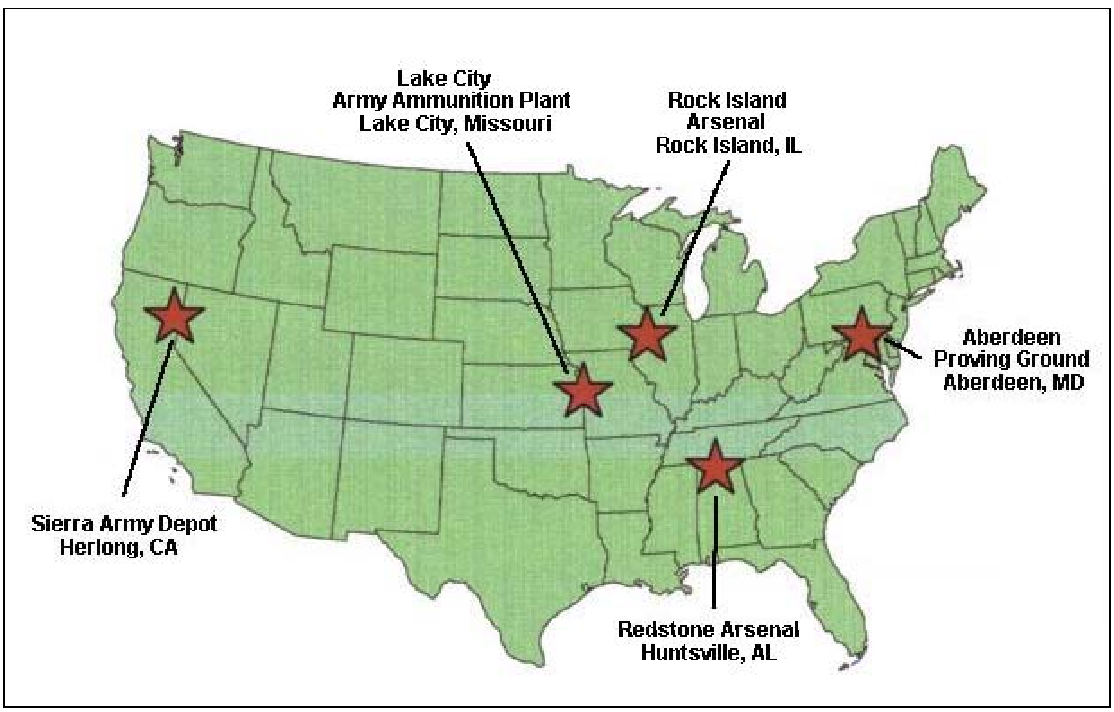

Figure 1. Locations of five surveyed Army industrial installations.

Aberdeen Proving Ground (APG)

Several hundred small compressors (e.g., under 20 hp capacity each) distributed across the base provide most of the facility's compressed air. Many buildings on the Aberdeen Proving Ground and the adjacent facility, the Edgewood Arsenal are not connected to the base natural gas distribution system. Since the objective of the survey was to focus on larger central compressed air systems that were in reasonable proximity to a natural gas line, this limited the survey to a few buildings. These were buildings 345, 4600, 338, 525, and 315. Of these buildings, 345 (boiler house), 315 (machine shop), and 525 (tank maintenance facility) were examined most closely, since they appeared to be potentially the most promising for a natural gas engine driven air compressor. With the exception of two buildings that are connected with an underground pipe, all of the buildings each have their own air compressor(s). The two buildings that are connected are buildings 315 (machine shop) and 345 (boiler house).

Building 315 has a 10-year-old 50 hp, 230 standard cu ft/minute (scfm) (estimated), direct drive screw compressor that leaks oil. Building 345 has a 5-year-old 25 hp Gardner Denver belt drive screw compressor. Facility staff indicated that the machine is rarely run, except when the boiler is burning oil, at which time the compressor is needed for atomizing air. Building 525 has three new 30 hp, 122 scfm @125 psig Ingersoll-Rand screw compressors that are piped together, networked and controlled by a single sequencing controller. The air dryer on the system looks like it has not operated in years. The drains are not piped, and there is no evidence of watermarks on the floor under the drains. The representative from Scales Air Compressor observed that the drain trap arrangement is virtually ineffective, and that the receiver needs a relief valve to meet code. The compressors appear to be lightly loaded, with observed loading of 15 to 40 percent and are used primarily for first shift operation. On occasion there are higher demands but these are generally short duration production ramp-ups.

Energy operating costs for the 399 air compressors at APG are estimated at $629,509 based on 10,853,611 kWh of energy use per year. This is based on an average price of electricity of $0.058/kWh. Reducing leaks appears to be the greatest overall opportunity for reducing energy use/operating costs. For example, one of the three compressors in Building 525 had a major leak which researchers observed and repaired. Our estimate was that the leak was about 20 scfm, which equates to an annual cost of nearly $1,000. On average it appears as though leaks may account for as much as 30 percent of the compressed air generated. Assuming an aggressive leak detection program was implemented, leaks could be cut in half. Based on this reduction—from 30 to 15 percent leakage—

APG could save an (approximately) estimated $94,426, based on a 1.628 million kWh reduction in electricity use.

Aberdeen Proving Ground does not appear to be a strong candidate for a natural gas engine driven air compressor demonstration due to the relatively small capacity of the compressors that are near natural gas lines, the low hours of operation and loading, limited heat recovery opportunity, and unfavorable economics. Table 1 lists the results of the economic analysis.

The information is based on typical part load operation observed during the survey, assuming a 25 hp NGEDAC replacing a 25 hp electric motor driven air compressor. Similar results would hold for a 30 hp or 50 hp unit—the capacity of the other surveyed units. Note that heat recovery is shown to indicate maximum benefits. Only limited heat recovery opportunity (e.g., space heating) was observed. Natural gas at $4/MBtu would give energy cost savings, but the results would still be marginal due to the added maintenance costs for the engine. The full Level I Audit report for APG is contained in Appendix G.

Lake City Army Ammunition Plant (LCAAP)

Description

The compressed air survey focused on Building 1 and 3, which are the largest users of compressed air. Generating compressed air for Building 1 and 3 costs $474,000/yr, or nearly 25 percent of LCAAP's electricity expenditures. This is based on an average electricity cost of $0.049/kWh. Building 1 has eight 500-hp Gardner Denver compressors, each with a design air supply capacity of 2500 scfm at 100 psig.

Table 1. APG annual energy use and operating costs at typical load (50% design load).

	Electric Air Compressor	NGEDAC	Net Savings
Energy Use	39,000 kWh	394 MBtu gas (engine) – 110 MBtu (engine heat recovery)	39,000 kWh (elec.) -394 MBtu gas 110 MBtu (gas including heat recovery)
Energy Operating Costs[1]	$2,262	$3,278	-$1016
Operation & Maintenance Costs	$600	$1,250	-$650
Heat Recovery Costs	0	-$915	$915
Total Costs (w/heat recovery)	$2,862	$3,590	-$728
Total Costs (w/o heat recovery	$2,862	$4,528	-$1,666

[1]Assumes average natural gas price of $8.32/MBtu and average electricity price of $0.058/kWh, including energy and demand charge components. This covers the period 3/00-2/01.

Typically, no more than four of these units are in operation, since the maximum compressed air demand is on the order of 9900 scfm. Building 3 has two-200 hp Gardner Denver compressors, each with a design air supply capacity of 1000 scfm at 100 psig. Typical compressed air demand is 1600 scfm total. The units are fairly recent vintage (1990s) and appear to be in reasonable operating condition. The compressors have sequential controllers to enable efficient distribution of the compressed air loads among the compressors. No operational problems were noted with the compressors or ancillary equipment.

LCAAP has an active leak detection program that has helped maintain compressed air leaks to approximately 15 percent of the total demand—a relatively low level of leakage. The air pressure provided by the system matches the load requirements well, consequently, no opportunities to reduce compressor operating pressures were noted. The most significant opportunity for cost savings from the compressed air system is from recovering waste heat from the compressor oil coolers and the compressed air after coolers in Building 1. Currently, waste heat is only being recovered from 2 of the units (oil cooler heat recovery). This heat would be used for process water heating within the building, offsetting natural gas fuel purchases for the boiler. The estimated annual savings is $32,437 and the associated fuel energy savings is 5,612 MBtu.

A natural gas engine driven air compressor can readily be accommodated at Lake City Army Ammunition Plant, with possible applications in either Building 1 or 3. While the current economics favor a Building 1 application, Building 3 has a potentially greater need for additional compressed air capacity (currently two portable diesel engine driven air compressors are being operated to meet specialized loads), and has greater heat recovery opportunities. Furthermore, the NGEDAC can be sited next to the natural gas station immediately outside the building, whereas a location serving Building 1 would require a more significant gas piping run (100 ft). The unit would be housed in its own heated weatherproof enclosure to protect it from the elements. The NGEDAC supply air would be tied into the existing supply system from Building 3, and make use of the existing receiver and 2000 scfm air dryer. The NGEDAC could potentially: (1) meet the full load supplied by the existing electric motor driven air compressors, or (2) be operated in combination with one or both of these units to meet load growth. In particular, the NGEDAC could be used in place of the two portable diesel engine driven air compressors. Waste heat from the NGEDAC would be recovered and used for process water heating applications. The NGEDAC would be installed in a manner that would not compromise the operation of other unit.

Economic Analysis

NGEDAC units ranging in size from 250 HP to 400 HP were evaluated with different operating schemes. The results for a 350 HP unit with an output of about 1670 cfm are provided below. Two cases are examined. In the first case, the NGEDAC is assumed to meet the building's full compressed air requirements, under typical operating conditions. In the second case, the NGEDAC is assumed to meet the additional load currently being met by a combination of two portable diesel engine driven air compressors.

Operating Cost Comparison—NGEDAC Displacing Nominal Demand Currently Met by Electric Motor Driven Air Compressors

Table 2 lists the annual energy use and operating costs associated with the proposed unit operating in a manner that meets full load (1600 scfm) for most of the operating day. For this period, about 5833 hours/yr, the NGEDAC would enable both existing electric motor driven air compressors to be shut down. For the few hours during the operating day when demand is low (400 scfm for 267 hours/yr), one of the electric units would be operated. Should demand increase above the nominal levels, one or more of the electric units could be brought on-line.

Table 2. LCAAP annual energy use and operating costs—NGEDAC displacing electric compressor.

Parameter	Electric Air Compressor	Hybrid NGEDAC/Electric	Net Savings
Energy Use	1,783,796 kWh	33,327 kWh (elec. air compressor)[2] 19,056 MBtu gas (engine) -7,075 MBtu (engine heat recovery)[3]	1,750,469 kWh (elec.) -11,981 MBtu (gas)
Energy Operating Costs[1]	$87,406	$111,779	-$24,373
Operation & Maintenance Costs	$20,000	$25,458	-$5,458
Heat Recovery Costs	0	-$40,892	$40,892
Total Costs	$107,406	$96,345	$11,061

1 Electricity Costs: $0.049/kWh—includes demand and energy charges
Natural Gas Costs: $5.78/MBtu

2 The electric unit is assumed to operate during the 267 hours per year when the load is 400 scfm and consumes 33,322 kWh. annually

3 Based on (0.295/.8) *heat value of natural gas into the engine, where 0.295 is the fraction of recoverable heat (engine coolant, exhaust, or compressor oil) and 0.8 is the assumed efficiency of the process water boiler displaced.

The results shown are based on the most recent electric and gas prices as indicated. While the $11,000 savings are modest, small changes in energy prices could significantly increase this figure. For example, if the price of electricity increased by 10 percent, the savings would increase to almost $20,000.

Operating Cost Comparison—NGEDAC Displacing Demand Currently Met by Diesel Engine Driven Air Compressors

Table 3 lists the energy performance and costs associated with the proposed unit operating in a manner that displaces the load currently being met by two portable diesel engine driven air compressors. These compressors operate about 60 hours/week (3000 hours/yr) to meet the air requirements of specialty equipment. While measurements of the air supplied by the portable units were not available, known fuel consumption information combined with the assumption that the compressors would provide about 5 scfm/hp, indicate an average output of the combined units of about 1100 scfm. It was assumed that the 350 hp NGEDAC, operating at part load would be used to meet this demand, eliminating the need to operate the diesel units.

The full Level I Audit report for LCAAP is contained in Appendix H.

Table 3. LCAAP annual energy use and operating costs—NGEDAC displacing diesel compressor.

	Diesel Engine Driven Air Compressor	NGEDAC	Net Savings
Energy Use	6300 MBtu	6861 MBtu gas (engine) —2530 MBtu (heat recovery)[2]	6300 MBtu (diesel) - 4331 MBtu gas
Energy Operating Costs[1]	$62,550	$39,655	$22,895
Operation & Maintenance Costs	$12.500	$12,500	0
Heat Recovery Costs	0	-$14,623	$14,623
Total Costs	$75,050	$37,532	$37,518

1 Natural Gas Costs: $5.78/MBtu
Diesel Fuel Costs: $9.93/MBtu (Based on $1.39 /gal/140,000 Btu/gal)

2 Based on (0.295/.8) *heat value of natural gas into the engine, where 0.295 is the fraction of recoverable heat (engine coolant, engine exhaust, and compressor oil) and 0.8 is the assumed efficiency of the process water boiler displaced.

Redstone Arsenal (RSA)

Compressed air requirements vary across RSA depending on application and virtually each building where compressed air is used has its own dedicated system. Three compressed air end-use systems were selected for the site survey based on total installed horsepower, annual hours of operation, proximity to a natural gas supply, and accessibility due to security requirements. The systems serve Buildings 5436 (calibration laboratory facility), 7159 (rocket testing/fuel grinding), and 3634 (motor pool vehicle maintenance shop. Each building is served by a primary compressor and has a back-up compressor(s). Primary compressor sizes are 25 hp, 150 hp, and 50 hp for Buildings 5436, 7159, and 3634, respectively.

The compressed air survey identified several compressed air system operational cost cutting opportunities (Table 4). Savings for these opportunities are summarized in the following table. Cost savings are calculated at a FY 2000 to date (April 2001) total facility average electricity cost of $0.047/kWh.

RSA is not a suitable site for installation of an NGEDAC for the following reasons:

1. The location's small compressor size
2. Each compressor system surveyed operated 2,000 hours/yr or less
3. No heat recovery applications were discovered near the compressed air systems surveyed.

The full Level I Audit report for RSA is contained in Appendix I.

Table 4. RSA compressed air annual savings opportunities.

Cost Cutting Opportunity	Annual Energy Savings (kWh)	Annual Cost Savings ($)
Building 5436 (Calibration Laboratory Facility)		
Reduce Generation Pressure Start Setpoint to 110 psig and Stop Setpoint to 120 psig	2,872	135
Refrigerated Air Dryers/Compressor Interlock Start/Stop Control and Reduced Desiccant Drying Tower Regeneration Cycling	86,830	4,081
Building 3634 (Motor Pool Vehicle Maintenance Shop)		
Install 10 Horsepower Compressor as Lead Compressor	46,277	2,175
Total Savings	135,979	6,391

Rock Island Arsenal (RIA)

The compressed air system survey focused on Buildings 220 and 222 that are the largest users of compressed air and house the main compressors. It is estimated that the compressed air costs RIA about $154,326 based on 3,486,960 kWh of energy use per year. This is based on an average electricity cost of $0.044/yr. Building 220 has 5 compressors capable of providing 14,000 scfm of air at 100 psig pressure. However, the 4200 scfm Ingersoll-Rand reciprocating compressor and the 3700 scfm Worthington reciprocating compressor alternate in providing most of the facility's demand—3000 scfm during normal production hours and 1800 scfm at all other times. These units are 1940s and 1950s vintage, but are capable of efficient operation. No operational problems were noted with the main compressors or ancillary equipment.

RIA does not have an active leak detection program, and could benefit from such an effort. It is estimated that leak reductions could save the installation $26,340 annually, based on 360 scfm of reduction in losses, and corresponding energy savings of 600,686 kWh. Additional cost cutting/energy savings opportunities identified include: (1) operating the most efficient compressor(s) rather than rotating use of the many compressors on hand—annual savings of $7,297 and 165,840 kWh and (2) heat recovery from compressor inter and after coolers for space heating—annual savings of $14,853 in natural gas fuel expenses. Implementation of item 1 could also reduce operator time spent on the various units, freeing up this individual for implementing the leak detection program.

RIA is a marginal site (near break-even) for installation of an NGEDAC, with the most favorable location just outside Building 222. A 400 hp, 1800 scfm NGEDAC unit was evaluated in terms of energy performance and economics for the intended application. The unit would be operated at design capacity during the utility on-peak period—about 3,942 hours/yr. This operating mode was chosen because it enables the NGEDAC to reduce electricity use during the most expensive periods. Table 5 below summarizes the energy performance and operating cost savings.

The results shown are based on 2-year average gas prices, which were used because they damp the effect of significant gas price increases in 2000 through 2001. Table 6 shows changes in the annual operating costs of the NGEDAC system based on possible changes in future electric rates or gas prices. Note also that the maintenance costs for the NGEDAC are a function of the hours of operation for a given size unit.

The full Level I Audit report for RIA is contained in Appendix J.

Table 5. RIA annual energy use and operating costs.

	Electric Air Compressor	NGEDAC	Net Savings
Energy Use	1,251,388 kWh	13,490 MBtu gas (engine) -1,872 MBtu gas (engine heat recovery)[2]	1,251,388 kWh (electricity) -11,618 MBtu (gas)
Energy Operating Costs[1]	$61,920	$63,499	-$1,579
Operation & Maintenance Costs	$12,614	$23,652	-$11,038
Heat Recovery Costs	$0	-$8,816	$8,816
Total Costs	$74,534	$78,335	-$3,801

1 Electricity Costs: $0.049/kWh—includes demand and energy charges
Natural Gas Costs: $4.71/MBtu
2 Based on (0.22/.8)*heat value of natural gas into the engine, where 0.22 is the fraction of recoverable heat and 0.8 is assumed efficiency of heating boiler displaced.

Table 6. RIA annual operating costs ($)—sensitivity to changes in energy prices.

Energy Price Assumptions	Electric Air Compressor	NGEDAC	Net Savings
Higher Elec. Rates/Base Case Gas Rates			
1) Elec.: $0.054/kWh and Gas: $4.71/MBtu	$80,726	$78,335	$2,391
2) Elec.: $0.059/kWh and Gas: $4.71/MBtu	$86,918	$78,335	$8,583
Base Case Elec. Rates/Lower Gas Rates			
1) Elec.: $0.049/kWh and Gas: $4.24/MBtu	$74,534	$72,867	$1,667
2) Elec.: $0.049/kWh and Gas: $3.77/MBtu	$74,534	$67,398	$7,136
Higher Elec. Rates/Lower Gas Rates			
1) Elec.: $0.054/kWh and Gas: $4.24/MBtu	$80,726	$72,867	$7,859
2) Elec.: $0.059/kWh and Gas: $3.77/MBtu	$86,918	$67,398	$19,520

Sierra Army Depot (SIAD)

The compressed air survey focused on the feasibility of installing a 125 hp, 600 scfm natural gas engine driven air compressor as a direct replacement for the existing electric motor driven air compressor of the same capacity. The existing unit is housed in Building 210 but serves as the central compressed air source for two other buildings (208 and 209). Adequate space exists within this building for the NGEDAC. Natural gas is available and heat recovery for space heating appears promising.

Table 7 below summarizes the energy performance and operating cost savings. The net annual operating cost savings are estimated to be $16,373.

Table 7. SIAD annual energy use and operating costs.

	Electric Air Compressor	NGEDAC	Net Savings
Energy Use	225,472 kWh	2,166 MBtu gas engine - 300 MBtu (engine heat recovery)[2]	225,742 kWh (elec.) -1,866 MBtu (gas)
Peak Demand	100.9 kW	1.041 MBtuh	
Energy Operating Costs	$23,675	$15,769	$7,906
Peak Demand Costs	$9,106		$9106
Operation & Maintenance Costs	$3,072	$5,899	-$2,827
Heat Recovery Costs	0	-$2,188	$2,188
Total Costs	$35,853	$19,481	$16,373

1 Electricity Costs: $0.145/kWh average—includes demand @$7/kW and energy charges @.105/kWh (2/6/01 rate)

Natural Gas Costs: $7.28/MBtu (Average for April 2000—March 2001)

2 Based on (0.22/0.8) *heat value of natural gas into the engine, where 0.22 is the fraction of recoverable heat and 0.8 is assumed efficiency of heating or process water boiler displaced.

The survey indicated that other opportunities for reducing energy and operating costs associated with the compressed air system, that had been identified previously have not yet been implemented (Lin, et al., *Compressed Air System Survey at Sierra Army Depot*, ERDC/CERL TR-00-37, November 2000). These six opportunities (other than the NGEDAC) included:

1. Repair compressed air leaks
2. Change the air compressor control to low demand mode
3. Disconnect the air receiver from the oil/water separator
4. Duct outside air into the air compressor room
5. Install sensor-type valves on the purifier pre-filters
6. Replace the timer-type drain valves with sensor-type valves.

Collectively, these six opportunities represented an annual cost savings of $15,541 in electricity costs, an energy savings of 181,409 kWh, and a demand reduction of 49.9 kW. Given the recent price increases in electricity, these savings opportunities should be increasingly attractive to SIAD.

The full Level I Audit report for SIAD is contained in Appendix K.

4 Discussion

Table 8 summarizes the opportunities uncovered during the compressed air surveys and NGEDAC installation assessments—over $390,000 in annual operating cost savings resulting from 5,260 MWh reductions in electricity use, and 8,765 million Btu (MBtu) of natural gas use. Compressed air leak management was identified as a key opportunity at the majority of the installations. Since the surveys focused primarily on the major compressed air systems, other opportunities may well warrant further investigation.

Of the 11 sites surveyed, three (CEGA, APG and RSA) were found to be technically unsuitable for the installation of an NGEDAC, due to either the difficulty or expense of accessing natural gas, or to insufficient local compressed air requirements. This latter point is significant since NGEDACs typically are available in 50 hp or larger capacities, and become more economic with increasing capacity.

Table 8. Compressed air annual savings opportunities.

Base	Opportunity	Energy Savings	Operating Cost Savings ($)
PICA	Leak Repair (515 cfm)	468,000 kWh	41,200
WVA	Leak Repair (774cfm)	688,000 kWh	61,920
CCAD	Leak Repair(450cfm)	645,000 kWh	41,934
LSAAP	Drain Replacement (50cfm)	43,000 kWh	2,480
PBA	Leak Repair (420cfm)	626,000 kWh	35,700
CEGA	Use smaller compressor (20HP)	25,000 kWh	1,900
APG	Leak Management Program	1,682,000 kWh	94,426
LCAAP	Heat Recovery from Compressors	5,612 MBtu (gas)	32,437
RSA	Optimized Controls	135,979 kWh	6,391
	Lead-Lag Compressor		
RIA	Leak Management Program	766,526 kWh	48,490
	Use Most Efficient Compressors	3,153 MBtu (gas)	
	Heat Recovery from Compressors		
SIAD	Leak Management Program	181,409 kWh	25,000
	Controls		
Total		5,260,914 kWh	391,887
		8,765 MBtu (gas)	

Another technical consideration is the reliability of electric service. The NGEDAC can offer an advantage in this regard, since it can be configured to operate virtually independent of the base power system. For example, in the case of SIAD, which has some power (voltage) problems, NGEDAC technology offers an opportunity to take the existing electric unit off-line, freeing up electrical capacity.

PICA, WVA, LCAAP and SIAD appeared best suited for the installation of a natural gas engine driven air compressor. The principal determinant in this assessment was project economics (Table 9). SIAD was ultimately selected for the NGEDAC technology demonstration since it was a government-operated rather than contractor-operated facility, i.e., savings attributable to the NGEDAC would flow to the government rather than to a private entity. Note that the net savings were based on a 2-year average natural gas price, which included the higher prices seen from November 2000 to April 2001. The savings would have been substantially larger had lower natural gas prices been assumed (e.g., $3/MBtu-$4/MBtu). Clearly, NGEDAC installations make economic sense in situations where there is a reasonable spread between the price of natural gas and electricity or other competing energy source.

Table 9. NGEDAC annual savings estimates.

			Utility Rates		Annual Operating Costs		
Base	**NGEDAC**	**Capacity**	**$/kWh Electric**	**$/MBtu Gas**	**Electric Compressor**	**NGEDAC**	**Net Savings (Nominal Case)**
PICA	145hp	925 cfm	$0.088/kWh	$3.41/MBtu	$69,412	$48,106	$21,306
WVA	362 hp	1480 cfm	$0.09/kWh	$5/MBtu	$321,000	$240,000	$81,000
CCAD	352 hp	1400 cfm	$0.065/kWh	$6/MBtu	$122,618	$125,893	($3,275)
LSAAP	153 hp	780 cfm	$0.045/kWh	$5/MBtu	$12,796	$19,505	($6,709)
PBA	234 hp	809 cfm	$0.057/kWh	$4/MBtu	$82,400	$79,000	$3,400
LCAAP	350 hp	1670 scfm	$0.049/kWh	$5.78/MBtu	$107,406	$96,345	$11,061
RIA	400 hp	1860 scfm	$0.049/kWh	$4.71/MBtu	$74,534	$78,335	($3,801)
SIAD	125 hp	450 scfm	$0.145/kWh	$7.28/MBtu	$35,853	$19,481	$16,373

5 Conclusions and Recommendations

Conclusions

This study conducted compressed air system surveys at 11 Army industrial sites to identify opportunities to reduce compressed air operating expenses and to determine the suitability of the site for the NGEDAC installation.

The study revealed substantial opportunities for savings during the compressed air surveys and NGEDAC installation assessments—over $390,000 in annual operating cost savings resulting from 5,260 MWh reductions in electricity use, and 8,765 million Btu (MBtu) of natural gas use. Compressed air leak management was identified as a key opportunity at the majority of the installations.

The study concluded that, of the 11 sites surveyed, three (CEGA, APG and RSA) were found to be technically unsuitable for the installation of an NGEDAC, due to either the difficulty/expense of accessing natural gas or insufficient local compressed air requirements.

PICA, WVA, LCAAP and SIAD appeared best suited for the installation of a natural gas engine driven air compressor. The principal determinant in this assessment was project economics. In general, NGEDAC installations make economic sense in situations where there is a reasonable spread between the price of natural gas and electricity or other competing energy source.

Recommendations

This work has identified opportunities to reduce compressed air operating costs, and specific applications where NGEDACs offer technical and economic benefits. To gain these benefits, it is recommended that Army facilities:

1. Conduct a compressed air system survey if one has not been performed. Air flow monitoring should be conducted to provide insights into system operation.
2. Consider periodic follow-up surveys.
3. Establish a leak management program if one has not been devised.

4. Consider NGEDACs as an alternative to electric motor driven air compressors. Make sure the specifications include an accurate load profile. Actual compressor loads can be very different from estimates based on operator's assumptions about their process equipment's air requirements. Make sure to evaluate the part load performance of the options being considered.
5. Maintenance should be tracked closely to isolate the differential costs between NGEDACs and electric motor driven air compressors.
6. Compare NGEDAC performance to better understand differences among different manufacturer's products, as well as increase the overall confidence in the data collected.
7. Conduct follow-up review on the implementation of the survey recommendations, and document the benefits.

Appendix A: Compressed Air System Survey at Corpus Christi Army Depot

Background

The compressed air system at the Corpus Christi Army Depot is a centralized system with three Ingersoll Rand three-stage centrifugal compressors as the primary air supply: 4,000 cfm, 2,500 cfm, and 1,200 cfm class.

CCAD is evaluating a proposal to decentralize this system and place dedicated compressors in many of the buildings now served by the central system. This will create several areas of savings:

- eliminating leaks by abandoning some 8,000 ft of older distribution lines connecting the buildings
- allowing the smaller horsepower satellite units to be optimum-sized to the local demand
- allow the satellite compressors to shut off when either the local air demand is low or that particular building is not in production.

The largest production area, Building #8, will set up its air system with two of the central centrifugals (2,500 and 1,200 class) along with potentially a 750 cfm and a 450-cfm Quincy rotary screw. These four units will be controlled by a networking capacity unloading control system. The objectives will be to assure that:

- no centrifugal is in blow off
- only one unit at part load; all others at full load or off.

This Compressed Air Decentralization Plan will also create energy savings by replacing current noncycling and other inefficient dryers with effectively sized cycling-type refrigerated dryers.

Since most staff believe this decentralization program will be implemented, the economic benefits of a natural gas engine driven unit is being evaluated in the context of the proposed decentralized system rather than the current centralized system.

Following is an outline of the basic air use in each selected building for the decentralization project and what machinery will be added:

Building 340

173 cfm demand (peak)

Has two 1,000 cfm noncycling dryers and one 250 cfm noncycling dryer.

Note: Two 1,000s are not needed.

Decentralized Project

Install two 20-hp rotary screw; run one or two as required—small cycling dryer

Run at 100 psig

Savings

Can be shut off from 100 to 50 percent when called for, small dryer only runs when needed.

Building 49

167 cfm demand (peak)

Decentralized Project

Two 25-hp rotary screw with cycling dryer

Savings

Can be shut off from 100 to 50 percent when called for, dryer only runs when needed.

Buildings 1808 & 1828

Demand (average) 1828—120 cfm

1808—100 cfm

Decentralized Program

Install two 180 cfm rotary screw compressors in 1828 with cycling dryer and receiver.

Remove old noncycling dryer

Savings

Can be shut off from 100 to 50 percent or less when called for, dryer only runs when needed.

Building 1880

225 cfm peak demand

Air to come from supply in 1828.

Building 206 and 127

625 Cfm

Now has one noncycling dryer for cold air in 127.

Decentralize Project

Add two new 288 cfm compressors in building 206

Move dryer between 206-127

Savings

Shut off utility from 100 to 50 percent when called for.

Building 339 (Motor Pool)

500 cfm demand at 125 psig

Decentralization Project

Add two 35 cfm compressors and dryer—125 psig

Savings

Shut off when not needed. Proper pressure without raising pressure of the whole system.

Hanger 45

100 cfm demand

Now have one 250 cfm dryer, noncycling

Decentralization Project

Add one 25-hp 100 cfm compressor at 125 psig

Auto start/stop controls—200 cfm cycling dryer

Savings

Shut off when not needed and uses cycling dryer. oversized for adequate summer performance.

Hanger 43

100 cfm demand

Decentralization Project

Add one 25-hp 100 cfm compressor at 125 psig

Auto start/stop controls—200 cfm cycling dryer

Savings

Shut off when not needed and uses cycling dryer. oversized for adequate summer performance.

Hanger 46

100 cfm demand

Decentralization Project

Add one 25-hp 100 cfm compressor at 125 psig

Auto start/stop controls—200 cfm cycling dryer

Savings

Shut off when not needed and uses cycling dryer. oversized for adequate summer performance.

Buildings 1209, 1219, 259, 30, 252 & Wheel Tower

Now part of Building #8 flow measurement.

Decentralization Project

Leave as part of Building #8; fed by underground lines.

Other pertinent areas of decentralization program

1,000-hp centrifugal will be abandoned in Building 13.

2,500 & 1,200 cfm class centrifugals will be moved to Building 8.

Quincy 450 rotary screw (100 hp) will be moved next to Quincy 750 rotary screw (both air cooled) under outside shed in northwest corner of Building 8. Both units will become part of Building 8 air supply (also serves Building 1209, 1219, 259, 30, 258, and two whirl towers).

Measured air in Building 8 flow during September 2000 investigation:

- weekday production (2,080 hours)—peak 2,400 cfm with an average about 2,200 cfm
- nonproduction (6,240 hours)—900 cfm average flow
- weekend production (440 hours)—1,450 cfm average flow

- estimated leaks (which will be repaired)—400-450 cfm.
- new average demand with leaks repaired: 1,750 cfm (weekday production), 450 cfm (nonproduction), and 1,000 cfm (weekend production).

System Baseline

The information listed in Tables A1 through A12 summarize the key characteristics describing the performance and economics of the current compressed air system. The tables below were developed based on the data collected during the site visit and with discussions with plant personnel. The estimates are conservative and reflect observed performance of each compressor compared to load cycle. The estimates reflect Building #8 flow as measured in September 2000.

Table A1. Key system characteristics: current system serving Building #8 Area.

Measure*	Weekday Production (electric)	Nonproduction (electric)	Weekend Production (electric)	Total
Average System Flow	2,200 cfm	900 cfm	1,450 cfm	NA
Average kW	367.2 kW	170.46 kW	263.11 kW	1943215 kWh
Operating Hours	2,080 hrs	6,240 hrs	440 hrs	8, 760 hrs
Specific Power	5.99 cfm/kW	5.27 cfm/kW	5.51 cfm/kW	NA
Electric Cost for Air (cfm/yr)	$22.56 /cfm/yr	$76.82 /cfm/yr	$5.19 /cfm/yr	$104.57 /cfm/yr
Electric Cost for Air (psig/yr)	Not applicable with centrifugal			
Total Annual Electric Cost for Air	$49,645 /yr	$69,138 /yr	$7,525 /yr	$126,308 /yr

* Assumes blended power rate = 0.065 kWh.
Centrifugals have an estimated 20% turndown.
Pressure = 115 psig.
Controls (Centrifugal)= Modulator Blow Off

Table A2. Compressor utilization—current system serving Building #8 Area.

	Manufacture	% of Load*	% of Power	Full Load kW x% of Power	Net kW	CFM
Weekday	IR Cent—600 hp	85%	90%	408 kW x 0.9	367.2	2,200
Non-Production	IR Cent—300 hp	72%	85%	200.6kW x 0.85	170.5	900
Weekend Production #1	Quincy 450	44%	78%	80.22 x 0.78	62.57	202
Weekend Production #2	IR Cent—300 hp	100%	100%	200.54 x 1	200.54	1,248

* Predicted load profile/energy use—Building 8 and (1209, 1219, 2593, 258 & two Whirl Towers) in projected decentralized air system without leak fixes.
Data from flow meter measurement taken September 20 to September 26, 2000.

Table A3. Key system characteristics: NGEDAC serving current Building #8 Area.

Measure	Weekday Production (NG)	Weekday Production (Electric)	Non-Production (NG)	Weekend Production (NG)	Total
Average System Flow	1,450 cfm	750 cfm	900 cfm	1,450 cfm	NA
Input Energy	352.45 BHP	121.73 kW	239 BHP	352.45 BHP	NA
Operating Hours	2,080 hrs	2,080 hrs	6,240 hrs	440 hrs	8760 hrs
Specific Power*	4.11 cfm/hp	6.16 cfm/kW	3.77 cfm/hp	4.11 cfm/hp	NA
Electric Cost for Air (flow)	$22.57 /cfm/yr	$21.94 /cfm/yr	$73.81 /cfm/yr	$4.79 /cfm/yr	$100.96 /cfm/yr
Annual Electric Cost for Air	$32,725 /yr	$16,455 /yr	$66,429 /yr	$6.922 /yr	$122,531/yr

* Blended Power Rate = $0.065 per kWh @ 115 psig.
Leaks not fixed
NG rate—$6.00 per Million BTU

Table A4. Compressor utilization: NGEDAC serving current Building #8 Area.

	Manufacture	% of Load	% of Power	Full Load kW x % of Power	Net kW	Actual cfm
Prod*	NGED	100	0.95	371 x 0.95	352.45 hp	1,450
Prod	Quincy QSI 750	100	100	121.73 x 1	121.73	750
Non	NGED	64.28%	64.28	371 x 0.6428	239 BHP	900
Wkd	NGED	100	0.95	371 x 0.95	352.45 hp	1,450

* Psig = 115; Operating Hours = 8,760.
Conditions/Comments
Based on the load profile without leaks fixed, the NGEDAC will save about $3,777 /yr in energy costs.

Table A5. Key system characteristics: decentralized system serving Building #8 area (electric).

Measure	Weekday Production	Non-Production	Weekend Production	Total
Average System Production	1,750 cfm	450 cfm	1,000 cfm	NA
Input Energy	347 kW	80.22 kW	170 kW	NA
Operating Hours	2,080 hrs	6,240 hrs	440 hrs	8,670 hrs
Specific Power	5.04 cfm/kW	5.60 cfm/kW	5.88 cfm/kW	NA
Electric Cost for Air (flow)	$26.80 cfm/yr	$72.42 cfm/yr	$4.86 cfm/yr	NA
Electric Cost for Air (pressure)	$234.57 /psig/yr	$169.94 /psig/yr	$24.31 /psig/yr	NA
Annual Electric Cost for Air	$46,914 /yr	$32,589 /yr	$4,862 /yr	$84,365 /yr
Blended Power Rate = $0.065 per kWh, Leaks fixed—450 acfm.				

Table A6. Compressor utilization: decentralized system serving Building #8 area (electric).

	Manufacture	% of Load	% of Power	FL kW x% of Power	Net kW	Actual cfm
Prod	IR 600 HP	69%	85%	408 x 347	347	1,750
Non	QuincyQSI 450	100%	100%	80.22 x 1	80.22	450
Wkd	IR 300 HP	79%	85%	200.54 x 0.85	170	1,000

Conditions/Comments

Building 8	Before Leak Repair	After Leak Repair
Prod Weekday	2,200 acfm	1,750 acfm
Non-Production	900 acfm	450 acfm
Weekend Production	1,450 acfm	1,000 acfm

Predicted load profile/energy use—Building 8 and (1209, 1219, 259, 3, 258, & two (whirl towers) in projected decentralized air system with 400 scfm of leaks fixed [see Report]. Data from flow meter measurement taken September 20 to September 26, 2000.

Table A7. Key system characteristics: decentralized system serving Building #8 area (NGEDAC/electric hybrid) "Alternative 1—Add One Large 1,400 cfm Class Natural Gas Engine Drive."

Measure	Weekday Production (NG)	Weekday Production (electric)	Non-Production (electric)	Weekend Production (NG)	Total
Average System Flow	1,400 cfm	350 cfm	450 cfm	1,000 cfm	NA
Input Energy	352.45 hp	68 kW	80.22 kW	264 BHP	NA
Operating Hours	2,080 hrs	2,080 hrs	6,240 hrs	440 hrs	8,760 hrs
Electric Cost for Air (flow)	$23.38 /cfm/yr	$26.26 /cfm/yr	$72.30 /cfm/yr	$5.19 /cfm/yr	NA
Annual Electric Cost for Air	$32,725 /yr	$9,193 /yr	$32,537 /yr	$5,185 /yr	$79,640 /yr
Blended Power Rate = $0.065 per kWh, Leaks fixed—450 acfm, BSFC = 7,440 BTU/hp/hr					

Table A8. Compressor utilization: decentralized system serving Building #8 area (NGEDAC/electric hybrid).

	Manufacture	% of Load	% of Power	Full Load hp x% of Power	BHP	Actual cfm
Prod#1	NGED 1400	100%	100%	371 hp x 0.95	352.45 hp	1,400
Prod#2	QSI 450	77%	85%	80.22 x 0.85	68 kW	350
Non	QSI 450	100%	100%	80.22 x 1	80.22 kW	450
Wkd	NGED 1400	71%	75%	352.45 x 0.75	264 BHP	1,000

Conditions/Comments:

352.45 x 7,440 x 2,080 ¸ 1,000,000 = 5,454.23 cfm x 6 = $32,725 /yr

264 x 7,440 x 440 ¸ 1,000,000 = $5,184 /yr

Additional savings from heat recovery total Btu on plant 25% recoverable = $9,000/yr.

Table A9. Key system characteristics: decentralized system serving Building #8 area (NGEDAC) "Alternate 2—Add One Large and One Small Natural Gas Engine Driven."

Measure	Weekend Production	Non-Production	Weekend Production	Total
Input Energy	7,440/7,859 Btu/hp/hr	7,859 Btu/hp/hr	7,440 but/hp/hr	NA
Operating Hours	2,080 hrs	6,240 hrs	440 hrs	8760 hrs
Horsepower	440.37 bhp	113.04 bhp	264 bhp	NA
Electric Cost for Air (flow)	$23.62 /cfm/yr	$73.91 /cfm/yr	$5.19 /cfm/yr	NA
Annual Electric Cost for Air	$41,348 /yr	$33,260 /yr	$5,185 /yr	$79,793/yr
Gas Rate = $6 per Million BTU, Leaks fixed—450 acfm				

Table A10. Compressor Utilization: Decentralized System Serving Building #8 Area (NGEDAC) -- Weekday Production.

	Manufacture	% of Load	% of Power	FL BHP x% of Power	BHP	Actual cfm
1	NGEDAC 1400	100%	100%	352.45 bhp	352.45 hp	1,400
2	NGEDAC 800	56%	56%	157 x 0.56	87.92	350

Pressure = 115 psig; Operating Hours = 2,080 hrs.

Operating Cost Estimate (During Production Hours):

Unit #1= 352.45 hp x 7440 BSFC x 2080 hrs ¸ 1000000 x $6 per million btu = 32,725/yr

Unit #2 = 87.92 x 7,859 x 2,080 ¸ 1,000,000 x 6 = $8,623 /yr

TOTAL $41,348 /yr

Table A11. Compressor utilization: decentralized system serving building #8 Area (NGEDAC) -- non-production period air.

	Manufacture	% of Load	% of Power	FL BHP x% of Power	Net BHP	Actual cfm
1	NGEDAC 800	72%	72%	157 x 0.72	113.04	450

Operating Cost Estimate (During Non-Production Hours):

Unit #1 = 113.04 hp x 7,859 BSFC x 6,240 hrs ¸ 1,000,000 X 6 = $33,260 /YR

Table A12. Compressor utilization: decentralized system serving Building #8 Area (NGEDAC) -- weekend production air.

	Manufacture	% of Load	% of Power	FL BHP x% of Power	Net BHP	Actual cfm
1	NGEDAC 1400	71%	75%	352.45 x 0.75	264	1,000
Operating Cost Estimate (During Weekend Hours): Unit #1 = 264 hp x 7,440 BSFC x 440 ¸ 1,000,000 x 6 = $5,185 /yr						

Savings Summary

Average electric rates at the plant are $0.065/kWh. The projected plant electric cost for air production, as running today, is probably in excess of $126,000/yr. The system pressure appears to run about 100 psig in the headers during production. This report identifies the “electric cost/hr/loaded cfm” of air used and the cost/cfm with a NGED compressor.

The leak reduction program associated with the Compressed Air Decentralization Plan will cut energy costs by $42,000. The various NGEDAC combinations will reduce energy costs by an additional $3,000 to $5,000.

It is important to note that other recoverable compressed air costs should also be considered, i.e., maintenance, water costs, depreciation, etc. Usually, the electric cost is between 70 and 90 percent of the total “variable compressed air costs.” Associated maintenance and other costs will be, in all probability, at least 20 percent or more of the identified electric cost. Existing plant records may already have these identified.

Supply-Side System Review (Primary Air Compressor Supply)

All the units the facility is considering using as base supply for Building 8 and its system are of basic current state-of-the-art technology and relatively power efficient compressors for their class of air compressor (Table A13). The two centrifugals are of three-stage type, and were not running during the site visit, but plant personnel say they run well; each runs about 20 percent turn down.

The two rotary screws are single-stage, lubricant-cooled, with relatively large rotors and effective performance for their type of unit. The QSI 750 has been set up for high pressure (175—190 psig) which may be nothing more than an oversized motor and starter or there may have to be other changes to the unit to modify it for 100 psig class service, such as: minimum pressure valve, separator, oil return or scavenge line, controller, etc.

Table A13. Compressor unit profile (rating at full, average pressure, 100 psig).

Type	Centrifugal	Centrifugal	Single-Stage Rotary Screw	Single-Stage Rotary Screw
Brand	Ingersoll Rand	Ingersoll Rand	Quincy	Quincy
Model	2CV29M30	2C12M3E	QSI 750	QSI 450
ACFM	2,538	1,248	750	450
Full Load Pressure	115	115	100	100
kW @ 100 psig	407.49	200.54	121.93	80.22
Cfm/kW/100 psig	6.23 cfm/kW	6.23 cfm/kW	6.15 cfm/kW	5.60 cfm/kW
Annual Elec Cost $/cfm	$91.39 cfm/yr	$91.39 cfm/yr	$92.58 cfm/yr	$101.68 cfm/yr
Annual Elec Cost $/psig	NA	NA	$347.20 psig/yr	$228.77 psig/yr
Based on Blended Power Rate = $0.065 per kWh; Operating Hours = 8,760 hrs/yr.				

Before actually set to run as a low pressure unit, the OEM (Quincy Compressors, Quincy, IL) should be contacted.

The primary compressed air supply will be produced by relatively efficient air compressors that are capable of delivering the 100 psig full load pressure in a continuous manner. The units are well applied. They appear to be in good operating order and well maintained. Key characteristics of the units are summarized in Table A-13. These are the units anticipated to become the compressed air supply for building #8 after the decentralization program is implemented.

Compressor Capacity Controls

The two most effective ways to run air compressors are at “Full Load” and “Off.”

Capacity controls are methods of restricting the output cfm delivered to the system while the unit is still running. This is always a compromise and is never as efficient as full load on a specific power (cfm/hp) basis.

Reciprocating Controls

Single-acting, air-cooled, tank-mounted reciprocating compressors may be the best selection in the future for some of the very lightly loaded with intermittent running times. These units use two-step unloading and should be equipped with dual control, constant speed, and auto start/stop.

Rotary Screw Controls

The two most common controls used are modulation and online/offline. Modulation is relatively efficient at very high loads—and inefficient at lower loads. Online/offline controls are very efficient for loads below 60 percent, when properly applied with adequate time for blow down.

There are several other control types (e.g., "rotor length adjustment" or "variable displacement" and "variable speed drive") that have very efficient turn down from 100 percent load to about 60 percent load. These controls must be installed correctly to operate efficiently. Piping and storage should be available close to the unit with no measurable pressure loss at full load to allow the signal to closely match the air requirements.

The current system has full modulating control with blow down (auto start/stop can be added if not on now) and idle. These appear to be very well applied.

Centrifugal Controls

The two most common controls used are modulation and blow off. Modulation is relatively efficient at very high loads, but will not work much below 80 to 85 percent load. After "modulation" or "turn-down," the compressor then just "blows off" excess air. The basic power draw at the blow off point then stays the same regardless of the load.

There are modern electronic control systems that can be applied today that will effectively close off the inlet and will blow the unit down to idle and significantly reduce the kW draw. Inlet guide valves are available to increase the effective turn down range from 15 to 20 percent to 25 to 30 percent and increase the unloaded efficiency.

The current system has modulation with blow off. The two small units can full unload and idle when the installation and loads allow this. It does not appear that they will do this often now. Care should be taken to consider this in the new Building #8 installation.

Central Networking Control System

The proposal for a decentralized air system calls for a full networking control system for Building #8. This microprocessor-driven centralized full networking electronic control system, when properly applied, will automatically place the most efficient machine online and assure no more than one partial loaded unit at a time. They can be much more effective than a standard sequencer.

Air Treatment and Air Quality

Dryers

The decentralized plan calls for all noncycling refrigerated compressed air dryers to be replaced with "oversized" (to allow for high ambient summer performance) cycling-type [will only use the kW input commensurate to the moisture load it sees] refrigerated air dryers. Installed and selected properly, these will not only give high performance but optimized energy usage.

Basic Air Drying

Refrigerated dryers require a refrigeration system to mechanically cool the air. The lowest possible consistent pressure dew point with a noncycling dryer is +40 °F. Cycling dryers not only save power (60 to 75 percent), but also can deliver a lower pressure dew point (down to 35 to 38 °F).

Desiccant Dryer Regeneration equipment removes moisture vapor by "adsorbing" it to desiccant beads. These dryers can consistently deliver a pressure dew point to –40 °F or lower, which removes much more water than conventional refrigeration units. To regenerate the wet tower while the other tower is drying requires the use of heat in some form and some dry air to "sweep" or "purge" the exchanged moisture out. Desiccant dryers are usually rated at the same 100 °F and 100 psig conditions.

Water or Oil Carryover in System

Water (condensate) and oil carryover problems in the current air system are significant and can be expected to increase in magnitude during the summer.

The correct way to eliminate water and oil in the air system is to clean and dry the air immediately after it is produced in the compressor room. Then clean dry air can be stored in a separate air receiver and flow it to the system, as required. Some guidelines for controlling oil and water carryover include the following:

Generally, it is best to eliminate the water and oil at the air source before it enters the air system.

Every 20 °F increase in temperature doubles the "moisture load" the compressed air will hold.

Compressed air dryers are usually capacity rated with 100 °F, 100 psig inlet air conditions. At 120 °F, 100 psig, the dryer's capacity rating is reduced 50 percent.

Putting "dry/or oil free" air into system 90 percent of the time and then allowing wet/oily air in sporadically 10 percent of the time will, in reality, make the system wet or oily all the time. The liquid water and/or oil will fall out in the piping system continuing to "re-entrain" and contaminate and/or collected in the "low spots" of the system; thus, recontamination as it is pulled into the flowing compressed air system. A wet/oily system may well take many months of continued flowing of clean dry air to "clean up."

Identify required pressure dew point and meet it. Monitor performance, if critical.

Aftercoolers

The centrifugal air-cooled after coolers are water cooled with cooling water from a closed radiator-type cooling system. The rotary screws have air-cooled aftercoolers. During the very hot summer weather, there is very little chance that they can deliver 100 °F air to the dryer. Therefore, the "oversizing" will help this. The cycling will eliminate potential "freeze-up" problems.

Pre-Filters and After-Filters

Pre- and after-filters are generally either particulate or coalescing type and their use depends on the type of dryer in use and various installation considerations.

Desiccant dryers always require a high-quality coalescing prefilter to keep liquid oil and water out of the drying tower. They also always require an effective particulate filter after the dryer to keep "desiccant dust" from migrating into the system.

Refrigerated dryers may or may not need pre and after filters depending on the piping, type of compressor, and desired degree of cleanliness. If the inlet air is apt to be dirty and fouled with carbon scale, etc., a particulate prefilter is called for. If it is liable to have significant liquid or heavy oil mist, a coalescing (or combination coalescing particulate) pre filter may be needed. If oil/water mist is leaving the dryer, a coalescing after filter may be in order.

Care in selection must be taken in all cases because:

- wasted air pressure costs energy dollars
- wasted air pressure neutralizes the operating pressure band early
- standard coalescers will usually not perform effectively at flows much below 20 percent of their rated capacity
- standard coalescers life will be significantly shortened by particulate load
- loose-packed, deep-bed mist eliminators (those with correct elements) will coalesce effectively throughout the total scfm range
- loose-packed, deep-bed mist eliminators (those with correct element) have very high particulate load capability.

Automatic Condensate Drains

Level-actuated, see-through drains should be installed at all central drain points. Most of the drains observed were mechanical level-operated and these were well installed and maintained. As long as this continues, there is no reason to change. Your maintenance personnel may find the see-through electric or pneumatic-actuated easier to maintain and enhance system integrity.

Most refrigerated dryers today have built-in dual timer electric drains. It is recommended that the site consider using level drains instead.

Demand-Side System Review

A review of demand-side issues is not a part of the scope for the Preliminary Site Analysis (Level l Audit). However, this section is included as background material should the Corpus Christi site be selected for the next phase.

Minimum Effective System Pressure

The cornerstone of any effective demand-side air conservation program is to identify and operate at the lowest acceptable operating pressure required at various sectors and operating units in the plant. This should be a continuing program and part of any training awareness procedures.

Regulator Usage

Some regulators are probably set at higher than necessary feed pressure to the process, with some wide open to full header pressure. The questions plant personnel ask are: Is there a minimum effective pressure at operation established at the unit for each product run? If so, is it being adhered to?

In this type of operation, it is very important that the actual inlet pressure to the process be known and that the lowest effective pressure be held steady for the proper product quality. Installation of storage bottles downstream of the regulator may be needed to “close up” the pressure readings at rest and at operation.

Compressed Air Condensate Handling

If the condensate goes to water treatment and the discharge condensate meets the requirements of the water treatment facility plant, there is no problem.

However, if the base discharging the condensate to a storm sewer or in some other manner to ground water (Federal EPA minimum is 10 ppm) or are required to separate it by your local water treatment facility. This should be discussed in detail.

Automatic Ball Valves

Some of the most significant areas for leaks in any high-production plant involve shutting off the air supply to machinery when not in use. When these are found, there are usually some very economical and easy methods to automatically do this. Electric-operated automatic ball valves that can be installed in the main feed line to a piece of equipment and be wired in so as to open and close when the machine is powered up or shut off.

Cabinet Coolers

Cabinet cooling is often required to obtain reasonable life and performance of the electronic equipment in control cabinets. There are various means of accomplishing this. Blowing straight compressed air into the cabinet is generally very inefficient. Vortex coolers can use chilled air with no moving parts and use less of it.

Vortex coolers should always:

- be regulated to the lower effective pressure
- be equipped with the lowest possible flow generator
- be equipped with automatic temperature controlled shutoffs.

Refrigeration units should be carefully selected and equipped with automatic Regulation Control. Heat tubes are the most energy efficient when applied and can cool a "sealed cabinet."

There may be some cabinet coolers in use in the plant. Some with refrigeration and some with compressed air-driven Vortex Coolers; and some just have compressed air blowing into them.

Blow Offs

The Depot may have ¼-in. lines running as blow off on units at 80 psig. These will use approximately 32 cfm each.

An alternate is an "air amplifier" which takes less compressed air and through "Venturi" action amplifies the usable air by pulling in significant amounts of ambient air and mixing it directly into the air stream. These have amplification ratios up to 25:1. Using 10 cfm of compressed air would generate a savings of 22 cfm compressed air per ¼-in. blow off and will supply 250 cfm blow air to the process.

Vacuum Generators

If the plant's current production system uses vacuum generators, they should be reviewed. Vacuum generators are very convenient, very responsive, and very inefficient compared to positive displacement pumps, i.e., rotary screw, reciprocating.

Energy cost escalates as vacuum goes down with Venturi generators. Energy cost falls as vacuum goes down after about 14 in. with positive displacement pump. It is very important to only run a Venturi vacuum generator to a minimum vacuum and a minimum acceptable "on time" cycle at the lowest possible pressure.

Air Operated Diaphragm Pumps

Although air-operated diaphragm pumps are not very energy efficient, they tolerate aggressive conditions relatively well and run without catastrophic damage even if the pump is dry. There are several areas to pursue in the future to perhaps generate significant air savings:

Is the air-operated diaphragm pump the right answer? An electric pump is significantly more power efficient. Electric motor driven diaphragm pumps are available. An electric motor drive progressive cavity pump may well work.

Consider the installation of electronic or ultrasonic controls to shut the pumps off automatically when they are not needed. Remember the pump uses the most air when it is pumping nothing

Is the base operating most of the time at the lowest possible pressure? The higher the pressure, the more air used. For example, often a filter pack operation, the pump does not need high pressure except during the final stages of the filter packing cycle. Controls can be arranged to accomplish lower pressure in the early stages and higher pressure later that may generate significant savings.

Misapplied High Pressure Air

High-pressure air being used for very low-pressure applications is not an efficient use of energy. A close review of the Depot's system should be made and measurements taken to identify if there is any potential energy savings in using an alternate source of low-pressure or high-pressure air in the production area. This could be a part of a future study of demand-side activities at Corpus Christi Army Depot.

NGEDAC Site Assessment

This section provides a preliminary assessment of the opportunity for gas engine driven compressors at Corpus Christi Army Depot. The assessment is based on five key design factors:

1. Operating the new system as a fully hybrid system with the existing electric system
2. Meeting or exceeding environmental requirements of the area
3. Improving the current demand system so that the air requirements for the new system are minimized, while reducing system operating costs for the Depot
4. Providing an effective component in the Depot's plan to implement a decentralized air system
5. Including an effective heat recovery application for NGEDAC.

NGEDAC System Design Factors

The conceptual design for the gas engine-driven system is based on two alternatives. The first alternative includes a 1400 cfm class NGEDAC unit that could serve three-quarters of the anticipated air flow during Weekday Production in the Building #8 system under the Decentralization Plan. The second alternative adds both a 1400 cfm class and an 800 cfm class NGEDAC unit that could serve the entire level of air flow requirements.

In either case, the NGEDAC system will be configured as a hybrid system in conjunction with the existing electric system. In this way, the existing electric system can serve as a back-up to the gas engine system, if the gas system has a planned or unplanned shutdown or if the air requirements of the base are suddenly increased. Using this approach, the Department of Defense can gain experience with not only operating a gas engine driven system, but also with integrating it with electric systems to improve overall compressed air system reliability and flexibility. This flexibility is important given the increasing uncertainty concerning the price and supply reliability of most energy sources.

Environmental issues are expected to be minimal in this application given the key areas to address in any major project on this nature, although less so in this part of the country. The assessment is based generating up to 2.6 gm/bhp/hr for NOx and 1.75 gm/bhp/hr for CO.

Foremost in the design and implementation plan for the NGEDAC unit is coordination with the plans to decentralize the air and steam system at Corpus Christi Army Depot. The preferred location for the NGEDAC unit(s) is with the new power house to be constructed as part of the Steam Decentralization Project. The timing for that construction is likely to be after the completion of the NGEDAC Program. For this reason, the design phase for both the Decentralization Project and the NGEDAC Project must be tightly coordinated and completed very soon. In addition, the goal of including a heat recovery system with this NGEDAC application will require further coordination.

Operating Cost Comparison

Table A14 displays the energy costs for various operating scenarios. Implementing the leak reduction program associated with the Building #8 area cuts flow by 450 cfm and energy costs by $42,000. The NGEDAC units save an additional $5,000 in energy costs over the All Electric System for the 1750 cfm production period flow under the Compressed Air Decentralization Plan.

Table A14. Annual energy cost comparison of current system, decentralized system, and proposed NGEDAC units.

	Current Flow—Bldg #8 (Pre-decentralization) 2200 cfm/Production	Planned Flow—Bldg #8 (Post-decentralization) 1750 cfm/Production
All Electric System	\$126,308 (Table A-1)	\$84,365 (Table A5)
NGEDAC/Electric Hybrid	\$122,531 (Table A-3)	\$79,640 (Table A7)
All Gas System: NGEDAC (2 units)		\$79,793 (Table A9)

The comparison is based on input energy costs of \$6/Million Btu for gas and \$65/MWh for electricity. Energy prices paid by the Army Depot to the Naval Air Station are expected to increase in the near-term. An increase from \$65 to \$85/MWh and from \$6 Million to \$7.20/Million Btu for gas will increase the relative annual energy savings for the Hybrid Gas/Elec System from \$5,000 to \$15,000 and for the All Gas System from \$5,000 to \$19,000.

Relative to the All Electric System, annual maintenance costs are \$8,000 higher for the Hybrid Gas/Elec System and \$17,000 higher for the All Gas System. Maintenance costs for the NGEDAC units are based on 1.5 cents/hp-hr.

Assuming the Steam Decentralization Plan is implemented, NGEDAC at Corpus Christi provides an excellent showcase for demonstrating the application of heat recovery. With a recovery rate of 25 percent, the benefit of heat recovery is \$9,000 for the Hybrid System and \$20,000 for the All Gas System.

The evaluation at Corpus Christi Army Depot needs to reflect the existence of two distinct sets of energy price signals: those provided by the Corpus Christi Naval Air Station to the Army Depot and those provided by the energy utilities to the Naval Air Station. The electric rate structure governing the cost of electricity paid by the Naval Air Station reflect an unusually high percentage of the electric bill. The rate structure is made up of a large demand charge based on the peak demand during regular business hours during four summer months and ratcheted for the remaining 8 months at a 90 percent level. Such rate structures are ideal candidates for summer peak load shaving, for which the NGEDAC unit is well-suited. However, the rate structure for the Army Depot includes no demand charge element and hence no savings associated with a peak load shavings strategy.

The benefit in peak shavings is estimated as being the value of avoiding the peak demand ratchet set during the summer peak and in place for the eight non-summer months at a 90 percent level.

Peak Shaving Value = Monthly Demand Charge X Summer Peak Shave X 90% X 8 Months
= $11/kW X Summer Peak Shave X 90% X 8 months
= $79.2 X Summer Peak Shave

In the case of the Hybrid System, the Summer Peak Shave = 280 kW or $22,000. In the case of the All Gas System, the Summer Peak Shave = 348 kW or $28,000.

Table A15 lists the key factors in comparing overall operating costs. Not including the potential energy price increase scenario, the Hybrid System shows a net annual savings of $28,000 relative to an All Electric System, while the All Gas System is estimated to have an annual savings of $36,000. Including the energy price increase scenario bumps the annual savings estimate to $38,000 and $50,000 for the Hybrid Gas System and All Gas System, respectively.

Capital Cost Assessment

Table A16 displays the capital cost estimates for the one NGEDAC unit (1400 cfm) included in the Hybrid System and the two NGEDAC units (2200 cfm) included in the All Gas System. The capital costs include the catalytic converter for the gas engine. The costs also include an estimate for all installation and freight costs and a budget estimate to erect an outside enclosure to house the NGEDAC unit in the vicinity of where the new power would be constructed under the Boiler Decentralization Plan. Without consideration to potential cost reductions resulting from negotiating or utility rebates, the capital costs are $290,000 for the All Gas System and $180,000 for the Hybrid System.

Table A15. Key factors in comparing overall operating costs.

Key Operating Cost Factors	"Hybrid System" (only one NGEDAC unit)	"All Gas System" (two NGEDAC units)
Net Energy Savings	5K	5K
Incremental NGEDAC Maint Cost	(8K)	(17K)
Heat Recovery Benefit	9K	20K
Summer Peak Shaving Benefit	22K	28K
Total Annual Operating Savings	$28K	$ 36K
Additional Energy Savings*	$ 10K	$ 14K
Total Annual Operating Savings	$ 38K	$ 50K
*Assumes that electricity prices increase by 30% and gas by 20%)		

Table A16. Capital cost estimate of NGEDAC alternatives.

	All Gas System 1400 cfm NGEDAC/ 800 cfm NGEDAC	Hybrid Gas System 1400 cfm NGEDAC
Average Flow	2200 cfm	1400 cfm
Required Horsepower	352/157	352
Fuel Consumption	7440/7859	7859 Btu/HP-Hr
Engine and Catalytic Converter Package	$225,000	$135,000
Building/Soundproofing	$35,000	$25,000
Cooling Tower	($35,000 potentially funded by Decentralization Project)	($25,000 potentially funded by Decentralization Project)
Heat Recovery System	($16,000 potentially funded by Decentralization Project)	($12,000 potentially funded by Decentralization Project)
Installation and Freight	$30,000	$20,000
Total Capital Costs	$290,000	$180,000

Appendix B: Compressed Air System Survey at Combat Equipment Group—Afloat

Background

The AMC Combat Equipment Group—Afloat at Goose Creek, SC overhauls and maintains the onboard U.S. Army equipment, when a Navy ship comes into port for a 90-day overhaul. This requires a lot of small tool and medium air uses spread out over a long period of time and over a large area. The working area consists of many buildings served by large fiberglass underground lines from three separate compressed air supplies:

1. 100-hp Ingersoll Rand rotary screw EP100—446 cfm, 125 psig, 110 BHP air-cooled with aftercooler
2. 75-hp Ingersoll Rand rotary screw EP75—320 cfm, 125 psig, 82.5 BHP
3. 40-hp Ingersoll Rand rotary screw EP40SE—157 cfm, 125 psig, 47.2 BHP.

The underground lines form a very effective storage for the entire system, approaching the equivalent of 10,000 gal:

System size = 157 cfm takes 60 minutes to raise to 100 psig.

Using the equation:

$$T = V(P_2 - V_1)/\text{flow cfm} \times 14.4 \text{ psi} \qquad \text{Eq 1}$$

where:
T = time in minutes

60 = V(100)/157 cfm x 14.4 psi = v(100)/2,260.8
Volume = 1,356.48 cu ft or over 10,000 gal

The facility has good control over its system leak level. The maintenance personnel have already implemented a far-reaching leak identification and repair program, not only on the demand side, but also to all the underground lines. As leaks were identified underground (in rain storms), the pipe was exposed and the pipe repaired or replaced. Since the pipe is fiberglass and not "black iron," the leaks are usually caused by ground movement, and not by overall deterioration.

This program has led to a very efficient compressed air system that generally runs all the time on a part loaded 40-hp compressor.

There is no apparent need for compressed air dryers for this work. Since the underground storage is also at the underground temperature, the air is cooled to about 50 °F or less. The lines are sloped and have automatic condensate drains to remove condensed water to control pits. At the compressor supply areas, all the receivers, aftercoolers, and risers also have mechanical, level-actuated automatic condensate drains. The electric rate charged the Arsenal is $0.075/kWh; the natural gas rate, $8.00/10^6 Btu.

Overall Performance Profile of Electric Units Versus NGEDAC

Tables B1 and B2 list the key characteristics describing the performance and economics of the current compressed air system and the potential NGEDAC unit. The tables were developed based on the data collected during the site visit and with discussions with plant personnel. The estimates are conservative and reflect observed performance of each compressor compared to load cycle.

The estimated annual operating cost for the electric and natural gas engine are about equal. The maintenance cost of an engine drive is on the order of $7,000 higher annually than the electric motor.

Performance Profile for Reconfigured System

One option the base may want to consider in reconfiguring its existing system is to install one or two 20-hp units and operate one as base load to supply the base's 80 air requirements. Tables B3 and B4 list the system's operating characteristics.

There is an approximate $1,900 annual savings in electric power by running a 20-hp rotary screw in lieu of the current 40-hp at an average demand of 80 cfm. One or two new 20-hp units will involve a turnkey cost of about $10,000-$16,000. The resulting payback probably does not meet the base's normal payback requirements for this type of project.

Table B1. Operating profile for current system and NGEDAC.

	Current Electric Unit	Potential Natural Gas Unit
Measure	All Shifts	All Shifts
Average System Production	80 cfm	80 cfm
Average Power Requirement	22.48 kW	27.5 BHP
Hours	8,760 hrs	8,760 hrs
Specific Power	3.56 cfm/kW	BSFC 7,500
Unit Electric (or NG) Cost for Air	$184.62 /cfm/yr	$180.66 /cfm/yr
Total Electric (or NG) Cost for Air	$14,769 /yr	$14,454 /yr

Table B2. Compressor use profile for electric unit (“A”) and with NGEDAC (“B”).

	Manufacture	% of Load	% of Power	Full Load as a % of Power	Net Power	Actual cfm
A	IR EP40SE	51%	65%	34.59 kW x 0.65	22.48 kW	80
B	Natural Gas 160 cfm/ 50 hp*	51%	55%	50 hp x 0.55	27.5 BHP	80

Table B3. Operating profile for reconfigured system—running two 20-hp Units.

Measure	All Shifts
Average System Flow cfm	80 cfm
kW Power	19.5 kW
Operating Hours	8,760 hrs
Specific Power*	4.10 cfm/kW
Unit Electric Cost for Air ($/cfm)	$160.14 /cfm/yr
Unit Electric Cost for Air ($/psig)	$64.06/yr
Total Annual Electric Cost for Air	$12,811/yr
* Blended Power Rate = $0.075 per kWh.	

Table B4. Compressor use profile for reconfigured system—running two 20-hp units.

	Manufacture	% of Load	% of Power	FL kW x% of Power	Net kW	Actual cfm
1	Ingersoll Rand	100%	100%	19.5 kW x 1	19.5	80

The flow meter readings are provided by the total of six different Fox-heated wire anemometer thermal mass flow meter readings installed with the Johnson Control central air management system. These read every 15 minutes and read the instantaneous sensed “rate of flow.” They do not average to obtain cfm.

For the future, the Arsenal might consider modifying the software to read every “5-minutes averaged” or something similar to get a more representative load profile.

With this type of reading when the compressor first loads in, it may give a short "high rate of flow" generated at the supply side, not necessarily reflective of demand.

Summary

The existing 40-hp compressor is relatively power efficient. The unloading controls work well with the extremely large volume of effective storage. No recommend any change to the air supply is recommended, but the 40-hp compressor should be the primary base load unit.

The unit has recently had trouble "throwing belts." The base might check with the IR service provider to see if there has been a modification or change to that drive system that may help alleviate this condition. Changing these belts should normally be no more than annual or an every other year event.

Average electric rates at the plant are $0.075/kWh; natural gas rate is $8.00/MBtu. The actual plant electric cost for air production, as running today, is probably in excess of $14,000/yr (Table B5). The natural gas alternative offers no significant operating cost reduction at this time, given the prevailing energy rates charged to the base.

The system pressure appears to run from 95 to 105 psig in the headers during production.

Table B5. Actual plant electric cost for air production.

	Electric	Natural Gas
Today's annual cost/(flow)	$184.62/cfm/yr	$180.66/cfm/yr
Today's annual operating cost	$14,769/yr	$14,454/yr

This report identifies the "electric cost per year per loaded cfm" of air used. Electric cost was selected as the key project evaluation factor, since it is a good overall indication of system costs and savings associated with potential measures. It is an absolute number and not a subjective opinion, i.e., if the compressed air is used, these dollars are spent. All paybacks are estimated using the "Full Load Operating Efficiencies," which are very conservative.

If the compressed air is not used, the compressor either shuts off or unloads. If it shuts off, there is a 100 percent saving of the electric cost. If it unloads, there is a 25 to 90 percent savings of the electric cost.

Current Supply-Side and Demand-Side System Review

Primary Air Compressor Supply

All of the existing Ingersoll Rand compressors are of current state-of-the-art technology. Although the packaging may change the basic compressor design remains similar. They are relatively power efficient and there is nothing in these smaller size classes that has significantly better performance to serve this apparent demand.

The primary compressed air supply is produced by relatively efficient air compressors that are capable of delivering the 100 psig full load pressure in a continuous manner. The units are well applied. They appear to be in good operating order and well maintained. Key characteristics of the units are summarized in Table B6.

Table B6. Rating at full, average pressure, 100 psig.

Type	SS Rotary Screw	SS Rotary Screw	SS Rotary Screw
Brand	Ingersoll Rand	Ingersoll Rand	Ingersoll Rand
Model	EP40SE	EP175	EP100
ACFM	160 cfm	322 cfm	448 cfm
FL Press	100 psig	100 psig	100 psig
kW @ 100 psig	34.59 kW	60.54 kW	80.15 kW
Cfm/kW/100 psig	4.63 cfm/kW	5.32 cfm/kW	5.59 cfm/kW
Annual Elec Cost $/cfm	$141.90 cfm/yr	$123.49 cfm/yr	$118.17 cfm/yr
Annual Elec Cost $/psig	$113.52 psig/yr	$198.82 psig/yr	$264.69 psig/yr

Compressor Capacity Controls

The two most effective ways to run air compressors are at “Full Load” and “Off.” Capacity controls are methods of restricting the output cfm delivered to the system while the unit is still running. This is always a compromise and is never as efficient as full load on a specific power (cfm/hp) basis.

Rotary Screw Controls

The two most common controls used are modulation and online/offline. Modulation is relatively efficient at very high loads—and inefficient at lower loads. Online/offline controls are very efficient for loads below 60 percent, when properly applied with adequate time for blow down. Several other control types (e.g., “rotor length adjustment” or “variable displacement” and “variable speed drive”) have very efficient turn down from 100 percent load to about 60 percent load.

Two-stage, oil-free, rotary screws generally are not applied with modulation, and therefore, use only two-step (full-load/no-load) unloading controls. These controls must be installed correctly to operate efficiently. Piping and storage should be available close to the unit with no measurable pressure loss at full load to allow the signal to closely match the air requirements.

The current system has IR Intellysis electronic microprocessor-based online/offline with upper range modulation and automatic control selection. This capacity control will automatically select the most power efficient operating mode to meet the sensed conditions. It requires proper effective storage to operate properly. The base's system has more than enough effective storage and the controls operate very well.

CEGA has an operating Johnson Control central energy management system which seems very well applied and operating effectively. CEGA has continued to fine tune it, and combined with high-quality compressors, effective storage, continued leak repair will keep the CEGA's system very energy efficient for many year to come.

Dryers

As described earlier, the 10,000 gal of underground storage effectively gives the base a stable 50 °F pressure dew point (approximately) even during the hottest, most humid months.

It is important that staff keep the condensate drains open and flowing to remove this water in a timely fashion; otherwise, it will evaporate back into the dry compressed air and recontaminate the air.

All the aftercoolers are air cooled and setting up any other kind of central drying system will be expensive and perhaps not as effective as what the base currently has. No changes in this area are recommended.

If there are particular parts of use in the production areas that experience trouble and the drains are working, then it recommended that the site:

1. Determine the flow to the process
2. Determine the required pressure dew point
3. Apply correct "point of use" dryer.

Auto Condensate Drains

Automatic drain traps come in three categories: Level-operated mechanically activated, dual timer electronic, and level-operated electronic.

Level-Operated Mechanically Activated Drains

Level operated mechanically activate do not waste air, but are prone to clogging and require continuing maintenance to assure operation. These work best in a "Power House Situation" where continuing regular attention is part of the system. Drain prices range from $65.00 each to $250.00 each.

Dual Timer Electronic

Dual timer electronic drains use an electronic timer to control the number of times per hour it opens and the duration of the opening. The theory is that you adjust the times to be sure to fully drain the condensate and minimize the open time without water which wastes compressed air. The reality is that the cycles either do not get reset from the original factory settings (which causes condensate build-up in the summer) or they get set wide open and not closed down later in cooler weather thus wasting more air. When they fail "stuck open," they blow at a full flow rate of about 100 cfm.

Consider that the usual "factory setting" is 10 minutes with a 20-second duration. 1500 scfm of compressed air will generate about 63 gal/day in average weather or 2.63 gal/hr. Each 10-minute cycle will have 0.44 gal to discharge. This will blow through a ¼-in. valve at 100 psig in approximately 1.37 seconds. Compressed air will then blow for 18.63 seconds each cycle, 6 cycles a minute will equal 111.78 seconds/hr of flow or 1.86 minutes/hr of flow. A 1/8-in. valve will pass about 100 cfm. The total flow will be 100 x 1.86 = 186 cu ft in 1 hour or 186 ÷ 60 minutes = 3.1 cu ft/min average.

Depending on the type of discharge valve (whether it is solenoid-operated or motorized ball valve-operated and whether its type of timer is dual with test button or remote alarm), these valve prices range from $89 to 425 each.

Level-Operated Electronic Drains/Pneumatic Drains

Level operated/electronic drains come in a number of varieties, including ones which receive the signal to open from the condensate high level and the signal to close from the condensate low level. These waste no air and from a power cost standpoint, are the best selection and their reliability is usually many times greater than the level operated mechanical. Prices on these range from $250 to $850 for standard products (more for specials).

CEGA is currently using mechanical, level-actuated automatic condensate valves and the few that were checked (6-8) were working well.

There are other types that may require a little less maintenance. As stated earlier, the continued operation of these valves is very important to maintaining acceptable air quality at the using point.

Compressed Air Condensate Handling

In reviewing the condensate handling system, the survey team was informed by plant personnel that the condensate is collected and then goes to water treatment. If this is true, and discharge condensate meets the requirements of the water treatment facility plant, there is no problem.

However, if the CEGA is discharging the condensate to a storm sewer or in some other manner to ground water (Federal EPA minimum is 10 ppm) or are required to separate it by your local water treatment facility, this should be discussed in detail with appropriate personnel on site.

Appendix C: Compressed Air System Survey at Lone Star Army Ammunition Plant

Review of Current Supply Side System

There are five primary air compressor sites: Areas B, F, G, P, and Q. The unit in Area F is currently not in production. There is also a small compressor in the machine shop. Use of production capacity at Lone Star is currently 10-20 percent.

The compressors currently in operation at the main sites are the same brand and size. They were all built in the early 1970s, but have been recently serviced along with a general updating of the entire drying system. Each of the compressor units is equipped with a 300 horsepower synchronous motor, five-step control, water-cooled after cooler and twin tower desiccant compressed air dryers. Each compressor is supported by a 1,660-gal receiver. Key characteristics of the basic unit are summarized in Table C1.

In these compressors, the airflow moves from the compressor through the aftercooler to the receiver. The route is then from the receiver to the pre-filter, twin tower desiccant dryer and the after cooler and thus out to the system. This is particularly good because the dryers should only see dry after the aftercooler, separator and receiver.

Table C1. Performance and cost profile of current compressor technology.

Type	Area G
Brand	Ingersoll Rand
Model	XLE 20½ & 12½ x 8
ACFM	1628
Full Load Pressure	100 psig
kW @ 100 psig	232
Cfm/kW @ 100 psig	7.01
Annual Elec Cost @ 8760 hours	$91,454
Annual Elec Cost @ 2080 hours	$21,715

The primary compressed air supply is produced by relatively efficient air compressors that are capable of delivering the 100 psig full load pressure in a continuous manner. The units are well applied. They appear to be in good operating order and well maintained. The aftercooler in Building P75 is most likely fouled.

Summarized in Tables C2 to C9 list the key characteristics describing the performance and economics of the current compressed air system for each the four main compressor areas that are in production: Areas B, G, P, and Q. All estimates are based on a blended electric rate of $0.045/kWh.

Table C2. Key air system characteristics—current system/Area B.

Measure	Production (10 hrs @ 4 days)	Non-Production	Total
Average System Flow (cfm)	1154 cfm	300 cfm	NA
Average Compressor Discharge Pressure (psig)	90 psig	100 psig	NA
Input Electric Demand (kW)	162.8 kW	59.9 kW	NA
Operating Hours of Air System (hrs)	2080 hrs	3488 hrs	5568 hrs
Specific Power	7.1 cfm/kW	5.0 cfm/kW	NA
Electric Cost for Air—per unit of flow ($/cfm/year)	$13.18 /cfm/yr)	$31.34 /cfm/yr	NA
Electric Cost for Air—per unit of pressure ($/psig/yr)	$76.19 /psig/yr	$47.00 /psig/yr	NA
Annual Electric Cost for Air ($/yr)	$15,238	$9,402	$24,640

Table C3. Compressor use profile—current system/Area B.

Unit #	Compressor—Manufacturer and Model	Percent of Load	Percent of Power	Full Load kW x Percent of Power	Net Demand (kW)	Actual Flow (cfm)
Production: Operating at 1154 cfm and 90 psig						
1	IR XLE 300 hp	71	74	220 x 0.74	162.8	1154
Non-Production: Operating at 300 cfm and 100 psig						
1	IR XLE 300 hp	18	25.8	232 x 0.258	59.9	300
The estimated electric cost to operate the air system at Area B is on the order of $24,600 annually. The estimate is based on loading/unloading and pressure measurements taken in the compressor room and reflect discharge pressure of 90 psig and an air flow of 1154 cfm during the 2080 hours of production each year.						

Table C4. Key air system characteristics—current system/Area G.

Measure	Production (10 hrs @ 4 days)	Non-Production	Total
Average System Flow (cfm)	749 cfm	300 cfm	NA
Average Compressor Discharge Pressure (psig)	90 psig	100 psig	NA
Input Electric Demand (kW)	110 kW	59.9 kW	NA
Operating Hours of Air System (hrs)	2080 hrs	3488 hrs	5568 hrs
Specific Power	6.8 cfm/kW	5.0 cfm/kW	NA
Electric Cost for Air—per unit of flow ($/cfm/year)	$51.48 /cfm/yr	$31.34 /cfm/yr	NA
Electric Cost for Air—per unit of pressure ($/psig/yr)	$13.74 /psig/yr	$47.00 /psig/yr	NA
Annual Electric Cost for Air ($/yr)	$10,296	$9,402	$19,698

Table C5. Compressor use profile—current system/Area G.

Unit #	Compressor Manufacturer and Model	Percent of Load	Percent of Power	Full Load kW x Percent of Power	Net Demand (kW)	Actual Flow (cfm)
Production: Operating at 749 cfm and 90 psig						
1	IR XLE 300 hp	46	50	220 x 0.50	110	749
Non-Production: Operating at 300 cfm and 100 psig						
1	IR XLE 300 hp	18	25.8	232 x 0.258	59.9	300
The estimated electric cost to operate the air system at Area G is on the order of $19,700 annually. The estimate is based on loading/unloading and pressure measurements taken in the compressor room and reflect discharge pressure of 90 psig and an air flow of 749 cfm during the 2080 hours of production each year.						

Table C6. Key air system characteristics—current system/Area P.

Measure	Production (10 hrs @ 4 days)	Non-Production	Total
Average System Flow (cfm)	1220 cfm	300 cfm	NA
Average Compressor Discharge Pressure (psig)	84 psig	100 psig	psig
Input Electric Demand (kW)	164 kW	59.9 kW	NA
Operating Hours of Air System (hrs)	2080 hrs	3488 hrs	5568 hrs
Specific Power	7.43 cfm/kW	50 cfm/kW	NA
Electric Cost for Air—per unit of flow ($/cfm/year)	$12.58 /cfm/yr	$31.34 /cfm/yr	NA
Electric Cost for Air—per unit of pressure ($/psig/yr)	$76.75 /psig /yr	$47.00 /psig/yr	NA
Annual Electric Cost for Air ($/yr)	$15,350	$9,402	$24,752

Table C7. Compressor use profile—current system/Area P.

Unit #	Compressor—Manufacturer and Model	Percent of Load	Percent of Power	Full Load kW x Percent of Power	Net Demand (kW)	Actual Flow (cfm)
Production: Operating at 1220 cfm and 84 psig						
1	IR XLE 300 hp	74.9	77	213 x 0.77	164	1220
Non-Production: Operating at 300 cfm and 100 psig						
1	IR XLE 300 hp	18	25.8	232 x 0.258	59.9	300
The estimated electric cost to operate the air system at Area P is on the order of $24,800 annually. The estimate is based on loading/unloading and pressure measurements taken in the compressor room and reflect discharge pressure of 84 psig and an air flow of 1220 cfm during the 2080 hours of production each year.						

Table C8. Key air system characteristics—current system/Area Q.

Measure	Production (10 hrs @ 4 days)	Non-Production	Total
Average System Flow (cfm)	1220 cfm	300 cfm	NA
Average Compressor Discharge Pressure (psig)	84 psig	100 psig	psig
Input Electric Demand (kW)	164 kW	59.9 kW	NA
Operating Hours of Air System (hrs)	2080 hrs	3488 hrs	5568 hrs
Specific Power	7.43 cfm/kW	5.0 cfm/kW	NA
Electric Cost for Air—per unit of flow ($/cfm/year)	$12.58 /cfm/yr	$51.34 /cfm/yr	$
Electric Cost for Air—per unit of pressure ($/psig/yr)	$76.75 /psig/yr	$47.00 /psig/yr	$
Annual Electric Cost for Air ($/yr)	$15,350	$9,402	$24,752

Table C9. Compressor use profile—current system/Area Q.

Unit #	Compressor—Manufacturer and Model	Percent of Load	Percent of Power	Full Load kW x Percent of Power	Net Demand (kW)	Actual Flow (cfm)
Production: Operating at 1220 cfm and 84 psig						
1	IR XLE 300 hp	74.9	77	213 x 0.77	164	1220
Non-Production: Operating at 300 cfm and 100 psig						
1	IR XLE 300 hp	18	25.8	232 x 0.258	59.9	300
The estimated electric cost to operate the air system at Area Q is on the order of $24,800 annually. The estimate is based on loading/unloading and pressure measurements taken in the compressor room and reflect discharge pressure of 84 psig and an air flow of 1220 cfm during the 2080 hours of production each year.						

All the areas except Area G are generally operating at ¾ load, 1200 cfm during production hours, and $25,000 in annual electric bills. Area G is operating at ½ load, 750 cfm, and $20,000. Total electric bill to operate the four areas is approximately $100,000. Only Area G has the potential of an available gas source and heat recovery for the NGEDAC unit.

Compressor Capacity Controls

The two most effective ways to run air compressors are at “Full Load” and “Off.”

Capacity controls are methods of restricting the output cfm delivered to the system while the unit is still running. This is always a compromise and is never as efficient as full load on a specific power (cfm/hp) basis.

Reciprocating compressors are double-acting, water-cooled units with multi-step unloading. This is an efficient compressed air unloading system, Reciprocating multi-step unloading will efficiently translate percentage of “less air used” into almost the same proportional reduction in energy cost.

The current system has five step, clearance control unloading devices. This is clearly the most efficient unloading system available for reciprocating compressors available today. Review of the part of the part load horsepower (kW) consumption at part load conditions is extremely attractive will verify this statement. During the audit, the amps (translate to kW) were recorded over a 20-minute period at each site. The demand load at each of these sites ranged from ¼ through ½ to ¾ load for each compressor. The reduction in kW matches very closely the reduction in compressed air flow. This is excellent efficiency condition, given the substantially reduced demand loads for each of the systems over time.

Air Treatment and Air Quality

Dryers

Current Drying Operation

An overview of the system’s current drying system is shown in Table C10.

Desiccant Dryer Regeneration equipment removes moisture vapor by “adsorbing” it to desiccant beads. These dryers can consistently deliver a pressure dew point to –40 °F or lower, which removes much more water than conventional refrigeration units. To regenerate the wet tower while the other tower is drying requires the use of heat in some form and some dry air to “sweep” or “purge” the exchanged moisture out. Desiccant dryers are usually rated at the same 100 °F inlet and 100 psig conditions.

Table C10. Comparison of current dryers.

Type	DSAC Tower Regenerative Desiccant		
Brand	Deltech	ZEKS	ZEKS
Model	PS1441CFHMR	Hydronix1910 HPS	1630HPS
Rating in scfm @100°F; 100 psig	1441	1900	1630
SCFM Purge	216 Max	285 Max	244.5 Max

The primary dryers are twin tower, regenerative, desiccant dryers capable of delivering a consistent –40 °F pressure dew point when:

- Air is delivered to the dryer at no more than 100 °F in all examined locations except P-75, which as at 108 °F
- The condensate driven out of the aftercooler, prefilter, dryer and afterfilter is immediately removed from the system and not allowed to retrain or build up.
- Regeneration is accomplished by using dried compressed air purge and the dryer is equipped with appropriate purge controls.

Aftercoolers

Aftercoolers are water cooled and appear capable of delivering 100 °F or lower temperature compressed air to the dryer. There is only a 4 to 5 °F heat gain across the aftercooler resulting in high entry temperatures to the desiccant dryer. At each of the other sites, the temperature gain across the aftercooler ranged from 15 to 20 °F range. Please bear in mind that each 20 °F rise in the inlet temperature to the dryer reduces the dryer capacity by 50 percent. Temperatures in the 110 to 108 °F were recorded at the site on 13 March 2001 about midday.

☑ ***Recommended Project—Reconfigure or modify aftercooler to correct performance—check aftercooler in P-75 for scale or mechanical problem.***

Water or Oil Carryover in System

Water (condensate) and oil carryover problems in the current air system are not significant. The dryers and filters appear to be in good to new condition except for P-75.

The correct way to eliminate water and oil in the air system is to clean and dry the air immediately after it is produced in the compressor room. Then clean dry air can be stored in a separate air receiver and flow it to the system, as required.

Pre-Filters And After-Filters

Pre- and after-filters are generally either particulate or coalescing type and their use depends on the type of dryer in use and various installation considerations.

Desiccant dryers always require a high-quality coalescing prefilter to keep liquid oil and water out of the drying tower. They also always require an effective particulate filter after the dryer to keep "desiccant dust" from migrating into the system.

Refrigerated dryers may or may not need pre and after filters depending on the piping, type of compressor, and desired degree of cleanliness. If the inlet air is apt to be dirty and fouled with carbon scale, etc., a particulate prefilter is called for. If it is liable to have significant liquid or heavy oil mist, a coalescing (or combination coalescing particulate) pre filter may be needed. If oil/water mist is leaving the dryer, a coalescing after filter may be in order.

Care in selection must be taken in all cases because:

- wasted air pressure costs energy dollars
- wasted air pressure neutralizes the operating pressure band early
- standard coalescers will usually not perform effectively at flows much below 20 percent of their rated capacity
- standard coalescers life will be significantly shortened by particulate load
- loose-packed, deep-bed mist eliminators (those with correct elements) will coalesce effectively throughout the total scfm range
- loose-packed, deep-bed mist eliminators (those with correct element) have very high particulate load capability.

The pre-filter(s) for the system are ZEKS and Deltech and are correctly applied. They are sized to handle the proper scfm and have an estimated performance of 1 to 2 psi average pressure loss.

The after-filter(s) are ZEKS and Deltech also, and are correctly applied. They are sized to handle the proper scfm and have an estimated performance of 1 to 2 psi average pressure loss.

Automatic Condensate Drains

Background

Automatic drain traps come in three categories: Level-operated mechanically activated, dual timer electronic, and level-operated electronic.

Level-Operated Mechanically Activated Drains

Level operated mechanically activate do not waste air, but are prone to clogging and require continuing maintenance to assure operation. These work best in a "Power House Situation" where continuing regular attention is part of the system. Drain prices range from $65.00 each to $250.00 each.

Dual Timer Electronic Drains

Dual timer electronic drains use an electronic timer to control the number of times per hour it opens and the duration of the opening. The theory is that you adjust the times to be sure to fully drain the condensate and minimize the open time without water which wastes compressed air. The reality is that the cycles either do not get reset from the original factory settings (which causes condensate build-up in the summer) or they get set wide open and not closed down later in cooler weather thus wasting more air. When they fail "stuck open," they blow at a full flow rate of about 100 cfm.

Consider that the usual "factory setting" is 10 minutes with a 20-second duration. 1500 scfm of compressed air will generate about 63 gal/day in average weather or 2.63 gal/hr. Each 10-minute cycle will have 0.44 gal to discharge. This will blow through a ¼-in/ valve at 100 psig in approximately 1.37 seconds. Compressed air will then blow for 18.63 seconds each cycle, 6 cycles/minute will equal 111.78 seconds/hr of flow or 1.86 minutes/hr of flow. A 1/8-in. valve will pass about 100 cfm. The total flow will be:

100 x 1.86 = 186 cu ft in 1 hour or 186 ÷ 60 minutes = 3.1 cu ft/min average.

Depending on the type of discharge valve (whether it is solenoid-operated or motorized ball valve-operated and whether its type of timer is dual with test button or remote alarm), these valve prices range from $89 to $425 each.

Level-Operated Electronic Drains/Pneumatic Drains

Level operated/electronic drains come in a number of varieties, including ones which receive the signal to open from the condensate high level and the signal to close from the condensate low level. These waste no air and from a power cost standpoint, are the best selection and their reliability is usually many times greater than the level operated mechanical. Prices on these range from $250 to $850 for standard products (more for specials).

Current Application

The configuration and performance of condensate drains in the plant's compressor building do need to be modified.

☑ *Recommended Project—Replace all timer drains with level activated drains.*

Be sure auto drains are set up to work effectively. Some examples are:

- drains should not be tied together to a common header
- be sure all drains can be checked easily for operation
- be sure all drains are properly "vented"
- level-actuated, see-through drains should be installed at all compressor locations.

Connect each drain's point (after-cooler, pre-filter, dryer, after-filter, receivers, and all risers) separately to individual level-activated electric or pneumatic drains to collect and direct the condensate to a proper handling point carry it in a large plastic vented line (4 or 6 in.). Be sure maintenance personnel can effectively and visually monitor the drain's action. Table C11 lists the costs and savings of the project.

Table C11. Costs and savings of drain replacement.

Parameter	Cost/Savings
CFM savings per drain	3.1 cfm/yr
Value of savings per drain	$50/cfm/yr
Estimated energy savings	$155/yr each
Total of number of drains	Four per compressor area (16 units)
Total annual savings	$2480/yr
Cost per drain (installed)	$300 each
Cost of 16 drains	$4800
Estimated project payback	2 years

NGEDAC Site Assessment

This section provides a preliminary assessment of the opportunity for gas engine driven compressors at Lone Star Army Ammunition Plant. The assessment is based on four key design factors:

- operating the new system as a fully hybrid system with the existing electric system
- meeting or exceeding environmental and safety requirements of the area

- improving the current demand system so that the air requirements for the new system are minimized, while reducing system operating costs for the plant
- including an effective heat recovery application for NGEDAC.

NGEDAC System Design Factors

The conceptual design for the gas engine-driven system is based on serving the load in Area G. None of the other areas had access to gas within a reasonable distance. The NGEDAC alternative is a 780 cfm class unit that could serve the entire air requirements of Area G based on current production levels. Operating and capital cost estimates are based on the Ingersoll-Rand PCD200-NG Platform with a Caterpillar G3306TA (780L) engine.

The NGEDAC system would be configured as a combined system in conjunction with the existing electric system. In this way, the existing electric system can serve as a back-up or supplement to the gas engine system, if the gas system has a planned or unplanned shutdown or if the air requirements of the base are suddenly increased.

Using this approach, the Department of Defense can gain experience with not only operating a gas engine driven system, but also with integrating it with electric systems to improve overall compressed air system reliability and flexibility. This flexibility is especially important given the increasing uncertainty associated with the price and supply reliability of most energy sources.

Environmental issues are expected to be minimal in this application given the key areas to address in any major project on this nature, although less so in this part of the country. The assessment is based generating up to 2.6 gm/bhp/hr for NOx and 1.75 gm/bhp/hr for CO.

Also important in this particular application is the opportunity to incorporate a heat recovery system to improve the overall efficiency and cost-effectiveness of the gas system.

Operating Cost Comparison

Table C12 displays the energy costs for the current XLE and proposed NGEDAC units.

The comparison is based on input energy costs of $5/Million Btu for gas and $45/MWh for electricity. Relative to the electric system, annual maintenance costs are $4,500 higher for the NGEDAC units. Maintenance costs for the NGEDAC units are based on 2.2 cents/hp-hr.

It costs about 20 percent more to operate the proposed NGEDAC unit at Lone Star than existing electric system.

Capital Cost Assessment

Table C13 displays the capital cost estimates for the NGEDAC unit (780 cfm class). The capital costs include the catalytic converter for the gas engine. The costs also include an estimate for all installation and freight costs and a budget estimate to erect an air line to link the NGEDAC unit in Area G with the production facilities in Area G.

Even with the heat recovery benefit, the estimated cost to operate the NGEDAC unit is 20 percent higher than the current electric unit—$12,600 versus $10,300. Estimated costs for a turnkey installation of a 780 cfm class NGEDAC unit at Lone Star total $170,000. The annual operating cost comparison summarized in Table C13, is based on an electric rate of $0.045/KWh and a gas rate of $5/million Btu. The incremental maintenance costs of $4,500 associated with the NGEDAC unit are negated by the credit of $4,400 given to the NGEDAC unit from the heat recovery application.

Table C12. Annual energy cost comparison of current system and proposed NGEDAC unit.

Parameter	Current Electric Unit	Proposed NGEDAC Unit
Average Air Flow	750 cfm (46% loaded) 1628 cfm (100% loaded)	750 cfm (96% loaded)
Production Hours	2080	2080
Discharge Pressure	90 psig	90 psig
Input Energy	110 kW (6.8 cfm/kW)	7,859 btu/hp/hr (153 bhp)
Energy Costs	$10,296 (@ $0.045 per kWh)	$12,505 (@ $5 per Million Btu)
Heat Recovery Credit	NA	$4,396 (@ 35%)
Incremental Maintenance Costs	NA	$4,500
Net Energy and Maintenance Costs	$10,296	$12,609

Table C13. Capital cost estimate of NGEDAC alternative.

Parameter	NGEDAC SYSTEM (780 cfm class)
Average Flow/Pressure	750 cfm (96% loaded)
Required Horsepower	153
Fuel Consumption	7859 btu/hp/hour
Engine and Catalytic Converter Package	$117,000
Heat Recovery System	$10,000
Installation (incl pipe extension) and Freight	$43,000
Total Capital Costs	$170,000

Appendix D: Compressed Air System Survey at Picatinny Arsenal

Introduction

The compressed air system at Picatinny Arsenal encompasses an extensive geographical area. Today, there are almost 27 miles of compressed air piping, joining 15 to 16 areas of production buildings. With air usage levels significantly less than those required during the height of production at the Arsenal, there are numerous opportunities to improve energy efficiency in the system and to further reduce system operating costs by implementing a gas engine driven compressed air system.

The main compressed air system is fed by a compressor plant in the main Power House Building 506. Average flow for the main system is 925 acfm at 80 psig with the system operating 8760 hours/yr. The air delivered from the Power House is dried only with a water-cooled aftercooler. When the site visit took place on a 79 °F ambient day, the compressed air system was delivering 80 °F saturated air at 80 psig to the system.

There are three other independent compressed air systems:

1. *Wind Tunnel*. A special application for projectile testing at supersonic, transonic, and subsonic speeds. This system requires higher pressure (110-120 psig) and significant storage (16,000 cu ft) for proper operation. Average flow for the Wind Tunnel system is 2200 acfm. The system operates 500 hr/yr.
2. *Building 3150 (Machine Shop)*. This building houses a large machine shop and runs with its own air compressor supply. Average flow is 20 acfm with the system operating 8760 hours/yr. There is no feeder line from the main air system to this building.
3. *Building 3028*. This building also has no feed from the main air supply and currently has its own air system. Average flow is 40 cfm with the system operating 8760 hours/yr.

There are also several dedicated systems in the Main Power House: one for engine starting of the 12-cylinder Caterpillar natural gas engine generator/fuel cell power system and one for the instrument air and HVAC control systems.

There are other dedicated control and fire suppression compressed air systems throughout the Arsenal. In general, these units are small horsepower duplex units with compressed air dryers. They do not normally run many hours a year and are not part of the main air system. These are not included in the system evaluation of this report.

In summary, Picatinny Arsenal has a large volume air system that is currently supplying a relatively small system demand. This “system downsizing” presents many opportunities for energy savings that are addressed in this report.

Previous System Improvements

The Arsenal personnel have already implemented several key programs that have successfully lowered the energy cost.

Power House (506)

Operating personnel have lowered the final discharge pressure to the system from 100 psig to 80 psig. This has reduced electrical demand by approximately 31.69 kW resulting in savings of $24,430/yr. Today the system still runs effectively at this reduced pressure level.

Building 3150

This system previously ran a 100 hp, 490 acfm Quincy Rotary Screw Compressor with apparent demand of 40 cfm or less, running 10 percent loaded continuously. This was a very inefficient mode of operation and resulted in excessive wear on the compressor. The system operated at 59.4 kW with an annual energy cost of $45,790.

Today the machine shop typically runs two small 4.7 hp tank-mounted units. Ten of these units (IMC) are located strategically around the building. The operating pairs are alternated as required. These units operate at 14.21 kW. Since they are commercial as opposed to industrial units, the motors are a low-efficiency, single-phase type. Today, the operating cost of $10,923/yr results in savings of $34,867/yr.

Building 3028

This building previously ran a 40 hp/35 kW 150 cfm Ingersoll Rand ESV nonlubricated, double-acting, water-cooled, single-stage compressor. At 40 cfm demand, this unit operated at 13.2 kW for a cost of $10,175/yr.

Today, air is supplied by a 25 hp Ingersoll Rand Model 3000, delivering 100 cfm at 100 psig at 27 bhp/22.8 kW. At 40 cfm average demand, this unit operates at approximately 8.8 kW over 8760 hours or $6791/yr operating cost, a net savings of approximately $3384/yr.

System Load Profile and Cost Analysis

Based on optimum performance of each compressor—compared to Load Cycle—and on discussions with plant personnel, load profiles and power usage assessments were developed for each of the main compressed air systems (Table D1).

Measured Flow

Flow and pressure were measured for 24 hours beginning the morning of 29 August 2000. The flow measurement was taken with a Sierra-heated wire anemometer (0-20,000 fpm ± 3 percent). Readings were taken (on average) every 11 seconds. The curve shown is with these readings averaging every 10 minutes (Figure D1).

Table D1. Load profiles and power usage assessments for main compressed air systems at Picatinny Arsenal.

	Main Power House	Wind Tunnel	Bldg 3028	Bldg 3150	Total
Average System Flow	900 acfm	2,200 acfm	40 acfm	20 acfm	NA
Average Prod kW	126.75 kW	356.28 kW	14.21 kW	8.8 kW	NA
Annual System Operating Hrs	8760 hrs	500 hrs	8760 hrs	8760 hrs	NA
Specific Power	7.1 cfm/kW	6.17 cfm/kW	2.81 cfm/kW	2.27 cfm/kW	NA
Energy Cost --$ cfm/yr	$108.57 cfm/yr	$7.13 cfm/yr	$274.33 cfm/yr	$339.95 cfm/yr	NA
Air Energy Cost – $ psig/yr	$488.54 psig/yr	$78.49 psig/yr	$54.77 psig/yr	$37.30 psig/yr	NA
Est Air Energy Cost – $/yr	$99,487 /yr	$15,689 /yr	$10,973 /yr	$6,791 /yr	$132,940/yr
Note: Blended Power Rate = 0.088 kWh; Power House Operating Pressure = 80 psig.					

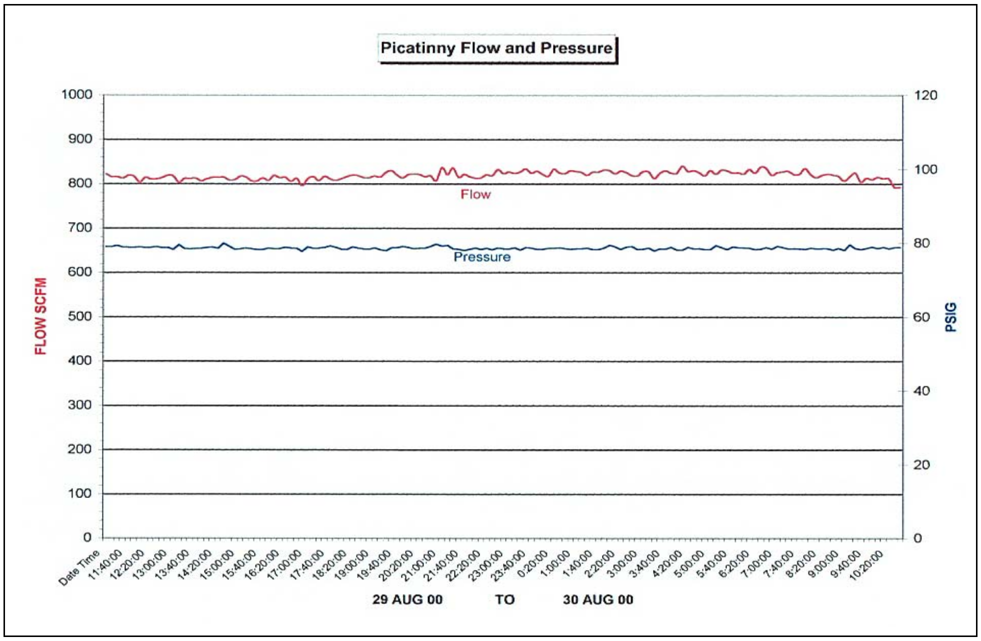

Figure D1. Picatinny CA flow and pressure.

The trended curve shows 800 scfm (900 acfm) in a continuous demand over the 24-hour period, both during production and nonproduction periods. Picatinny is essentially a one-shift operation. This indicates a significant number of leaks and/or process air "left on" during nonproduction hours. Both of these conditions represent an energy savings opportunity (refer to the Leak Management section). Arsenal personnel are in the process of determining which production activities were operating and which were not. This information will help determine the source and level of opportunity.

Pressure

The pressure was recorded at the same trending rates and at the same point as the flow. The pressure transducer was zeroed out against a calibrated Helcoid DP250 digital test gauge. The pressure held a steady 79 to 80 psig during the entire test.

The actual plant electrical power cost for the combination of the main system and satellite subsystems, as running today, is in excess of $130,000/yr. The load profile or demand of this system is almost like "process air" and is relatively stable during all shifts. The full load operating range is 365 days a year, 24 hours a day, 8760 hours a year (see flow meter readings).

There are no significant cost savings within the current air supply configuration for the main Power House, except for the potential for moving to gas engine drives. Moving the subsystems associated with Buildings 3028 and 3150 to the main system would save roughly $13,000 annually based on being to supply at an incremental cost of $80/cfm relative to the current cost averaging $300/cfm for the 60 cfm requirement.

Other Issues

The electrical power cost/hr/"loaded cfm" of air used was determined. Electrical power cost is used as a qualifying factor since it is "real bottom line dollars." This is an absolute number and not a subjective or opinion. All paybacks for savings projects are estimated using the "full load operating efficiencies," which are very conservative. If the compressed air is not used, the compressor either shuts off or unloads. If it shuts off, there is a 100 percent saving of the power cost. If it unloads, there is a 25 to 90 percent savings of the power cost.

It is important to note that all recoverable compressed air costs should also be considered, i.e., maintenance, water costs, depreciation, etc. Usually, the electrical power cost is between 50 and 75 percent of the total "variable compressed air costs." Associated maintenance and other costs will be, in all probability, at least 50 percent or more of the identified electrical power cost. Existing records may be available to estimate these more accurately.

Plant Compressed Air Survey

The primary objective of the survey was to review the basic system dynamics and identify the current basic load/power profile and then to project what it will be when optimized with this data. The objective is to size and recommend an appropriate natural gas engine-driven compressor to effectively carry the base load and optimize the natural gas engine savings over conventional electric driven units. This action is to evaluate this Arsenal's operating characteristics to reflect accurate and effective results with a natural gas engine driven air compressor demonstration unit.

Some specific selected steps were to:

- determine what follow-up plans and actions would be appropriate to lower the overall compressed air energy cost in the continuing short and long terms
- evaluate the potential energy cost savings in compressed air demand side conservation programs:
  - leak control/management

  - specific demand side requirements
- review appropriateness of all compressed air equipment to produce proper quality and quantity of usable compressed air power at the acceptable efficiency
- identify a relatively accurate load profile.
- identify your current electric power cost/cfm and per psig to calculate anticipated return.
- key concepts to consider if a Level II Audit is implemented:
- identify and target opportunities for compressed air savings on the demand side.
- outline plans for point of use pressure and quality management.
- evaluate characteristics and appropriateness of central compressed air control system.
- identify savings potential in use of air saving devices—nozzles and auto drains
- identify savings potential in replacement or re-evaluation of "misapplied air"—cabinet coolers, vacuums, pumps, and bearing cooling
- review total piping system and leaks. Develop action plan to remove as much pipe as possible, then repair leaks on what is left.

Above-Ground Leak Needing Immediate Repair

Note that during the site visit, a significant air leak was identified in an above-ground, rusted-out distribution line under enclosed walkway between Building 807 and Building 810. This caused oil accumulation in ground is a significant "safety issue" (possible blowout). The audit team pointed this out to Arsenal personnel on site and at the "wrap-up" meeting and recommend this leak and any others like it be "corrected immediately."

Current System Review

Power House Building

506 Main Compressor Room Supply

The basic air supply consists of running either compressor Unit 1 or 2 (Figure D2). Unit 1 is currently not operational. Both units are 18 ½-in. and 11 ½-in. x 8 ½-in. stroke, double-acting reciprocating Ingersoll Rand 200 bhp (1130 acfm at 100-110 psig) compressor with five-step unloading.

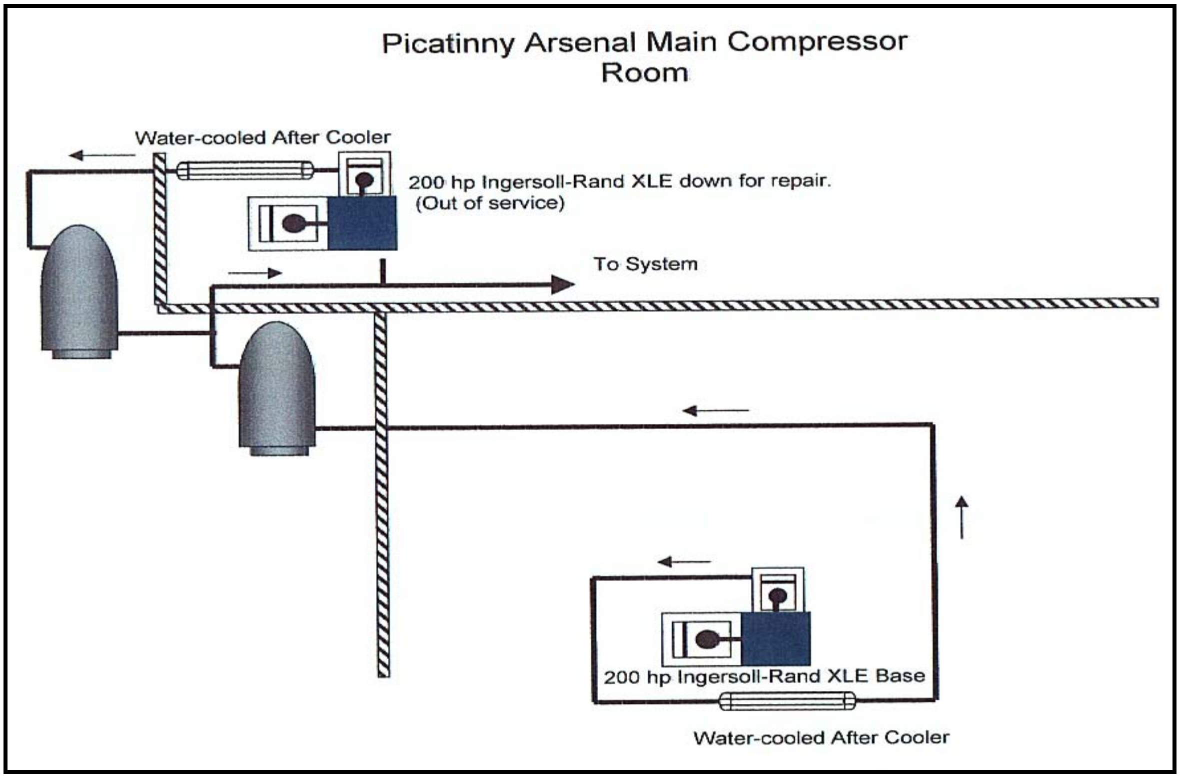

Figure D2. Main Compressor Room at PICA.

These units are the most power efficient units on the Arsenal and have a capacity control system, which effectively translates lower air demand with lower input energy.

The audit team observed Unit 2 running and except for a little too much oil from the oiler, it appeared to be in very good shape. These units are applied excellently and there are no more power efficient units available in this size class. They are still state-of-the-art systems.

The back-up air is supplied by:

- 75 hp Atlas Copco two-stage, water-cooled, single-acting dr two-compressor with water-cooled heads and jackets. This unit is smaller and is 10 percent less driver efficient than the XLE. Parts are usually hard to obtain for these since they are manufactured in Belgium. The unit should be run as little as possible.
- 40 hp Gardner Denver WXE air-cooled delivering 157 acfm at 41 bhp. It is also a two-stage, single-acting unit and relatively old. It is also 10 percent less efficient than the XLE and should be run as little as possible.

- 25 hp Champion R70-12 25 hp two-stage, single-action, tank-mounted (120 gal) reciprocating compressor delivering 91 cfm at 27 bhp. This unit is 15 percent less efficient than the XLE and should be run as little as possible.

These three back-up units will certainly not be required in the future if the NGEDAC unit is installed and the system is optimized, unless a significant low load condition occurs. Consideration should be given to using these units at selected places within the production areas, if required. Tables D2 and D3 list the efficiency ratings of the primary compressors.

Compressor Capacity Controls

The most effective way to run an air compressors is either to let it run at full load, or to turn it off. Capacity controls are methods of restricting the output air volume delivered to the system, while the unit is still running—in other words, by running the compressor at less than full load. This is always a compromise, and on a specific power (cfm/hp) basis, is never as efficient as full load.

Table D2. Efficiency rankings of primary compressors (100 psig).

	#2—Base	Back Up	Back Up	110 psig
Brand	I-R	GD	Champion	Sullair
Model	XLE	WXE1000	R70-12	32/25 400L
ACFM	1130	157	91	2200
FL Press	100	100	100	110
kW@100 psig	165.69 kW	35.60 kW	22.89 kW	356.78 kW
Cfm/kW/100 psig	6.82	4.4	3.98	6.17
ANN PWR CST/cfm	$113.03	$175.20	$193.68	$7.13
ANN PWR CST/psig	$638.64	$137.22	$88.23	$78.49

Table D3. Efficiency rankings of primary compressors (85 psig).

	#2—Base	Back Up	Back Up
Brand	I-R	GD	Champion
Model	XLE	WXE1000	R70-12
ACFM	1130	157	91
FL Press	85	85	85
kW@80 psig	144.49 kW	31.15 kW	20.03 kW
Cfm/kW/100 psig	7.82	5.04	4.54
ANN PWR CST/cfm	$98.57	$152.95	$169.80
ANN PWR CST/psig	$556.92	$120.06	$77.20

Reciprocating Controls

The main Power House base reciprocating compressor is a double-acting, water-cooled unit with five-step unloading. This is an efficient compressed air unloading system, reciprocating five-step unloading will efficiently translate percentage of "less air used" into almost a comparable reduction in energy cost.

Rotary Screw Controls

The two most common controls used are modulation and online/offline. Modulation is relatively efficient at very high loads—and very inefficient at lower loads. Online and offline is a very efficient commercial control available for loads below 60 percent when properly applied with adequate time for blow down. There are several other ("rotor length adjustment" or "variable displacement," and "variable speed drive") that have very efficient turn down from 100 percent load to about 60 percent load.

These controls must be installed properly to operate correctly and efficiently. The installation should have piping and storage available close to the unit with no measurable pressure loss at full load to allow the signal to closely match the air requirements. Also the systems at Picatinny have some modulation units (Sullair) and some online/offline (Atlas Copco). All appear to be installed properly and run correctly.

Recommendations—Short Term

All of the units involved have or are very close to having unloading controls capable of translating "less air used" into a comparable reduction in power cost. These controls will work effectively with your current piping and air receiver storage situation.

Recommendations—Long Term

With the system stabilized and balanced in the main Power House (506), consider a microprocessor-driven centralized full networking electronic control system. This will automatically place the most efficient machine online and assure no more than one partial loaded unit at a time.

Air Treatment and Air Quality

General Air Treatment Concepts

Eliminating Water/Oil in Air Systems

The correct way to eliminate water and oil in your air system is to clean and dry the air immediately after it is produced in the compressor room. Then store clean dry air in a separate air receiver and flow it to the system as required.

Addressing Water and Oil Carryover Problems in a Compressed Air System

The water (condensate) and oil carryover problems in an air system are real, and can be expected to increase in magnitude in the extreme weather. Some guidelines regarding water and oil carryover control in compressed air systems are:

1. Generally, it is best to eliminate the water and oil at the air source before it enters the air system.
2. Every 20 °F increase in temperature doubles the "moisture load" the compressed air will hold.
3. Compressed air dryers are usually capacity rated with 100 °F, 100 psig inlet air conditions. At 120 °F, 100 psig, the dryer's capacity rating is reduced by 50 percent.
4. Putting "dry or oil free" air into your system 90 percent of the time and then allowing wet/oily air in sporadically 10 percent of the time will, in reality, give you a "wet/oily" system all the time. The liquid water and/or oil will fall out in the piping system continuing to "re-entrain" and contaminate and/or collected in the "low spots" of the system, thus re-contaminating as it is pulled into the flowing compressed air system. A wet/oily system may well take many months of continued flow of clean dry air to "clean up."
5. Identify required pressure dew point.

Refrigerated Air Dryers

Refrigerated dryers require a refrigeration system to mechanically cool the air. The lowest possible consistent pressure dew point with a noncycling dryer is +40 °F. Cycling dryers not only save power (60 to 75 percent), but also can deliver a lower pressure dew point (down to +35 °F to + 38 °F). Picatinny has some refrigerated dryers throughout the system, most in the dedicated control/fire air systems.

Desiccant Dryers

Desiccant dryer regeneration types remove moisture vapor by "adsorbing" it to activated alumina desiccant beads. These dryers can consistently deliver a pressure dew point to –40 °F or lower, which removes much more water than conventional refrigeration units. To regenerate the wet tower while the other tower is drying, requires the use of heat in some form and some dry air to "sweep" or "purge" the exchanged moisture out. Desiccant dryers are usually rated at the same 100 °F inlet, 100 psig conditions.

Current Air Treatment System

The only dryers noted were in the dedicated systems:
- Wind tunnel—desiccant
- Engine starting—desiccant
- Instrument air/control air—desiccant and refrigeration.

All these dryers are sized to their specific application and must have their own air supply. These units normally run a very limited number of hours/yr, and therefore, offer few significant opportunities for energy recovery. Nothing observed would change this opinion. If in the future this changes, then that operation should be reviewed again.

The main Power House air is dried by water-cooled aftercoolers delivering 80 °F saturated on a 79 °F day, about as good as one might expect. The smaller units throughout the system have appropriate air- or water-cooled aftercoolers that appear to be satisfactory.

The Wind Tunnel use outside-mounted air-cooled after cooler to a dryer. This appears to work very well.

There are no compressed air line filters in the main Power House air (506).

Basic System Header/Piping and Interconnecting Piping Between the Primary Air Compressors and the Distribution System

Basic Header Piping

Headers were checked at appropriate points with a single test gauge and there was little or no pressure loss in the header systems. Consequently, it is believed that the header system today can deliver the required air to any area without any significant pressure loss. Any low-pressure problems encountered will, in all probability, be in the feeds from the header to the area. The header runs between building is long, extensive, and old. Leaks resulting from holes rusted through the pipe not only lose air, but create safety problems as well.

Interconnecting Piping

Air is being delivered from the compressors to the interconnecting piping ranges between 78 and 80 psig and getting into the main air system at 78 to 80 psig. This is an apparent pressure loss of 0 psig, which is very good.

Flow Regulation At The Process

Some flow regulators are probably set higher than the feed pressure required by the process, and some are left wide open to full header pressure. In this type of operation, it is very important that the actual inlet pressure to the process be known and that the lowest effective pressure be held steady for the proper product quality. Picatinny may need to install storage bottles downstream of the regulator to "close up" the pressure readings at rest and at operation. The minimum effective pressure at operation for each product run, established at the unit, needs to be established and adhered to.

Auto Condensate Drains

Automatic drain traps come in three categories. Level Operated Mechanically Activated Drains do not waste air, but are prone to clogging and require continuing maintenance to assure operation. These work best in a "Power House situation" where continuing regular attention is part of the system.

Dual Timer Electronic Drains use an electronic timer to control the number of times/hr it opens and the duration of the opening. The theory is that you adjust the times to be sure to fully drain the condensate and minimize the open time without water that wastes compressed air. The reality is that the cycles either do not get reset from the original factory settings (which causes condensate build up in the summer) or they get set wide open and not closed down later in cooler weather thus wasting more air. When they "fail open," they blow at a full flow rate of about 100 cfm.

Consider that the usual factory setting is 10 minutes with a 20-second duration. Consider that 1500 scfm of compressed air will generate about 63 gal/day in average weather or 2.63 gal/hr. Each 10-minute cycle will have 0.44 gal to discharge. This will blow through a ¼-in. valve at 100 psig in approximately 1.37 seconds. Compressed air will then blow for 18.63 seconds each cycle, 6/minute will equal 111.78 seconds/hr of flow or 1.86 minutes/hr of flow. This will waste about 3.1 cfm. A 1/8-in. valve will pass about 100 cfm. The total flow will be 100 x 1.86 = 186 cu ft in 1 hr x 60 minutes = 3.1 cu ft/min average. Energy cost/lost air = $310/yr/valve.

Level Operated/Electronic Drains can receive the signal to open from the condensate high level and the signal to close from the condensate low level. These waste no air and (from a power cost standpoint) are the best selection and their reliability is usually many times greater than the level operated mechanical.

There is no doubt that automatic drain traps are a much better idea than manual drains for Picatinny's circumstance. The Arsenal should take the following action:

- For air conservation and enhanced performance, all dual timer electronic drains and manual drains should be replaced by level-actuated electronic or air-operated drains. Timer-activated drains or dual-timer drains may not be able to handle "heavy loads" of condensate unless continuously "monitored during the summer conditions."
- Be sure your auto drains are set up to work effectively, for examples:
  - drains should not be tied together to a common header
  - be sure all drains can be checked easily for operation
  - be sure all drains are properly "vented."

The survey of the condensate handling system revealed several issues. Arsenal personnel stated that the condensate goes to a mechanical oil/water separator and then to the storm sewer and lake. According to plant personnel, the discharge is monitored constantly to assure no USEPA violation. If this is always in effect, there is no apparent problem.

If Picatinny is discharging filtered condensate to a storm sewer or in some other manner to ground water (the USEPA minimum is 10 ppm), or if the Arsenal is required to separate it by local water treatment facility, this issue should be discussed in detail.

Leak Management Programs

With a campus facility of this type, an effective leak control program could well save in the average range of 300 to 400 cfm, which could potentially result in an annual power cost savings of $30,000 to $40,000. The estimated recoverable value is $25,000/yr.

To effectively control and manage leaks in such an extensive operation as Picatinny Arsenal, a continuing economical program must be in place. Generally speaking, the most effective programs are those that involve the production supervisors and operators working positively with the maintenance personnel.

Accordingly, the TMSI Team recommends:

- In the short-term, set up a continuing leak inspection by Maintenance Personnel so that for a while, each primary sector (see drawing) of the plant is inspected once a quarter or at a minimum, once every 6 months to identify and repair leaks. A record should be kept of these findings and overall results.
- In the long-term consider setting up programs where the production people (particularly the operators and their supervisors) are positively motivated to identify and repair these leaks. A very effective ultrasonic leak locator quote can be provided from the Team.
- The Team can also lists some electric-operated automatic ball valves that can be installed in the main feed line to a piece of equipment and be wired in so as to open and close when the machine is powered up or shut off and thus eliminate off-production leaks and open air left on.

Cabinet Coolers

There may be cabinet coolers in use in the facility. Some with refrigeration (1500 Btu), some with compressed air-driven vortex coolers; and some may just have compressed air blowing into them. These all may be able to be replaced with “heat tube” cabinet coolers with a potential savings of 3.5 to 4 kW each.

The initial cost for this range is usually in the $700 to $750 range with a potential resultant electric savings of $1000 to $2000/yr each.

Blow Offs

Picatinny may have 1/8-in. and ¼-in. lines running as blow off on units at 80 psig. These will use 8 to 35 cfm each.

An alternate is an air amplifier that takes less compressed air and through Venturi action amplifies the usable air by pulling in significant amounts of ambient air and mixing it directly into the air stream. These have amplification ratios up to 25:1. Using 10 cfm of compressed air would generate a savings of 25 cfm compressed air per ¼-in. blow off and flow 250 cfm total air at the process.

Table D4. Costs associated with tube / nozzle change to alter blowoff configuration.

Parameter	Cost
Annual power cost of one ¼-in. tube (continuous)	$3500 /yr/each
Annual power cost of one venturi nozzle (continuous)	$1000 /yr/each
Energy cost saved	$2500 /yr/blow off
Recoverable energy cost	$1750 /yr/blow off
Nozzle cost	$17
Annual power cost of one ¼-in. tube (10% use)	$350 /yr/each
Annual power cost of one venturi nozzle (10% use)	$100 /yr/each
Energy cost saved	$250 /yr/blow off
Recoverable energy cost	$175 /yr/blow off
Nozzle cost	$17

For example, ¼-in., 1-ft-long tube will flow 35 cfm at 80 psig inlet, at an annual cost power of $3500/yr/ea. Place a variable flow Venturi nozzle to amplify flow on the end of this tube and it will now only use 10 cfm and flow 250 cfm at the work. Table D4 summarizes the associated costs.

Vacuum Generators

Production may use vacuum generators, which are:
- convenient
- responsive
- inefficient compared to positive displacement pumps, e.g., rotary screw, reciprocating.

Note that energy cost escalates as vacuum goes down with Venturi generators. Energy cost also falls as vacuum goes down after about 14 in. with positive displacement pump. It is very important to only run a Venturi vacuum generator to a minimum vacuum and a minimum acceptable “on time” cycle at the lowest possible pressure.

For example, if generator uses 60 scfm at 80 psig, it can pull a 20-in. vacuum in about 0.25 seconds. If shut off at 20-in. vacuum, total air demand will be about 0.25 scfm with Energy Cost = $25/yr. If allowed to run continuously, air usage 60 scfm with Energy Cost = $6000/yr.

Air-Operated Diaphragm Pumps

Air-operated diaphragm pumps are generally used because they tolerate aggressive conditions relatively well and run without catastrophic damage even if the pump is dry. Efficiency is not usually considered.

There are several areas to pursue here in the future to perhaps generate significant air savings:

- Is the air-operated diaphragm pump the right answer? An electric pump is significantly more power efficient. Electric motor driven diaphragm pumps are available.
- Consider the installation of electronic or ultrasonic controls to shut the pumps off automatically when they are not needed. Remember the pump uses the most air when it is pumping nothing.
- Is Picatinny running most of the time at the lowest possible pressure? The higher the pressure, the more air used. For example, in a filter pack operation, the pump often does not need high pressure except during the final stages of the filter packing cycle. Controls can be arranged to accomplish lower pressure in the early stages and higher pressure later, which may generate significant savings.

Misapplied High Pressure Air

High pressure air being used for very low pressure applications is not an efficient use of energy. A close review of your system should be made and measurements taken to identify if there is any potential energy savings in using an alternate source of low pressure air in the production area.

Gas Engine-Driven System Assessment

This section provides a preliminary assessment of the opportunity for gas engine driven compressors at Picatinny Arsenal. The assessment is based on three key design factors:

- operating the new system as a fully hybrid system with the existing electric system
- meeting environmental requirements of the area
- improving the current demand system so that the air requirements for the new system are minimized, along with producing system operating costs for the Arsenal.

Gas Engine-Driven System Design Factors

The conceptual design for the Picatinny gas engine driven system is based on providing the full requirements of the main system at Picatinny, but configured as full hybrid system in conjunction with the existing electric system. In this way, the existing electric system can serve as a back-up to the gas engine system if the gas system has a planned or unplanned shutdown or if the air requirements of the base are suddenly increased.

Using this approach, the Department of Defense can gain experience with not only operating a gas engine driven system, but also integrating it with electric systems to improve overall compressed air system reliability and reduce operating costs. This flexibility is especially important given the increasing uncertainty associated with the price and supply reliability of most energy sources.

Environmental issues are key areas to address in any major project on this nature, particularly on the East Coast. The assessment is based generating up to 0.70 gm/bhp/hr for NOx and 0.48 gm/bhp/hr for CO.

The current demand for compressed air of 925 acfm can be reduced by an estimated 500 acfm. This reduction can be accomplished by reducing leaks (a potential reduction of 300 acfm throughout the day) and shutting off equipment during nonproduction periods (a potential reduction of 200 acfm during the nonproduction period from 6 pm to 6 am).

The gas engine driven system evaluated in this study reflects the reduced demand level, even though the original demand level would make the gas-driven system appear to be even more cost-effective.

Operating Cost Comparison

Table D5 lists the electrical energy costs for the Main Power House System for both the current demand level and the reduced demand level. Annual electrical costs are $99,487 for the current system and $57,612 for the modified system, a savings of $32,000 annually. Maintenance contract costs for both are estimated at $11,800 annually given the age and condition of the machines. The sum of the operating costs for energy and maintenance costs is $111,287 for the current demand level and $69,412 for the modified level.

Table D6 displays the estimated operating cost for two types of gas engines: 3406TA and the slightly smaller 3306TA. Annual fuel costs for the 3406TA system are $28,976 annually and $26,206 for the 3306TA system. The 3306TA system is some $32,000 less in energy costs than the current electric system based on the modified air demand level. The number increases to $56,000 annually based on the original air demand level.

Contract maintenance costs are $28,908 annually for the 3406TA system and $21,900 for the 3306TA system. This is some $10,000-17,000 higher than with the electric system.

Table D5. Electrical energy costs (annual)—Main Power House.

	Current Demand Level	Modified Demand Level		
		1st Shift	2nd Shift	Total
Average Prod Flow	925 cfm	Prod600 cfm	Non-Prod330 cfm	
Average Prod kW	126.75 kW	79.5 kW	56.4 kW	
Prod Air Oper Hours	8,760 hrs	4,380 hrs	4,380 hrs	
Specific Power	7.1 cfm/kW	6.29 cfm/kW	5.32 cfm/kW	
Energy Cost $/cfm/yr	$108.59 /cfm/yr	$61.28 /cfm/yr	$72.45 /cfm/yr	
Air Energy Cost/psig	$488.34 /psig/yr			
Est Air Energy Cost/yr	$99,487 /yr	$33,704 /yr	$23,908 /yr	$57,612 /yr
Maintenance Contract	$11,800 /yr	—	—	$11,800 /yr
Sum of Operating Costs for Energy and Maintenance Contract	$111,287	—	—	$69,412
NOTE: Blended Power Rate = 0.088 $/kWh; Overall system pressure = 80 psig				

Table D6. Cost comparison: natural gas engine 3406TA/3306TA.

	3406TA			3306TA		
	1st Shift	2nd Shift	Total	1st Shift	2nd Shift	Total
Average Prod Flow	600 cfm	330 cfm		600 cfm	330 cfm	
Average Production	121 bhp	104 bhp		121 bhp	104 bhp	
Prod Air Oper Hours	4,380 hrs	4,380 hrs		4,380 hrs	4,380 hrs	
BSFC Fuel Consumption	8,390	8,485		7,745	7,860	
Est Air Energy Cost/yr	$15,163 /yr	$13,813 /yr	$28,976 /yr	$13,997 /yr	$12,209 /yr	$26,206 /yr
Maintenance Contract	—	—	$28,908 /yr	—	—	$21,900 /yr
Sum of Operating Costs for Energy and Maintenance Contract	—	—	$57,884 /yr	—	—	$48,106 /yr
Note: Blended Power Rate = 0.088 $/kWh; Overall system pressure = 80 psig, Natural gas rate = 3.41 $/MBtu. (Based on NG/water-cooled engine @ 80 psig with 1,000 Btu per ft^3).						

The sum of the operating costs for energy and maintenance contract are $57,884 for the 3406TA system—some $12,000 less than the electric system on an annual basis. The sum of the operating costs for energy and maintenance contract are $48,106 for the 3306TA system—some $21,000 less than the electric system on an annual basis.

Estimated Fuel Costs: Gas Engine System 3406TA:

121 x 8390 x 4380 x 3.41 ÷ 1,000,000 =	**15,163**
109 x 8485 x 4380 x 3.41 ÷ 1,000,000 =	**13,813**
	28,976

Estimated Fuel Costs: Gas Engine System 3306TA:

121 x 7745 ÷ 1,000,000 x 4380 x 3.41 =	**13,997**
104 x 7860 ÷ 1,000,000 x 4380 x 3.41 =	**12,209**
	26,206

***Based on BSFC = 7,500 and 27.5 BHP x 7,500 x 8,760 ÷ 1,000,000 x $8/Million Btu = $14,454/yr.**

Maintenance Contract: (Gas Engine System 3406TA):

$3.30/operating hour x 8760 hours/yr

(based on 2-year agreement and 100 hrs portal to portal/60 mi) = $28,908

Maintenance Contract: (Gas Engine System 3306TA):

$2.50/operating hour x 8760 hours/yr

(based on 2-year agreement and 100 hrs portal to portal/60 mi) = $21,900

Maintenance Contract: (Current Electric System):

$1.35/operating hour x 8760 hours/yr

(based on relatively worn machines—comparative purposes only) = $11,800

Capital Cost Assessment

Table D7 displays capital cost estimates for three configurations of the natural gas engine-driven systems: IR 3406, GD3406, and GD3306. The capital costs include the catalytic converter and a placeholder estimate for installation costs.

Without consideration to potential cost reductions resulting from negotiating or utility rebates, the capital costs for the natural gas systems are on the order of $150,000.

Table D7. Capital expenditure need for Picatinny.

Basic Unit	Model		
	IR 3406	**GD 3406**	**GD 3306**
925 cfm @ 80 psig	PCD250NG 925L	GRS-200LW	GRS200LW
1300-1800 rpm			
Noise level load dBA			
	$110,000	$143,000	$125,000[1]
Catalytic converter[2]	23,500	13,400	12,200
	$133,500	$156,400	$135,200
Installation & Freight	$ 15,000	$ 15,000	$ 15,000
Total	$148,500	$171,400	$150,200

[1] Add $4,000 for air-cooled engine and compressor and water-cooled after-cooler -- air-cooled installation may be preferable with adequate ventilation.

[2] Based on a generation limit of NOx 0.70 gm/bhp/hr and CO 0.48 gm/bhp/hr.

Appendix E: Compressed Air System Survey at Pine Bluff Arsenal

Background

The main compressed air system serves six different production areas—Area 3 (Sections 1, 2, 3, and 4) and Area 4 (Sections 2 and 4). There are six Ingersoll Rand "remanufactured" two-stage, double-acting, water-cooled air compressors 16 in. and 10 x 7 in. stroke. Three are 150-hp units at 585 rpm. Three are 200-hp units at 705 rpm. These units are less than 1 year old and are very power efficient and very responsive to demand changes.

There is a pair of 150-hp and 200-hp units in Buildings 32-060; 33-060, and 34-140. These units appear to be well installed and maintained. However, the cooling water system seems to be acting a bit unstable and perhaps should be reviewed.

The air from these units goes through water-cooled aftercoolers and then to air receivers (1,000 gal) and to a heatless-type regenerative desiccant dryer. These also were recently purchased.

According to plant personnel, the air has been satisfactory since the system upgrade. Each building today has the following total air volume available for production:

150 hp unit	**+ 809 acfm**
200 hp unit	**+1,000 acfm**
	1,809 acfm
Less purge for dryer	**286 acfm**
Net	**1,523 acfm**

There are several dedicated air systems on post:

Incinerary (Building 42-980). The older 75-hp IRLLE has been replaced with a new Sullair 75-hp rotary screw.

LAP Building (Building 44-120). The old Worthington in HBs have been replaced with four Ingersoll Rand EP75, lubricant-cooled (1993) rotary screw compressors. These are excellent units. On site inspection revealed only one of four units was on and it was loaded about 7 to 8 percent (31 cu ft). It was mostly at idle with an average 38 kW and annualized electric cost of $18,000/yr. Consideration could be given to installing a 20-hp unit to handle this low load.

System Baseline

The key characteristics describing the performance and economics of the current compressed air system are summarized in Tables E1, E2,and E3. They were was developed based on the data collected during the site visit and with discussions with plant personnel. The estimates are conservative and reflect observed performance of each compressor compared to load cycle.

Table E1. Key characteristics of existing system.

Measure	1st Shift	2nd Shift	Total	Non-Production or Holidays	Total
Average Air Production Flow	3118 scfm	2541 scfm		2058 scfm	
Average Production kW	423 kW	365 kW		368 kW	
Production Air Operating Hours	754 hrs	1321 hrs	2080 hrs	6680 hrs	8760 hrs
Specific Power	7.37 cfm/kW	6.96 cfm/kW		5.59 cfm/kW	
Energy Cost for Air ($/cfm/year)	$1.17 /cfm /yr	$15.80 /cfm /yr		$68.12 /cfm /yr	$85.09/cfm/yr*
Total Annual Energy Cost for Air	18,180 /yr	$27,588 /yr	$45,768 /yr	$140,191/yr	$185,959 /yr
Energy Cost for Air ($/psig/year)	$90 /psig/yr	$137.94 /psig/yr		$706.68 /psig/yr	$928.62 /psig/yr

Table E2. Existing air compressor during production hours.

	Manufacture	% of Load	% of Power	FL kWx% of Power	Net kW	Actual cfm
1	XLE/150	100%	100%	139 x 1	139	809
2	XLE/150	100%	100%	139 x 1	139	809
3	XLE/200	50%	60%	87 kW	87	500
4	XLE/200	100%	100%	145 kW	145	1000
Psig: 110 for 2080 hrs. Total cfm: 3118 cfm/754 hr/423 kW; 2541 cfm/ 1326 hr/ 365 kW Estimated non-production flow=1200 cfm in leaks + purge from 3 dryers = 286 x 3=2058 cfm.						

Table E3. Existing air compressors during weekend/holiday hours.

	Manufacture	% of Load	% of Power	FL kWx% of Power	Net kW	Actual cfm
1	XLE/150 hp	100%	100%	139 kW	139	809
2	XLE/150	100%	100%	139 kW	139	809
3	XLE/150	54%	65%	90 kW	90	440
	TOTAL				368	2058

Average electric rates at the plant are 0.057 kWh. The actual plant electric cost for air production, as running today, is on the order of $186,000/yr.

The load profile or demand of this system is relatively stable during all shifts. The full operating range is 365 days a year, 24 hours a day, 8,760 hours a year. There are no flow meters in the system.

The system pressure appears to run from 110 to 115 psig to the headers during production.

There are a number of potential measures that are recommended in this review as in the Phase II Action Plan to reduce electric costs to operate the compressed air system.

The report identifies the "electric cost per hour per loaded cfm" of air used. Electric cost was selected as the key project evaluation factor, since it is a good overall indication of system costs and savings associated with potential measures. It is an absolute number and not a subjective opinion, i.e., if the compressed air is used, these dollars are spent. All paybacks are estimated using the "Full Load Operating Efficiencies," which are very conservative.

If the compressed air is not used, the compressor either shuts off or unloads. If it shuts off, there is a 100 percent saving of the electric cost. If it unloads, there is a 25 to 90 percent savings of the electric cost.

It is important to note that other recoverable compressed air costs should also be considered, i.e., maintenance, water costs, depreciation, etc. Usually, the electric cost is between 75 and 90 percent of the total "variable compressed air costs." Associated maintenance and other costs will be, in all probability, at least 20 percent or more of the identified electric cost. Existing plant records may already have these identified.

Today's annual cost/(flow)	**$ 85.09/cfm/yr**
Today's annual cost/(pressure)	**$ 928.67/psig/yr**
Today's annual estimated operating cost	**$ 185,959/yr**

NGEDAC Application

The logical implementation sector for a supplementary need rotary screw air compressor would be at Building 34-140 in the side yard at the end.

In addition to purchasing and installing the compressor, it would also require:

- Cooling water at 124 gpm 90 °F water at the appropriate pressure. The water connection exists at that end of the building which used to feed the boiler that is now out of service.
- Compressed air piping connection of discharge air to the main header back to the air receiver in the building for two XLEs.
- Piping natural gas to the unit with an appropriate regulator—approximately 400—450 ft away.
- Contractor of an appropriate steel building to protect the open unit from the elements. It will have to be properly vented and built to applicable codes.

The suggested basic units 925/H will basically be set to run instead of one current 150-hp XLE (809 cfm) and can be piped with its water-cooled aftercooler, so it can also go to the Building 34 dryer. Note that it will not be possible to run the two XLEs and the NGED to the dryer. This would overload it.

Table E4 lists data that can be used to make a the comparison of the annual operating cost of 150-hp XLE and the NGED 925/H—when supplying 809 acfm at 110 psig.

Energy Cost Baseline

Table E5 shows a recent history of energy expenditures at Pine Bluff Arsenal. Electric usage peaks in the summer for space cooling applications and natural gas usage peaks in the winter for space heating applications.

Table E4. Annual operating cost comparison.

	IR 150 HP XLE w/ Induction Elec Motor	IR Model 925/H w/ CAT 3406 TASCAC Engine
CFM Flow	809	809
FL Pressure	110 psig	110 psig
Input Power	139 kW	At 0.008020 MMBtu/hp/hr BHP = 234 x 0.87 x 0.925 = 189
Operating Hours	8760	8760
Energy Cost	$0.057 per kWh	$4.00/MMBtu $6.00/MMBtu
Annual Operating Energy Cost	$69,405 /yr (139 X 0.057 X 8760)	$53,112 /yr $79,668 /yr .008020 x 189 x 8760 = 13,278 MMBtu
Annual Maintenance Cost	$13,000	$26,280
2-year Fixed Maintenance Cost via IR	$26,000	$52,560

Table E5. Energy cost summary.

	Electric			Natural Gas		
Month	Use (kWh)	Cost ($)	Rate ($/kWh)	Use (MMBtu)	Cost ($)	Rate ($/MMBtu)
Oct-99	1,335,600	67,345	0.0504	37,827	119,896	3.1696
Nov-99	1,086,400	52,808	0.0486	38,264	140,965	3.6840
Dec-99	1,195,600	51,389	0.0430	51,329	125,145	2.4381
Jan-00	1,237,600	52,299	0.0423	60,004	159,051	2.6507
Feb-00	1,058,400	47,983	0.0453	49,417	144,189	2.9178
Mar-00	1,078,000	50,719	0.0470	37,948	111,472	2.9375
Apr-00	1,019,200	49,095	0.0482	23,594	77,316	3.2770
May-00	1,145,200	69,977	0.0611	22,764	79,047	3.4725
Jun-00	1,615,600	92,000	0.0569	18,907	91,399	4.8342
Jul-00	1,663,200	94,734	0.0570	17,919	86,080	4.8039
Aug-00	1,871,100	103,751	0.0554	16,313	69,985	4.2901
Sep-00	1,607,760	82,462	0.0513	18,600	95,120	5.1140
TOTAL	15,913,660	814,561	0.0512	392,886	1,299,667	3.3080

Gas costs averaged $3.31 during the October 1999 to September 2000 time period. At $5.11/MMBtu, the September figures reflect the beginning of the recent run-up in gas prices. The Arsenal does enjoy low LDC rates which are in the area of 30 cents/million Btu. The analysis in this report is based on $4.00/million Btu. Operating costs were also derived based on using $6.00/MMBtu.

Electric costs averaged 5.1 cents/kWh and ranged from 4.2 cents to 6.1 cents. Some pressure to increase rates could likely occur, although large jumps of more than 40 percent are not expected. The analysis in this report is based on 5.7 cents/kWh.

Supply-Side System Review

Primary Air Compressor Supply

The basic air supply is with very high-quality, durable, and power-efficient double-acting reciprocating compressors. There are less than a year old and with proper installation and timely maintenance such as they are now getting they will give many years of service. There are no other types of compressors for this application that would allow as power efficient full load and part load capabilities.

The primary compressed air supply is produced by relatively efficient air compressors that are capable of delivering the 110 psig full load pressure in a continuous manner. The units are well applied. They appear to be in good operating order and well maintained. Table E6 lists key characteristics of the units.

Compressor Capacity Controls

The two most effective ways to run air compressors are at “Full Load” and “Off.”

Capacity controls are methods of restricting the output cfm delivered to the system while the unit is still running. This is always a compromise and is never as efficient as full load on a specific power (cfm/hp) basis.

Table E6. Key characteristics of existing air compressors.

Type	Double-acting Reciprocating	Double-acting Reciprocating	SS Rotary Screw
Brand	Ingersoll Rand	Ingersoll Rand	Ingersoll Rand
Model	16” & 10” x 7” XLE 150hp	16” & 10” x 7 XLE 200hp	EP75 75hp
ACFM	809 acfm	1000 acfm	320 acfm
FL Press	110 psig	110 psig	110 psig
kW @ 110 psig (measured)	139 kW	145 kW	76 kW
Cfm/kW/110 psig	5.82 cfm/kW	6.89 cfm/kW	4.2 cfm/kW
Annual Elec Cost $/cfm	$85.79 /cfm/yr	$72.47 /cfm/yr	$118.88 /cfm/yr
Annual Elec Cost $/psig	$347.03 /psig/yr	$362.01 /psig/yr	$189.74 /psig/yr

Reciprocating Controls (Main Air System)

Reciprocating compressors are double-acting, water-cooled units with three-step unloading. This is an efficient compressed air unloading system, Reciprocating three-step unloading will efficiently translate percentage of “less air used” into almost the same proportional reduction in energy cost.

The current system has three-step unloading with electronic Intellysis model. The electronic control will allow these six units to link in a network system in the future.

Rotary Screw Controls (LAP Building)

The two most common controls used are modulation and online/offline. Modulation is relatively efficient at very high loads—and inefficient at lower loads. Online/offline controls are very efficient for loads below 60 percent, when properly applied with adequate time for blow down. There are several other control types (e.g., “rotor length adjustment” or “variable displacement” and “variable speed drive”) that have very efficient turn down from 100 percent load to about 60 percent load.

These controls must be installed correctly to operate efficiently. Piping and storage should be available close to the unit with no measurable pressure loss at full load to allow the signal to closely match the air requirements.

The current system has an electronic Intellysis system which combines online/offline with upper range modulation. The Intellysis also has automatic control selection which will automatically move to the appropriate control depending on the sensed load conditions.

The units involved have capacity controls capable of translating “less air used” into a comparable reduction in electric cost. These controls will apparently work effectively with current piping and air receiver storage situation. None of the systems could be reviewed at maximum load.

Central Networking Control System

With the system stabilized and balanced, consider a microprocessor-driven centralized full networking electronic control system. This will automatically place the most efficient machine online and assure no more than one partial loaded unit at a time. All of the Ingersoll Rands have the electronic Intellysis which is designed to become part of a full networking system.

Air Treatment and Air Quality

Current Drying Operation

Table E7 gives an overview of the system's current drying system.

Aftercoolers

Aftercoolers are water cooled on the XLEs and appear incapable of delivering 100 °F or lower temperature compressed air to the dryer.

Refrigerated dryers require a refrigeration system to mechanically cool the air. The lowest possible consistent pressure dew point with a noncycling dryer is +40 °F. Cycling and variable speed-driven dryers not only save power (60 to 75 percent), but also can deliver a lower pressure dew point (down to 35-38 °F).

Desiccant Dryer Regeneration equipment removes moisture vapor by "adsorbing" it to desiccant beads. These dryers can consistently deliver a pressure dew point to –40 °F or lower, which removes much more water than conventional refrigeration units. To regenerate the wet tower while the other tower is drying requires the use of heat in some form and some dry air to "sweep" or "purge" the exchanged moisture out. Desiccant dryers are usually rated at the same 100 °F inlet 100 psig conditions.

The primary dryers are a twin tower, heatless, regenerative, desiccant dryer capable of delivering a consistent -40 °F pressure dew point when air is delivered to the dryer at no more than 100 °F (Air was 74 °F). (Dryer was at +31 °F at time of the site visit.)

Table E7. Current drying system.

Type	Heatless Desiccant
Brand	ZEKS
Model	1910
Rating	1910 scfm @100°F; 100 psig
SCFM Purge	286
Estimated Annual Electric Cost of Purge	$29,310 /yr
Heater kW/Refrigeration kW	N/A
Annual Operating Electric Cost for Current Dryers	$29,310 yr x 3 dryers = $87,930 elastic cost to product purge air
Rating	$12.73/ cfm
Actual Pressure Loss	N/A
Annual Electric Cost to Produce psig Lost	N/A

The condensate driven out of the aftercooler, prefilter, dryer and afterfilter is immediately removed from the system and not allowed to retrain or build up.

Regeneration is accomplished by heatless purge air (15 percent) and the dryer is not equipped with appropriate purge controls.

Recommended Measure—Add dew point demand purge controller.

This will reduce total purge at 50 percent. For three dryers from 858 to 429 cfm average:

Estimated Electric Energy Savings	**$30,000/yr**
Estimated Cost of 3 Controllers	**$45,000/yr**

Water Or Oil Carryover in System

Water (condensate) and oil carryover problems in the current air system are significant and can be expected to increase in magnitude during the summer.

The correct way to eliminate water and oil in the air system is to clean and dry the air immediately after it is produced in the compressor room. Then clean dry air can be stored in a separate air receiver and flow it to the system, as required. Some guidelines for controlling oil and water carryover include the following:

- Generally, it is best to eliminate the water and oil right at the air source before it enters the air system
- Every 20 °F increase in temperature doubles the "moisture load" the compressed air will hold
- Compressed air dryers are usually capacity rated with 100 °F, 100 psig inlet air conditions. At 120 °F, 100 psig, the dryer's capacity rating is reduced 50 percent.
- Putting "dry/or oil free" air into system 90 percent of the time and then allowing wet/oily air in sporadically 10 percent of the time will, in reality, make the system wet or oily all the time. The liquid water and/or oil will fall out in the piping system continuing to "Re-entrain" and contaminate and/or collected in the "Low Spots" of the system, thus recontamination as it is pulled into the flowing compressed air system. A wet/oily system may well take many months of continued flowing of clean dry air to "clean up."
- Identify required pressure dew point.

Automatic Condensate Drains

Background

The configuration and performance of condensate drains in the plant's system should be reviewed to determine whether it needs to be modified.

The automatic condensate drains currently all go through closed lines to a several drain pipes to a collection point. The drains on the dryer are almost all timer-activated. Many of the timer drains are operating uncontrolled and blocking other drains from operating correctly. Many of the drains do not appear to be working at all.

☑ *Recommended Measure—Replace all timer drains with level activated drains.*

Connect each drain's point (after-cooler, pre-filter, dryer, after-filter, receivers, and all risers) separately to individual level-activated electric or pneumatic drains to collect and direct the condensate to a proper handling point carry it in a large plastic vented line (4 or 6 in.). Be sure maintenance personnel can effectively and visually monitor the drain's action.

CFM savings per drain	**3.1 cfm/yr each**
Estimated Energy Savings	**$264 yr each**
Total of 15 drains replaced	**$3,960 yr total**
Cost per Drain	**$350 each**
Cost 15 Drains	**$5,250**

Demand-Side System Review

Basic System Header And Piping

It is the job of the main header system to deliver compressed air for production use from the compressor area to all sectors of the plant with little or no pressure loss—with 1-3 psig being a reasonable target. It is also desirable that the compressed air velocity in the main headers be kept below 20 fps to allow effective drop out of contaminants and to minimize pressure losses caused by excessive turbulence. The magnitude of the turbulence effect also depends on piping design and layout.

Headers were not checked during this visit but should be in any Phase II operation.

Regulator Setting

Some regulators are probably set at higher than necessary feed pressure to the process, with some wide open to full header pressure. Is there a minimum effective pressure at operation established at the unit for each product run? If so, is it being adhered to?

In this type of operation, it is very important that the actual inlet pressure to the process be known and that the lowest effective pressure be held steady for the proper product quality. Installation of storage bottles downstream of the regulator may be needed to "close up" the pressure readings at rest and at operation.

Regulators were not checked during this visit, but they should be a part of a Phase II operation.

Compressed Air Condensate Handling

If (as site personnel indicated) the condensate (in the condensate handling system) goes to water treatment, then the discharge condensate should meet the requirements of the water treatment facility plant, there is no problem.

However, if you are discharging the condensate to a storm sewer or in some other manner to ground water (Federal EPA minimum is 10 ppm) or are required to separate it by your local water treatment facility, this should be discussed in detail.

Leak Identification and Repair

With a plant of this type, an effective leak control program could save 1200 cfm or the equivalent of repairing 300 leaks averaging 4 cfm each. On a percentage basis, this leak level is about the same as leak levels in other plants. Leaks totaling 1200 cfm translate into an annual loss of $102,000 in electric cost. A comprehensive leak management program could reduce such levels by 70 percent or $71,400 annually.

☑ *Recommended Measure—Implement a continuing leak identification and repair program with ultrasonic locators.*

There should be a continuing economical program in place. Generally speaking, the most effective programs are those that involve the production supervisors and operators in a positive manner working in concert with the maintenance personnel. Accordingly, it is suggested that the program consist of the following:

- *Short Term*—Set up a continuing leak inspection by Maintenance Personnel so that for a while, each primary sector of the plant is inspected once a quarter or at a minimum, once every 6 months to identify and repair leaks. A record should be kept of these findings and overall results.
- *Long Term*—Consider setting up programs where the production people (particularly the operators and their supervisors) are positively motivated to identify and repair these leaks. One method that has worked well with other operations is to monitor the airflow to each responsible section (perhaps with the use of recording the nonrecording flow meters) and to identify the air usage as a measurable part of the operating expense of that area. This usually works best when combined with an effective "In House" Training And Awareness Program.

Table E8 lists savings associated with implementing a leak management program.

Table E8. Savings associated with implementing a leak management program.

Parameter	Cost
Estimated number of leaks	300 leaks
Estimated average leak size	2 cfm/leak
Estimated leak level	600 cfm
Estimated unit electric savings (from system baseline chart)	$85.09/cfm
Estimated potential electric savings	$51,000
Recoverable leak losses	70 percent
Calculated electric savings from leak program	$35,700/yr
Costs associated with implementing a leak management program include:	
Leak detection equipment	$2,800
Leak program development and detection equipment training	$1,000
Leak repair (300 leaks @ $30 materials/leak and $30 labor/leak)	$18,000
Total program cost (annually for ongoing repairs)	$21,800 plus $1,000

Cabinet Coolers

Cabinet cooling is often required to obtain reasonable life and performance of the electronic equipment in control cabinets. There are various means of accomplishing this. Blowing straight compressed air into the cabinet is generally very inefficient. Vortex coolers can use chilled air with no moving parts and use less of it.

Vortex coolers should always:

- be regulated to the lower effective pressure
- be equipped with the lowest possible flow generator
- be equipped with automatic temperature controlled shutoffs.

Refrigeration units should be carefully selected and equipped with automatic Regulation Control. Heat tubes are the most energy efficient when applied and can cool a "sealed cabinet."

No check was done for cabinet cooler or other vortex cooling on this visit. These items should be part of a Phase II operation.

Vacuum Generators

The plant's current production system may use vacuum generators. Vacuum generators are very convenient, very responsive, and very inefficient compared to positive displacement pumps, i.e., rotary screw, reciprocating.

Energy cost escalates as vacuum goes down with Venturi generators. Energy cost falls as vacuum goes down after about 14 in. with positive displacement pump. It is very important to only run a Venturi vacuum generator to a minimum vacuum and a minimum acceptable "on time" cycle at the lowest possible pressure.

No check of vacuum generators was done during this visit. They should be part of any Phase II operation.

Air Operated Diaphragm Pumps

Although air-operated diaphragm pumps are not very energy efficient, they tolerate aggressive conditions relatively well and run without catastrophic damage even if the pump is dry. There are several areas to pursue in the future to perhaps generate significant air savings:

Is the air-operated diaphragm pump the right answer? An electric pump is significantly more power efficient. Electric motor driven diaphragm pumps are available. An electric motor drive progressive cavity pump may well work.

Consider the installation of electronic or ultrasonic controls to shut the pumps off automatically when they are not needed. Remember the pump uses the most air when it is pumping nothing

Are you running most of the time at the lowest possible pressure? The higher the pressure, the most air used. For example, often a filter pack operation, the pump does not need high pressure except during the final stages of the filter packing cycle. Controls can be arranged to accomplish lower pressure in the early stages and higher pressure later which may generate significant savings.

No check of the diaphragm pumps was made during this visit. They should be part of any Phase II operation.

Misapplied High-Pressure Air

High-pressure air being used for very low-pressure applications is not an efficient use of energy. A close review of your system should be made and measurements taken to identify if there is any potential energy savings in using an alternate source of low-pressure or high-pressure air in the production area.

This could be part of a future study of demand-side activities at Pine Bluff.

Gas Engine Driven System Assessment

This section provides a preliminary assessment of the opportunity for gas engine driven compressors at Pine Bluff Arsenal. The assessment is based on three key design factors:

Operating the new system as a fully hybrid system with the existing electric system

- meeting or exceeding environmental requirements of the area
- improving the current demand system so that the air requirements for the new system are minimized, while reducing system operating costs for the arsenal.

Gas Engine Driven System Design Factors

The conceptual design for the gas engine-driven system is based on replacing one IR 150hp XLE and providing about half of the requirements of the main system at Pine Bluff. It will be configured as a hybrid system in conjunction with the existing electric system. In this way, the existing electric system can serve as a back-up to the gas engine system, if the gas system has a planned or unplanned shutdown or if the air requirements of the base are suddenly increased.

Using this approach, the Department of Defense can gain experience with not only operating a gas engine driven system, but also integrating it with electric systems to improve overall compressed air system reliability and reduce operating costs. This flexibility is especially important given the increasing uncertainty associated with the price and supply reliability of most energy sources.

Environmental issues are expected to be minimal in this application given the key areas to address in any major project on this nature, although less so in this part of the country. The assessment is based generating up to 2.60 gm/bhp/hr for NOx and 1.75 gm/bhp/hr for CO.

Operating Cost Comparison

Table E9 displays the electrical energy costs for the IR 150hp XLE compressor and the proposed NGEDAC system. Annual electrical costs are $69,400 for the current system and $53,000 for the proposed system, a savings of $16,400 annually, based on the cost of gas at $4/Million Btu. Annual comprehensive maintenance contract is about $13,000 higher for the proposed system. Net annual savings for the proposed system incorporating both the lower energy costs but higher maintenance costs are $3,400.

Capital Cost Assessment

Table E10 lists capital cost estimates for a Caterpillar 3406 TASCAR engine. The capital costs include the catalytic converter for the Caterpillar engine. The costs include a placeholder estimate for all installation and freight costs. The capital costs also include a budget to erect an outside enclosure to house the NGEDAC unit.

Without consideration to potential cost reductions resulting from negotiating or utility rebates, the capital costs for the natural gas system is on the order of $220,000 to $230,000.

Table E9. Operating cost comparison of current and proposed NGEDAC units.

Parameter	Current—IR 150hp XLE	Proposed NGEDAC—IR 925/H
Average Air Flow	809 cfm	809 cfm
Operating Hours	8760	8760
Annual Energy Cost	$69,400	$53,000 (Gas @ $4)
Maintenance Contract	$13,000	$26,000
Total Energy and Maintenance Contract Costs	$82,400	$79,000 (Gas @ $4)

Table E10. Capital cost estimate of NGEDAC unit.

Parameter	NGEDAC Unit 1
Model	IR 925/H
Engine	3406 TASCAR (Caterpillar)
Discharge Pressure	100 psig
Required Horsepower	234 BHP
Operating Speed	1800-1300 rpm
Environmental Discharges (GMS/BHP-hr)	NO_X 2.0 CO 2.0
Fuel Consumption	8020 Btu/HP-Hr
Package	$145,000
Catalytic Converter	$21,000
Separate Enclosure	$25,000
Installation and Freight	$37,000
Total Capital Costs	$228,000

Appendix F: Compressed Air System Survey at Watervliet Arsenal

Background

The Watervliet Arsenal has a very extensive compressed air system linking many separate buildings and spread over a large geographical area. The air system reaches most production sectors and runs building to building, eventually completing a full "loop" system. The compressed air supply is primarily generated in Building 110 with one large 2000 cfm (450 hp) class Joy centrifugal compressor and two 125-hp Ingersoll-Rand XLE (650 cfm/machine) reciprocating compressors.

- There are six other major compressors tied in to the main air system in surrounding buildings. There are also a number of smaller air-cooled reciprocating units throughout the Arsenal either as part of the separate "controls air system" or dedicated air to a particular process.
- Air drying is provided by both desiccant and refrigeration units and appears to be working well according to plant personnel and survey results. Most of the compressors are water cooled, but some have their own air-cooled, radiator-type, closed-cooling systems, which also appear to be working well.
- The complete air system appears to be very well laid out, well maintained and operated consistently with the type of controls on each compressor unit. However, on the demand side of the system, there are a number of areas that should be reviewed in the future in more detail, as they appear to be significant opportunities reducing air consumption.
- The overall usage in the full system today is on the order of 2000 to 2500 cfm. In the past, when there was a higher level of production at the site, the overall usage was larger. The results of the preliminary site survey suggests there are leaks amounting to at least 300 cfm that could be identified and repaired, which would reduce annual electric costs by over $22,000. There seem to be some tank agitation applications that could perhaps be powered by low-pressure air compressors or blowers rather than costly high-pressure air. These and other demand-side savings opportunities will be enumerated in the Level II Assessment if Watervliet Arsenal is selected as a NGEDAC demonstration site.

Current and Reconfigured System Baseline

The key characteristics describing the performance and economics of the current and proposed compressed air system are summarized in Tables F1, F2, and F3. The table was developed based on the data collected during the site visit and in discussions with plant personnel. The proposed system estimates are technically and economically conservative and reflect the observed performance of each compressor compared to load cycle.

Table F1. Surveyed air compressor performance characteristics.

Bldg	Unit	FL kW	FL acfm	% Load	Net cfm	Net kW
25	IR LLE-5	100.80	653	75%	490	80.64
20	ED 100	11.79	446	85%	379	74.68
35	(3) WN112	<< Off >>	—	—	—	—
110	XLE-2	100.51	687	100%	687	100.51
110	Joy TA18	<< Off >>	—	—	—	—
110	XLE-2	100.51	687	90%	618	95.48
125	WN112	<< Off >>	—	—	—	—
Total					2,174	351.31

Table F2. Estimated air compressor performance characteristics.

Bldg.	Unit[1]	FL kW	FL acfm	% Load	Net cfm	Net kW
25	IR LLE-5	100.80	653	27%	174	35.28
110	Joy TA18	353.37	2,000	100%	2,000	353.37
Total					2,174	388.65
1 Assumes all other units off.						

Table F3. Estimated energy costs — current and proposed systems (all shifts).

Performance Measure	Current Systems		Proposed NGED1 System		
	31Oct00	Typical Day	Electric	Natural Gas	Total
Average Air Flow (cfm)	2,174	2,174	674	1,500	2,174
Input Power (kW)	351.31	388.65	100	NA	NA
Operating Hours	8,760	8,760	8,760	8,760	8,760
Specific Power (cfm/kW)	6.18	5.56	6.74	NA	NA
Annual Energy Cost for Air ($/cfm/yr)	$127.57	$141.79	$116.97	$65.08 @ $4/MCF $97.63 @ $6/MCF	$81.38 @ $4/MCF $102.82 @ $6/MCF
Annual Energy Cost for Air ($/yr)	$276,972	$306,411	$78,840	$97,630 @ $4/MCF $146,445 @ $6/MCF	$176,470 @ $4/MCF $225,285 @ $6/MCF

[1]Proposed system includes 1,500 cfm natural gas engine drive and one existing IR 125 hp XLE operating at full load. Natural gas system parameters include 8,170 Btu/hp-hr, 341 BHP and 8,760 hours annually.

Observations of Plant Personnel

At current load, the Joy centrifugal 450-hp will "carry the plant" with some assistance from the IR LLE-5 (Building 25), which provides on the order of 100 cfm for several hours a day. Usage levels for the second and third shifts do not seem to fall much, probably due to high use of aeration air, vortex cooling, leaks, etc., which occur 24-hours/day. The estimated average system flow is approximately 2000 to 2500 acfm.

Observations of Audit Team

The main air supply was on and running during the plant survey period from 10:30 am to 1:00 pm on 31 October 2000. The system supply pressure was observed in the operating range of 83 to 85 psig. The pressure at the compressors was observed in the operating range of 90 to 100 psig. In most cases, the centrifugal unit is operated at full load. However, on the day of the visit, the centrifugal unit was not operating.

The blended electric rate equals $0.09/kWh. Average annual electric rates at the plant are $0.09/kWh. The actual plant electric cost for air production, as currently operated, is in excess of $300,000/yr. The load profile or demand of this system is relatively stable during all shifts. The full load operating range is 24 hours a day, 365 days a year, for 8760 hours a year. The system pressure appears to operate in the range of 83 to 85 psig at the headers during production periods. There are no flow meters in the system.

The standard performance measure used for this analysis is "electric cost/hr/loaded cfm" of air. Annual electric cost was selected as the key project evaluation factor, since it is a good overall indication of system costs and savings associated with potential measures. It is an quantitative number and not a subjective opinion, i.e., if the compressed air is used, these dollars are spent.

All paybacks are estimated using "Full Load Operating Efficiencies," which are very conservative. If the compressed air is not used, the compressor either shuts off or unloads. If it shuts off, there is a 100 percent saving of the electric cost. If it unloads, there is a 25 to 90 percent savings of the electric cost.

It is important to note that other recoverable compressed air costs should also be considered, e.g., maintenance, cooling water costs, and depreciation. Usually, the electricity cost is between 75 and 90 percent of the total "variable compressed air costs." Associated maintenance and other costs will be, in all probability, at least 20 percent or more of the identified electric cost. Existing plant records may already have these identified.

Energy Cost Baseline

Table F4 lists recent history of energy expenditures at Watervliet Arsenal.

Gas costs averaged $4.21/million Btu in Fiscal Year 1999 (FY99). This average was up about 10 percent over Fiscal Year 1998 (FY98) and by about 20 percent over Fiscal Year 1997 (FY97). These gas prices include $0.60/million Btu transportation costs. An estimate of $5/million Btu was used as the baseline for this assessment with $4 and $6/million Btu used as a sensitivity analysis. A $1 increase in gas price increases operating costs by about $25,000 for the NGEDAC.

Electric costs averaged $0.83/kWh during FY99. At the end of December 2000, a special contract that Watervliet had with NIMO expired. The net impact of this change will be an increase to $0.09/kWh as the average rate for Watervliet in moving forward. This level of impact was provided by Watervliet staff and confirmed by project staff. The value of $0.09/kWh was used in the project assessment.

Table F4. Energy cost summary.

Month	Electric			Natural Gas		
	Use (kWh)	Cost ($)	Rate ($/kWh)	Use (MMBtu)	Cost ($)	Rate ($/MMBtu)
FY-97	32,240,516	2,751,330	0.0853	32,963	116,433	3.5322
FY-98	31,404,550	2,322,902	0.0740	24,611	97,380	3.9568
Oct-99	2,573,874	185,904	0.0722	15,100	57,609	3.8152
Nov-99	2,363,428	179,699	0.0760	23,690	108,690	4.5880
Dec-99	2,430,821	178,911	0.0736	34,303	119,123	3.4727
Jan-00	2,705,661	213,373	0.0789	20,593	77,053	3.7417
Feb-00	2,417,303	183,645	0.0760	43,095	177,984	4.1300
Mar-00	2,501,397	191,840	0.0767	27,822	111,626	4.0121
Apr-00	2,344,946	193,010	0.0823	10,389	43,444	4.1817
May-00	2,557,299	216,039	0.0845	3,862	31,089	8.0500
Jun-00	2,646,735	273,949	0.1035	0	1,014	0.0000
Jul-00	2,371,271	212,614	0.0897	0	1,014	0.0000
Aug-00	2,938,037	275,736	0.0939	0	23,350	0.0000
Sep-00	2,503,698	218,487	0.0873	29	1,921	66.2414
FY-99	30,354,470	2,523,207	0.0831	178,883	753,917	4.2146

Supply-Side System Review

Primary Air Compressor Supply

The following is an overview of the compressed air supply system as observed on 31 October 2000.

Building 110

Units 110N and 110S are each 125-hp class Ingersoll Rand, two-stage, water-cooled, double-acting reciprocating XLE compressors. They are also of a continuous duty design. These are the most power-efficient air compressors at full load and when at part load to meet varying demand. They appear to be in good operating condition, although the inspection team did not perform any tear down inspection. There is no reason from a power efficiency standpoint to replace these units.

Unit 110 Center is a 450-hp Joy three-stage centrifugal (oil-free) TA18 compressor delivering 1850 to 2000 acfm at 100 psig at 450 bhp. This is a dynamic compressor, and actual air delivered and performance will vary with operating conditions. From a full load power efficiency standpoint, the TA18 is about the same as the XLE. However, the TA18 does not unload or meet part load demands as efficiently in "turndown" much below 25 percent when operating correctly. This unit is very power efficient from about 2000 to 1500 acfm flow. This TA18 is equipped with inlet guide vanes (IGVs), which allow almost "perfect turndown" from 2000 to 1500 acfm load. At flows below 1750, it will be less efficient, and at lower loads it will be very inefficient with the current installed control system. Other than a more efficient unloading central controller, there is no reason from a power standpoint to replace or modify this unit.

Because of its central location on the system, proximity to other compressor units, available physical space, and easy access to gas, Building 110 is the leading candidate as the site for the proposed NGEDAC system. Preferred location is probably along the south wall of building.

Building 125

Building 125 houses a Joy WN112 75-hp two-stage, double-acting, water-cooled compressor delivering 405 acfm at 100 psig at 77.3 bhp. This unit also appears to be in excellent shape and, according to plant personnel, runs very well. Even though it is an older unit (circa 1956), it is of the best designs for its type. There is no reason from a power efficiency or application standpoint that it should have to be replaced.

Building 35

Building 35 has three Joy WN112 compressors, the same as described above. One unit is a 75-hp (405 acfm @ 100 psig) and the other two are 100-hp (564 acfm @ 100psig). They all appear to be in good working order and well maintained.

Building 25

Building 25 houses a 125-hp Ingersoll Rand LLE-5 two-stage, double-acting, water-cooled compressor delivering 653 acfm @ 100 psig @ 125 bhp. This is the newest of the double-acting, water-cooled units and is of "leading edge technology." Key characteristics of this "balanced drive" include:

- extra large valve area—shorter lift—cooler running
- large cooling jackets
- built-in high performance intercooler and aftercooler.

As in the case of the rest of the double-acting units, the unit runs well and appears to be in good shape, and is very well maintained. There is no reason to replace this unit based on power efficiency. As in the case of the other compressors, it is continuous duty rated.

Building 20

Building 20 has a new Ingersoll Rand EP100 single-stage, lubricant-cooled, rotary screw air compressor. This unit is air cooled, but it is also continuous duty. The EP100 is obviously state-of-the-art and very conservatively applied. Its 100-hp motor is designed to run with a 1.15 service factor and the basic unit delivers 446 acfm at 125 to 135 psig at full load. It has been applied in the system very professionally with an operating band of 90 to 100 psig. This puts a load of 96.25 bhp or less on the 115-hp rated motor. It should do very well in the long run and, of course, save energy.

General Comments on the Air System

The above listed units are the main or primary air compressors used to support manufacturing and test operations at Watervliet. All but one (a rotary screw) are water-cooled units and each unit has its own polyglycol closed-cooling system. This use of available equipment is an excellent operational strategy and appears to be working well. This type of operation eliminates many of the problems associated with water-cooled units. The 450-hp Joy centrifugal has a closed-radiator-type system also, and according to plant personnel, it works well except for several hours a day during extremely hot weather (>90 °F). To alleviate this problem, there is a manually operated spray line set up to super cool when necessary. Centrifugal and rotary screws are more sensitive to cooling conditions in both useful life and performance than industrial reciprocating units. The sprayer is currently working. In the future, some consideration could be given to an automatically controlled high-performance secondary inline cooler between the radiator discharge and the compressor water inlet.

Buildings 133 and 40 have are some Worthington M-Line, single-acting, air-cooled reciprocating units which are not operating under continuous duty. These type units are not well suited to industrial production applications. They are rated very low in power efficiency. One of these is inoperable now; these units should be kept only for emergency backup air, if at all.

In addition to these 50- and 100-hp air-cooled units, there are at least nine 25-hp air-cooled Ingersoll Rand compressors in Building 15; one 15-hp air-cooled Wayne compressor in Building 120; and one 25-hp Champion (Speedair) compressor in Building 120. These types of units are well applied at or near the point of end-use production, particularly where higher than the 85 psig systems pressure is needed, to feed an intermittent demand. They are not continuous duty and should be applied on about a 50 percent duty cycle for normal life, operating, and maintenance costs. They are not particularly power efficient and should not be run in place of general system units unless higher pressure is required.

Well over 20, 5-hp and smaller air-cooled reciprocating compressors are set up on appropriately sized horizontal air receivers and refrigerated air dryers throughout the Arsenal. Most of these are not part of the control system and are separate from the main system air. Where a 5-hp or fractional-hp unit is run instead of the general air system, use of these units should be questioned unless it is for higher air pressure than the main system. These units are not even close in power efficiency performance to the main air system units.

There is also a Breathing Air compressor and system in Building 110 South and 123 for painting processes. These are well applied and only used when painting is in progress.

All units have their own local capacity control system and all, except the 450-hp Joy centrifugal, are set up to start automatically when air is needed and to shut off automatically when not needed. This control strategy appears to work very well and is a very positive step in air conservation already taken.

The primary compressed air supply is produced by relatively efficient air compressors that are capable of delivering the 100 psig full load pressure continuously. The units are well applied. They appear to be in good operating order and well maintained. Table F5 lists key characteristics of the units.

Compressor Capacity Controls

The two most effective ways to run air compressors are at “Full Load” and “Off.” Capacity controls are a means of restricting the output cfm delivered to the system while the unit is still running. This is always a compromise and it is never as efficient as full load on a specific power (cfm/bhp) basis.

Controls for Reciprocating Compressors

Reciprocating compressors are double-acting, water-cooled units with multi-step unloading. This is an efficient compressed air unloading system. Reciprocating multi-step unloading will efficiently translate percentage of “less air used” into almost the same proportional reduction in energy cost.

Table F5. Key performance characteristics by compressor type.

Performance Characteristics	Double Acting Recip (2 units)	Centrifugal (1 unit)	Double Acting Recip (2 units)	Double Acting Recip (2 units)	Double Acting Recip (1 unit)	Single-stage Rotary Screw (1 unit)
Brand	IR	Joy	Joy	Joy	IR	IR
Model	XLE	TA18	WN112	WN112	LLE-5	EP100
Air Capacity (acfm)	687	2,000	564	405	653	446
FL Press	100	100	100	100	100	100
FL kW @ 100 psig	100.51	353.37	88.69	64.79	100.8	77.79
Cfm/kW/100 psig	6.83	5.66*	6.36	6.25	6.48	5.73
Annual Electric Cost ($/cfm)	$115.43	$139.29	$123.92	$126.14	$121.66	$137.59
Annual Electric Cost ($/psig)	$396.25	NA	$349.61	$255.41	$397.35	$306.64
For more precise performance measures, see OEM curve or measure actual flow and input kW—compare in scfm, unit was down for repairs during the site visit. Data were obtained from plant personnel. Blended electric rate equals $0.09/kWh.						

The current system has two-step, free air unloading on the Ingersoll Rands and two-step total closure on the Joys. They are very responsive and power efficient. There are also newer electronic Intelysis controls on the IRs.

Controls for Rotary Screw Compressors

The two most common controls used rotary screw compressors are modulation and online/offline. Modulation is relatively efficient at very high loads—and inefficient at lower loads. Online/offline controls are very efficient for loads below 60 percent, when properly applied with adequate time for blow down. There are several other control types (e.g., "rotor length adjustment" or "variable displacement" and "variable speed drive") that have very efficient turndown from 100 percent load to about 60 percent load.

These controls must be installed correctly to operate efficiently. Piping and storage should be available close to the unit with no measurable pressure loss at full load to allow the signal to closely match the air requirements.

The current system has online/offline controls with an automatic electronic upper range modulator on the new IR rotary screw. It is very well applied and installed and appears to be working well.

Controls for Centrifugal Compressors

The two most common controls used for centrifugal compressors are modulation and blow off. Modulation is relatively efficient at very high loads, but will not work much below 75 percent load. After modulation or turndown, the compressor then just blows off excess air. The basic power draw at the blow off point then stays the same regardless of the load. The Watervliet unit uses these types of controls, and also uses IGVs to allow efficient turndown.

Today's modern electronic control systems can be applied to effectively close off the inlet and blow the unit down to idle, significantly reducing the kW draw. The Quad II control system installed now is somewhat limited, but the new Quad 2000 by Cooper (Joy) would do this with some system storage and piping modification. There is no reason to pursue this as long as the unit stays in base load and does go into continuing blow off.

The centrifugal units involved have capacity controls capable of translating "less air used" into a comparable reduction in electric cost. These controls will work effectively with current piping and the air receiver storage situation at today's conditions.

Long-Term Recommendation

With the system stabilized and balanced and with the primary air supply centrally located, consider a microprocessor-driven, centralized, full networking electronic control system. This will automatically place the most efficient machine online and assure use of no more than one partial loaded unit at a time. It will operate at a fixed system target pressure.

Air Treatment And Air Quality

Aftercoolers

Aftercoolers are mostly water cooled and appear capable of delivering 100 °F or lower temperature compressed air to the dryer during all seasons. The new rotary screw unit has a high performance air-cooled aftercooler.

Dryers

Refrigerated dryers require a refrigeration system to mechanically cool the air. The lowest possible consistent pressure dew point with a noncycling dryer is +40 °F. Cycling and variable speed-driven dryers not only save power (60 to 75 percent), but also can deliver a lower pressure dew point (down to 35 to 38 °F) when:

- air is delivered to the dryer at no more than 100 °F
- the condensate driven out of the aftercooler, prefilter, dryer and afterfilter is immediately removed from the system and not allowed to re-entrain or build up
- the dryer is not overloaded in volume (scfm).

Desiccant dryer regeneration equipment removes moisture vapor by “adsorbing” it to desiccant beads. These dryers can consistently deliver a pressure dew point to –40 °F or lower, which removes much more water than conventional refrigeration units. To regenerate the wet tower while the other tower is drying requires the use of heat in some form and some dry air to “sweep” or “purge” the exchanged moisture out. Desiccant dryers are usually rated at the same 100 °F inlet, 100 psig conditions. They also require:

- that air is delivered to the dryer at no more than 100 °F
- that the condensate driven out of the aftercooler, pre-filter, dryer and afterfilter is immediately removed from the system and not allowed to retrain or build up.

The current system has a refrigerated dryer on most of the air compressors, and they all appeared to be well applied and maintained. Those that were in use were running well. There are also two heatless, twin-tower regenerative dryers (670, 730 scfm each), which deliver dryer air to specific areas. These are also relatively well applied and, even though they use 15 percent purge air, they are equipped with new point removal purge controllers, which will usually reduce this by about 50 percent.

The centrifugal goes through a 2500 scfm rated Van Air internally heated twin-tower regenerative dryer, which is the most energy efficient type of dryer available except heat of compressors. It takes less intensive energy because of induction compared to the condition heating of the bead with other types and uses much less purge air. It is also equipped with a dew point demand purge controller.

Water or Oil Carryover in System

Water (condensate) and oil carryover problems in the current air system are not significant. The correct way to eliminate water and oil in the air system is to clean and dry the air immediately after it is produced in the compressor room. Then, clean dry air can be stored in a separate air receiver and delivered to the system, as required. Some guidelines for controlling oil and water carryover include:

1. Generally, it is best to eliminate the water and oil at the air source before it enters the air system.
2. Every 20 °F increase in temperature doubles the "moisture load" the compressed air will hold.
3. Compressed air dryers are usually capacity rated with 100 °F, 100 psig inlet air conditions. At 120 °F, 100 psig, the dryer's capacity rating is reduced 50 percent.
4. Putting "dry or oil free" air into the system 90 percent of the time and then allowing wet or oily air in sporadically 10 percent of the time will, in reality, make the system wet or oily all the time. The liquid water or oil will fall out in the piping system continuing to "re-entrain" and contaminate or collect in the "low spots" of the system, thus causing recontamination as air is pulled into the flowing compressed air system. A wet or oily system may well take many months of continuous flowing of clean dry air to "clean up."
5. Identify required pressure and dew point.

Pre-Filters and After-Filters

Pre- and after-filters are generally either particulate or coalescing type, and their use depends on the type of dryer being used and various installation considerations.

Desiccant dryers always require a high-quality coalescing pre-filter to keep liquid oil and water out of the drying tower. They also always require an effective particulate filter after the dryer to keep "desiccant dust" from migrating into the system.

Refrigerated dryers may or may not need pre- and after-filters depending on the piping, type of compressor, and desired degree of cleanliness. If the inlet air is apt to be dirty and fouled with carbon scale, etc., a particulate pre-filter is required. If the inlet air is liable to have significant liquid or heavy oil mist, a coalescing (or combination coalescing particulate) pre-filter may be needed. If oil or water mist is leaving the dryer, a coalescing after-filter may be in order.

Care in selection must be taken in all cases because:

- Wasted air pressure costs energy dollars.
- Wasted air pressure neutralizes the operating pressure band early.
- Standard coalescers will usually not perform effectively at flows much below 20 percent of their rated capacity.
- Standard coalescers life will be significantly shortened by particulate load
- Loose-packed, deep-bed mist eliminators (those with correct elements) will coalesce effectively throughout the total scfm range.
- Loose-packed, deep-bed mist eliminators (those with correct elements) have very high particulate load capability.

The pre- and after-filter(s) in this system are well applied and apparently well maintained.

Automatic Condensate Drains

The configuration and performance of condensate drains in the plant's system do not need to be modified. However, there still are some dual-timer drains that should ultimately be replaced with level-actuated ones.

Demand-Side System Review

It is the job of the main header system to deliver compressed air for production use from the compressor area to all sectors of the plant with little or no pressure loss—with 1 to 2 psig being a reasonable target. It is also desirable that the compressed air velocity in the main headers be kept below 20 fps to allow effective dropout of contaminants and to minimize pressure losses caused by excessive turbulence. The magnitude of the turbulence effect also depends on piping design and layout.

Basic System Header and Piping

Headers were checked at appropriate points with a single test gauge and there is a pressure loss of approximately 1 psig or less in the header systems. This indicates that the header system today can deliver the required air to any area without any significant pressure loss. Low-pressure problems encountered may be in the feeds from the header to the area.

Minimum Effective System Pressure

The system is currently running at 83 to 85 psig. However, there are additional direct power cost savings that will accrue from continuing to lower the overall system operating pressure. A steady delivered pressure to the system will allow follow-up programs at each process to establish the lowest effective pressure. This will enhance productivity, quality, and continue to reduce air usage.

The cornerstone of any effective demand-side air conservation program is to identify and operate at the lowest acceptable operating pressure at various sectors and operating units in the plant. This conservation program should be a part of any training, operating, and maintenance procedures.

Check Regulator

Some regulators are probably set at higher feed pressures than necessary for the process, with some regulators set for wide open to full header pressure. Arsenal personnel should always keep certain questions in mind. Is there a minimum effective pressure at operation established at the unit for each product run? If so, is it being adhered to?

In this type of operation, it is very important that the actual inlet pressure to the process be known and that the lowest effective pressure be held steady for the proper product quality. Installation of storage bottles downstream of the regulator may be needed to “close up” the pressure readings at rest and at operation.

Recommended Investigation

Determine whether regulators and regulated flow at process can be modified to reduce overall system pressure.

Compressed Air Condensate Handling

If (as site personnel indicated) understand that the condensate (in the condensate handling system) goes to water treatment, then discharge condensate should meet the requirements of the water treatment facility plant, there is no problem. However, if condensate is discharging to a storm sewer or in some other manner to ground water (Federal EPA minimum is 10 ppm), or is required to be separated it by the local water treatment facility, this practice should be investigated in detail.

Recommended Investigation

Review compressed air condensate handling system to ensure compliance with environmental regulations.

Leak Identification and Repair

With a plant of this type, an effective leak control program could save 300 cfm or the equivalent of repairing 100 leaks averaging 3 cfm each. On a percentage basis, this leak level is then about the same as leak levels in other plants. A leak level of 300 cfm translates into an annual loss of $30,000 in electric cost, at $100/cfm. A comprehensive leak management program could reduce such levels by 75 percent, saving up to $22,000 annually.

Recommended Investigation

Consider implementing a continuing leak identification and repair program with ultrasonic locators.

There should be a continuing cost minimization program in place. Generally speaking, the most effective programs are those that involve the production supervisors and operators working positively with the maintenance personnel. Accordingly, it is suggested that the program consist of the following:

Short Term

Set up a continuing leak inspection by maintenance personnel so that for a while, each primary sector of the plant is inspected once a quarter, or at a minimum, once every 6 months, to identify and repair leaks. A record should be kept of the findings and overall results.

Long Term

Consider setting up programs where the production people (particularly the operators and their supervisors) are positively motivated to identify and repair leaks. One method that has worked well with other operations is to monitor the airflow to each responsible section (perhaps with the use of recording the nonrecording flow meters) and to identify the air usage as a measurable part of the operating expense of that area. This usually works best when combined with an effective in-house training and awareness program. Table F6 lists costs and savings associated with implementing a leak management program.

Automatic Ball Valves

Some of the most significant areas for leaks in any high-production plant involve shutting off the air supply to machinery when not in use. When these opportunities are found, there are usually some very economical and easy methods to automatically shut off machinery air supply when not in use.

Table F6. Costs and savings associated with implementing a leak management program.

	Parameter	Cost / Savings
Costs	Leak detection equipment	$2,800
	Leak program development and detection equipment training	$1,000
	Leak repair (100 leaks @ $30 materials per leak and $50 labor per leak)	$3,000
Savings	Calculated electric savings from leak program	$22,000 per year
	Estimated number of leaks	100 leaks
	Estimated average leak size	3 cfm per leak
	Estimated leak level	300 cfm
	Potential value of leak reduction	$100 per cfm
	Estimated unit electric savings	$30,000 per year
	Recoverable leak losses	75%
	Total Program Cost	$5,000 plus $1,000 annually for ongoing repairs

Cabinet Coolers

Cabinet cooling is often required to obtain reasonable life and performance of the electronic equipment in control cabinets. There are various means of accomplishing this cooling: blowing compressed air into the cabinet, and by using vortex coolers, refrigeration units, or heat tube cabinet coolers. Blowing straight compressed air into the cabinet is generally very inefficient.

Vortex coolers can use chilled air with no moving parts and use less air. Vortex coolers should always:

- be regulated to the lowest effective pressure
- be equipped with the lowest possible flow generator
- be equipped with automatic temperature controlled shutoffs.

Refrigeration units should be carefully selected and equipped with automatic regulation control. Heat tubes are the most energy efficient when applied and can cool a "sealed cabinet." There are some cabinet coolers in use in the plant. These may all be replaced with "heat tube" cabinet coolers with a potential savings of 3.5 to 4 kW each.

Gas Engine-Driven System Assessment

This section provides a preliminary assessment of the opportunity for gas engine driven compressors at Watervliet Arsenal. The assessment is based on three key design factors:

- operating the new system as a fully hybrid system with the existing electric system
- meeting environmental requirements of the area
- improving the current demand system so that the air requirements for the new system are minimized, while reducing system operating costs for the Arsenal.

Gas Engine-Driven System Design Factors

The conceptual design for the gas engine-driven system is based on providing about two-thirds of the requirements of the main system at Watervliet. It will be configured as a hybrid system in conjunction with the existing electric system. In this way, the existing electric system can serve as a back-up to the gas engine system, if the gas system has a planned or unplanned shutdown or if the air requirements of the base are suddenly increased.

Using this approach, the Department of Defense can gain experience not only with operating a gas engine driven system, but also with integrating it with electric systems to improve overall compressed air system reliability and reduce operating costs. This flexibility is especially important given the increasing uncertainty associated with the price and supply reliability of most energy sources.

Environmental issues are expected to be minimal in this application given the key areas to address in any major project on this nature, particularly on the East Coast. The assessment is based generating up to 2.60 gm/bhp/hr for NOx and 1.75 gm/bhp/hr for CO.

Operating Cost Comparison

Table F7 displays the electrical energy costs for the Main Power House System for both the current system (with the centrifugal compressor operating) and the proposed NGEDAC system. Annual electrical costs are $306,000 for the current system and $210,000 for the modified system, a savings of $96,000 annually, based on the cost of gas at $5/Million Btu. Adding or reducing the gas cost by $1/Million Btu would change the savings level by about $25,000 annually.

A 2-year comprehensive maintenance contract is about $15,000 higher for the proposed system. Quoted maintenance contract levels range from $3.15 to $3.85/hr. The estimate is based on $75.00/hr and 17,000 hours for a 2-hour operation. The price includes all parts, fluids, scheduled and unscheduled maintenance. Net annual savings for the proposed system incorporating both the lower energy costs but higher maintenance costs are $81,000.

Capital Cost Assessment

Table F8 displays capital cost estimates for two configurations of the natural gas engine-driven systems: a Caterpillar G3408SITA engine and a Waukesha F18GLD engine. The capital costs include the catalytic converter for the Caterpillar engine, but not the Waukesha since it is a specially built lean machine. The costs include a placeholder estimate for all installation and freight costs. The capital costs also include a budget to erect special sound-proofing enclosure to reduce noise levels below resulting from the system even with the hospital mufflers.

Without consideration to potential cost reductions resulting from negotiating or utility rebates, the capital costs for the natural gas systems are on the order of $350,000 to $400,000.

Table F7. Operating cost comparison of current NGEDAC units.

	Current	Proposed NGEDAC
Average Air Flow	2174 cfm	2174 cfm
Operating Hours	8760	8760
Annual Energy Cost	$306,000	$210,000 (Gas @ $5)
Maintenance Contract	$15,000	$30,000
Total Energy and Maintenance Contract Costs	$321,000	$240,000 (Gas @ $5)

Table F8. Capital cost comparison of NGEDAC units.

	NGEDAC Unit 1	NGEDAC Unit 2
Model	GRS-300LW (Water-cooled)	GRS-300LA (Air-cooled)
Engine	G3408SITA (Caterpillar)	F18GLD (Waukesha)
Capacity	1480 cfm	1480 cfm
Discharge Pressure	100 psig	100 psig
Required Horsepower	341 BHP	362 BHP, incl fan
Operating Speed	1800-1300 rpm	1800-1500 rpm
Environmental Discharges (GMS/BHP-hr)	NOX 2.0 CO 2.0	NOX 2.60 CO 1.75 NMHC 0.75 HC 5.00
Fuel Consumption	7885 Btu/HP-Hr	7390 Btu/HP-Hr
Package	$330,000	$275,000
Catalytic Converter	$17,000	0
Soundproofing	$25,000	$25,000
Installation and Freight	$45,000	$45,000
Total Capital Costs	$417,000	$345,000

Appendix G: Compressed Air System Survey at Aberdeen Proving Ground

Overview of Facility

Base Mission

Aberdeen Proving Grounds is less than 100 miles northeast of Washington DC. The base is used for testing munitions and weapons systems, and consists of dozens of small to medium sized buildings. Numerous shops and facilities that can manufacture prototypes and repair various types of weapons systems are located here. No production manufacturing is done here; the primary task is building prototypes and testing.

Electric Use and Expenditures

The Base purchases electricity from Baltimore Gas and Electric on rate schedule P: Primary Voltage Service. The rate has seasonal demand charge variation and a year round energy charge components varied by time-of-day. Table G1 lists the rates.

Table G1. Baltimore Gas and Electric Rate Schedule P: Primary Voltage Service.

Monthly Maximum On-Peak Demand	Generation	Transmission	Total
Summer (1 June– 30 September)	$10.52/kW	$1.17/kW	$11.69/kW
Non-Summer (1 October– 31 May)	$4.65/kW	$1.17/kW	$5.82/kW
Electricity Usage Costs	**Energy**	**CTC***	**Total**
Summer On-Peak	$0.04069/kWh	$0.00522/kWh	$0.04591/kWh
Summer Intermediate Peak	$0.03059/kWh	$0.00522/kWh	$0.03581/kWh
Summer Off-Peak:	$0.01831/kWh	$0.00522/kWh	$0.02353/kWh
Non-Summer On-Peak	$0.02592/kWh	$0.00522/kWh	$0.026422/kWh
Non-Summer Intermediate (Peak)	$0.02382/kWh	$0.00522/kWh	$0.02902/kWh
Non Summer (Off-Peak)	$0.01548/kWh	$0.00522/kWh	$0.02070/kWh
*Competitive Transition Charge			

The rating periods are:

Summer On-Peak:	Mon-Fri., 10 am-8 pm, excluding national holidays
Summer Intermediate Peak:	Mon-Fri., 7 am-10 am, 8 pm-11 pm, excluding national holiday
Summer Off-Peak:	Times other than On-Peak or Intermediate Peak
Non-Summer On-Peak:	Mon-Fri., 7 am-11 am, 5 pm-9 pm, excluding national holiday
Non Summer Intermediate:	Mon-Fri., 11 am-5 pm, excluding national holidays
Non-Summer Off-Peak:	Times other than On-Peak or Intermediate Peak

Recent data (Tables G2 and G3) indicates APG annual electricity expenditures of approximately $7.5 million, with an average cost of electricity of about $0.058/kWh. The cost includes both electricity energy use ($/kWh) and peak electric demand charge ($/kW) components.

Natural Gas Rates

APG purchases natural gas on the spot market through the Defense Energy Support Center. Gas bills for the last 2 years are presented in the Tables G4 and G5. The data in the tables show that the cost of natural gas purchased by APG approximately doubled in price over the past year to about 8.32/MBtu. However, prices have started to come down due to a combination of seasonal factors and some increases in supply.

Table G2. Year 2000/2001 electricity use and expenditures.

Month/Year	Electricity Used (kWh)	Electricity Expenditures ($)	$/kWh
Feb 2001	10721459	548183	0.051
Jan 2001	11253643	570948	0.051
Dec. 2000	12469015	595756	0.048
Nov. 2000	10018058	522991	0.052
Oct. 2000	8807191	464873	0.053
Sep. 2000	11329680	815619	0.072
Aug. 2000	12151656	852869	0.070
July 2000	12025331	831178	0.069
June 2000	10793088	783014	0.073
May 2000	10060960	493749	0.049
April 2000	9668631	471393	0.049
March 2000	9857328	574696	0.058
Annual	129156040	7525270	0.058

Table G3. Year 1999/2000 electricity use and expenditures.

Month/Year	Electricity Used (kWh)	Electricity Expenditures ($)	$/kWh
Feb 2000	11195428	551733	0.049
Jan 2000	11468746	570970	0.050
Dec. 1999	11306690	540970	0.048
Nov. 1999	9388507	469026	0.050
Oct. 1999	9432475	458542	0.049
Sep. 1999	10783727	791456	0.073
Aug. 1999	11995238	766267	0.064
July 1999	14239764	946160	0.066
June 1999	10354947	777175	0.075
May 1999	9650005	454848	0.047
April 1999	9001787	446520	0.050
March 1999	11169500	551167	0.049
Annual	129986814	7324834	0.056

Table G4. Year 2000/2001 natural gas use and expenditures.

Month/Year	Natural Gas Used (Therms)	Natural Gas Expenditures ($)	$/Therm	$/MBtu
Feb 2001	505510	512856	1.015	10.15
Jan 2001	729305	1070729	1.468	14.68
Dec. 2000	465512	469944	1.010	10.10
Nov. 2000	548934	328960	0.599	5.99
Oct. 2000	247101	176273	0.713	7.13
Sep. 2000	108484	71832	0.662	6.62
Aug. 2000	86466	57744	0.668	6.68
July 2000	98138	68936	0.702	7.02
June 2000	97695	65567	0.671	6.71
May 2000	123492	65552	0.531	5.31
April 2000	375969	157324	0.418	4.18
March 2000	519550	205614	0.396	3.96
Annual	3906156	3251330	0.832	8.32

Table G5. Year 1999/2000 natural gas use and expenditures.

Month/Year	Natural Gas Used (Therms)	Natural Gas Expenditures ($)	$/Therm	$/MBtu
Feb 2000	877292	371747	0.424	4.24
Jan 2000	838720	376888	0.449	4.49
Dec. 1999	646763	233212	0.361	3.61
Nov. 1999	453316	198011	0.437	4.37
Oct. 1999	248030	96778	0.390	3.90
Sep. 1999	97047	48578	0.501	5.01
Aug. 1999	84976	41213	0.485	4.85
July 1999	76676	35875	0.468	4.68
June 1999	86708	19191	0.221	2.21
May 1999	94306	62793	0.666	6.66
April 1999	293012	96530	0.329	3.29
March 1999	754367	265639	0.352	3.52
Annual	4551213	1846453	0.406	4.06

Compressed Air Survey

On 23-24 April 2001 a compressed air system survey was conducted by Science Applications International Corporation (SAIC) and the U.S. Army Construction Engineering Research Laboratory (CERL) personnel. The purpose of the survey was two-fold:

To identify opportunities for reducing energy operating costs associated with the existing compressed air system

To evaluate the site as a candidate for a CERL-funded project to demonstrate the operation of a natural gas engine driven air compressor.

APG personnel interviewed during the survey included:

- Gary Testerman—Energy manager (DPW)
- Tom Vincenti— Facilities Manager (DPW)
- Jack Phipps (Bldg. 4600): HVAC Lead
- Rachael Swearingen—Environmental.

Small compressors distributed across the base provide most of the facility's compressed air. Many buildings on the Aberdeen Proving Ground, and the adjacent facility, the Edgewood Arsenal are not connected to the base natural gas distribution system. Since the objective of the survey was to focus on larger central compressed air systems that were in reasonable proximity to a natural gas line, this limited the survey to a few buildings. These were buildings 345, 4600, 338, 525, and 315. Of these buildings, 345 (boiler house), 525 (tank maintenance facility), and 315 (machine shop) were examined most closely, since they appeared to be potentially the most promising for a natural gas engine driven air compressor. With the exception of two buildings that are connected with an underground pipe, all of the buildings each have their own air compressor(s). The two buildings that are connected are buildings 315 (machine shop) and 345 (boiler house).

The survey involved a "walk-through" inspection of the facilities to obtain information with regard to major components (compressors, dryers, coolers, controls), distribution systems, and operational strategies. Information on utility rates, maintenance practices, and compressed air requirements (loads) was also collected. Spot measurements and/or readings of air flow, and compressor power consumption and loading/unloading intervals were taken to help quantify baseline air and power requirements.

Building 4600: Rodman Research Facility Compressed Air System Overview

Building 4600's compressed air load is primarily for building heating, ventilating, and air conditioning (HVAC) equipment and bench air for the various research projects in the many small laboratories. The air requirements vary widely based on the particular experiments that are underway at any given time. The facility has the following equipment:

Compressor:	Aerovac oil-free triplex direct-drive screw compressor (3-30 hp units)
Load served	Lab air
Pressure required	86-90 psig for lab air
Motor efficiency (nameplate):	90.2 percent
Loading	30 percent (typical)
Compressor	15 hp duplex two-stage reciprocating compressor
Load served	HVAC damper and valve actuators that runs
Pressure required	90 psig
Loading	50 percent (typical)
Compressor	25 hp duplex two-stage reciprocating compressor
Load served	General lab air
Motor efficiency	88.5 percent
Power factor	0.82
Loading	Very lightly loaded (not running during survey)

According to the APG staff, leaks were minimal, although it was acknowledged that inspection of the compressed air lines was difficult due to their location.

Although there is space for an NGEDAC outside the machine room in the parking lot, and gas is available in the building, there is no year round process use for heat (some boiler feed water in summer, but minimal). Building operating staff were also concerned about spilling water in the parking lot from an oil free compressor.

Building 338: Vehicle Maintenance Compressed Air System Overview

Two-20 hp two-stage Gardner Denver (model APMOMA9BJA9A) reciprocating compressors in the basement generate compressed air for this facility. One machine was running with a loud knock. The machine would load up for 34 seconds out of 230, or 15 percent. APG staff should inspect the engine to make sure that significant damage will not occur. After reviewing the compressor operation, the nearest gas line was seen to be about 1500 ft away, and the trenching would have to cut through a concrete apron 3-ft thick. This alone precluded further consideration of the building for an NGEDAC.

Building 525: Tank Maintenance Compressed Air System Overview

Building 525 has three new 30 hp, 122 scfm @125 psig Ingersoll Rand screw compressors that are piped together, networked and controlled by a single sequencing controller (Figures G1 and G2). The units have heat recovery for space heating. Facility staff indicated that sometimes more than one compressor was needed to carry the load, and sometimes they even used all three. During our visit, only one machine was partly loaded, and one of the three machines had a major leak. Typical operation is one shift only. Facility staff reported that for a few weeks leading up to armed forces day, the paint shop is very busy. The rest of the year the load is low. On the second day, the leak was traced to a missing hose clamp on the drain line internal to the unit, which the Scales Air Compressor representative, who was part of our survey team, repaired. Given that the leak was so loud that researchers could hear it halfway across the plant even when the compressor room door was closed, it was probably 20 scfm or more. The output of the machine that was running was 26 scfm (21 percent of 122 scfm), There were no other apparent uses of air the day of the site visit, so it may be concluded that the leak was the load. By the time researchers repaired the leak, there was no time to revisit the machine and time the loading/unloading cycle. Assuming the leak was 20 scfm, and had not been repaired, the leak would have cost the facility about $1000/yr.

Figure G1. Building 525: 30 HP IR compressor.

Figure G2. Building 525: dryer and receiver.

Additional compressor information

- Quantity: three
- Size: 30 hp
- Ultra-air dryers
- 109 psi on control panel
- Heat recovery (ducts to spill air to room or outdoors)
- Gas: right outside building
- No thermal loads other than space heating.
- Usually one shift operation, although they might sometimes run evenings or weekends

Air uses

- Paint booth
- Hand tools (grinders, polishers, etc.)

Performance of Compressor

Table G6 lists the performance characteristics of the subject compressor.

Table G6. Performance characteristics of the Ingersoll Rand SSR-EP-30SE compressor.

				scfm/kW			
SCFM	Brand	Type	Year	100% Load	75% Load	50% Load	25% Load
122	Ingersoll Rand	Screw model SSR-EP-30SE	1998	5.3	4.87	3.95	1.97

Compressed Air Load Profile

There was no reliable information about the loading of the compressors. However, based on our observations, it appears that the compressors are lightly loaded; and on average only one unit operating at 30 percent load (36 scfm) is in use. Typical annual operation is first shift only—9 hr/day or 2340 hr/yr.

Building 315: Machine Shop

Building 315 has a 10-year-old, 50 hp, 230 scfm (estimated), direct drive screw compressor that leaks oil (Figure G3). Facility staff indicated that the machine is virtually problem free. Although no air dryer was observed on the system the operator indicated that the only time there was a moisture problem is if they were being fed from Building 345.

Figure G3. Building 315: 50 hp Rotary-Aire screw compressor.

Additional compressor information

- Quantity: one
- Size: 50 hp
- No dryer
- 40 percent load on control panel (gauge suspect)
- 104 psi on receiver
- 140 psi sump pressure (could indicate a plugged separator or a broken gauge)
- High discharge temperature
- Plugged radiator
- No heat recovery
- No thermal loads other than space heating.
- Usually one shift operation, although they might sometimes run evenings or weekends

Air Uses

- Machine tools (lathes, drill presses, milling machines)
- Glass bead blasting booth
- Hand tools (cut off saw, grinders, polishers, etc.)
- Blow off hoses

Compressed Air Load Profile

Facility staff reported that occasionally they are busy and a few people will need to stay for an evening, but usually the load is low. Typical first shift operation is 7:10 to 16:10 (9 hours/week day), although perhaps twice a week the building is used until 20:10 (13 hr/weekday). Based on this information, it can be concluded that, on average, the machine is providing about 90 scfm during this period (2340 hours/yr- 2756 hours/yr).

Performance of Compressor

During our visit, the machine was only partly loaded (40 percent) according to a gauge, and one of the milling machines, number 52 “Lucas” (S42B-84) had an audible leak. The measured amps and voltage from the control panel were as follows:

- 227V, 75A
- 227V, 75A
- 226V, 81A

Assuming a 0.85 Power Factor, the readings indicate an input power requirement of about 26 kW, an electric power input of about 60 percent of design power. This implies the machine was generating compressed air at about 40 percent of the design output, or about 90 scfm. Researchers attempted to load up the machine by operating the bead blaster, but there was no appreciable effect on the capacity gauge, although the receiver pressure did drop about 4 psi. Table G7 lists the performance characteristics of the subject compressor.

Table G7. Performance characteristics of the Gardner Denver compressor.

				scfm/kW			
SCFM	Brand	Type	Year	100% Load	75% Load	50% Load	25% Load
230 (est.)	Rotary Aire	Direct drive 50 hp air cooled screw	1991	5.3	4.87	3.95	1.97

Building 345: Boiler Plant

Building 345 has a 5-year-old 25 hp Gardner Denver belt drive screw compressor (Figure G4). Facility staff indicated that the machine is hardly ever run, except when the boiler is burning oil, at which time the compressor is needed for atomizing air. The air dryer on the system looks like it has not operated in years (Figure G5). The drains are not piped, and there is no evidence of watermarks on the floor under the drains. The representative from Scales Air Compressor observed that the drain trap arrangement is virtually ineffective, and that the receiver needs a relief valve to meet code.

Figure G4. Building 345: 25 hp GD compressor.

Figure G5. Building 345: ineffective air dryer.

There was a slight hiss from the air receiver, which was being fed by a 1-1/2 hp reciprocating machine upstairs. The 1-1/2 hp machine is all that is needed for the boiler controls when they burn natural gas, which is most of the time. The day of the site visit, the boiler was shut down, and the small compressor was lightly loaded.

Additional compressor information

- Quantity: one
- Size: 25 hp
- Dryer was antiquated and obviously not in use
- Machine was operating in its most efficient mode (shut off)
- Currently heat recovery (dumps compressor heat in confined room in basement)
- Could serve bldg. 345 through existing 4-in. pipe

Thermal loads: could use waste heat to preheat boiler feed water in the winter. Waste heat could also be used to preheat boiler combustion air also wintertime load. Boilers already have stack economizer (possibly for preheat of feed water). Boilers run all winter (15 October to 15 April nominally). A portion of that time they may burn oil

Air uses

- boiler controls
- atomizing air

Compressed Air Load Profile

Given that the elapsed hours of operation reading was 31,685 and the compressor is 1992 vintage, and that the compressor operates an estimated 3520 hours/yr. This compares reasonably well with an assumption of 24 hr/day operation during the heating season. The assumption is that the compressor runs at its design load and provides about 125 scfm.

Performance of Compressor

Table G8 lists the performance characteristics of the subject compressor.

Table G8. Performance characteristics of the Gardner Denver compressor.

				scfm/kW			
SCFM	BRAND	TYPE	YEAR	100% Load	75% Load	50% Load	25% Load
125[1]	Gardner Denver	Belt drive 25 hp air cooled Screw	1997	5.3	4.87	3.95	1.97

[1]The full load scfm is based on 5 scfm/hp. Assumed scfm/kW performance similar to the 50 hp Rotary-Aire screw compressor in Building 315.

Compressed Air Energy Use and Energy Operating Costs

Table G9 summarizes the energy operating costs of the compressors at APG based on typical capacity and performance assumptions. The total number of compressors was obtained from data sheets provided by APG personnel. For compressors under 20 hp, it was assumed reciprocating units predominated and are controlled using an on/off scheme. For compressors over 20 hp it was assumed screw units predominated and were controlled and performed in a fashion similar to those surveyed. Total compressor energy operating costs are estimated to be $629,509 based on 10,853,611 kWh of energy use per year.

Compressed Air System Operational Cost Cutting Opportunities

Summary of Opportunities

Generating compressed air for APG costs $629,509/yr, or nearly 10 percent of APG's electricity expenditures. A number of opportunities were examined as a means of reducing these expenditures. The most significant opportunity for savings is from reducing compressed air leaks—an estimated $94,426 annually in electricity expenditures. This is described more fully below.

Table G9. Compressed air energy use and energy operating costs for compressors.

	Under 20 HP	20 HP or Larger	Total
Total Number of Compressors	370	29	
Average HP	10	41.7	
Total HP	3700	1208	
Design SCFM/HP	5	5	
Total SCFM at Design Load	18500	6040	
Supply Efficiency at Design Load (scfm/kW)	N/A	5.3	
Load Factor	N/A	0.5	
Total HP at Load Factor	N/A	838.5	
Total SCFM at Load Factor	N/A	3,020	
Supply Efficiency at Load Factor (scfm/kW)		3.95	
Total Input Power (kW)	3373.6	764.6	
Annual Hours of Operation	8,760	2,600	
Fraction Hours Compressor On	0.3	1	
Energy Use (kWh)	8,865,762	1,987,848	10,853,611
Total Energy Cost ($)	514,214	115,295	629,509
Unit Energy Cost ($/kWh)	0.058	0.058	
($/scfm)	58.70	38.18	

Minimize Compressed Air Leaks

There are a number of things staff can do to reduce the amount of air used. Changes such as fixing leaks and using venturi blow-off nozzles, are important because the sum of their reductions taken together adds up to a large reduction.

Reduce Air Leaks

Although the walk-through survey did not involve measuring specific leak rates, no obvious leaks were seen. Research indicates that facilities with an aggressive leak control program can reduce the leakage rate to 5 or 10 percent. Facilities without a leak control program commonly have a leakage rate that exceeds 40 percent. For the entire base, information collected from base personnel indicates that there are approximately 400 air compressors totaling approximately 4600 hp. Many of these operate 24 hours feeding air to HVAC control systems for the operation of dampers, valves, and other control devices.

To estimate the benefit of repairing the leaks, the small compressors (less than 20 hp) are conservatively estimated to be on 30 percent of the time, and fully loaded during the on period. For the larger compressors, an average loading of 50 percent was assumed during all hours of operation—for the most part first shift operation, for industrial operations, and longer for any HVAC operations. The leakage rate was also assumed to be currently 30 percent, and could be cut in half, to 15 percent of supply air—by implementing a vigorous leak elimination program. This would result in savings of approximately $94,426, based on a 1.628 million kWh reduction in electricity use. This is based on a one for one reduction of electricity use with a reduction in leakage—savings = (0.30—0.15) x Total Energy Used by Compressors.

To effectively capture these savings, it is not enough to find, tag, and fix the leaks. For screw machines with a throttled inlet, you must also modify the controls to operate the machine in a load/no load mode, or the power reduction will be only a small fraction of the air flow reduction.

An aggressive leak control program to reduce air waste requires several tactics:

- Institute a plant wide compressed air training and awareness program
- Institute a split tag leak reporting system.
- Priority A leaks are those audible to the human ear during production.
- Priority B leaks are those that are audible to the human ear during shutdown.
- Priority C leaks are those that can be detected with an ultrasonic leak detector.

- Purchase an ultrasonic leak detector and periodically (once or twice a year) check all parts of the compressed air system during production, and tag, log and repair any leaks found.
- Wire every production machine and HVAC device with a solenoid valve to cut off the air to the machine when it is shut down.

See the following suggestions for a more complete description of a comprehensive leak reduction program

Suggestions for an Aggressive Leak Control Program

An aggressive leak control program requires several tactics be employed simultaneously:

- Institute a plant-wide training and awareness program to inform production and maintenance staff of the cost of leaks and inappropriate uses of compressed air. Let them know what you are doing to reduce the compressed air costs, and let them know what you need them to do.
- Institute a split tag leak reporting system. Brightly colored, perforated, wired tags are issued to production, maintenance, and security personnel. Anyone who hears or sees a leak should initial, date and attach the tag, noting on the tear-off portion the location and severity of the leak. The tear-off portion is then turned in to maintenance, where the leak is logged and prioritized for repair.
- Priority A leaks are those audible to the human ear during production since they may be 20 scfm or more. Repair them as soon as possible.
- Priority B leaks are those that are audible to the human ear during times when the production machines are not running, and should be fixed within a week.
- Priority C leaks are those that can be detected with an ultrasonic leak detector. Check hoses, filters, regulators and lubricators, quick-connect fittings, and any screwed connections. Repair Priority C leaks within a few weeks.
- Purchase an ultrasonic leak detector and periodically (once or twice a year) check all parts of the compressed air system (during production) and tag and log the leaks for repair. The ultrasonic leak detector can "hear" leaks that are out of the range of human hearing. Ultrasonic leak detectors range in cost from $150 to $15,000. For your purposes, a $150 model will probably work just fine.
- To make the leak detection and repair tasks more manageable, you might break the plant down into a half dozen production zones, and then check one zone every month on a rotation that you can work into your preventive maintenance schedule. As an alternative, you could hire a local company to periodically perform a leak survey for you. The disadvantage of hiring someone

to find and tag the leaks (besides the cost) is that you get a large number of leaks being reported for repair all at the same time.

- Each machine should have a switch wired to a solenoid valve to cut off the air when the machine is not in production. Operators need training and reinforcement to shut off the air when the machine is shut off. Better still, interlock it electrically, so that when the machine shuts off, the valve shuts also, unless overridden (temporarily only) for machine set-up. As an alternative to this approach, divide the production areas by zones or departments so that each zone or department is controlled by an automatic shut off switch. When there is no production in the zone, turn off the air.

Potential for Natural Gas Engine-Driven Air Compressor

Site Suitability

APG has a limited number of buildings that have close proximity to natural gas and compressed air loads of sufficient capacity for the purposes of demonstrating a natural gas engine-driven air compressor. The survey of candidate buildings determined that the larger machines (25 hp—50 hp) generally operated substantially below design load during normal operation. Furthermore, normal operation for these compressors is estimated to be no more than 2600 hours/yr on average.

Economic Analysis

The primary difference between a natural gas engine driven compressor and an electric motor driven compressor is the technology that drives the compressor. The compression process remains unchanged.

The natural gas engine driven system is more costly to purchase than an electric motor driven system. In addition, the maintenance on the engine is greater than the maintenance on the electrical motor. The benefits of the natural gas engine include the ability to use natural gas as an energy source, the ability to reclaim large amounts of waste heat from the natural gas engine, and the independence of the energy source from the electrical system.

To justify the additional expense of the natural gas engine, the application requires a significant benefit from fuel savings. Assuming 30 percent natural gas engine efficiency, and a 92 percent electrical motor efficiency, with comparable compressors, the ratio of NGEDAC costs to electric motor driven air compressor energy costs would need to be about 1/3 or less (e.g., 30/92 percent), to begin to realize savings in energy operating costs. At the most recent APG rates, the ratio is $8.32/MBtu gas/$17.00/MBtu electric—about ½. If gas prices were to return to their previous levels of about $4/MBtu, there would be some savings. However, the additional maintenance cost would need to be factored in as well. Typically, this is on the order of $0.01/hp-hr to $0.02/hp-hr, depending on the capacity of the unit and the operating hours. This would be for engine maintenance alone. For example, assuming a 25 hp NGEDAC operating 2600 hours/yr $0.01/hp-hr would equate to about $650/yr of added maintenance costs.

Recovery of waste heat from a NGEDAC offers an opportunity to for additional savings. The amount of heat that could be captured is approximately 22 percent of the heating value of the natural gas, and fuel savings would be about 28 percent, assuming the recovered heat offsets fuel required for an 80 percent efficient boiler (22%/0.8). Using the APG cost of $8.32/MBtu for natural gas the cost savings from heat recovery would be valued at $2.33/MBtu (8.32 * 0.28). For a 25 hp NGEDAC operating at design load, fuel input would be about 0.23 MBtu/hr and annual gas use would be on the order of 494 MBtu for 2600 hours of operation. The savings from heat recovery would be $1151, assuming the waste heat could be used (2.33*494). At lower gas prices, the heat recovery savings would be reduced proportionally.

Table G10 lists the economics based on typical part load operation observed during the survey, assuming a 25 hp NGEDAC replacing a 25 hp electric motor driven air compressor. Similar results would hold for a 30 hp or 50 hp unit—the capacity of the other units surveyed. Note that heat recovery is shown to indicate maximum benefits. Only limited heat recovery opportunity (e.g., space heating) was observed (Table G11).

As discussed previously, the economics currently look unattractive. Natural gas at $4/MBtu would give energy cost savings, but the results would still be marginal ($951 net savings with heat recovery and $36 without heat recovery) due to the added maintenance costs for the engine.

Table G10. Compressor performance characteristics at design load.

	Electric Air Compressor	NGEDAC
Compressed Air Capacity	120	120 scfm
Motor/Engine Power	25 hp (27.5 bhp)	25 hp (27.5 bhp)
Full Load Power	22.8 kW	.233 MBtuh
Efficiency	5.3 scfm/kW	515 scfm/MBtuh

Table G11. Annual energy use and operating costs at typical load (50% design load) baseline energy price assumptions.

	Electric Air Compressor	NGEDAC	Net Savings
Energy Use	39,000 kWh		39,000 kWh (elec.)
		394 MBtu gas	-394 MBtu gas
		- 110 MBtu gas (engine heat recovery)	110 MBtu gas (engine heat recovery)
Energy Operating Costs[1]	$2,262	$3,278	-$1016
Operation & Maintenance Costs	$600	$1,250	-$650
Heat Recovery Costs	0	-$915	$915
Total Costs (w/heat recovery)	$2,862	$3,590	-$728
Total Costs (w/o heat recovery	$2,862	$4,528	-$1,666

[1] Average natural gas price of $8.32/MBtu and average electricity price of $0.058/kWh, including energy and demand charge components. This covers the period 3/00-2/01.

Appendix H: Compressed Air System Survey at Lake City Army Ammunition Plant

Overview of Facility

Base Mission

The Lake City Army Ammunition Plan (LCAAP) is currently the only active small caliber ammunition manufacturing facility within the Department of Defense, and produces 5.56mm, 7.62mm, caliber 0.50, and 20mm ammunition. LCAAP employs about 750 people, under an operations contract with Alliant TechSystems (ATK). Compressed air is one of the primary energy input streams into the production process.

Energy Use and Expenditures

Electric Rates and Consumption

Recent data indicates LCAAP annual electricity expenditures of approximately $2 million, with an average cost of electricity of about $0.049/kWh. The cost includes both electricity energy use ($/kWh) and peak electric demand charge ($/kW) components. Demand related charges account for about 20 to 25 percent of the total electricity expenditures (Table H1).

The current rate schedule is a declining block type rate structure where the unit electricity charges decrease as the usage goes up. The rates also vary seasonally, although not based on time-of-day, with higher charges in the summer vs. winter (Table H2).

Table H1. LCAAP annual electricity expenditures.

Month / Year	Electrical Energy Used (MWh)	Peak Electric Demand (kW)	Expenditures ($)	Effective Electricity Rate ($/kWh)
Apr 2000	2814	8422	$127,193	0.0452
May 2000	3528	8770	$189,806	0.0538
Jun 2000	3738	9324	$199,983	0.0535
Jul 2000	3402	10130	$197,316	0.0580
Aug 2000	4116	10307	$218,148	0.0530
Sep 2000	3318	9601	$175,854	0.0530
Oct 2000	3612	8795	$158,928	0.0440
Nov 2000	3360	9173	$155,568	0.0463
Dec 2000	3276	9173	$153,644	0.0469
Jan 2001	4116	9223	$172,460	0.0419
Feb 2001	3444	9349	$158,768	0.0461
Total	38724		$1,907,669	
Average	3520	9297	$173,424	0.0493
Average demand factor = monthly kWh / (demand times hrs in a month) or 52%.				

Table H2. LCAAP current rate schedule.

Electrical Demand Block (kW)*	May 16—September 15 Demand Charge($/kW)	September 16—May 15 Demand Charge ($/kW)
First 2541 kW	6.883	4.678
Next 2541 kW	5.507	3.652
Next 2541 kW	4.613	3.222
Over 7623 kW	3.367	2.479
Electrical Energy (hrs)*	***($/kWh)***	***($/kWh)***
First 180 hours	0.04311	0.03655
Next 180 hours	0.02999	0.02727
Over 360 hours	0.02151	0.02131
* Source: Kansas City Power & Light, Schedule LPS **hrs = (Monthly Electrical Energy Use/Monthly Peak Electric Demand)		

Natural Gas Rates

Natural gas has doubled in price over the past year, although prices have started to come down due to a combination of seasonal factors and some increases in supply (Table H3).

Table H3. Natural gas prices April 2000—February 2001.

Month / Year	Natural Gas ($/MMBTU)
Apr 2000	3.26
May 2000	3.52
Jun 2000	4.83
Jul 2000	4.79
Aug 2000	4.38
Sep 2000	5.27
Oct 2000	5.84
Nov 2000	5.32
Dec 2000	6.89
Jan 2001	11.22
Feb 2001	7.02
Average	***5.78***

Compressed Air Survey

On 5 April 2001 a compressed air system survey was conducted by Science Applications International Corporation (SAIC) and the U.S. Army Construction Engineering Research Laboratory (CERL) personnel. The purpose of the survey was two-fold:

To identify opportunities for reducing energy operating costs associated with the existing compressed air system

To evaluate the site as a candidate for a CERL-funded project to demonstrate the operation of a natural gas engine driven air compressor.

LCAAP staff that were interviewed during the survey included:

- Gregg Ashley—Facilities Engineering
- Allen Fails—Senior Plant Engineer, Facilities
- Burton Volkers—Manager, Facilities Engineering

The focus of the survey was on the largest compressed air systems—in particular, the compressor systems that served Building 1 and Building 3. The survey involved a “walk-through” inspection of the facilities to obtain information with regard to major components (compressors, dryers, coolers, controls), distribution systems, and operational strategies. Information on utility rates, maintenance practices, and compressed air requirements (loads) was also collected. Spot measurements of air flow, and compressor power consumption were taken to help quantify baseline air and power requirements.

Building #3 Compressed Air System Overview

The compressed air system on Building 3 is the smaller of the two compressed air systems evaluated. The Building 3 system consists of two screw compressors, a receiver, a refrigerated drier and air distribution system. Figure H1 shows the layout. Figures H2 to H6 show details of the compressed air system. The air compressors are identical and have design air flow capacities of 1000 cfm each. Specific nameplate data on the major components follows.

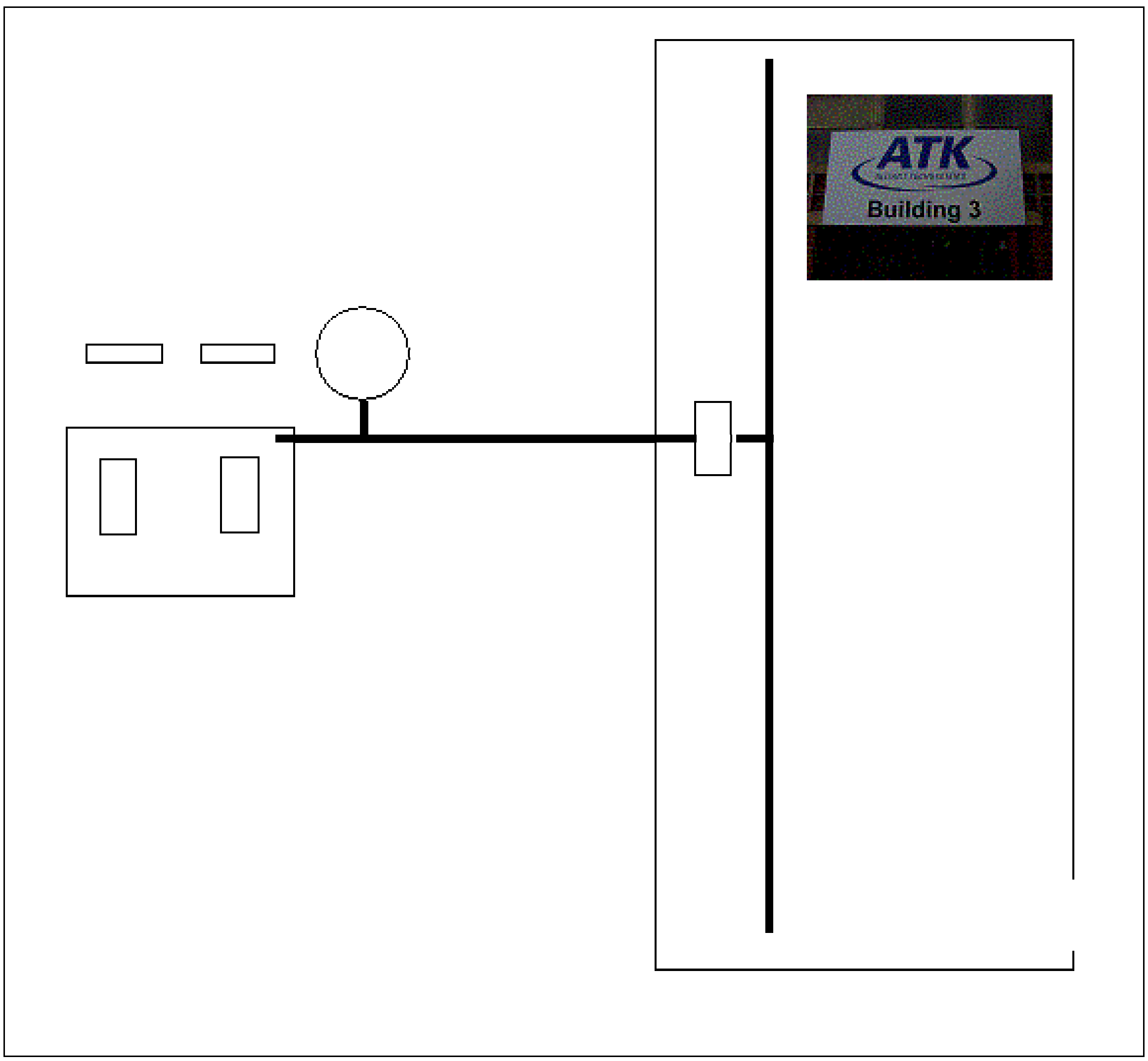

Figure H1. Compressed air supply for Building 3.

Figure H2. One of two 200 Hp screw compressors that supplies building #3.

Figure H3. Inter and after coolers for compressors supplying building #3.

Figure H4. Building #3 air storage tank outside of compressor building.

Figure H5. Refrigerated compressed air dryer located inside building #3.

Figure H6. Example section of Building #3 compressed air distribution piping.

Compressors #1 & #2

Manufacturer:	Gardner-Denver	
Model #:	EAUQPC	
Motor:	200 HP	
RPM:	1741	
Pressure:	Minimum:	65 psig
	Maximum:	100 psig
Electric:	460 VAC/3 phase/60 Hz	
FLA:	238 amps	
Power Factor:	90	
Date of Manufacture:	8/85	

Refrigerated Drier

Manufacturer:	Pneumatch
Model:	AD-2000
Rated Air Flow:	2000 cfm
Electric:	460 VAC/3 phase/60 Hz
Compressor Motor:	10 HP
Refrigerant:	R-22

Measurements and operational readings were taken to determine performance (Table H4).

Table H5 lists the efficiency of the compressor in terms of air supplied (cfm) per unit of electric power input (kW).

Table H4. Measurements and operational readings for Pneumatch refrigerated drier.

Parameters	Compressor #1:		Compressor #2:	
Hours of Operation:	30,892		25,540	
Leaving Air Pressure:	105 psig		108 psig	
Leaving Air Temperature	90 °F		85 °F	
Voltages	L1—L2	468 volts	L1—L2	470 volts
	L1—L3	469 volts	L1—L3	469 volts
	L2—L3	470 volts	L2—L3	470 volts
Currents	L1	177 amps	L1	235 amps
	L2	185 amps	L2	235amps
	L3	180 amps	L3	232 amps

Table H5. Efficiency of the compressor in terms of air supplied (cfm) per unit of electric power input (kW) (efficiency = scfm/kW).

cfm	Brand	Type	Year	100% Load	75% Load	50% Load	25% Load
1000	Gardner-Denver	Screw with Turn Valve	1985	5.70	5.24	4.25	2.13

Building #3 Compressed Air Load Profile

Based on conversations with the Facilities Engineering Staff, the load profile shown in Figure H7 represents the best estimate of the compressed air load profile Monday through Friday. The compressors are shut off Saturday morning at 3:30 am and are off until approximately 6:00 am on Monday morning.

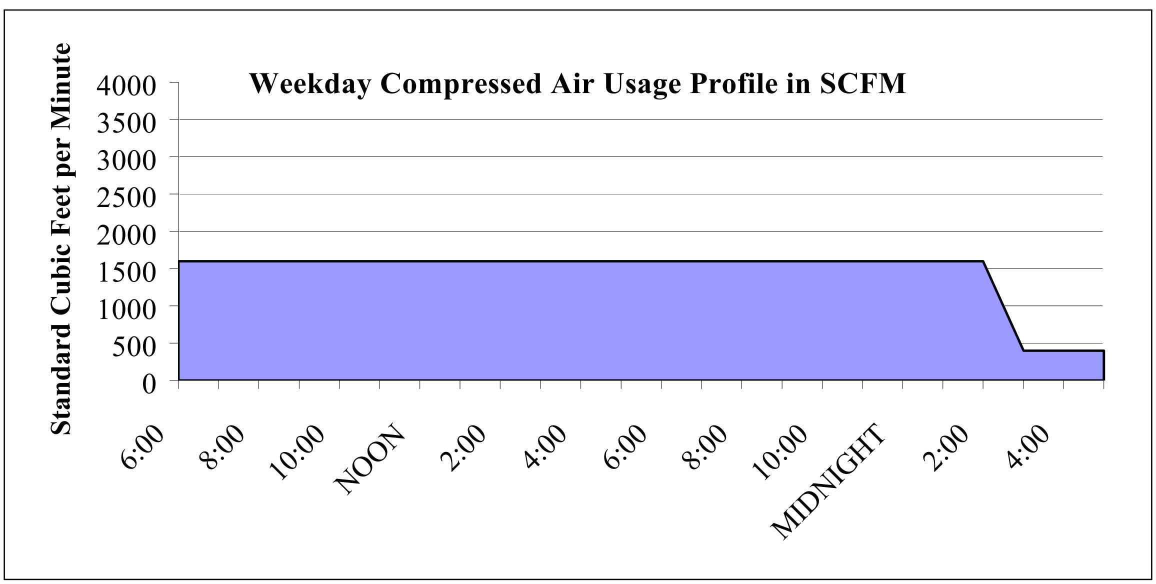

Figure H7. Estimate of the compressed air load profile Monday through Friday.

Building #3 Air Compressor Controls

The two screw compressors that supply Building #3 were operating independently of each other during the day of the site visit. Based on amperage readings taken during the site visit, one of the compressors was fully loaded (234 amps), and the other compressor had a motor load of approximately 75 percent (180 amps). The screw compressors have turn valve part load control, which provides economical operation above 50 percent load.

Building #3 Generation and Distribution Pressure/Condensate and Oil Elimination

The compressed air system is reported to operate as desired with no condensate or oil carryover identified as a problem. Pressure drop through the refrigerated dryer was only 4.0 psig, and pressure drop from the outlet of the refrigerated dryer to end use points was undetectable. These observations lead to a conclusion that the system piping is properly sized and does not contain any significant restrictions.

Building #3 Compressed Air Energy Use and Energy Operating Costs

Table H6 lists the key energy—related operational information for the two units. The operating assumption is that one unit operates at full load (1000 scfm), while the other unit operates at part load (400 scfm—600 scfm). The energy cost information is based on the $0.049 average rate. Note that this does not include nonenergy operation and maintenance costs, which would increase the totals shown.

Building #1 Compressed Air System Overview

The compressed air system on Building 1 is the larger of the two compressed air systems evaluated. Figure H8 shows the system layout, and Figures H9 to H14 show system details. The Building 1 system consists of eight screw compressors, two receivers, two refrigerated dryers and air distribution system. The air compressors are identical, with design capacities of 2580 cfm at 100 psig, assuming 550 bhp input power requirements. The motor purchased with the system is 600 HP, which exceeds these requirements. Consequently, the motor is typically loaded to 92 percent of its rated design capacity when the compressor is providing the design air flow.

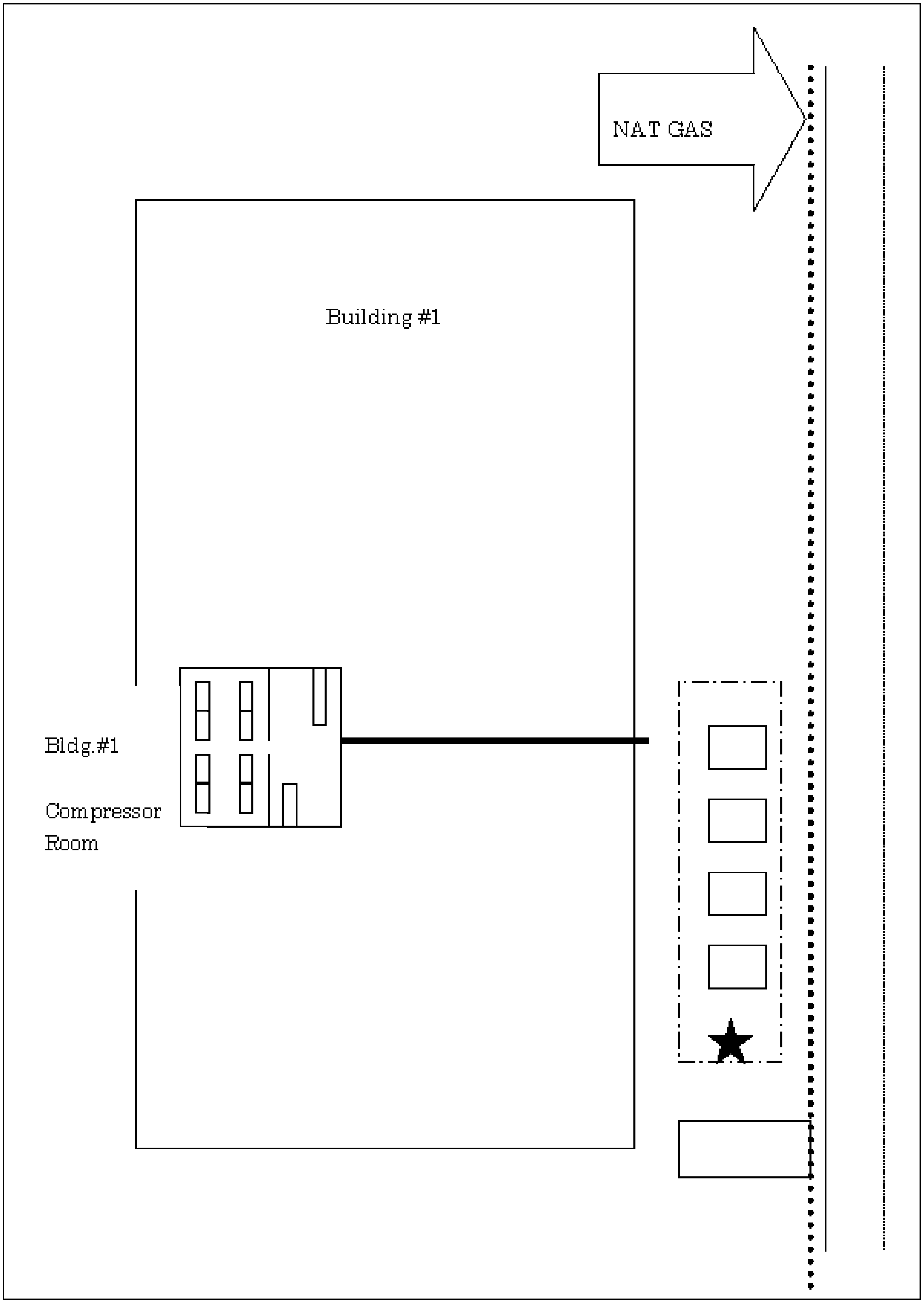

Figure H8. Compressed air supply for Building 1.

Figure H9. One of eight 500 hp screw compressors that supplies Building #1.

Figure H10. Inter and after coolers for compressors supplying Building #1.

Figure H11. Air storage tank for Building #1 compressor room.

Figure H12. One of two refrigerated compressed air dryers for Building #1.

Figure H13. Oil heat recovery in Building #1 Compressor Room.

Figure H14. Potential outdoor site for natural gas engine driven air compressor.

Table H6. Key energy-related operational information for the two units in Building 3.

	Compressor at Full Load[1]	Compressor at Part Load[2]	Total / Composite
Average air supplied (scfm)	1,000	575	1,575
Average input power (kW)	175.3	124.8	300.1
Supply efficiency (scfm/kW)	5.70	4.61	5.25
Annual hours of operation	5,833	6,100	6100
Energy use (kWh)	1,022,525	761,540	1,784,065
Total energy cost ($)	50,104	37,315	87,419
Unit energy cost ($/scfm)	50.10	64.90	55.51

[1]Assumes 1 compressor operates at full load.
[2]Assumes 1 compressor operating in a "trim" mode providing part power as needed to supplement the compressor operating at full (design) load.

Specific nameplate data on the major components is as follows:

Compressors #1 – #4

Manufacturer: Gardner-Denver
Model #: EAYQVD
Motor: 500 HP
Electric: 4060 VAC/3 phase/60 Hz
FLA: 80 amps
Date of Manufacture: 6/92

Compressors #5 – #8

Manufacturer: Gardner-Denver
Model #: EAYQVD
Motor: 500 HP
Electric: 4060 VAC/3 phase/60 Hz
FLA: 80 amps
Date of Manufacture: 6/95

Refrigerated Dryers #1 & #2

Manufacturer: TEK Engineering
Model#: 50-10000-480-3-60-TH
Rated Air Flow: 10,000 cfm
Electric: 460 VAC/3 phase/60 Hz
Compressor Motor: 40 HP

Measurements and operational readings taken at the site are:

Compressor #1: OFF
Leaving Air Pressure: N/A psig
Leaving Air Temperature: N/A °F
Currents: L1: 0 amps

Compressor #2:
Leaving Air Pressure: 102 psig
Leaving Air Temperature: 182 °F
Currents: L1: 70 amps

Compressor #3:
Leaving Air Pressure: 101 psig
Leaving Air Temperature: 182 °F
Currents: L1: 69 amps

Compressor #4:
Leaving Air Pressure: OFF
Leaving Air Pressure: N/A psig
Leaving Air Temperature: N/A °F
Currents: L1: 0 amps

Compressor #5:
Leaving Air Pressure: OFF
Leaving Air Pressure: N/A psig
Leaving Air Temperature: N/A °F
Currents: L1: 0 amps

Compressor #6:
Leaving Air Pressure: 100 psig
Leaving Air Temperature: 175 °F
Currents: L1: 54 amps

Compressor #7:
Leaving Air Pressure: 103 psig
Leaving Air Temperature: 197 °F
Currents: L1: 62 amps

Compressor #8:
Leaving Air Pressure: 100 psig
Leaving Air Temperature: 181 °F
Currents: L1: 70 amps

Building #1 Compressor Performance

Table H7 lists performance characteristics for the Building No. 1 compressor

Table H7. Performance characteristics for Building No. 1 compressor (efficiency: scfm/kW).

scfm	Brand	Type	Year	100% Load	75% Load	50% Load	25% Load
2500	Gardner-Denver	Screw with Turn Valve	1992, 1995	5.61	5.16	4.18	2.09

Building #1 Compressed Air Load Profile

Based on conversations with the LCAAP Facilities Engineering Staff, the load profile shown in Figure H15 represent the best estimate of the compressed air load profile Monday through Friday. The compressors are shut off Saturday morning at 3:30 am and remain off until approximately 6:00 am Monday.

Building #1 Air Compressor Controls

During the site survey, five of the eight screw compressors that supply Building #1 were operating. The compressors have a sequencer that determines the order of compressor operation. Typically, no more than four compressors are needed to meet the loads. In general, the units are operated at design capacity, with no more than one unit operated at part load, as necessary.

Building #1 Generation and Distribution Pressure/Condensate and Oil Elimination

The compressed air system is reported to operate as desired with no condensate or oil carryover identified as a problem. Pressure drop through the refrigerated dryers was 5.0 psig, and pres-sure drop from the outlet of the refrigerated dryer to end use points was undetectable. These observations lead to a conclusion that the system piping is properly sized and does not contain any significant restrictions.

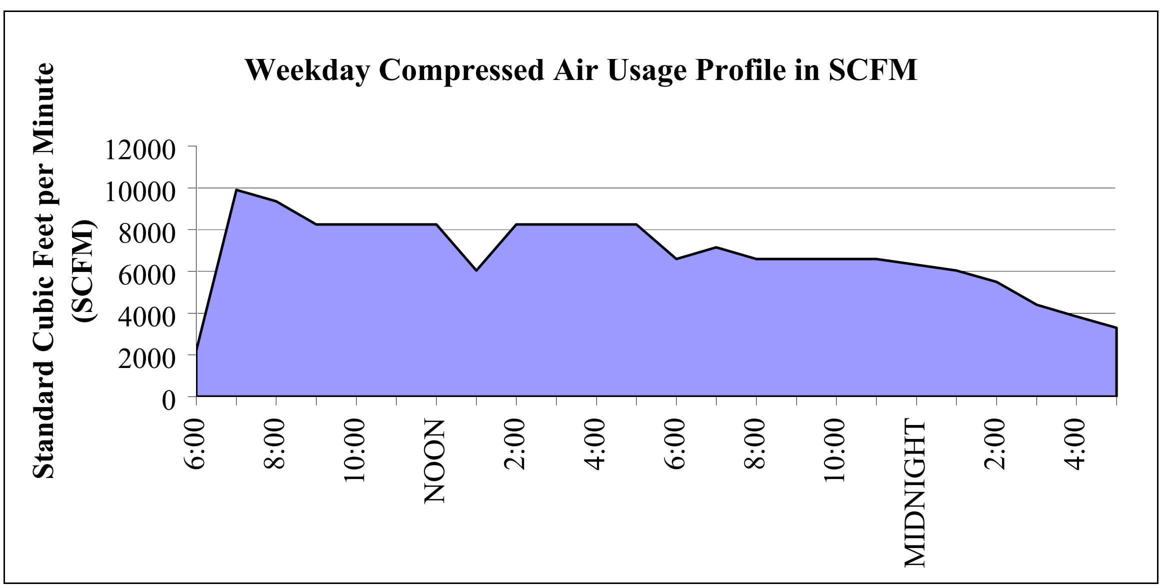

Figure H15. Estimate of the compressed air load profile Monday through Friday at LCAAP.

Building #1 Compressed Air Energy Use and Energy Operating Costs

Table H8 lists the key energy related operational information for the system. The operating assumption is that one to three of the four units that are used in a typical day, operate at full load (2500 scfm), while the other unit operates at part load (500 cfm—2400 cfm). The energy cost information is based on the $0.049 average rate. Note that this does not include nonenergy operation and maintenance costs, which would increase the totals shown.

Compressed Air System Operational Cost Cutting Opportunities

Summary of Opportunities

Generating compressed air for Building 1 and 3 costs $474,000/yr, or nearly 25 percent of LCAAP's electricity expenditures. A number of opportunities were examined as a means of reducing these expenditures. The most significant opportunity for savings is heat recovery from the compressors in these buildings — an estimated $51,000 in natural gas fuel savings for space heating.

Other opportunities initially considered included compressed air distribution system leak reductions, optimum sequencing of the controllers, and compressed air optimization. However, the survey revealed that this was already being considered. More specifics follow.

Table H8. Key energy related operational information for the compresses air system.

	Compressors at Full Load[1]	Compressors at Part Load[2]	Total /Composite
Average Air Supplied (scfm)	5,761	1,274	7,035
Average Input Power (kW)	1,027	321.6	1348.6
Supply Efficiency (scfm/kW)	5.61	3.96	5.22
Annual Hours of Operation	5,846	6,100	6100
Energy Use (kWh)	6,003,045	1,961,996	7,965,041
Total Energy Cost ($)	294,149	96,138	390,287
Unit Energy Cost ($/scfm)	51.06	75.46	57.44

[1]Assumes up to 3 of the 8 compressors, each operating at full (design) load (2500 cfm each). The average air supplied is the sum of the air supplied over a typical day by these compressors operating at full (design) load, divided by the hours of operation. The average input power is based on the compressors operating at the full (design) load, based on the supply efficiency at full (design) load.

[2]Assumes 1 compressor operating in a "trim" mode providing part power (less than 2500 scfm) as needed to supplement the compressors operating at full (design) load.

Table H9. Summary of annual heat recovery cost savings.

Heat Recovery Method	Savings
Heat rejected by oil cooler	2274 Btuh/kW
Heat rejected by after cooler	442 Btuh/kW
Heat recovery efficiency	50%
Heat recovery potential	1358 Btuh/kW
Average Input Power	1348.6 kW
Total heat recovery	0.92 Mbtuh[1]
Annual heating energy saved	5,612Mbtu[2]
Annual heat recovery cost savings	$32,437[3]

[1]Heat Recovery Potential x Average Power Input x 0.5. The 0.5 reduces the potential to account for the fact that the demand for process hot water is 1.2 MBtuh, which could be met by heat recovery on only 4 of the 8 units (2 already have heat recovery).

[2](Total Heat x 6100 operating hours)/.80 boiler efficiency

[3]Annual Heating Energy Saved x $5.78/MBtu natural gas fuel

Recover Heat From Compressed Air Oil and After Coolers for Process Water Heating

Heat generated by the compressors is removed from the oil and the after cooler by water that is piped to fan/radiator units outside the building. It is possible to use this heat to supplement the process water heating in Building 1 prior to rejecting the remaining heat outdoors. This reduces the overall facility's utility costs because it reduces the amount of boiler-generated steam necessary for process water heating. The estimated savings is $32,437 with the details (Table H9). The heat rejection rates are based on the manufacturer's specifications for the 500 hp compressors.

Minimize Compressed Air Distribution Leaks

LCAAP has an active leak maintenance program that appears to be working well. Consequently, leaks are not likely to exceed 15 percent of the total load of Buildings 1 and 3. Based on normal usage, it is difficult to reduce system wide leaks much below this level. Therefore, no additional actions related to this measure are recommended.

Compressor Sequencing Control

The compressors in Building 1 have sequencing control to establish which compressors should turn on and under what conditions. One compressor serves as the lead compressor, and the others follow based on pressure (demand) signals. It was observed that five compressors were in operation and only three of the five were running at design capacity. Since operating compressors at their design load is the most efficient operating point, it is recommended that at most one compressor be operated at part load at any given time. If there are significant hours of low part load operation, this would indicate a need for a more appropriately sized compressor.

Compressed Air Optimization

This opportunity refers to matching compressed air supply pressures to the load. Savings result if it's possible to reduce the supply air pressure. For LCAAP there does not appear to be a mismatch, therefore no savings opportunity exists.

Potential for Natural Gas Engine-Driven Air Compressor

Site Suitability

A natural gas engine driven air compressor can readily be accommodated at Lake City Army Ammunition Plant, with possible applications in either Building 1 or 3. While the current economics favor a Building 1 application, Building 3 has a potentially greater need for additional compressed air capacity (currently two portable diesel engine driven air compressors are being operated to meet specialized loads), and has greater heat recovery opportunities. Furthermore, the NGEDAC can be sited next to the natural gas station immediately outside the building, whereas a location serving Building 1 would require a more significant gas piping run (100 ft). The unit would be housed in its own heated weatherproof enclosure to protect it from the elements. The NGEDAC supply air would be tied into the existing supply system from Building 3, and make use of the existing receiver and 2000 scfm air dryer. The NGEDAC could potentially 1) meet the full load supplied by the existing electric motor driven air compressors or 2) be operated in combination with one or both of these units to meet load growth. In particular, the NGEDAC could be used in place of the two portable diesel engine driven air compressors. Waste heat from the NGEDAC would be recovered and used for process water heating applications. The NGEDAC would be installed in a manner that would not compromise the operation of any other unit. The principal benefits of the NGEDAC unit for LCAAP include:

- net savings in operating costs (depending on the price of natural gas vs. electricity or vs. diesel fuel).
- hedge against power disruptions—operates on natural gas, not electricity.
- added capacity/redundancy for the compressed air system.

Economic Analysis

NGEDAC units ranging in size from 250 HP to 400 HP were evaluated with different operating schemes. The results for a 350 HP unit with an output of about 1670 cfm are provided below. Two cases are examined. In the first case, the NGEDAC is assumed to meet the building's full compressed air requirements, under typical operating conditions. In the second case, the NGEDAC is assumed to meet the additional load currently being met by a combination of two portable diesel engine driven air compressors.

Operating Cost Comparison—NGEDAC Displacing Nominal Demand Currently Met by Electric Motor Driven Air Compressors

Table H10 below summarizes the energy performance and costs associated with the proposed unit operating in a manner that meets full load (1600 scfm) for most of the operating day. For this period, about 5833 hours/yr, the NGEDAC would enable both existing electric motor driven air compressors to be shut down. For the few hours during the operating day when demand is low (400 scfm for 267 hr/yr), one of the electric units would be operated. Should demand increase above the nominal levels, one or more of the electric units could be brought on-line.

Table H10. Compressor performance characteristics at design load.

	Electric Air Compressor	NGEDAC
Compressed air capacity	2-1000 cfm each	1670 cfm
Motor/engine power	2–200 hp (216 bhp) each	350 hp (385 bhp)
Full load power	2–175.3 kW each	3.267 MBtuh (HHV)
Efficiency	2–5.70 cfm/kW each	511.2 cfm/MBtuh

Annual Energy Use and Operating Costs

Table H11 lists baseline energy price assumptions. The results shown are based on the most recent electric and gas prices as indicated. Table H12 shows changes in the annual operating costs of the NGEDAC system based on possible changes in future electric rates or gas prices. Note also that the maintenance costs for the NGEDAC are a function of the hours of operation for a given size unit.

Table H11. Baseline energy price assumptions.

	Electric Air Compressor	Hybrid NGEDAC/Electric	Net Savings
Energy Use	1,783,796 kWh[1]	33,327 kWh (elec. air compressor)[2] 19,056 MBtu gas (engine) -7,075 MBtu (engine heat recovery)[3]	1,750,469 kWh (elec.) -11,981 MBtu (gas)
Energy Operating Costs	$87,406	$111,779	-$24,373
Operation & Maintenance Costs	$20,000	$25,458	-$5,458
Heat Recovery Costs	0	-$40,892	$40,892
Total Costs	$107,406	$96,345	$11,061

[1]Electricity Costs: $0.049/kWh—includes demand and energy charges
Natural Gas Costs: $5.78/MBtu

[2] The electric unit is assumed to operate during the 267 hours per year when the load is 400 scfm and consumes 33,322 kWh. annually

[3] Based on (0.295/0.8) *heat value of natural gas into the engine, where 0.295 is the fraction of recoverable heat (engine coolant, exhaust, or compressor oil) and 0.8 is the assumed efficiency of the process water boiler displaced.

Table H12. Annual operating costs ($)—sensitivity to changes in energy prices.

Energy Price Assumptions	Electric Air Compressor	NGEDAC	Net Savings
Higher Elec. Rates/Base Case Gas Rates			
1) Elec.: $.054/kWh and Gas: $5.78/MBtu	116,325	96,508	19,817
2) Elec. $.059/kWh and Gas: $5.78/MBtu	125,244	96,674	28,569
Base Case Elec. Rates/Lower Gas Rates			
1) Elec.: $.049/kWh and Gas: $5.25/MBtu	107,406	89,991	17,415
2) Elec. $.049/kWh and Gas: $4.82/MBtu	107,406	84,839	22,567
Higher Elec. Rates/Lower Gas Rates			
1) Elec.: $.054/kWh and Gas: $5.25/MBtu	116,325	90,158	26,167
2) Elec. $.059/kWh and Gas: $4.82/MBtu	125,244	85,173	40,071

Operating Cost Comparison—NGEDAC Displacing Demand Currently Met by Diesel Engine Driven Air Compressors

Table H13 lists performance characteristics Table H14 lists energy performance and costs associated with the proposed unit operating in a manner that displaces the load currently being met by two portable diesel engine driven air compressors. These compressors operate about 60 hours/week (3000 hr/yr) to meet the air requirements of specialty equipment. While measurements of the air supplied by the portable units were not available, known fuel consumption information combined with the assumption that the compressors would provide about 5 scfm/hp, indicate an average output of the combined units of about 1100 scfm. It was assumed that the 350 hp NGEDAC, operating at part load would be used to meet this demand, eliminating the need to operate the diesel units.

Table H13. Annual energy use and operating costs.

Baseline Energy Price Assumptions 1	Diesel Engine Driven Air Compressor	NGEDAC	Net Savings
Energy use	6300 MBtu	6861 MBtu (gas engine) —2530 MBtu (heat recovery)2	6300 MBtu (diesel) - 4331 MBtu gas
Energy Operating Costs	$62,550	$39,655	$22,895
Operation & Maintenance Costs	$12.500	$12,500	0
Heat Recovery Costs	0	-$14,623	$14,623
Total Costs	$75,050	$37,532	$37,518

1 Natural Gas Costs: $5.78/MBtu
Diesel Fuel Costs: $9.93/MBtu (Based on $1.39 /gal/140,000 Btu/gal)
2 Based on (0.295/0.8) heat value of natural gas into the engine, where 0.295 is the fraction of recoverable heat (engine coolant, engine exhaust, and compressor oil) and 0.8 is the assumed efficiency of the process water boiler displaced.

Table H14. Compressor performance characteristics.

	Diesel Engine Driven Air Compressors	NGEDAC
Full Load Compressed Air Capacity	2-1575 cfm combined	1670 cfm
Full Load Motor/Engine Power	2–315 hp combined	350 hp (385 bhp)
Full Load Input Power	2–2.940 MBtuh combined (HHV)	3.267 MBtuh (HHV)
Full Load Efficiency	535.7 cfm/MBtuh	511.2 cfm/MBtuh
Average Compressed Air Demand	1100 scfm	1100 scfm
Part Load Input Power	2.150 MBtuh combined (HHV)	2.29 MBtuh (HHV)

Capital Cost for the NGEDAC

Table H15 lists the Capital Cost for the NGEDAC.

Table H15. Capital cost for the NGEDAC.

Cost Element	Cost ($)
350 hp NGEDAC	281,000
Compressor Enclosure	33,000
Heat Recovery	22,000
Installation	36,000
Freight	3,000
Total	375,000

Appendix I: Compressed Air System Survey at Redstone Arsenal

Overview of Facility

Base Mission

Redstone Arsenal (RSA) is located in Huntsville, Alabama. RSA is the home of the U.S. Army Aviation and Missile Command (AMCOM), established on 1 October 1997, through a merger of the U.S. Army Missile Command (MICOM) and the U.S. Army Aviation and Troop Command (ATCOM). RA provides testing, research, and development.

Energy Expenditures

Electric Rates

Table I1 provides total facility average electricity cost for RSA for FY 2000, to date, by month.

Natural Gas Rates

Table I2 lists total facility average natural gas cost for RSA for FY 2000, to date, by month. Table I3 lists additional historical rate schedule information.

Table I1. Total facility average electricity cost.

FY2000 to Date	$/kWH
Oct-99	$0.049
Nov-99	$0.046
Dec-99	$0.047
Jan-00	$0.046
Feb-00	$0.046
Mar-00	$0.047
Apr-00	$0.045
May-00	$0.048
Jun-00	$0.047
Jul-00	$0.047
Aug-00	$0.046
Sep-00	$0.047
Oct-00	$0.049
Nov-00	$0.047
Dec-00	$0.046
Average	$0.047

Table I2. Total facility average natural gas cost.

FY2000 to Date	$/Million BTU
Oct-99	$5.35
Nov-99	$4.52
Dec-99	$4.26
Jan-00	$4.27
Feb-00	$5.25
Mar-00	$4.41
Apr-00	$4.45
May-00	$4.48
Jun-00	$4.52
Jul-00	$4.63
Aug-00	$4.31
Sep-00	$4.43
Oct-00	$5.35
Nov-00	$4.52
Dec-00	$4.76
Average	$4.63

Table I3. Natural gas rate schedule.

	Firm Supply ($/Million BTU)	Interruptible Supply ($/Million BTU)
Up to 12/29/00	$3.89	$3.32
12/29/00–1/30/01	$4.48	$4.95
1/30/01 to Present	$8.58	$8.01

Compressed Air Survey

On 11 April 2001 a compressed air system survey was conducted by Science Applications International Corporation (SAIC), the U.S. Army Construction Engineering Research Laboratory (CERL), and General Machinery Company personnel. The purpose of the study was two-fold:

1. To identify opportunities for reducing energy operating costs associated with the compressed air system
2. To evaluate the site as a candidate for a CERL-funded project to demonstrate the operation of a natural gas engine driven air compressor (NGEDAC).

RSA staff who were interviewed during the survey included:

- Morton Archibald—Facility Energy Manager
- Tim Smith—Facility Mechanical Engineer
- Ronnie Starky—Facility Compressed Air Maintenance Contractor

Compressed air is a necessary energy input stream into the operations carried out at the Redstone Facility. Because compressed air requirements vary across RSA depending upon application, each building where compressed air is used has its own system. Three compressed air end-use systems were selected for the site survey based on total installed horsepower, annual hours of operation, proximity to a natural gas supply, and accessibility due to security requirements:

- calibration laboratory facility (Building 5436)
- rocket testing/fuel grinding (Building 7159)
- motor pool vehicle maintenance shop (Building 3634)
- calibration Laboratory Facility (Building 5436).

Compressed Air System Overview

Figures I1 and I2 provide schematic diagrams of the compressed air system in Building 5436 (TMDE Activity). Compressed air is used to operate vibration free tables, air conditioning controls, parts cleaning, and for liquid flow metering equipment calibration. Table I-4 provides air compressor descriptive details. Service is provided by one 25 horsepower air compressor located outside at the back of the building. This compressor is dedicated to supplying compressed air to the main portion of the building. An additional 25 horsepower air compressor is available solely for back-up. Should both the main and back-up compressors be unavailable, a 3 horsepower back-up compressor is used to operate pneumatic controls to maintain continuity in the operation.

Measured amperage during operation of the 25 horsepower compressor indicated the motor was loaded to 26.4 horsepower, assuming the nameplate power factor of 83 percent and full load efficiency of 91 percent (32.5 amps, 463 volts). Assuming 5 scfm of compressed air produced per horsepower, compressed air output is 132 scfm.

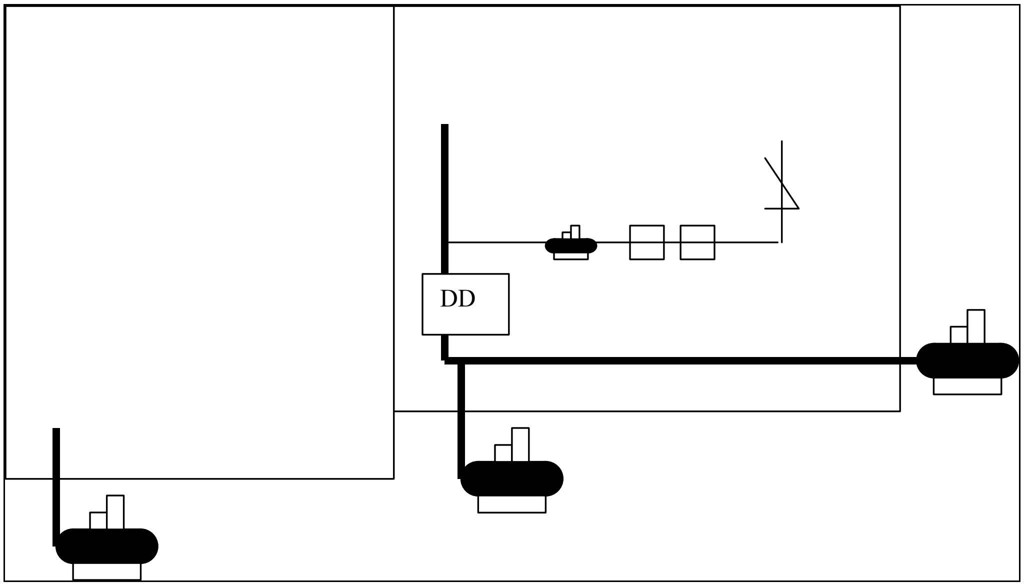

Figure I1. Building 5436 (Calibration Laboratory Facility) CA System.

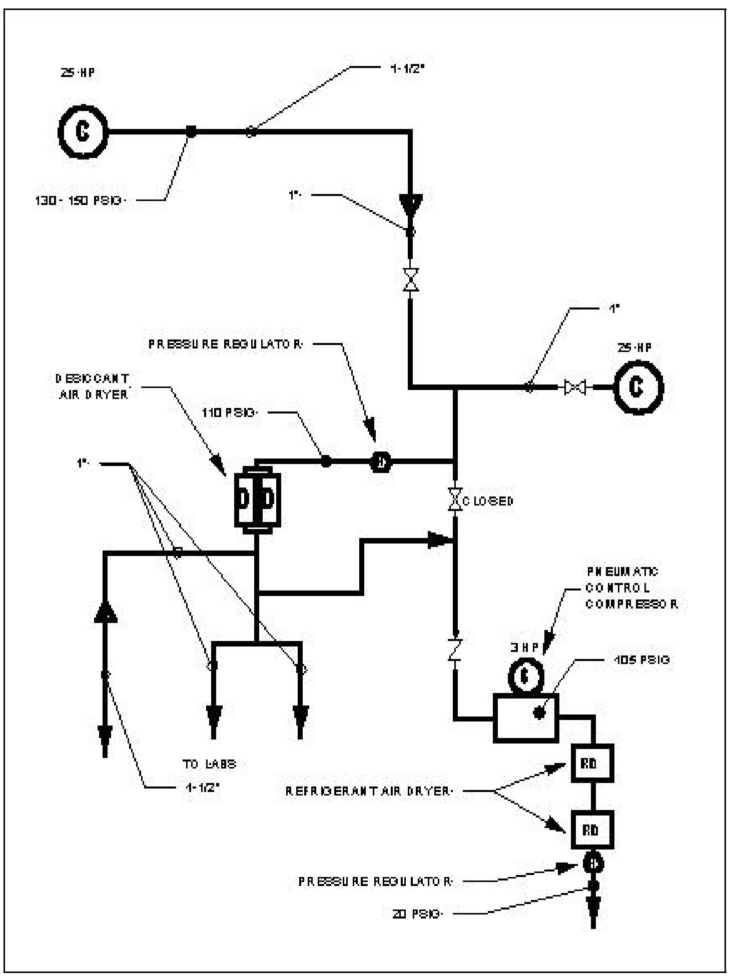

Figure I2. CA system diagram TMDE activity, Building 5436 (Calibration Lab).

Table I4. Operational compressor inventory, TMDE activity—Building 5436 (Calibration Laboratory Facility).

scfm	Hp	Brand	Type	Year
132	25	Ingersoll-Rand IR3000	Recip—single acting	2001
132	25	Ingersoll-Rand IR3000	Recip—single acting	2001
16	3	N/A	Recip—single acting	N/A

This compressor operates for 30 seconds, then shuts down for 60 seconds (equivalent to 33 percent compressor runtime); therefore, the average compressor output is 44 scfm.

The 25 horsepower compressor operates for one 8-hour shift, 5 days/week, 50 weeks/yr. During one 8-hour shift, the motor would start/stop 320 times (recommendation addressing short cycling will follow).

The air compressor does not operate during the lunch period. Compressed air consumption was undetectable when measured during this period, a good indicator that air leak loads on this building's compressed air system are minimal.

Compressed Air Moisture Removal

Moisture in the compressed air is required to be maintained at a—40 °F dewpoint. A Kemp model number 7030ORIAD twin tower desiccant air drying unit has been installed to remove moisture from the compressed air (Table I-5 lists specifications). Based on employee interviews, the desiccant tower is not successfully removing the moisture down to a—40 °F dewpoint. Based on the observation that the dried compressed air was leaving the desiccant dryer at 150 °F, there is a problem with control of the dryer's heating elements. With a high temperature of the desiccant in the dryer, successful moisture removal is not possible.

Assuming the drying unit is not shut down when the staff leaves at the end of the day, the annual energy cost of the desiccant compressed air dryer is $4,242/yr. Table I-6 lists the estimate of annual costs to operate desiccant dryer

Table I5. Kemp desiccant air dryer specifications.

Parameter	Specification
Inlet temperature (maximum)	100 °F
Inlet pressure (maximum)	110 psig
Heater energy input	5.85 kW
Purge rate	7 % of 490 scfm full load dryer capacity

Table I6. Estimate of annual cost to operate desiccant dryer.

Parameters / Total	Cost
Annual purge air energy consumed	49,264 kWh[1]
Annual heating element energy consumed	40,997 kWh[2]
Total energy cost	$4,242[3]

[1](((0.07x490 scfm purge rate)/5 scfm/hp)x0.746 kW/hp)/0.91 motor efficiency x 8,760 hours/year, assuming 5 scfm/hp compressed air production efficiency.

[2]5.85 kW heater energy input x 8,760 hours/year x 0.8 annual operating hours fraction

[3]Energy cost for purge air and heating element assuming facility-wide average FY 2000 electricity cost of $0.047/kWh.

Rocket Testing/Fuel Grinding (Building 7159)

Compressed Air System Overview

Figure I3 provides a schematic diagram of the compressed air system serving Building 7159. The system consists of the following two compressors:

Primary Compressor: 150 HP Worthington, four stage 2,500 psig reciprocating compressor

Back-Up Compressor: 100 HP Rix, four stage 2,500 psig oil-free reciprocating compressor (under repair during site survey)

Compressed Air End Use

Rocket motor testing: High pressure compressed air is used for testing typically one time per week. In preparation for each motor test, air compressor is pumped into a 2,500-gal air storage tank to achieve a pressure of about 2000 psig. This process takes four to 6 hours.

Rocket fuel grinding: High pressure (about 700 psig) compressed air is used for fuel grinding. Grinding is performed as a batch operation, with one batch equal to one to 4 days of grinding. Batch production frequency ranges from weekly to once every other month dependant on the testing schedule.

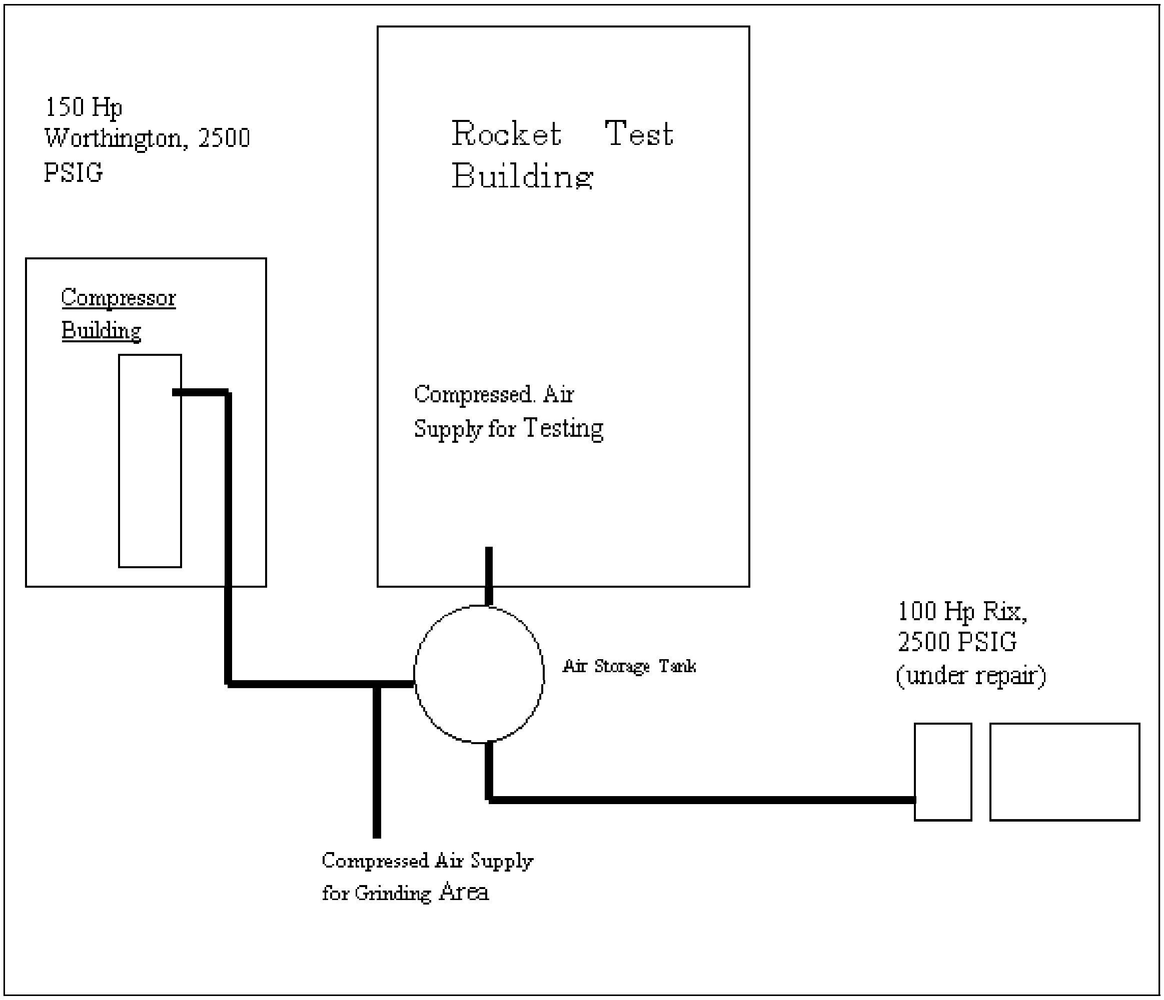

Figure I3. Compressed air system overview Building 7159 (rocket testing / fuel grinding).

Compressed Air System Performance

The system operators want to reduce condensate and oil present in the high-pressure system. These contaminants are especially detrimental in the rocket fuel grinding process. Presently, the moisture and oil are removed by condensing them out in the high-pressure air storage tank.

Annual Energy Cost

> **4 hours/week x 50 weeks/yr x (150 hp/0.91 motor efficiency) x 0.746 kW/hp x $0.047/kWh = $1,156**

Usage of this equipment, even for short periods, could increase the electrical billing cost for peak demand. However, the facility load profile needed to make this assessment was unavailable.

Given the low number of hours that these compressors are used, there are no recommended energy efficiency measures.

Methods to reduce pressure by adding storage capacity or to reduce electrical demand by pressurizing the tank over a longer period of time with a smaller compressor would not have a reasonable return on investment.

Due to the reported critical nature of oil and condensate free air, proactive measures necessary to recommission the oil-free reciprocating air compressor are warranted. Additionally, desiccant drying needs to be added to reduce the air moisture content.

Motor Pool Vehicle Maintenance Shop (Building 3634)

Compressed Air System Overview

Figure I4 shows the compressed air system supplying the maintenance shop. The system consists of one 50 horsepower, air cooled, rotary screw air compressor. No nameplate was found to identify the compressor manufacturer or model number. The compressor has inlet modulation control, and no rotor shortening control. When the compressor reaches its upper set-point, the unit unloads and the oil sump is depressurized to 40 psig. The compressor was set to the “on” position, which kept it running continuously. The alternative control setting was labeled “auto.” This is assumed to shut down the compressor motor after a specified period of compressor idling.

The compressed air is discharged from the compressor into a 1,000-gal compressed air storage tank. The Maintenance Shop is fed directly from this storage tank. Condensate and oil removal filters have been installed only at the end points of compressed air use.

Typical compressed air operation is 8 hours/day, 5 days/week. Annual operation is 2,000 hours. Based on our observations, the compressor is loaded 10 percent of the time and unloaded 90 percent of the time. The compressed air generation pressure control was set to load the compressor at 130 psig and to unload the compressor at 120 psig.

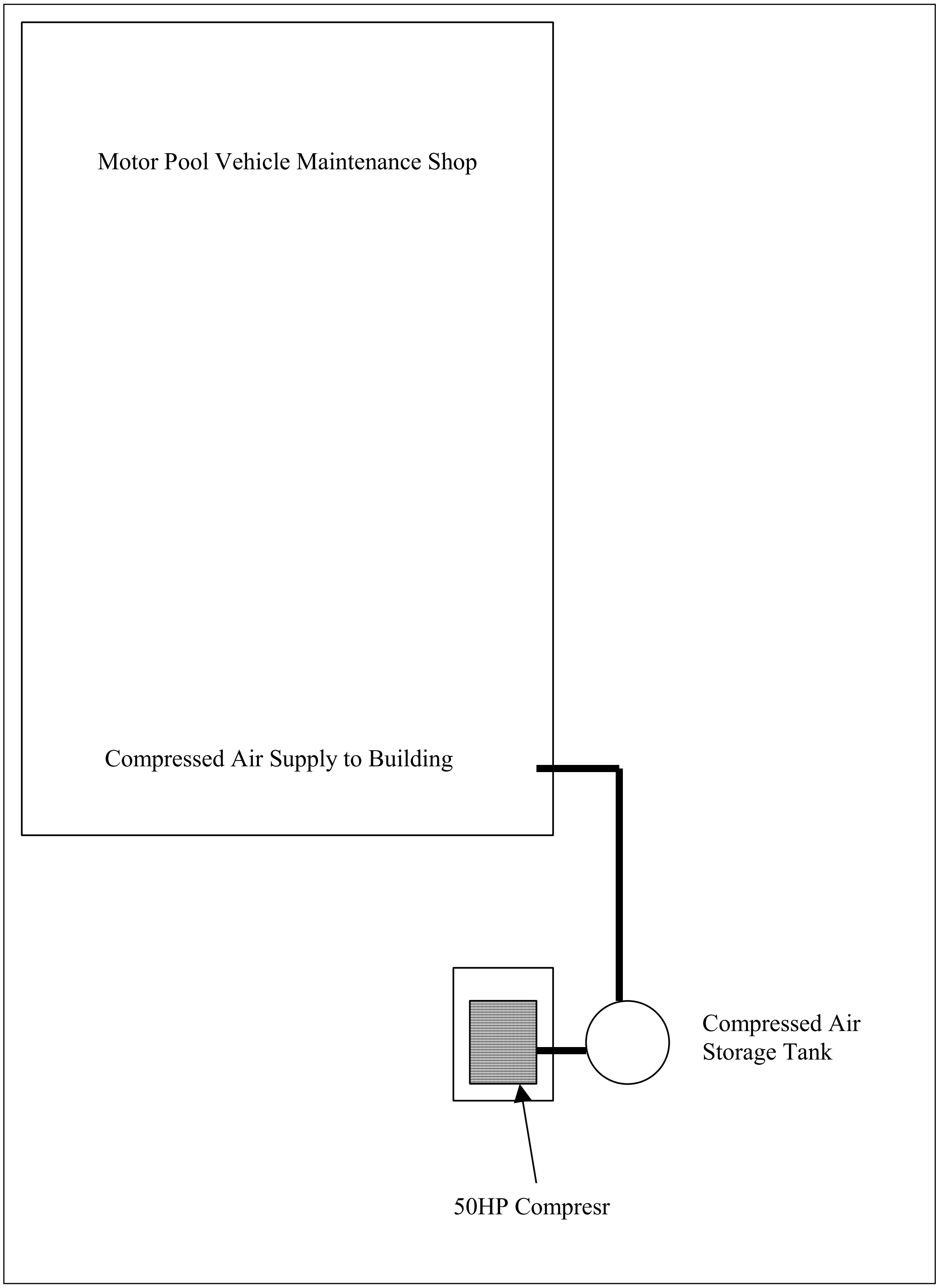

Figure I4. Compressed air system overview Building 3634 (motor pool vehicle maintenance shop).

Compressed Air Water/Oil Removal

The facility staff reported no problems with water or oil in the compressed air. It is assumed that the water dropout occurs in the relatively large receiver tank, and the compressor oil separator at the end use is sufficient to address oil that escapes the compressor.

Annual Compressed Air Energy Cost

Unloaded (measured): 36.5 amps at 465 volts, three phase, 85 percent power factor, 90 percent motor efficiency

Electrical consumption during unloaded period = 465 volts x 36.5 amps x (1 kW/1000 W)x

SqRt 3 x 0.85 pf = 25 kW

Compressor full load includes 15 percent service factor of motor capacity for air compressors.

Full Load Power = (50 hp x1.15x 0.746 kW/hp)/0.90 efficiency = 48 kW

Annual Operation Cost: 2,000 hr x (0.90x25 kW + 0.10x48 kW) x $0.047/kWh = $2,560

Compressed Air Energy Use and Energy Operating Costs

Table I7 lists the energy operating costs of the compressed air systems.

Compressed Air System Operational Cost Cutting Opportunities

Summary of Opportunities

A number of opportunities were examined as a means of reducing compressed air costs at RSA and are described below.

Table I7. Compressed air energy use and energy operating costs.

Parameter	Shift Operating Load
Building 5436	
Air supplied (scfm)	132
Input power to compressor (kW)	21.6
Supply efficiency (scfm/kW)	6.1
Annual hours of operation	660
Energy use (kWh)	14,256
Total energy cost ($)	670
Unit energy cost ($/scfm)	5.08
Building 7159	
Average air supplied (scfm)	750
Average input power to compressor (kW)	123.0
Supply efficiency (scfm/kW)	6.1
Annual hours of operation	200
Energy use (kWh)	24,600
Total energy cost ($)	1,156
Unit energy cost ($/scfm)	1.54
Building 3634	
Average air supplied (scfm)	165
Average input power to compressor (kW)	27.3
Supply efficiency (scfm/kW)	6.0
Annual Hours of Operation	2,000
Energy Use (kWh)	54,510
Total Energy Cost ($)	2,562
Unit Energy Cost ($/scfm)	15.53

Calibration Laboratory Facility (Building 5436)

Prevent Short Cycling of Compressor Motor

The 25 horsepower air compressor located outside at the back of the building was solely supplying compressed air to the main portion of the building. The compressor operated 30 seconds, and shut down for 60 seconds. With this type operation, the motor would start/stop 320 times/8-hour day.

The most straightforward option to address this situation may be to replace the lead 25 horsepower compressor with a 10 horsepower compressor. Prior to doing this, the compressed air contractor should witness the operation of the system to ensure that the operation mode of the compressor during our visit is truly characteristic of normal operation. The advantage of downsizing the compressor as compared to installing no-load control for the existing compressor is that that a compressor still consumes approximately 20 percent of it full load power even when it is under a no-load idle operation.

Reduce Generation Pressure Start Setpoint to 110 psig and Stop Setpoint to 120 psig

The efficiency of compressing air is improved 1 percent for each 2 psi that the generation pressure is reduced. The present start set point is 150 psig and the current stop set point is 170 psig. Because the desiccant dryer that follows the compressor has a maximum inlet pressure of 110 psig, the compressed air pressure is regulated down to approximately 100 psig prior to entering the dryer. The only advantage of this elevated pressure operation is that the compressor storage tank can store a greater amount of air. In turn, the benefit of increased storage is reduced compressor starts and stops. Because the compressor currently sees an excessive amount of starts and stops, this higher pressure is not benefiting the system. If the pressure is reduced to an average of 115 psig as opposed to the current 160 psig average, the annual energy cost reduction is estimated at $135 (Table I8).

Table I8. Estimated annual energy cost reduction.

Parameter	Cost
Current annual energy cost of operation	$600
Percent of energy cost reduced from 45 psig average pressure reduction	22.5%
Annual energy cost savings from pressure reduction	$135

Ensure Desiccant Dryer Tower Heating Element Control Is Working Properly

Desiccant compressed air dryers have two towers. One tower is used to absorb moisture from the compressed air while the second tower is being heated to a temperature of about 250 °F and having about seven percent of the dried compressed air being used to purge the water vapor out. At the end of the tower's regeneration cycle, the heating element shuts off but the purge air continues to flow so the temperature of the desiccant is brought down to the temperature of the dried compressed air.

The survey team observed that the compressed air was leaving the drying tower at 150 °F. Based on this observation, it appears that the heating element is not properly shutting off. Although the capacity of this desiccant drying tower is nine times greater than the load on it, it does not appear to be properly drying the compressed air based on the staff reports of water in the compressed air. The elevated drying tower temperature is believed to be a cause of this less than desired operation. By addressing this situation, it is felt that the desiccant dryer will be able to effectively dry the air to the designed—40 °F dew point temperature.

Refrigerated Air Dryers/Compressor Interlock Start/Stop Control and Reduced Desiccant Drying Tower Regeneration Cycling

Currently, three compressed air dryers are operating with the compressed air system. The main compressed air dryer is the desiccant tower, which is rated at 490 scfm and is designed to bring the dew point down to—40 °F. The energy to operate the desiccant dryer is 90,261 kWh/yr (see Compressed Air Moisture Removal in previous Section). If the purge rate is at the factory setting of 7 percent of full load dryer capacity, it would be consuming 34 scfm. Since the total compressed air produced is 44 scfm (see Compressed Air System Overview in previous Section), the actual building average consumption for other loads is only 10 scfm. Since the survey team's observations were for a short time, it is necessary for the compressed air maintenance contractor to time the system to quantify its true operation during and after working hours.

The second two compressed air dryers are refrigerated dryers designed to bring the compressed air dew point down to + 40 °F. They dry compressed air for the 20 psig control air system if the main compressed air system fails, resulting in start-up of the three horsepower compressor as a backup source. Both units are of the same design; each is rated to dry 10 scfm at 100 psig and to use 0.32 kW.

By interlocking the refrigerated dryer's power with the operation of the 3 horsepower control-air backup compressor, the dryers will not consume any power unless the backup compressor is in operation. This will reduce the continuous 0.32 kW loads on each of the refrigerated dryers.

Additionally, the desiccant drying tower has about 90 percent excess capacity; the tower has a 490 scfm full load capacity but only operates at 44 scfm. The excess capacity results in too frequent operation of a timed tower regeneration cycle. The result is that 90 percent additional purge air and 90 percent more electrical element heating is used than is needed. By working with a representative of the desiccant dryer manufacturer, it should be possible to identify a control scheme that will better match the regeneration cycle with the actual need.

By electrically interlocking the two refrigerated dryers with the backup compressor and by better matching the regeneration cycle of the desiccant towers with the actual need for regeneration, the annual cost savings is estimated at $3,500 (Table I9).

Table I9. Annual cost savings of electrically interlocking the two refrigerated dryers.

Parameter	Cost
Interlock Refrigerated Dryer Stop/Start Control	
Annual electrical energy usage from refrigerated dryers	5,606 kWh[1]
Annual electrical cost from refrigerated dryers	$263[2]
Reduce Tower Regeneration Cycling	
Annual electrical heater energy excess usage from desiccant dryer	36,897 kWh[3]
Annual electrical heater excess energy cost	$1,734[4]
Annual energy to deliver 34 SCFM	44,337 kWh[5]
Annual energy cost to provide 90% more purge air than necessary	$2,084[6]
Total cost reduction potential (both measures)	$4,081

[1]0.32 kW/dryer x 2 dryers x 8760 hours of compressed air system “on” time/yr

[2]5,606 KWH X $0.047/KWH average RSA electricity cost

[3]5.85 KWX8760 hr/yr X 80% operation time X90% excess capacity

[4]36,897 KWH X $0.047/KWH average RSA electricity cost

[5](((0.07X490 SCFM purge rate)/5 SCFM/HP)X 0.746 KW/HP)/0.91 motor efficiency x 8760 hr/yr x 90% excess capacity, assuming 5 SCFM/hp compressed air production efficiency.

[6]44,337 kWh x $0.047/kWh average RSA electricity cost

Motor Pool Vehicle Maintenance Shop (Building 3634) (Install 10 Horsepower Compressor as Lead Compressor)

The 50 horsepower screw compressor supplying the motor pool vehicle maintenance shop appears to run on idle mode for 90 percent of the time based on observations during the site survey. The power consumption during the idle mode, 25 kW (see Annual Compressed Air Energy Cost in previous Section), is about 50 percent of the full loaded energy consumption based on electrical measurements taken.

By adding an additional 10 horsepower compressor (50 percent duty cycle expected) to serve as the lead compressor, power costs currently paid to idle the existing 50 horsepower compressor can be reduced. The existing 50 horsepower compressor would serve as a backup that would automatically start up should the 10 horsepower compressor not supply the compressed air demand. The estimate of annual savings for this measure is $2,175 (Table I10).

Table I10. Estimate of annual savings for installing a 10-hp compressor as lead compressor.

Parameter	Cost
Annual cost to operate compressor (current)	$2,560[1]
Annual cost to operate the 10 Hp compressor on a 50% duty cycle for 2000 hr/yr (1 shift)	$385[2]
Annual cost of operation reduction	$2,175[3]
Annual kWh reduction	46,277 kWh[4]

[1]See Annual Compressed Air Energy Cost in previous Section
[2]((10 HPX0.746 KW/HP)/0.91)X2000 HR/YRX50% DUTYX 0.047/KWH
[3]$2560-$385
[4]$2,175/($0.047/kWh)

Potential for Natural Gas Engine-Driven Air Compressor

Site Suitability

The survey team determined that RSA does not offer good application opportunities for the natural gas engine driven air compressor for the following reasons.

- Small compressor size: 25 hp (Building 5436); 150 hp (Building 7159); 50 hp (Building 3634)
- Hours of operation: Each compressor system surveyed operated 2,000 hours/yr or less.
- No heat recovery applications were discovered near the compressed air systems surveyed.

Economic Analysis

For reasons cited before, no economic analysis was justified.

Appendix J: Compressed Air System Survey at Rock Island Arsenal

Overview of Facility

Base Mission

Rock Island Arsenal (RIA) is located on the Mississippi River on the border of Illinois and Iowa in the Quad Cities area. RIA represents the only general purpose metal manufacturing complex for the Department of Defense. RIA provides production capability for artillery/gun mounts, equipment integration, spare parts, and other equipment for the Armed Forces. RIA has a complete in-house metal parts forge, foundry and plating shop with an extensive inventory of fabrication machinery.

Energy Use and Expenditures

Electric Rates and Consumption

RIA has an electric rate structure with demand and energy components, shown in Table J1. The demand charge varies seasonally; the energy charge varies both seasonally and by time-of-day.

Forty-five percent of 8,760 total annual operating hours, equivalent to 3,942 hours, is on-peak. The remaining 55 percent of annual operating hours are off-peak. RIA provided the historic electric bills listed in Table J2:

Natural Gas Rates and Consumption

RIA purchases natural gas on the spot market through the Defense Energy Support Center. Gas bills for the last 2 calendar years are presented in Tables J3 and J4. The data in the tables show that the cost of natural gas purchased by RIA nearly doubled from 1999 to 2000, following trends similar for the entire United States. A 2-year average price is shown in Table J5.

Table J1. Electric rate structure.

Rate*	Charge	
	Summer (Jun–Sep)	Winter (Oct–May)
Monthly Maximum On-Peak Demand Charge ($/kW)	9.14	4.98
Electrical Energy Charge ($/kWh)		
On-Peak: Monday–Friday, 8:00 am to 8:00 pm	0.0301	0.0301
Off-Peak: All other hours and specified holidays.	0.0185	0.0185
*Source: MidAmerican Energy Company, Rate Schedule #53: Commercial and Industrial Electric Service		

Table J2. RIA historic electric bill summary.

Parameter	Consumption	Cost
August 2000 Bill Summary		
Demand	13,331 kW	$121,845.34
On-Peak	2,777,639 kWh	$ 84,468.00
Off-Peak	2,677,487 kWh	$ 50,363.53
Totals	5,455,126 kWh	$259,346.16
Average Electric Cost		$0.0475/kWh
December 2000 Bill Summary		
Demand	14,087 kW	$70,153.26
On-Peak	1,905,542 kWh	$57,947.53
Off-Peak	2,220,322 kWh	$41,762.26
Totals	4,125,864 kWh	$169,146.13
Average Electric Cost	:	$0.0410/kWh

Table J3. Gas cost for 1999.

Month	Therms	Cost	$/Therm
Jan-99	87,300	$25,019.80	$0.2866
Feb-99	47,880	$14,290.71	$0.2985
Mar-99	57,240	$15,082.74	$0.2635
Apr-99	35,400	$9,324.08	$0.2634
May-99	15,500	$7,204.49	$0.4648
Jun-99	15,000	$7,341.60	$0.4894
Jul-99	10,230	$3,281.74	$0.3208
Aug-99	10,230	$3,589.46	$0.3509
Sep-99	9,300	$3,693.12	$0.3971
Oct-99	16,430	$5,989.50	$0.3645
Nov-99	51,000	$20,182.00	$0.3957
Dec-99	58,630	$18,209.00	$0.3106
Total	***414,140***	***$133,208.24***	***$0.3217***

Table J4. Gas cost for 2000.

Month	Therms	Cost	$/Therm
Jan-00	75,080	$25,910.07	$0.3451
Feb-00	64,330	$49,359.35	$0.7673
Mar-00	58,590	$44,898.95	$0.7663
Apr-00	36,000	$7,916.89	$0.2199
May-00	15,500	$6,058.81	$0.3909
Jun-00	15,000	$7,751.31	$0.5168
Jul-00	8,524	$4,299.86	$0.5044
Aug-00	11,160	$5,441.41	$0.4876
Sep-00	10,500	$6,064.21	$0.5775
Oct-00	14,790	$16,096.98	$1.0884
Nov-00	40,500	$25,597.53	$0.6320
Dec-00	78,640	$62,771.41	$0.7982
Total	***428,614***	***$262,166.78***	***$0.6117***

Table J5. Two-year average gas prices (1999–2000).

Month	Price ($/Therm)
January	0.316
February	0.533
March	0.515
April	0.242
May	0.428
June	0.503
July	0.413
August	0.419
September	0.487
October	0.726
November	0.514
December	0.554
Two-Year Average	0.471

Compressed Air Survey

On 2 April 2001 a compressed air system survey was conducted by Science Applications International Corporation (SAIC) and the U.S. Army Construction Engineering Research Laboratory (CERL) personnel. The purpose of the survey was two-fold:

1. To identify opportunities for reducing energy operating costs associated with the existing compressed air system
2. To evaluate the site as a candidate for a CERL-funded project to demonstrate the operation of a natural gas engine driven air compressor (NGEDAC).

RIA staff that were interviewed during the survey included:

- Joe Behan—Maintenance Supervisor
- Tom Sawvell—Compressed Air System Operator
- Jay Richter—Mechanical Engineer Public Works
- Dave Osborne—Energy Manager
- David Foss—Environmental

Compressed air is one of the primary energy input streams into the production process. Most of the facility compressed air is provided through a central compressed air distribution system that is supported by eight compressors. The compressor systems that were evaluated during the site survey were those located in Buildings 220 and 222 where manufacturing, foundry, forge, and plating processes are housed (Figure J1).

Compressed Air System Overview

Building 220 and 222 compressed air distribution piping is interconnected. Pressure loss after the refrigerated drying to the most distance usage is undetectable. Pressure drop is a concern when it exceeds 5 psig. Secondary distribution piping is also generously sized which is very desirable for a compressed air system. Repair of leaks presents a significant opportunity for cutting compressed air system operational costs.

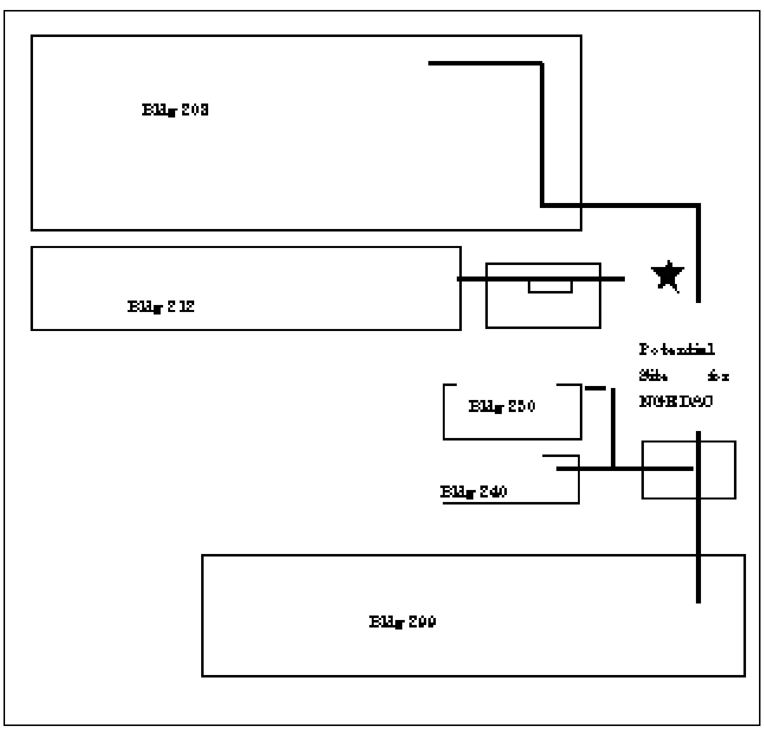

Figure J1. Compressed air system main piping.

Performance of Electric Air Compressors

Tables J6 and J7 provide performance specifications for the existing electric air compressors in Buildings 220 and 222. Figures J2 and J3 show the (models) 4200 and 2500 Ingersoll-Rand reciprocating compressors. Figure J4 shows the three air compressors in Building 3. Figure J5 shows the compressor heat recovery ducts used for space heating in the winter months.

Table J6. Building 220 operational compressor inventory—performance.

cfm	Manufacturer	Type	Year	scfm/kW			
				100% Load	75% Load	50% Load	25% Load
2500	Ingersoll Rand	Recip	1985	N/A	N/A	N/A	N/A`
4200	Ingersoll Rand	Recip	1951	6.43	6.35	5.96	4.21
3700	Worthington	Recip	1941	5.91	5.83	4.84	4.35
1200	Ingersoll Rand	Recip	1953	5.58	4.93	3.68	2.37
2400	Worthington	Recip	1919	5.8	N/A	N/A	N/A

Table J7. Building 222 compressor inventory.

CFM	Brand	Type	Year	scfm/kW			
				100% Load	75% Load	50% Load	25% Load
3000	Ingersoll Rand	Screw w/ IM	1992	N/A	N/A	N/A	N/A
2500	Ingersoll Rand	Screw w/IM	1992	6.12	N/A	N/A	N/A
2500	Ingersoll Rand	Screw w/IM	1992	6.12	N/A	N/A	N/A

Figure J2. 4200 Ingersoll Rand reciprocating compressor.

Figure J3. 2500 Ingersol-Rand reciprocating compressor.

Figure J4. Three air compressors in building 222.

Figure J5. Compressor heat recovery ducts used for space heating during winter months.

Compressed Air Load Profile

Based on conversations with the compressor operator, the weekday air demand load profile was established (Figure J6). Saturday, Sunday, and holiday compressed air consumption remains at the 1800 scfm baseload. Amp meter readings were used during the site visit to determine the loading on the reciprocating compressor in operation. The load remained at 2500 scfm during the period observed.

Air Compressor Controls

The reciprocating compressors in Building 220 have 100/75/50/25 percent loading capabilities for the compressor used to do load "trimming" or "modulating." Each machine is controlled individually. Multiple machine operation is manually controlled by the compressed air operator.

The screw compressors in Building 222 have inlet modulation part-load control and unload control. No rotor shortening part-load control mechanism is in place. A lead-lag control mechanism is in place to even out operation hours on the compressors. After 1 hour of unloaded operation, the compressor will totally shut down.

Each compressor is controlled by its own pressure sensor. The master lead-lag control serves as the compressor sequencing control. The existing reciprocating compressor uses a dual control. The existing screw compressors have straight modulation control and have ratings in the report for 100 percent load.

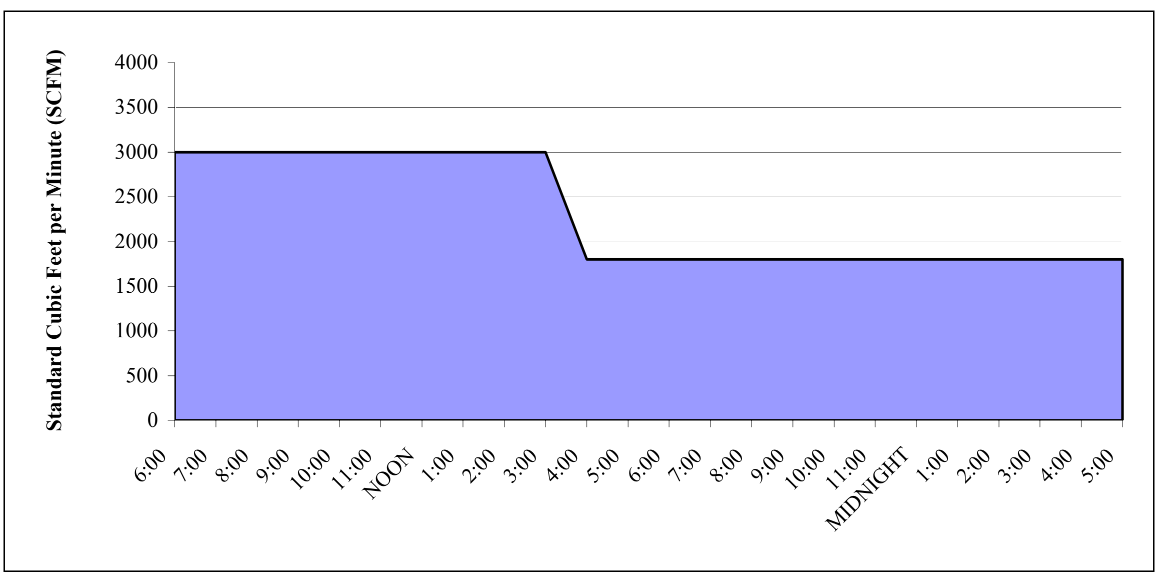

Figure J6. Weekday air demand load profile.

Generation and Distribution Pressure/Condensate and Oil Elimination

The compressed air system is reported to operate as desired by all the end users interviewed. No difficulties with water or oil in the lines or large pressure fluctuations were reported. Past problems with these difficulties have been addressed. With the size of compressors in this building, typically only one of the compressors is operated at any one time. Occasionally, a pneumatic sand transport system in the foundry requires more than one compressor to be operated.

Using a single air pressure gauge testing unit, the pressure was surveyed at key points in the distribution system. A pressure drop of only two psig was measured from a point after the after-cooler to a point after the refrigerated-dryer. No pressure drop was measured from the point after the refrigerated-dryer to the furthest point of distribution piping away from the dryer. The maximum tolerable pressure drop is typically five psig.

The only times condensate is reported to be present in the system is when there is a breakdown of the air-drying equipment. Similarly, the only time compressor oil has been noticed in the system is when the oil filter was damaged.

Compressed Air Energy Use and Energy Operating Costs

Table J8 summarizes the energy operating costs of the compressed air systems assuming Worthington and Ingersoll Rand composite performance characteristics. Compressor operating costs are estimated to be $154,326 based on 3,486,960 kWh of energy use per year.

Table J8. Compressed air energy use and energy operating costs.

Parameter	Shift Operating Load	Off-shift Operating Load	Total/ Composite
Average Air Supplied (scfm)	3000	1800	2120
Average Input Power to Compressor (kW)	486	366	398
Supply Efficiency (scfm/kW)	6.17	4.92	5.25
Annual Hours of Operation	2,340	6,420	8,760
Energy Use (kWh)	1,137,240	2,349,720	3,486,960
Total Energy Cost ($)	50,039	103,388	153,426
Unit Energy Cost ($/scfm)	16.68	57.44	

Summary of Compressed Air System Operational Cost Cutting Opportunities

A number of opportunities were examined as a means of reducing compressed air costs at RIA and are described below.

Minimize Compressed Air Distribution Leaks

Compressed air leaks can represent 30 percent of a facility's compressed air load if the facility does not have a scheduled proactive leak detection and repair program. With a static "baseload" of 1800 SCFM of compressed air and the leaks witnessed during the survey, a 30 percent leak-load estimate is believed to be conservative. By reducing the leak percentage to 10 percent, annual savings would be $26,430/yr (Table J9).

To address this leak-load, a quarterly leak detection and repair program is recommended. RIA owns an ultrasonic leak detector that can be used for the leak detection work. This tool will allow swift detection of leaks by simply walking through the plant.

Leaks in hard-piped distribution lines are rare, and were not witnessed during the site survey. Leaks in connections between hard pipe and flexible lines were witnessed during the survey, as were leaks in pneumatic cylinder seals, and small-orifice drain traps. There were also leaks witnessed that fall into a category known as "planned leaks." These are uses of compressed air that could be replaced with a much lower operational cost technology.

Table J9. Annual savings from minimizing compressed air distribution leaks.

Parameter	Savings
Current leak load	540 scfm[1]
Potential reduction in load via leak ID and repair	360 scfm[2]
Annual kWh due to repairable leaks	600,686[3]
Annual cost savings potential from routine leak repair	$26,430[4]

[1]2120 scfm composite daily load x 30% leak load, using total/composite assumptions in Table J8.

[2]2120 scfm composite daily load x 20% leak load, assuming repairs reduce leak load from 30% to 10%, using total/composite assumptions in Table J8.

[3](360 scfm/5.25 scfm/kw)x8,760 hours/year, using total/composite assumptions in table J8.

[4]600,686 kwhx$0.044/kwh effective RIA rate.

Table J10. Savings resulting from exclusive use of best efficiency compressor.

Parameter	Savings
Demand savings from operating at 3000 scfm	16 kW[1]
Demand savings from operating at 1800 scfm	20 kW[2]
Annual electric energy savings	165,840 kWh[3]
Annual operational energy cost savings, full-time operation	$7,297[4]

[1]3000 scfmx((1/6.08 scfm/kW)—(1/6.30 scfm/kW))
[2]1800 scfmx((1/5.14 scfm/kW)—(1/5.47 scfm/kW))
[3](9 hr/dayx5 day/wkx52 wk/yrx16 kW)+((8760 hr/yr-(9 hr/dayx5 day/wkx52 wk/yr))x20 kW)
[4]Assuming annual electric energy savings valued at year 2000 average electricity price of $0.044/kWh, including both energy and demand costs.

Condensate draining based on timed solenoid operation is a common response to failure of first generation automatic drain traps. Due to the common problem of drain failure, a second generation of reliable "large-orifice" automatic drain traps now exists which will drain condensate without loss of compressed air.

Use of Best Efficiency Compressor Exclusively

RIA has older electric reciprocating compressors that are currently cycled to maintain operability. The analysis shown below illustrates the operating cost savings from operating a more efficient compressor continuously throughout the year. The comparison is between the Ingersoll-Rand 4200 scfm and Worthington 3700 scfm reciprocating compressor, the most energy efficient RIA compressors. Based on prior engineering measurements of RIA compressor performance,* the Ingersoll-Rand compressor has the best energy efficiency at loads of 3000 scfm (6.3 scfm/kW), which is the output during shift operation, and at 1800 scfm (5.47 scfm/kW), which is the continuous level of compressor output. The Worthington compressor is second best, producing 5.85 scfm/kW at the 3000 scfm load and 4.80 scfm/kW at the 1800 scfm load. Since this data is from a site survey conducted in 1982, both compressors may be expected to have some energy efficiency reduction. Thus, the savings shown in Table J10 are a maximum.

These compressors can be maintained to retain their "as new" operating efficiency. By overhauling the reciprocating compressor that best matches the load, energy cost savings closer to the maximum shown below could be realized. To value the cost savings, the original efficiency specifications of the machine can be

* Missman, Stanley and Associates, Compressed Air Survey, May 1982.

compared with the current efficiency of the machine. The efficiency of the machines can be determined by putting an electrical sub-meter on the compressor room and correlating this reading with the output flow readings of each machine being considered for overhaul.

Recover Heat from Compressed Air Inter and After Coolers for Winter Space Heating

Air compressors produce compressed air at an energy efficiency of approximately 20 percent. The remaining 80 percent of the energy is converted into heat, which is carried away with cooling water. Currently, RIA rejects this heat through a cooling tower. Mixed compressor cooling water leaves the compressor room at 130 °F and a 30 gpm flow rate. Instead, this heat could be used to supplement space heating during winter months prior to sending it to the cooling tower. The heating season at RIA is assumed to be half of the year (4,380 hours). The benefit would be reduction in RIA's overall utility costs because the amount of steam necessary for space heating would be reduced (Table J11).

Table J11. Savings resulting from recovery of heat from compressed air inter and after coolers for winter space heating.

Parameter	Savings
Rate of heat transfer to heating system	576,000 Btu/hr[1]
Annual value of natural gas fuel displaced by recovery of compressed air cooling water heat	$14,853[2]

[1] 30 gpm x 8.33 lb/gal x 60 min/hr x (130 °F—90 °F) x1 Btu/lb-of, assuming cooling water temperature drops from 130 °F to 90 °F when passing across heat exchangers in the heating system.

[2] ((0.576 mBtu/hr x 4,380 hr/yr)/0.8 boiler efficiency) x $4.71/mBtu natural gas, assuming a 1999-2000 2-yr average natural gas price.

Provide Monthly Compressed Air Billings or Usage Reports to Production Areas

Currently there are compressed air orifice plate meters installed to monitor four main compressed air lines distributed to the plant. It is recommended to install electrical metering in the compressor room and to calibrate the existing orifice plate compressed air mains sub-metering. The metering would quantify the electrical consumption by the compressed air room, including air-drying units.

The benefits of this submetering would be:

1. The compressed air operator could determine operating efficiency on a daily basis, and make adjustments to optimize the system.

2. The compressed air operator could provide monthly billings or consumption/cost reports for each sector of the plant based on actual meter readings. Once quantified in economic terms, compressed air no longer is a free resource for the facility. This leads to leak identification, and minimization of nonessential uses of compressed air.
3. Any energy cost efficiency project targeted at reduction of compressor electrical costs can be quantified. The measures recommended in this report such as a proactive leak identification and repair program will quickly show up on this metering if carried out. This provides justification of such "additional maintenance" programs.

Potential for Engine-Driven Air Compressor

Site Suitability

An 1,860 scfm natural gas engine driven compressor could be installed at RIA. The NGEDAC could be sited either inside, or just outside, Building 222 (see Figure J1, where a "star" designates the potential site). The natural gas main is 50 ft away, and a separate line would have to be run to the NGEDAC. A cement pad would be needed. The unit would be housed in its own heated weatherproof enclosure to protect it from the elements. An electrical supply for the engine heater would be needed. The NGEDAC supply air would be tied into the central compressed air distribution system that serves the buildings. The NGEDAC would be operated in conjunction with the existing electric motor driven compressors because RIA wants to continue operation of these units. RIA cycles operation of these compressors to even out operational wear and tear. Currently, these compressors provide more than enough capacity to meet load requirements, and the cycling schedule leaves no capacity that could be served by the NGEDAC.

Waste heat for building space heating could be recovered from the NGEDAC engine jacket coolant and from air compressor oil. The percentage of the fuel input heat value that is useable for space heating is 22.2 percent. This percentage is based on 30 percent of the input fuel heat value being rejected through engine jacket cooling and 14.4 percent of input fuel heat value being rejected through compressor coolant oil.* Fifty percent of this waste heat is assumed useable.

* Heat balance information from Robin Wall, Dearing Compressor and Pump Company, Youngstown, OH.

The waste heat could be used 6 months per year space heating. There is no significant requirement for hot water, eliminating this use of waste heat from consideration.

Economic Analysis

Operating Cost Comparison

NGEDAC units ranging from 400 HP to 500 HP were evaluated with different operating schemes. Based on the RIA electric rate structure, the cost of natural gas, and capital cost considerations, a 400 HP unit with an output of about 1800 scfm is proposed. Tables J12 and J13 summarize the energy performance and costs associated with the proposed unit operating to meet RIA's 1800 scfm baseload during daily utility peak period hours, equivalent to 3,942 hours/yr. This operating mode was chosen because it enables the NGEDAC unit to displace the highest value electric air compressor operating hours, those with the highest electricity cost.

The results shown are based on 2-year average gas prices, which were used because they damp the effect of significant gas price increases in 2000 through 2001. Table J14 shows changes in the annual operating costs of the NGEDAC system based on possible changes in future electric rates or gas prices. Note also that the maintenance costs for the NGEDAC are a function of the hours of operation for a given size unit. The capital costs for the NGEDAC is shown in Table J15.

Table J12. Compressor performance characteristics at design load.

Parameter	Electric Air Compressor*	NGEDAC
Compressed Air Capacity	3950 scfm	1860 scfm
Motor/Engine Power	700 hp (790 bhp)	400 hp (414 bhp)
Power	640 kW	3.422 MBtuh
Efficiency	6.17 scfm/kW	544 scfm/MBtuh

*Based on composite characteristics of the Worthington 3700 scfm and Ingersoll-Rand 4200 scfm reciprocating compressors, assuming each operated for six months per year.

Table J13. Annual energy use and operating costs baseline energy price assumptions.

Parameter	Electric Air Compressor[1]	NGEDAC	Net Savings
Energy Use	1,251,388 kWh	13,490 MBtu gas (engine)	1,251,388 kWh (electricity)
		-1,872 MBtu gas (engine heat recovery)[2]	-11,618 MBtu (gas)
Energy Operating Costs	$61,920	$63,499	-$1,579
Operation & Maintenance Costs	$12,614	$23,652	-$11,038
Heat Recovery Costs	$0	-$8,816	$8,816
Total Costs	$74,534	$78,335	-$3,801

[1]Electricity costs: $0.049/kwh—includes demand and energy charges
Natural gas costs: $4.71/mbtu

[2] Based on (0.22/.8)*heat value of natural gas into the engine, where 0.22 is the fraction of recoverable heat and 0.8 is assumed efficiency of heating boiler displaced.

Table J14. Annual operating costs ($)—sensitivity to changes in energy prices.

Energy Price Assumptions	Electric Air Compressor	NGEDAC	Net Savings
Higher Elec. Rates/Base Case Gas Rates			
1) Elec.: $0.054/kWh and Gas: $4.71/MBtu	$80,726	$78,335	$2,391
2) Elec.: $0.059/kWh and Gas: $4.71/MBtu	$86,918	$78,335	$8,583
Base Case Elec. Rates/Lower Gas Rates			
1) Elec.: $0.049/kWh and Gas: $4.24/MBtu	$74,534	$72,867	$1,667
2) Elec.: $0.049/kWh and Gas: $3.77/MBtu	$74,534	$67,398	$7,136
Higher Elec. Rates/Lower Gas Rates			
1) Elec.: $0.054/kWh and Gas: $4.24/MBtu	$80,726	$72,867	$7,859
2) Elec.: $0.059/kWh and Gas: $3.77/MBtu	$86,918	$67,398	$19,520

Table J15. Capital costs for the NGEDAC.

Cost Element	Cost ($)
400 hp NGEDAC	$276,650
Compressor enclosure	$33,000
Heat recovery	$42,476
Installation	$28,958
Freight	$3,300
Total	$384,384

Appendix K: Compressed Air System Survey at Sierra Army Depot

Overview of Facility

Base Mission

The Sierra Army Depot is located in Herlong, CA approximately 50 miles from Reno, NV. The base mission is to provide Operation Project Stock services to customers including storage, repair and issue of equipment and various equipment components. In addition, the base stores, maintains and demilitarizes conventional ammunition.

Energy Use and Expenditures

Electric Rates and Consumption

The Base purchases electricity from Lassen Municipal Utility District on rate schedule #70: Industrial Service. A summary of the rate is as follows:

- demand: $7.00/kW
- energy: $0.1050/kWh (Basic Charge + Rate Adjustment)
  - basic charge: $0.0650/kWh
  - rate adjustment (2/6/01):$0.0400/kWh

The demand charge is based on the maximum average power taken during any 15-minute interval in a month.

March 2001 Bill Summary (2/28–3/30)

Demand:	1,981.7 kW	=	$ 13,867.70
Energy:	809,907 kWh	=	$ 85,040.28
Total Bill:			$99,137.98
Average Electric Cost:			$0.122/kWh ($99,137.98/809,907 kWh)

Gas Rates and Consumption

The Base purchases natural gas from Texas Ohio Energy. Gas bills for the last 2 calendar years are presented in the following tables. The data in Tables K1 and

K2 show that the cost of natural gas purchased by the base nearly doubled from 1999 to 2000.

Table K3 shows gas usage and costs for the first 3 months of 2001. During the month of March 2001, the base negotiated a natural gas rate of $0.795/therm with their gas supplier so that the natural gas cost would be competitive with the cost of diesel.

Table K1. Gas usage and costs (1999).

Month	Therms	Cost	$/Therm
Jan-99	80,960	$19,835.00	$0.2450
Feb-99	67,540	$13,913.00	$0.2060
Mar-99	65,600	$12,005.00	$0.1830
Apr-99	59,100	$11,761.00	$0.1990
May-99	12,650	$3,074.00	$0.2430
Jun-99	590	$145.00	$0.2458
Jul-99	0	$0.00	
Aug-99	0	$0.00	
Sep-99	430	$133.00	$0.3093
Oct-99	18,910	$5,541.00	$0.2930
Nov-99	66,730	$22,421.00	$0.3360
Dec-99	93,480	$24,211.00	$0.2590
Total	465,990	$113,039.00	$0.2426

Table K2. Gas usage and costs (2000).

Month	Therms	Cost	$/Therm
Jan-00	91,145	$22,786.00	$0.2500
Feb-00	80,999	$21,546.00	$0.2660
Mar-00	72,881	$20,188.00	$0.2770
Apr-00	47,700	$15,693.00	$0.3290
May-00	7,640	$2,468.00	$0.3230
Jun-00	70	$33.00	$0.4714
Jul-00	0	$0.00	
Aug-00	620	$286.00	$0.4613
Sep-00	4,770	$3,143.00	$0.6589
Oct-00	34,970	$19,478.00	$0.5570
Nov-00	87,940	$46,784.00	$0.5320
Dec-00	37,450	$58,122.00	$1.5520
Total	466,185	$210,527.00	$0.4516

Table K3. Year to date gas usage and costs (2001).

Month	Therms	Cost	$/Therm
Jan-01	41,650	$61,850.00	$1.4850
Feb-01	31,870	$34,069.00	$1.0690
Mar-01	55,440	$66,000.70	$1.1905
Total	128,960	161,920	$1.2556

Compressed Air Survey

On 19 April 2001 a compressed air survey was conducted by Science Applications International Corporation (SAIC) and the U.S. Army Construction Engineering Research Laboratory (CERL) personnel. The purpose of the survey was primarily to evaluate the site as a candidate for a CERL-funded project to demonstrate the operation of a natural gas engine driven air compressor (NGEDAC). It was also intended to identify opportunities for reducing energy operating costs associated with the existing compressed air system. Mr. Dan Moore was the SIAD point-of-contact for the survey and provided information about the compressed air system and energy usage and costs.

Compressed Air System Overview

The survey covered the compressed air system for buildings 208 (maintenance shop), 209 (metal shop), and 210 (paint shop). The compressed air system consists of three compressors, of which only one is normally operated. Operation of the backup compressors is undesirable, since their use appears to be related to a problem with moisture in the system that has an adverse affect on the breathing air supply. The compressor that is typically operated is located in Building 210. There is a 3.5-in. compressed air line that serves as the compressed air distribution for the three buildings.

The compressed air loads consist of paint booths, shot blast booths, paint sprayers, and various hand-held air tools. The paint booths use air for both the paint sprayers and breathing air. Breathing air is provided by the central air compressor and is then processed through a purifier that consists of filters for moisture, carbon monoxide, and odor removal. The system has two purifying systems for breathing air. Only one system is operated at any given time with the other used for backup in the case of a failure or the need to replace filters.

The focus of the survey was on the electric compressor located in Building 210 and the potential of installing an engine-driven unit to operate in its place. Specific nameplate information on this compressor (Figure K1) is as follows:

Manufacturer:	Gardner Denver
Model:	EDMQNA
Nominal Power:	125 HP
Rated Capacity:	600 icfm
Rated Supply Pressure:	100 psig
Voltage:	460 VAC/3phase/60 Hz
Full Load Current:	154 amps
Date of Manufacture:	December 1994

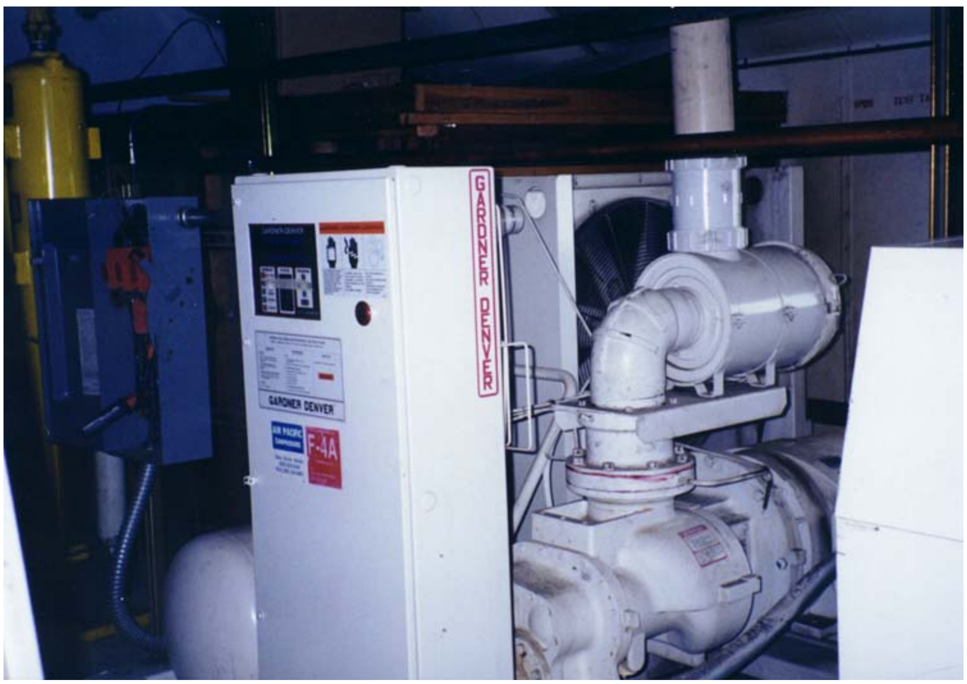

Figure K1. Building 210 Gardner Denver air compressor.

The specifications for the refrigerated dryer (Figure K2) is as follows:

Manufacturer:	Gardner Denver
Model #:	7000100
Rate Capacity:	800 icfm
Compressor Size:	5 HP
Compressor FLA:	18 amps
Fan Size:	1/3 HP
Fan FLA:	3.4 amps
Electric:	230 VAC/3 phase/60 Hz

Figure K2. Building 210 Gardner Denver refrigerated dryer.

The facility currently operates for one 10-hour shift/day, 4 days/week. The compressor operates during the shift (2100 hours of operation per year). The compressed air load varies significantly based on the work being processed through the three buildings. The largest impact occurs when the shot blast booths are being used. They use both process and breathing air. The shot blast booth usage is very sporadic and infrequent.

Measurements taken at the site are as follows:

Compressor:

Voltage:	L1–L2:	445
	L2–L3:	448
	L1–L3:	448
Current:	10:45:	130 amps
	11:00:	125 amps
	11:15:	130 amps
	11:30:	140 amps
	12:00:	160 amps
	12:30:	130 amps

The higher reading at 12:00 took place at the end of the lunch break.

Compressor output: 100 psig @ 175 °F

Compressor Hours of Operation: 11,890 Hours

Dryer Inlet: 104 psig @ 75 °F

Dryer Outlet: 102 psig @ 65 °F

Pressure of air supply in Building 208 (furthest point from compressor): 100 psig

Since there is a small pressure drop between the compressor and Building 208, the pressure of air leaving the compressor should be above 100 psig. Therefore, the gauge on the compressor for output pressure may be reading low.

Summary of Compressed Air System Operational Cost Cutting Opportunities

A detailed compressed air system survey conducted by CERL in 2000 identified seven opportunities to reduce energy operating costs, including the installation of a natural gas engine driven air compressor (see Lin, et al., *Compressed Air System Survey at Sierra Army Depot*, ERDC/CERL TR-00-37, November 2000). The six opportunities other than the NGEDAC included:

- repair compressed air leaks
- change the air compressor control to low demand mode
- disconnect the air receiver from the oil/water separator
- duct outside air into the air compressor room
- install sensor-type valves on the purifier pre-filters
- replace the timer-type drain valves with sensor-type valves.

Collectively, these six opportunities represented annual cost savings of $15,541 in electricity costs, energy savings of 181,409 kWh, and a demand reduction of 49.9 kW. Based on our survey, it does not appear that all of these opportunities have been implemented. Given the recent price increases in electricity, these savings opportunities are increasingly attractive to SIAD.

Potential for Natural Gas Engine-Driven Air Compressor

Site Suitability

The installation of an engine-driven air compressor to operate in place of the existing electric air compressor is straightforward for this site. The NGEDAC that is appropriate for this site is a 125 HP (137 bhp) unit that is capable of producing 600 icfm at 100 psig. The main compressor room has a second room located in the structure that has the appropriate space for installing the engine-driven unit (Figures K3 and K4). The room dimensions are 19 ft long, 20 ft wide, and 9 ft tall. The room contains a unit heater that can be used for freeze protection during the winter months (if needed). The condition of the structure is below average and will require some refurbishing. Areas of focus are the ceiling and an access door.

Figure K3. Main compressor room enclosure.

Figure K4. Second room in main compressor room enclosure: proposed NGEDAC location.

Natural Gas Access

Natural gas is accessible at the building. The gas line and regulator is located outside, at a distance of approximately 120 ft from the main compressor room (Figure K5). The current gas regulator is set at 14 in. w.c. The NGEDAC will require gas pressure of 2–5 psig. Increasing the pressure at the main pressure regulator will require that the regulators for the existing space heating units be replaced to accommodate the increased pressure. Base personnel indicted that this can be easily accommodated. The gas line would need to be extended into Building 210 approximately 20 ft and then run along the ceiling down to the compressor room and back to the location of the unit. The estimated total natural gas piping run is 200 ft.

Compressed Air Piping Interface

Since the NGEDAC capacity is the same as the electric unit's capacity, the compressed air output from the engine-driven unit can be interfaced to the compressed air output of the existing electric unit. The point of interface would be the piping between the output of the electric unit and the input into the receiver. Thus, the existing receiver, filter, dryer, and breathing air purifiers can be used. The total compressed air piping run is estimated to be 25 ft.

Figure K5. Stub-up for natural gas line.

Heat Recovery

There are no process hot water loads within the three buildings. The proposed heat recovery option is to install a hydronic space heater in Building 210. The building is approximately 18,000 sq ft and does not appear to be insulated. The ceiling in the center of the building is in excess of 30 ft high. There are currently gas-fired space heaters suspended from the ceiling throughout the building that supply space heat from October through April. The proposed heating system using heat recovered from the engine-driven air compressor would directly offset the heating requirement of the existing suspended gas heaters. The heat recovery equipment would consist of the heat recovery heat exchanger option on the engine-driven air compressor, interface copper piping, a circulation pump, a forced air hydronic heating unit, and controls. The hydronic heating unit would be suspended from the ceiling near the entrance of the main compressor room. The heat recovery heat exchanger would be capable of recovering approximately 600,000 Btuh of heat from the engine coolant and the compressor oil of the new compressor.

Ducting Modifications

The installation of the engine-driven air compressor will require some ducting to vent the engine exhaust from entering the compressor inlet air. The engine exhaust may trip the alarm for the breathing air purification system. To accommodate this requirement, it is proposed that ducting for the inlet air to both the new compressor and the existing electric unit be installed. The inlet ducting would be run towards the building and extended upward. The resulting distance between the inlet air and the engine exhaust would be approximately 50 ft.

Economic Analysis

The principal benefits of the NGEDAC unit for SIAD include:

- net savings in operating costs
- hedge against power disruptions—operates on natural gas, not electricity
- added capacity/redundancy for the compressed air system

The following estimates the operating costs and benefits.

Operating Cost Comparison

A 125 hp NGEDAC unit capable of providing 600 icfm at 100 psig was evaluated to serve 100 percent of the load currently met by the existing 125 hp Gardner-

Denver electric motor drive-air compressor. Tables K4 and K5 below summarize the energy performance and costs associated with the proposed unit. The load is based on a 10 hr/day, 4 day/week schedule (2080 hours/yr.), assuming an average loading of 90 percent of design capacity (540 icfm).

The results shown are based on the most recent electric and gas prices as indicated. Table K6, below, shows changes in the annual operating costs of the NGEDAC system based on possible changes in future electric rates or gas prices. Note also that the maintenance costs for the NGEDAC are a function of the hours of operation for a given size unit. Table K7 summarizes NGEDAC system capital cost components.

Table K4. Compressor performance characteristics at design load.

Parameter	Electric Air Compressor	NGEDAC
Compressed air capacity	600 icfm	600 icfm
Motor/engine power	125 hp (137 bhp)	125 hp (137 bhp)
Full load power	100.9 kW	1.041 MBtuh
Efficiency	6.00 icfm/kW	576.4 icfm/MBtuh

Table K5. Annual energy use and operating costs baseline energy price assumptions.

Parameter[1]	Electric Air Compressor	NGEDAC	Net Savings
Energy Use	225,472 kWh	2,166 MBtu gas (engine)	225,742 kWh (elec.)
		- 300 MBtu (engine heat recovery)[2]	-1,866 MBtu (gas)
Peak demand	100.9 kW	1.041 Mbtuh	
Energy operating costs	$23,675	$15,769	$7,906
Peak demand costs	$9,106		$9106
Operation & maintenance costs	$3,072	$5,899	-$2,827
Heat recovery costs	0	-$2,188	$2,188
Total costs	$35,853	$19,481	$16,373

[1]Electricity Costs: $0.145/kWh average—includes demand @$7/kW and energy charges @.105/kWh (2/6/01 rate)
Natural Gas Costs: $7.28/MBtu (Average for April 2000—March 2001)

[2]Based on (0.22/.8) *heat value of natural gas into the engine, where 0.22 is the fraction of recoverable heat from the engine coolant and compressor oil and 0.8 is the assumed efficiency of the heating unit.

Table K6. Annual operating costs ($)—sensitivity to changes in energy prices.

Energy Price Assumptions	Electric Air Compressor	NGEDAC	Net Savings
Higher Elec. Rates/Base Case Gas Rates			
1) Elec.: $0.16/kWh and Gas: $7.28/Mbtu	39,131	19,480	19,651
2) Elec. $0.175/kWh and Gas: $7.28/Mbtu	42,409	19,480	22,929
Base Case Elec. Rates/Lower Gas Rates			
1) Elec.: $0.145/kWh and Gas: $6.62/Mbtu	35,853	18,245	17,608
2) Elec. $0.145/kWh and Gas: $6.07/Mbtu	35,853	17,216	18,637
Lower Elec. Rates/ Gas Rates			
1) Elec.: $0.10/kWh and Gas: $4/Mbtu	25,680	13,361	12,319

Table K7. NGEDAC capital costs.

Cost Element	Cost ($)
125 hp NGEDAC	144,500
Compressor enclosure	10,000
Heat recovery	34,000
Installation	28,000
Freight	4,000
Total	211,500

Appendix L: Scope of Work, CAS Survey Level I & II

Level I Compressed Air System Survey

Objectives:

1. Provide the plant with a compressed air system overview at a cost commensurate with the system size, complexity, and potential savings recovery.
2. Generate short/longer term plans to establish basic control and management of the air system. Focus on what is generally needed to pull together all of the interrelated parts of the system and allow the user to understand the "basics" of these components and their relationship.
3. Create a general guide that the user can follow to continue to increase the efficiency of the system.
4. Identify specific programs and/or actions to be implemented with estimated costs and pay back.
5. Discuss with key plant staff known problems with CAS performance, operation, and capacity and identify plans for plant modifications that directly or indirectly impact the compressed air system.

Initial review should often lead to additional follow-up programs, or even to completely controlled and fully managed control systems. The success should lead the user to more in-depth programs, as they continue to press for efficiency and cost reductions.

Scope of Work: Level I—Supply Side Review

Step 1. Evaluate the Existing Air Compressors as to:

- suitability for application
- general apparent performance and condition (without disassembly or mechanical work)
- efficiency ratings
- suitability of unloading controls
- capable of translating power demand into lower power cost
- potential for modification if required

- capability for system sequencing, etc.
- installation and support systems (i.e., cooling water, ventilation, etc.)
- general appraisal of alternate types of equipment and controls that may be more preferable and/or more power efficient
- staff concerns regarding supply system performance and company plans that may impact supply system.

Step 2. Evaluate the Compressed Air Treatment Equipment—as to installation, general apparent conditions and performance (without disassembly or mechanical work); suitability for application general effect on efficiency and energy costs. Specifically:

- *Aftercoolers*—Effectiveness to reach 100 °F for dryer inlet; possible use of auxiliary coolers; installation critique
- *Dryers*—Suitability for application/sizing/efficiency/pressure loss/controls; possible modifications to improve performance/efficiency
- *Filters*—Suitability for application/efficiency/pressure loss/alternates
- *Auto Drains*—Are they used?; applied correctly?; alternates
- *Supply Side Piping*—(from compressor to system storage vessel) suitability for application/efficiency/pressure loss/alternates
- *Air Receiver Placement*—Ability to control of air; ability to store dry air; ability to serve function.

Scope of Work: Level I—Demand Side Review

- Distribution Piping: (particularly main headers)
- Efficiency ratings/pressure loss/moisture control/modification
- Identify the apparent "Load Profile" per shift by total plant/by sector (is possible)
- Identify the lowest pressure required to operate the production equipment at optimum performance. Identify which sectors limit this
- Evaluate the potential effectiveness of a demand side control main system and sub systems
- Look into specific areas of low pressure and air distribution problems/moisture and oil carryover problems, if applicable
- Identify the electric power cost of compressed air at the facility; i.e., the cost per cfm: cost per psig. Translate this to the cost of leaks and otherwise wasted air. Locate, identify, quantify, and assign a "cost of loss" or "recovery" to specific examples of "leaks," i.e., open drains, unregulated flows, blow offs
- Identify an accurate estimate of the total plant leakage by any of several methods
- Recommend the implementation of a continuing "leak management" program supervisor and line personnel training, flow identification, etc.

- Identify areas where high pressure air may not being used productively: open blow, low pressure, vacuum generators, etc.
- Identify potential uses for air saving devices such as Venturi nozzles, heat tubes, etc.
- Identify any "demand events" that might be handled by more effective storage application rather than continual loading/unloading of compressors.
- Identify any areas of potential thermal energy recovery using the generated "heat of compression."
- All compressed air energy conservation recommendations will be listed in the executive overview, giving the estimated implementation cost and predicted fiscal recovery.
- Staff concerns with demand system performance and company plans that may impact demand system.

Level II Compressed Air System Survey (for the selected demo sites)

The Level II Review adds four key activities not included in the "Level I" work scope:

1. A fully measured and trended evaluation of air flows and pressure to help establish usage baselines needed to document savings levels on a "before/after" basis
2. A leak survey of all major compressed air uses, as required
3. Verification of overall leak levels based on pressurizing the air system during nonproduction times
4. Creation of project specification write-ups that include a description of the recommended project, estimate of project savings, and estimate of project costs based on vendor quotes.

Flow and pressure measurements taken over an extended period of time provide the best-detailed load profile available, but is expensive to implement. In addition, effort would be spent on measuring a system soon to be changed. Generally, the measuring equipment is or can be left in place for post-verification but sometimes this is not practical and/or may no longer be measuring the right parameters for the modified system.

The Level II audit includes setting up a full trending analysis of selected measurements and readings over a 24-hour period. In this case, the equipment will be supplied and researchers will either perform the installation or supervise the installation of the transducers, meters, etc.

The second key activity in the Level II audit is a leak survey of all major compressed air uses, as required. Detailed attention is given to locating, tagging and quantifying leaks and open blow-offs. When performed without an overview and/or system analysis (i.e., a Level I Audit), particularly controls, piping, etc., the result can often be less air used with no significant change in the power bill.

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14. ABSTRACT

Compressed air (CA) is an indispensable commodity used in manufacturing and maintenance facilities. As in the private sector, Department of Defense (DOD) facilities make widespread use of air compressors. Although CA is a convenient power source, CA systems are not cheap to operate. An analysis of the cost breakdown of a CA system shows that as little as 10 percent of the input power supplied to the compressor is delivered as CA to the system. The electricity used by these air compressors is a major contributor to annual energy operating costs. The use of natural gas engine driven air compressors (NGEDACs) in place of conventional electric motor driven air compressors offers an opportunity to reduce these costs. The savings offered by this alternative technology can more than offset the capital and maintenance costs of its installation. This survey was undertaken to evaluate CA systems at Army industrial facilities to identify opportunities for energy savings—by improving system efficiencies and by identifying suitable candidates for installation of NGEDACs.

15. SUBJECT TERMS

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