1 of 2

INDUSTRIAL COMPRESSED AIR SYSTEM ENERGY EFFICIENCY GUIDEBOOK

Prepared Under

Bonneville Power Administration

Contract No. DE-AC79-90^B₁P03940

Prepared by

Carroll, Hatch & Associates, Inc.

Portland, Oregon

MASTER

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

872

Disclaimer

This report was prepared by Carroll, Hatch and Associates as an account of work sponsored by the United States Government. Neither the United States, nor the Bonneville Power Administration, nor any of their contractors, subcontractors, or their employees make any warranty, express or implied, or assumes any legal responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product or process disclosed within this report.

Reference herein to any specific commercial product, process, trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply an endorsement, recommendation or favoring by the United States Government or any agency thereof.

The views or opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

BPA Report Summary - Industrial Technology

SUBJECT

Industrial Compressed Air Systems

TITLE

INDUSTRIAL COMPRESSED AIR SYSTEM ENERGY EFFICIENCY GUIDEBOOK

SUMMARY

Energy efficient design, operation and maintenance of compressed air systems in industrial plants can provide substantial reductions in electric power and other operational costs. This guidebook will help identify cost effective, energy efficiency opportunities in compressed air system design, re-design, operation and maintenance.

The guidebook provides: 1) a broad overview of industrial compressed air systems, 2) methods for estimating compressed air consumption and projected air savings, 3) a description of applicable, generic energy conservation measures, and, 4) a review of some compressed air system demonstration projects that have taken place over the last two years.

The primary audience for this guidebook includes plant maintenance supervisors, plant engineers, plant managers and others interested in energy management of industrial compressed air systems.

BPA PERSPECTIVE

Conservation of electricity is a cornerstone of the Bonneville Power Administration's (BPA's) resource program. BPA is interested in promoting the efficient use of electric power. The industrial sector is the largest electric power consuming sector in the Pacific Northwest. Within the industrial sector, compressed air systems are one of the largest users of power. One study estimated that 160 average megaWatts of electric power savings is possible through the energy efficient use of compressed air systems.

BACKGROUND

Most industrial facilities throughout the world use compressed air to operate large equipment, hand tools, conveying systems, control systems, etc.. Compressed air may operate stamping presses, air wrenches, aerating equipment, sanders, vacuum systems, conveying and pneumatic controls, and at the same time be part of a chemical or manufacturing process.

Compressed air systems provide numerous advantages such as: 1) lower tool weight to horsepower ratio, 2) reduced fire hazard, 3) compressed air tools do not overheat or overload, and 4) compressed air equipment and tools require less maintenance.

There are disadvantages to compressed air systems: 1) For every unit of useful work, four units of electrical energy have been used in the preparation of the air. In other words, compressed air systems are around 25% efficient; 2) When industrial facilities are expanded, compressed air systems are generally not re-designed which can cause a significant waste of electric power.

OBJECTIVE

To assist the industrial sector in the cost effective, energy efficient design, re-design, operation and maintenance of compressed air systems.

APPROACH

Energy conservation opportunities exist in at least three general areas. The first area considered will be compressor design for part load operation and efficient sequencing (turning compressors' on and off). The second area for consideration is the distribution system design, re-design and leak prevention. Finally, the way air is used or misused will offer conservation opportunities.

PROJECT LEAD

Craig Ciranny
Technology Development Team - RMCC
Bonneville Power Administration
905 N. E. 11th Avenue
Portland, Oregon 97208
(503) 230-5865

ORDERING INFORMATION

REPORT NUMBER: DOE/BP-03940-1

For additional copies of this report, contact the Public Information Office, ALP-22, Bonneville Power Administration.

Document request lines

Portland, Oregon: 230-3474
Outside of Portland: 1-800-622-4520

TABLE OF CONTENTS

SECTION NO.		PAGE
1.0	Introduction	1 - 1 to 1 - 2
1.1	Purpose of the Guidebook	1 - 1
1.2	Expected User Audience	1 - 2
1.3	Scope and Limitations	1 - 2
2.0	The Compressed Air Utility	2 - 1 to 2 - 2
2.1	Why Use Compressed Air	2 - 1
2.2	Cost of Compressed Air	2 - 1
3.0	The Compressed Air System	3 - 1 to 3 - 6
3.1	Components of a System	3 - 1
3.1.1	Compressor and Drive	3 - 1
3.1.2	Equipment for Conditioning Air	3 - 2
3.1.3	Air Distribution System	3 - 3
3.1.4	Air Consuming Equipment	3 - 4
3.2	Cost of Operating and Maintaining Systems	3 - 4
3.2.1	Electrical Costs	3 - 4
3.2.2	Amortization of First Costs	3 - 5
3.2.3	Maintenance	3 - 5
3.3	Multiple Compressor Operations	3 - 6
3.3.1	Versatility in Volume	3 - 6
3.3.2	Versatility in Control	3 - 6
4.0	Compressors	4 - 1 to 4 - 17
4.1	Types of Compressors in Common Industrial Use	4 - 1
4.1.1	Reciprocating Air Compressors	4 - 1
4.1.2	Rotary Screw Compressors	4 - 2
4.1.3	Centrifugal Compressors	4 - 3
4.1.4	Other Types of Compressors	4 - 4
4.2	Compressor Drives and Electrical Systems	4 - 4
4.3	Full Load and Part Load Operation	4 - 5
4.3.1	Efficiency at Full Load	4 - 5
4.3.2	Part Load Operation	4 - 6
4.4	Compressor Performance Curves	4 - 6
4.4.1	Volume/Power Effectiveness at Part Load	4 - 6
4.4.2	Importance of Understanding Performance Curves	4 - 6
4.5	Methods of Controlling Compressor at Part Load	4 - 8
4.5.1	"On - Off" Control	4 - 8
4.5.2	Low Unloaded Power Control	4 - 9
4.5.3	Modulation Methods of Part Load Control	4 - 11
4.6	Sequencing Multiple Compressors	4 - 14
4.6.1	Coordinating Resources and Matching Load	4 - 15
4.6.2	Sequencing Controllers	4 - 15

TABLE OF CONTENTS CONTINUED

SECTION	PAGE
4.7 Air Demand Profile	4 - 17
4.7.1 How to Develop a Demand Profile	4 - 17
4.7.2 How to Use a Demand Profile	4 - 17
5.0 Equipment for Conditioning Air	5 - 1 to 5 - 5
5.1 Inlet Air Filter	5 - 1
5.2 Intercooler	5 - 1
5.3 Air/Oil Separator	5 - 1
5.4 Aftercooler	5 - 2
5.5 Air Dryer	5 - 2
5.6 Reheaters	5 - 4
5.7 Filters	5 - 4
5.8 Pressure Gauges	5 - 5
6.0 Other Compressed Air System Equipment	6 - 1 to 6 - 3
6.1 Drive Motors	6 - 1
6.2 Motor Starter	6 - 1
6.3 Pressure Switches and Pressure Transducers	6 - 1
6.4 Relief Valves	6 - 2
6.5 Traps and Drains	6 - 2
6.6 Heat Recovery Heat Exchangers	6 - 3
7.0 Compressor System Environment	7 - 1 to 7 - 2
7.1 Space	7 - 1
7.2 Access to Free Air	7 - 1
7.3 Light	7 - 1
7.4 Proximity to Other Equipment	7 - 2
8.0 Maintenance	8 - 1 to 8 - 3
8.1 Tests for Standard Performance	8 - 1
8.1.2 Compressor Performance Tests	8 - 1
8.1.2 Pressure Drop Tests	8 - 1
8.2 Regular Maintenance Program	8 - 2
8.2.1 Items to Include	8 - 2
8.2.2 Frequency of Attention	8 - 2
9.0 Compressed Air Distribution System	9 - 1 to 9 - 4
9.1 Purpose of the Distribution System	9 - 1
9.2 Allowable Pressure Drop	9 - 1
9.3 Piping Design	9 - 2
9.4 Receivers	9 - 4

TABLE OF CONTENTS CONTINUED

SECTION	PAGE
10.0 Air Consumption	10 - 1 to 10 - 3
10.1 Sizing Devices, Design Volume and Pressure	10 - 1
10.2 Intended Losses	10 - 3
10.3 Unintended Losses	10 - 3
11.0 Calculating and Valuing Savings	11 - 1 to 11 - 7
11.1 Establishing Compressor Performance Curves	11 - 1
11.2 Establishing A Demand Profile	11 - 5
11.3 Establish Air Demand Curves	11 - 6
11.4 Illustration of Savings	11 - 6
11.5 Value Savings	11 - 7
11.6 Section Summary	11 - 7
12.0 Investment Justification	12 - 1 to 12 - 3
12.1 Cost of Improvement	12 - 1
12.2 Value of Improvement	12 - 1
12.3 Method of Evaluation	12 - 1
13.0 Generic Conservation Measures	13 - 1 to 13 - 4
13.1 Repair System Leaks	13 - 1
13.2 Sequencing of Multiple Compressors	13 - 1
13.3 Proper Selection of Compressor Capacity Control Options	13 - 1
13.4 Compressor Performance Improvement	13 - 1
13.5 Optimization for End-Uses of Compressed Air	13 - 2
13.6 Switch End Use Device to Another Utility	13 - 2
13.7 Reduction of Piping and Distribution System Losses	13 - 2
13.8 Reduction of System Discharge Pressure	13 - 2
13.9 Compressor Lubrication	13 - 2
13.10 Proper Selection and Operation of Air Dryers and Purge Air Controls	13 - 3
13.11 Relocation of Air Intake to Outdoors	13 - 3
13.12 Improvement of Filter Condition	13 - 3
13.13 Improvement of Intercooler, Oil-cooler and Aftercooler Condition	13 - 3
13.14 Installation of Dew Point Monitoring	13 - 3
13.15 New Compressor	13 - 4
14.0 Selected Plant Studies	14 - 1
Case Study No. 1 - 25 hp ASD Demonstration	14 - 1
Case Study No. 2 - Add Air Dryer and Receiver to System	14 - 8
Case Study No. 3 - Synthetic Lubricants	14 - 14
Case Study No. 4 - Equip a 150 hp Compressor with an ASD	14 - 17
15.0 Bibliography	15 - 1

List of Tables and Figures

Tables	PAGE
Table 3 - 1 Electrical Costs	3 - 5
Table 9 - 1 Desired Air Flows	9 - 1
Table 10 - 1 Air Consuming Devices	10 - 1
Table 11 - 1 Field Data Form	11 - 2
Table 11 - 2 Field Data Form	11 - 3
Table 12 - 1 Simple Payback Determination	12 - 1
Table 12 - 2 Loan Terms	12 - 2
Table 12 - 3 Five Year Loan Schedule	12 - 2
Table 12 - 4 Five Year Cash Flows Projection	12 - 3

Figures	
Figure 3 - 1	Compressed Air System 3 - 1
Figure 4 - 1	Reciprocating Compressor 4 - 2
Figure 4 - 2	Rotary Screw Compressor 4 - 3
Figure 4 - 3	Centrifugal Compressor 4 - 4
Figure 4 - 4	Compressor Performance Curve 4 - 7
Figure 4 - 5	VPER Performance Curve 4 - 8
Figure 4 - 6	"On-Off" Control 4 - 9
Figure 4 - 7	Low Unload Power 4 - 10
Figure 4 - 8	Inlet Damper Modulation 4 - 12
Figure 4 - 9	ASD Speed Control 4 - 14
Figure 4 - 10	ASD and Unloaded Control 4 - 14
Figure 9 - 1	Pressure Drops in Pipes 9 - 2
Figure 9 - 2	Air Distribution System Layouts 9 - 3
Figure 11 - 1	Test Rig Setup for Non Modulating Machines 11 - 1
Figure 11 - 2	Performance Curve Data Points 11 - 2
Figure 11 - 3	Test Rig Setup for Modulating Machines 11 - 3
Figure 11 - 4	Performance Curve Data Points 11 - 4
Figure 11 - 5	Power Demand Curve 11 - 5
Figure 11 - 6	Air Demand Curve 11 - 6
Figure 11 - 7	Modified Air Demand Curve 11 - 7
Figure 11 - 8	Modified Power Demand Curve 11 - 7

1.0 Introduction

1.1 Purpose of the Guidebook

This book is intended to provide direction to an industrial type end user who is involved in an electrical energy conservation program. Specifically, energy conservation for compressed air systems is the subject of the book.

The book provides a broad overview of compressed air systems. Reasons for using compressed air as a utility are included. Typical compressed air systems are divided into logical segments of equipment groups. Each segment is discussed.

Types of compressors are reviewed. Advantages and limitations of various types of compressors are described. Common methods for controlling the operation of compressors at less than full volume are presented. Measures of energy performance effectiveness are described and illustrated.

Ancillary equipment commonly used in the industrial compressed air system is described. Environmental considerations for compressor or compressor room locations are reviewed. Maintenance practices for proper compressor operation are presented.

Methods for determining compressed air consumption are included. Compressed air conservation measures are presented with an emphasis on estimating air savings. The book presents an explanation for converting projected air savings into projected electrical energy savings.

The user is guided through the process of establishing a value on projected air savings so that a realistic economic justification of conservation efforts can be developed.

There is a section of the book devoted to describing generic energy conservation measures that can be applied to compressed air systems. The user may find one or more of the generic measures readily applicable to his own system.

Finally, there is a section of the book devoted to reviewing energy conservation demonstration projects that have taken place over the last two years. The projects include new techniques as well as traditional practices used in compressed air systems.

Energy conservation opportunities may be found in at least three general areas. The first area considered will be the compressor part load control and sequencing areas. Many compressors in the industrial arena are being operated in a wasteful fashion. The second area will concern itself with opportunities in the air distribution systems. Well designed distribution systems will see that air is delivered to a site at the lowest possible cost. Finally, the ways air is used or misused will offer conservation opportunities.

By reading through the book the user should begin to acquire a feel for how energy interfaces with compressed air. The feel should then extend to how the written material may apply to the user's situation.

1.2 Expected User Audience

The guidebook is intended for use by plant maintenance supervisors, plant engineers and others who have responsibility for the design, operation and maintenance of compressed air systems.

1.3 Scope and Limitations

Most industrial compressed air systems are operated in a range of 80 psig to 150 psig. This is the user system pressure range for which the book is intended. The book is not intended to cover air blower systems, which often operate in the range of 4 psig to 12 psig.

Suggestions contained in the text assume the use of typical test instruments normally found in an industrial setting. These include multimeters for measuring electrical power system parameters. Some instrumentation, i.e. air flow meters and recording power meters, may have to be rented or otherwise acquired by the user in order to carry out some suggestions.

The book assumes the user is familiar with typical industrial plant operation. There are a number of safety practices that must be exercised by people who work around industrial equipment. This book is not a safety manual for compressed air systems. The user must use prudent judgment and employ normal industrial safety practices when carrying out any suggestion made in the guidebook.

Any potential user of information in this book who is not completely familiar with the industrial environment should not attempt to carry out suggestions. There is always a danger of self injury when the user is working in unfamiliar areas. Execution of suggestions is intended for use by people with industrial air compressor system experience.

2.0 The Compressed Air Utility

2.1 Why Use Compressed Air?

Compressed air is a utility, similar to power, steam and water, that is found in most industrial facilities throughout the world. There are numerous manufacturing and service advantages available by users of compressed air.

Compressed air is used in virtually every phase of manufacturing in the United States and other industrially developed countries. Compressed air may actuate stamping presses, air wrenches, aerating equipment, conveying and pneumatic controls, and at the same time be part of a chemical or manufacturing process. At one time a rule of thumb in manufacturing said that at least 1 hp of compressor capacity was needed for each 100 square feet of manufacturing area, regardless of the product or process. Industries that are more air-intensive than the ordinary would adjust this figure accordingly.¹

An investigation into the major industrial users of compressed air becomes an investigation into the largest industries. Rank ordering industries by compressed air use merely rank orders the industries by total size of operations. Compressed air is essentially a form of energy. It ranks in breadth of use only below electricity, gas and water.²

Tools employing compressed air have certain advantages over electrically-driven tools. Tools using compressed air as a motive force have a lower weight to horsepower ratio than do electric tools. Consequently, they are the tool of choice in a factory situation in which an employee is required to spend an entire shift using the tool.

Compressed air products and tools do not produce sparks or objectionable exhaust. They are safe to use in an environment that has flammable or explosive liquids, and they are also safe to use in confined spaces.

Compressed air tools require less maintenance. They do not have a tendency to self destruct, like electrically driven tools, if they are overloaded.

There is, however, a substantial cost associated with this utility. Compressed air is an expensive form of energy when compared to the cost of other forms of utility services.

2.2 Cost of Compressed Air

In modern industrial plants an air compressor is driven by way of an electric motor. Air is compressed in a machine with a certain volumetric efficiency. During the compression process a great deal of heat is developed at the compressor. The heat of compression is carried away by a forced air or forced water cooling system. In most plants that heat is discarded.

By the time compressed air has been processed and is ready for use as a motive force, about 4

¹ Purchasing, July 11, 1979, (31)

² Brand, Horst, & Huffstuler, Clyde, Productivity in the Pump and Compressor Industry, Monthly Labor Review, December 1982

hp of electrical energy has been put into preparation of the air for every 1 hp of useful work that can be expected from the air. When the amortized cost of installed compressor equipment is added to the operating cost, the use cost of compressed air is often four to five times more expensive than the cost of electrical energy.

An example of how one task can be addressed in two different ways will make the point about relative costs. A mixture in a bucket needs to be agitated. The agitator motor needs to be 1 hp in capacity. An electric motor will use about 0.75 Kwh of electrical energy per hour to drive the motor. An air motor needs about 20 cfm of compressed air for operation. At full capacity a typical air compressor uses 1 hp to develop 4 cfm. A five hp motor is needed to drive a compressor which delivers 20 cfm. A five hp motor will use about 3.7 kWh to operate for one hour. With the air motor, electrical energy use is five times higher than with an electrically driven motor.

A prudent manager will be very careful about where and how compressed air is to be used as a utility in the plant. An electric motor can accomplish the same task for about 25% of the cost of an air system. Hydraulic power systems can often accomplish what a compressed air system does for half the energy cost. An internal combustion engine can be used in some cases more efficiently than compressed air motivation.

There are some tasks that compressed air systems clearly address more completely and more satisfactorily than any other form of energy. A compressed air system should be designed to address these uses.

3.0 The Compressed Air System

A compressed air system is customarily made-up of several parts. This section of the guidebook presents an overview of the system. Later sections of the book will detail the individual parts of the system.

Figure 3 - 1 Compressed Air System

Figure 3 - 1 is a schematic representation of a total compressed air system.

3.1 Components of a System

3.1.1 Compressor and Drive

The compressor and drive are the first significant parts of the system. There are several different types of compressor designs. Each design is intended for a particular type of use. Compressors are discussed in Section 4.1. In the industrial environment, most compressors are driven via electric motors. Occasionally a unit is driven via a steam turbine, but these are rare exceptions. In construction situations, compressors are often driven by gasoline or diesel internal combustion engines.

This book is intended to address industrial situations where compressors are driven with electric motors. Motors are frequently supplied with the compressor on a common frame. Many compressors are directly driven, in that the compressor drive shaft is coupled to the motor armature shaft. Other units are driven through v-belts and sheaves.

In order to function, the motor must be supplied with electricity. The industrial user pays for the electricity supplied to the motor. For maximum energy efficiency, it is important for the motor to deliver energy to the compressor with a minimum of losses. There are two sources of

energy loss between the input to the motor and the compressor. One is the electric efficiency loss through the motor and the second is the power transmission loss between the motor and the compressor.

Efficiency loss at the motor must be addressed when the drive is acquired for the compressor. Energy efficient drives are available at premium prices. The user is well advised to investigate the use of the most energy efficient drive available. Once a drive is in place, the user must live with that drive efficiency for the length of the drive's useful life. A 1% efficiency difference can amount to thousands of dollars of operating cost difference over the operating life of a medium sized compressor.

If an existing drive fails and the owner chooses to replace it with an energy efficient drive, care must be taken to see that the new motor has all of the operating characteristics that the old one did, including the same or a greater service factor. Do not put a motor on a compressor with a smaller service factor, since compressor manufacturers typically use that portion of the motor's operating range.

Power transmission losses can be minimized through proper initial design and proper equipment maintenance. Good maintenance practices will be emphasized throughout this book.

3.1.2 Equipment for Conditioning Air

Compressed air conditioning equipment makes up the second group of items in the system. Conditioning equipment includes: air/oil separators, heat exchangers, filters, dryers and other kindred items. Each one of these devices is intended to improve the usefulness of the air.

The extent to which a system is fitted with conditioning devices depends on the use for which compressed air is intended. In an industrial plant where the principal use of air is for hand tools and there will be no adverse effects to a dryer, there is little need to filter out all traces of lubricating oil. As a matter of fact, the oil is often beneficial for the tools. Conversely, if the compressed air is going to be used as instrument air, it is necessary to have a very high degree of cleanliness so that the instruments are not contaminated from air-borne oil and water.

Immediately after compression, air temperatures may reach 212°F or more. It is desirable to reduce air temperature to a safer and more manageable level. Most systems utilize an after-cooler to remove the heat of compression from the compressed air stream. The after-cooler is a heat transfer device used to reduce air temperature.

When air is compressed, water vapor contained in the ambient air supply is carried along with the compressed air. Compressed air often becomes saturated as it is cooled in an aftercooler. There is almost always a condensate stream associated with the aftercooler. Additional cooling after the aftercooler will result in water condensing within the system.

Since additional cooling will almost always occur, it is usually necessary to deal with water in the system. Most modern systems use an air dryer to remove the bulk of water from the compressed air before the air is delivered to the distribution system. There are a number dryer designs. Some are more energy-efficient than others. Some remove more water than do others. Again, the use for which the compressed air is intended will dictate the type and extent of air drying equipment for the system.

Once the air stream is cooled and dried, it is common to raise the air stream temperature so that additional condensation is not likely to occur. The heat of compression developed at the compressor is frequently used to raise air temperature in the reheater. Water-cooled compressors accommodate this feature better than air-cooled machines, since it is a relatively easy task to pipe the hot water to the reheater.

In order to assure efficiency in the system, the air conditioning equipment must be continuously maintained. Dirty filters and neglected dryers will cause higher energy use and reduce system performance. When a system is designed the designer must consider the future maintenance tasks.

3.1.3 Air Distribution System

After compressed air is properly conditioned, it must be delivered to the using devices. Delivery is accomplished via the distribution system. Often the system begins with a receiver tank located close to the compressor. The receiver tank feeds a piping system that carries air throughout the plant.

A well designed piping system balances the cost of building the system with the performance requirements of the system. Performance is described in terms of delivering a required amount of air to a specific point in the system at a desired pressure. When compressed air travels within a pipe there is friction between the fluid and the pipe walls. The amount of friction depends upon the velocity of the fluid in the pipe. Friction causes pressure loss.

When more air volume is forced through a given diameter pipe, the air velocity increases. There is a limit to the amount of volume that can be forced through a given diameter. The limit is usually defined in terms of allowable pressure drop. Interaction between air velocity and pipe wall develops pressure drop in an exponential relationship. Eventually, small increases in velocity are met with very large pressure drops.

The piping designer must select a large enough diameter pipe to prevent the air pressure from falling below a critical level before it reaches the destination. The designer must not use too large a pipe, unless there is a high probability the plant will be expanded in the future, because that causes the cost of the system to be unnecessarily high.

Many air consuming loads are intermittent. Often there will be a short term high demand and then periods of little demand. Receiver tanks strategically located throughout the distribution system act as reservoirs, which provide short term air supply to boost the delivery capability of the system. Receiver tanks tend to keep air pressure high by addressing momentary demands.

Receiver tanks also enable some compressors to operate more efficiently during times of lower air demand. With a large receiver, an unload control will allow the compressor to remain in the unloaded condition for longer periods of time. A receiver tank also acts as a cushion for smoothing-out air delivery from pulsating-type compressors.

Typically, the initial air distribution system in an industrial plant is adequate for the plant. Problems often begin to be encountered when the system demands increase. Many plants add compressor capacity to meet the increased need without addressing the increased distribution system load. This often works against the desired intent because of large pressure drops that

occur when too much air is forced through existing pipes.

Unless the air distribution system is well maintained, the system can be responsible for major air losses through leaks. It is not unusual to encounter systems where 50% of the load is required to feed system leaks. Industrial users face a continuing task of maintaining system integrity by fixing leaks. This task never goes away!

3.1.4 Air Consuming Equipment

The final group of elements in the compressed air system are the consuming devices. These devices consist of: air cylinders, machine tools, indexing devices, air motors, instruments, nozzles and countless other items of hardware. The most important consideration for this group of items is, ". . . is the device really necessary? . . . Is there another way to accomplish the task that is less energy intensive?

Once the plant engineer or manager has determined that the use of a compressed air device is the optimum way to proceed, the question becomes, ". . . do I have the most energy effective device?".

Finally, it is important to keep the devices operating at their most efficient levels. Again, good maintenance practice becomes an on-going challenge.

3.2 Cost of Operating and Maintaining Systems

3.2.1 Electrical Cost

By-and-large, energy costs associated with industrial air compressors are costs for electricity. The operating cost for a given machine is made up of a matrix of electrical costs. Typically, a utility has a number of elements in the cost schedule. Often there is an administrative fee that is a fixed value payable on a monthly basis. Second, there is usually a demand cost which is a function of the largest total demand experienced at the plant for a specified period of time, sometimes as short as 15 minutes, most frequently, 30 minutes. This is referred to as "peak" demand.

In most cases the largest cost element is the energy cost. This is a value for each kilowatt-hour that is consumed by the compressor drive. Total energy consumption is dependent upon the length of time the unit is operated and dependent upon the demand load experienced while the unit is operating. An integrated value of demand over time provides the energy consumed by the compressor drive.

Cost of energy is a function of the amount of energy consumed and the value assessed per unit of energy.

A poorly loaded air compressor will contribute to a lower plant power factor. When the power factor is below a required minimum, the plant is penalized with a low power factor charge.

A typical monthly cost for electricity to operate a 100 hp compressor is presented in Table 3 - 1. The unit operates 2 shifts per day, 5 days per week at an average level of 80% capacity, (90% power level or 67 KW). This compressor utilizes inlet valve modulation for part load

control.

Cost Element	Value	Experience	Monthly Cost
Demand Charge	\$5.50/KW	75 KW Peak	\$412.50
Energy Charge	\$0.025/kWh	23,208 kWh/mo	\$580.20
Total Cost			\$992.70

Table 3 - 1

On an annual basis the electrical cost will be about \$11,912 for operating the compressor. Not included in the table are portions of other costs, i.e., basic charge and power factor penalty fee.

It is clear that the cost of operation depends on both demand level and energy consumption as well as on the value of electrical energy. It is quite likely in many industrial areas that energy costs for heavily loaded compressors will exceed the original compressor cost on an annual basis. This is good reason to make efforts to obtain energy efficient machines and drives in the initial stages.

Actual electrical costs for a compressor system are dependent on the utility rates that prevail in the area in which the plant is located. It is incumbent upon the plant engineer to understand exactly how the power bill is calculated for his/her facility. The understanding must be extended to include how the use of the air compressor(s) effects electrical costs.

3.2.2 Amortization of First Costs

The capital cost of a 100 hp compressor with drive, ancillary equipment and distribution system on an installed basis, is in the range of \$50,000. For internal costing purposes (non-tax considerations), it would be logical to depreciate this asset on a straight-line basis over 10 years (minimum expected life of the equipment). Annual amortization expense would be \$5,000 per year.

This value is much less than the cost of energy to operate the machine. One can legitimately conclude that compressor ownership cost is much less than the cost to operate the compressor.

3.2.3 Maintenance Costs

Most industrial air compressors will operate for many years if they are properly maintained. The type and degree of maintenance depends on both the type of compressor involved and the duty to which it is subjected.

Annual maintenance costs can be expected to be in the range of 5% of initial capital cost. The previously described system could be expected to have maintenance costs of \$2,500 per year. This is less than 20% of the cost to operate the machine.

One can conclude from these discussions that the principal cost of operating air compressors is the energy cost.

3.3 Multiple Compressor Operations

Some new air systems are designed with more than one compressor to meet system demand. When the system demand is expected to be highly variable by time of day, a multiple compressor system makes economic sense. This allows only the needed aggregate compressor capacity to be turned on at any one time.

3.3.1 Versatility in Volume

A common characteristic of air compressors is that each machine is at maximum efficiency when the machine is delivering rated capacity. If all system demands were constant, one compressor could supply the demand at optimum energy efficiency. However, most system demands are variable, so a compressor must operate at less than full capacity (part load) some of the time. Depending upon the type of compressor, i.e. rotary screw, centrifugal, reciprocating, and type of part load control used on the compressor, operation at part load can be very energy inefficient.

When part load operation occurs over a significant portion of the operating day, it may be advisable to use more than one compressor. With multiple compressors, several units may be operating at full capacity and maximum efficiency while only one smaller unit is in part load operation. In section 4.4 of this book there will be discussions about the technique of sequencing compressors via automatic control systems.

3.3.2 Versatility in Control

No matter how diversified a system becomes, there will always be at least one machine that will need to operate in a part load mode. Compressor manufacturers have adopted a number of methods which facilitate part load operation. Some of these methods are energy efficient, but most are energy wasters. Sections 4.2 and 4.3 address the task of efficient part load operation.

4.0 Compressors

This section of the guidebook is all about compressors. Some of the more popular design types will be reviewed. Advantages and limitations to each type will be covered.

There will be a discussion of both full load operation and part load operation. Methods of part load compressor control will be reviewed. The concept of compressor sequencing will be covered.

The critically important concept of a "compressor performance curve" will be introduced and discussed. A method for measuring and recording compressor performance will be presented and explained.

Upon completing the reading of this section of the guidebook, the user will have been exposed to the most important aspects of compressor design and operation.

4.1 Types of Compressors in Common Industrial Use

The compressor manufacturing industry produces a multitude of compressor designs. There are, however, only three types of compressors that are predominant today in industrial air systems at pressures up to about 150 psig. The three types are: reciprocating machines, rotary screw machines and centrifugal machines.

4.1.1 Reciprocating Air Compressors

Reciprocating air compressors are similar in design to an internal combustion engine, in that there are pistons moving back and forth inside a cylinder. These compressors are made in single-action and double-action units. In a single-action compressor one motion of the piston compresses the air in the cylinder. A valve is opened at the end of the piston stroke and the compressed air is released to a receiver. The double-action compressor works in the same manner, except that compression is achieved in both stroke directions.

Reciprocating compressors can be single-stage or multiple-stage. In a multiple-stage machine, the discharge of compressed air from the first stage is input air for the second stage. If there are more than two stages, the process repeats with each stage increasing the compression from the former stage.

Single stage compressors are usually rated with discharge pressure ranges of 25 psig to 100 psig. Two stage compressors have discharge pressures of 100 to 250 psig.

Reciprocating compressors can be operated at part load in a very energy efficient manner with the proper controls. This will be addressed in Section 4.3.3.2.

Historically, reciprocal-designed compressors were the most popular type used in industrial plants. However, these units are more costly to manufacture and maintain than are other designs. In addition, often the stroke of this type of machine imposes a large turning force on the support structure under the machine. Foundations must be designed to withstand that turning force. This contributes to the overall expense of the installation.

Figure 4 - 1 Reciprocating Compressor

Today, reciprocal compressors are typically used in smaller manufacturing applications with smaller air loads. The single-action machines are very popular in service stations and auto repair facilities.

4.1.2 Rotary Screw Compressors

Rotary screw compressors have replaced other designs as the machine of preference in many industrial plants. These units are available in ranges from 0 to 25 cubic feet per minute (cfm), up to as high as 3,000 cfm. While most machines operate at less than 150 psig, there are machines available at pressures up to 600 psig. Machines come in single, two and three stage designs.

Rotary screw machines are less costly to manufacture than reciprocating designs. The turning moment in a rotary machine is a balanced moment, so there is no need for special foundations under the machine.

The design of this type of compressor consists of a pair of rotors meshing in a dual-bore cylinder. The male rotor has a number of lobes and the female rotor has a number of corresponding helical grooves. Air is drawn into the cavity between the main rotary lobes and the secondary rotary grooves. As the two rotors turn, air is compressed as it is forced into a smaller and smaller area. (See Figure IV - 2).

The most widely used design of rotary screw machines uses an oil circulation system to seal between rotors and the housing. Oil also serves as a heat transfer medium to assist with removing the heat as compression and cooling the machine. Droplets of oil are frequently carried over with the compressed air stream. These machines must be equipped with an air/oil separator to remove oil mist out of the air stream.

However, not all of the oil is eliminated. In systems where the air must be oil-free, rotary screw machines are manufactured for dry operation. Dry machines must be made to much closer manufacturing tolerances and as a result are much more costly than are the oil-flooded units.

Figure 4 - 2 Rotary Screw Compressor

Figure 4 - 2 presents the "air end" of a rotary screw air compressor. The two rotors are shown inside a cutaway of the housing.

Because of the few moving parts in these compressors, the units offer long life with minimal maintenance.

Rotary screw machines are quite efficient at full load. When part load is encountered the efficiency drops significantly. Later in this section of the book, there will be a description of part load techniques for rotary screw machines. In general, high part load efficiency is more difficult to achieve with rotary screw design machines compared to reciprocal machines.

4.1.3 Centrifugal Compressors

Centrifugal compressors (also referred to as dynamic machines) are machines in which air is compressed by the mechanical action of rotating impellers imparting velocity to air within a confined housing. (See Figure 4 - 3). These compressors are not unlike squirrel cage (centrifugal) fans in operating principle.

Centrifugal machines are often used as the central air suppliers in large manufacturing operations. They put out volumes ranging from 1,000 to 30,000 CFM at a discharge pressure of 100 to 125 psig. The large volume capabilities enable the extensive use of robotics and pneumatic tool equipment.

Figure 4 - 3 Centrifugal Compressor

Centrifugal machines are about as efficient as other designs at full load operation. At less than full load (part load) the efficiency deteriorates. Through the use of a butterfly valve in the air intake, centrifugal machines can be modulated in a range from 100% down to 85% capacity. Modulation beyond this range risks damage to the machine.

In the past, part load was achieved by simply venting any excess air not needed in the process at the time. While this is an effective means of matching load demand to capacity, it is energy-wasteful. That technique is no longer tolerable from an energy efficiency point of view.

4.1.4 Other types of Compressors

There are a number of other compressor designs. Most of these designs are for special or dedicated uses.

Some dynamic compressors are radial design and some are axial design. A jet engine on an airplane uses an axial design compressor. Single rotor, rotary designs include vane compressors, liquid ring compressors and screw machines. Each of these devices has a use niche where it meets the needs of a particular demand.

The compressors addressed in this guidebook are the type of machines most commonly found in industrial situations. No attempt is made to address the use of the many specialized compressor designs.

4.2 Compressor Drives and Electrical Systems

All three types of compressors in common industrial use can be driven by a number of prime movers. This guidebook concerns itself with electric motor drives.

Induction motors are the most frequently used compressor drivers in typical industrial plants. Synchronous, induction motors are often used when larger (500 hp and up) drives are required.

Most drives of less than 200 hp are low voltage class. Low voltage is defined as less than 600 volt. Beginning at 250 hp and above, medium voltage motors are often encountered. Medium voltage is defined as 2,400/4,160 volt systems. In newer facilities with large motors it is not unusual to find 7,200/12,470 volt systems.

Plant electricians are expected to work carefully around all voltage machines. Within this guidebook, electricians may be directed to measure the various parameters which describe the electrical system, i.e., voltage, current, power factor, power and so forth. Directions emanating from discussions in the guidebook will be based on the that assumption the power system is in

the 600 volt or less class.

Medium and higher voltage electrical systems should be worked on only by people who are fully qualified to do so.

4.3 Full Load and Part Load Operation

While some industrial compressed air uses are constant, most air demands are variable. Tools and cylinders often use air intermittently. An air nozzle used to blow dust from a work area is used occasionally. Positioning devices operate on a cyclic basis. The accumulation of air demands in an industrial plant most frequently present a variable profile of air demand.

The compressed air system is sized to meet the highest demand. This results in the compressor(s) operating at less than full capacity part of the time. When compressors operate at less than full capacity there is a high probability that the operation will be energy-wasteful. The degree of probability depends upon the type of compressor in use and the control system employed on the compressor for operation at part load.

4.3.1 Efficiency at Full Load

All three of the compressor types previously discussed develop between 3.8 and 5.0 cubic feet of compressed air per brake horsepower (BHP) at full load. This ratio is referred to as the Volume/Power Effectiveness Ratio (VPER). For example, a rotary screw compressor with a 100 BHP drive load delivers 435 cfm at full load. The VPER is 4.35 cfm/BHP (435 cfm/(100 BHP)).

It is appropriate to make the distinction between input power to the motor (electrical power) and input power to the compressor (mechanical power). Shaft output power of the motor is the brake horse power input to the compressor when the shafts are directly connected. In the event of a chain or v-belt drive, power transmission losses may occur between motor output shaft and compressor input shaft. There are electrical energy losses in every motor. Losses range from a low of about 4% (input power) in larger energy efficient motors to as high as 12% in smaller, older motors. Power transmission losses may be in the range of 1% - 5%, depending upon the type and condition of the equipment.

Brake horse power required by a compressor is net of the motor efficiency loss and power transmission loss. Therefore, if a compressor requires 90 BHP and the drive motor has an efficiency of 93% (loss of 7%) at that load and power transmission loss is 2%, the input electrical power to the motor must be the equivalent of 98.9 hp [$90 \text{ hp} / (1 - (7\% + 2\%))$].

A value related to VPER is referred to as the Compressor Specific Efficiency (CSE). This ratio is defined in terms of BHP per unit of compressed air. The common unit used for compressed air is 100 cfm. The CSE of the compressor discussed above at full load would be 22.98 BHP/100 cfm ((100 BHP)/4.35 cfm). This value is the reciprocal of the VPER multiplied by 100.

The values of the VPER and/or CSE are normally published for full load conditions. This means an operating level when the compressor is delivering rated volume at the desired operating pressure.

The VPER and CSE ratios are typically used to size a compressor drive when the system air load is known. For example, if the system load is to be 1,000 cfm, the compressor load will be 250 bhp (1000 cfm/4.0 cfm/hp). Different style compressors and different manufacturers of the same type compressors use slightly different values for the VPER.

4.3.2 Part Load Operation

Part load operation is defined as compressor operation at a level of air output that is less than full capacity.

When energy conservation calculations are performed, there is a tendency to project energy savings determined by using the VPER or CSE to relate air volume savings to energy reduction. Since the published ratios are valid only at full load, that tendency is frequently inaccurate.

4.4 Compressor Performance Curves

To be accurate in making energy-use forecasts, it is necessary to develop a performance curve for each compressor. This curve relates level of power demand to air delivery capacity over the entire air delivery range. The published VPER and CSE are accurate at only one point (100% capacity) on the curve. All other points on the curve are considered "part load" operation.

4.4.1 Volume/Power Effectiveness at Part Load

In all but the simple "on - off" part load control, VPER changes (diminishes) with various types of part load controls. In the era of inexpensive power, not much interest was given to efficient part load controls. In the recent past, compressor manufacturers have begun to provide more energy efficient part load controls, but there is a long way to go before they achieve effective energy efficiency.

4.4.2 Importance of Understanding Performance Curves

When attempting to establish electrical energy conservation measures, it is often necessary to assign a value to the resultant energy savings. It is the value of the savings that will drive the process of spending capital to achieve the savings.

Figure 4 - 4 is an example of a performance curve for a 100 hp, 450 cfm compressor. This curve is characteristic for rotary screw design machines with modulating inlet damper part load control.

As the output of air from the compressor is reduced (modulated), the power required to drive the machine falls. The power, however, changes from full load down to only 70 hp over the entire range of air capacity. At zero output the compressor still requires 70% of full power output. This part load control is successful in matching compressor output to system demand, but it does so at a very significant sacrifice of energy efficiency.

When estimates of compressed air conservation savings are made, energy savings forecasts can be estimated accurately through the use of the performance curves. An example will illustrate proper use of the curve.

Suppose the system load is normally 400 cfm. Required power is 96 bhp at 400 cfm. An engineer finds that about 100 cfm will be saved through implementation of conservation measures. This means the new operating point on the curve will be at 300 cfm (400 cfm - 100 cfm). At 300 cfm the load is 90 bhp. While the air consumption was reduced by 25%, power was reduced only 6%. Energy savings will be calculated by extending the power savings over a time period to determine kWh. This major savings in air use results in a very modest energy savings.

In retrospect, in the example, a critical energy decision was made by whomever purchased and installed the compressor. That decision carried with it the fact that once the compressor was installed and operating there would always be a minimum power demand of 70% of full load. That demand would exist whether or not there was an air need. This power demand can be regarded as an energy "overhead" expense.

The engineer who attempts to implement air savings measures with this type of a control system on the compressor will not be rewarded with a significant measure of success. A cost benefit analysis for these measures will not justify much of an expenditure.

Of course, once the compressor is in operation other air loads can be added to the system with very little additional operating expense. Incremental air additions can continue until the capacity of the machine is fully utilized. Incremental costs are very low.

When additional air needs occur over and above the capacity of the existing machine(s), the owner/manager faces the task of providing more compressor capacity. Selection of the lowest initial cost system may again result in installation of a compressor with a high energy overhead. Diligence used in the investigation of higher performance equipment can result in acquisition of a far more energy efficient part load control, albeit, at a higher initial cost. It will be up to the owner to decide if an increase in initial cost is justified by future energy savings.

Figure 4 - 5 adds a second curve to compressor performance for compressors equipped with

modulating inlet damper control. This curve presents the behavior of the Volume/Power Effectiveness with changes in compressor output. Observation of the curve shows that at full capacity the compressor delivers 4.5 cfm per brake horse power. This is often a value that is presented in the manufacturer's literature.

Figure 4 - 5, VPER Performance Curve

Energy savings calculations are often made on the basis of mistakenly using full load VPER. In the example where 100 cfm were conserved, a power savings of 22.5 bhp ($100 \text{ cfm} / 4.5 \text{ bhp/cfm}$) would mistakenly be forecast. Extending this into energy would yield an estimate that is almost four times too optimistic. When the compressor goes into part load, the VPER begins to fall rapidly. By using the performance curve to make savings forecasts, far more accurate economic analyses will result.

4.5 Methods of Controlling Compressor at Part Load

Means of varying the volume output of a machine range from shutting the compressor "off", or venting excess air when supply exceeds demand, to using a variable speed drive on the compressor motor. The most common methods will be reviewed herein.

Plant air systems are designed to operate in a relatively narrow fixed pressure range. System pressure is the common parameter used to control air volume delivery from a compressor. Typically, when the compressor delivers more air than current requirements demand, the system pressure builds. Conversely, when system demand calls for more air than is being delivered, system pressure falls. Pressure is therefore chosen as the easily measurable variable that is used to make adjustments to air delivery rates.

4.5.1 "On - Off" Control

The local service station air compressor is typically controlled through an "on" or "off" control system. When the machine is "on" it operates at full capacity, which is also at maximum VPER.

When it is "off", no power is used. This is an example of using an air compressor in the most energy-efficient fashion.

The control scheme is simple. A pressure switch has a high and a low setting. When the pressure falls to the low setting the compressor is turned "on". The unit operates until the pressure reaches the high setting, at which time it is turned "off". The user realizes the compressed air will always be available at a pressure between the low and high limits.

Figure 4 - 6 is a graph with a curve that relates the power required to drive an "on - off" controlled compressor to the capacity of the compressor. The relationship is linear. The VPER appears on the graph as a horizontal line, indicating the VPER value is constant over the capacity

range of the compressor.

Energy conservation measures projected for "on-off" controlled systems may use the VPER full load value to calculate power and energy savings. This is the only case where the published values may legitimately be used.

4.5.2 Low Unloaded Horsepower Control (Load/Unloaded Control)

Industrial air needs are frequently continuous, albeit, at varying levels of demand. Because of the severe mechanical and electrical stress that starting and stopping imposes on the drive, an "on - off" system isn't appropriate for most continuous needs. This is particularly applicable for drive sizes 25 hp and above. Low, unload horsepower control is available on all rotary screw machines, on some reciprocating machines and on some centrifugal machines.

Rotary screw compressors deliver compressed air to a sump which must be an integral part of the machine. The sump in turn delivers compressed air to the system. It is possible to isolate the sump from the air distribution system and then proceed to decompress the sump which unloads the compressor.

This control methodology is a take-off from "on - off" control. The compressor operates at either full delivery capacity or at zero delivery capacity. The drive motor operates continuously but at two significantly different load levels. When the system pressure is satisfied, an air valve isolates the compressor from the system. In the case of rotary screws, after the compressor is isolated another valve opens and vents the sump to a low pressure, sometimes even atmospheric pressure.

The compressor now operates at a very low compression ratio, a value of unity if the sump is vented to atmosphere. The corresponding motor load is just enough power to turn the

compressor and overcome windage and friction. Figure 4 - 7 presents the performance curves for this type of part load control. Depending upon the brand of machine, unloaded power is between 17% and 25% of full load power.

As air is used from the system, system pressure drops. When the pressure reaches a preset minimum level, the compressor sump vent is closed and the sump pressure is brought up to system pressure. Once the sump is up to pressure the compressor is reconnected to the system and the unit begins delivering air. The compressor operates at full capacity and brings the system up to the high pressure limit. The process then repeats at a frequency dictated by rate of system air consumption.

Figure 4 - 7 Low Unloaded Power

This system depends upon a significant pressure differential for successful operation. It is also highly desirable to have receiver tank capacity in the system which will provide an inventory of compressed air during the period the machine is in the unloaded condition. One gallon of storage capacity for each cfm of compressor capacity is recommended.¹ The tank acts as a storage buffer that keeps the load/unload cycle from repeating too frequently. Systems without adequate storage cannot take advantage of the "low unloaded power" control system.

When the compressor goes into the unloaded cycle it should remain in that condition for some minimum amount of time. That minimum is dependent on a number of factors, such as compressor size, capacity of receiver and system storage and length of time in loaded condition. Each time the machine allows the sump to blow down, some air that was previously compressed is vented and lost. The sacrifice of that loss must be off-set by a significant reduction in power level for an additional period of time or there will be no significant advantage (energy use reduction) to the control method.

Reciprocal compressors do not have a sump, but they can operate in a controlled unloaded fashion. When system pressure is satisfied, the compressor is isolated from the system. The air in-take valves remain in the open position as the piston begins its compression cycle. The air that was just drawn in will be exhausted. No compression takes place. The machine consumes energy at a level that is adequate to overcome windage and friction.

When a reciprocating compressor has several stages and is double acting, the machine can operate at many levels of output with a high degree of efficiency at each level.

¹. Van O'er, Capacity Control Influences Power Cost, Compressors, Power Magazine, December 1976.

4.5.3 Modulation Methods of Part Load Operation

It is often desirable to maintain a relatively constant pressure in a system. This is best accomplished through the use of a continuously operating compressor that is equipped with a control that will vary (modulate) the machine's output. Part load modulation controls take a number of forms, depending upon the type of compressor and the approach used by the manufacturer.

4.5.3.1 Inlet Damper Modulation

One technique used with both rotary screw machines and with centrifugal compressors merely closes a restricting device in the air inlet when the system pressure begins to reach desired operating pressure. The compressor operates at normal speed with normal oil pressure against full system air pressure. The throttling action restricts the area of the air intake passage. This results in a diminished air flow. With less air coming into the compressor, less compressed air is delivered by the machine.

There is a penalty associated with this type of control. The throttle restriction causes a pressure drop to occur so that the pressure at the intake valve is less than atmospheric (vacuum). However, the compressor must achieve the same output pressure (system pressure). Therefore, the compression ratio increases when throttling occurs. Since drive power varies as a function of compression ratio, the power input increases as compression ratio increases.

The net effect of reduced volume with increased compression ratio is a net reduced power consumption. But the power reduction is much less than proportional to the volume reduction. The power versus volume relationship is close to linear, however, the slope of the line is much different than the "on - off" line slope. Also, the line is displaced upward on the vertical axis as a result of the energy overhead or threshold power required to drive the compressor.

Centrifugal machines can use this technique to modulate down to about 85% of full capacity. Rotary screw machines can modulate output from full capacity to zero capacity. Power decrease may be linear but the value will drop on 30% from full flow to zero flow. In other words, it takes 70% of the drive power to operate the compressor against system pressure with no flow. With this method of modulated control, VPER and CSE values will be dramatically less than full load values.

Figure 4 - 8 presents a graph with a typical curve-relating power and capacity for inlet damper modulation part load control on a rotary screw compressor. The second curve shows the relationship between capacity and VPER with this type of control.

In this control scheme a pressure-activated valve senses system pressure. When the system pressure approaches set point, the valve sends an air signal to the modulating control damper. The inlet damper moves from fully open to fully closed over a pressure range. Depending upon compressor brand, that range may be as tight as 2 psig or as broad as 10 psig.

From energy considerations, the tight range is better, because the compressor is operating at highest efficiency through most of the pressure band. When the pressure range is broad, the compressor operates in a modulated fashion (lower energy efficiency) more of the time.

Figure 4 - 8 Inlet Damper Modulation

Pressure sensors that control the inlet damper are usually activated by one pressure level when the system pressure is moving in the increasing direction. It often takes a significantly different and lower pressure to deactivate the inlet valve as the system pressure moves lower. This differential is purposefully built-in to prevent the compressor control from "hunting" for a stable position by vacillating in and out of modulation control.

4.5.3.2 Turn-valve Modulation

More sophisticated modulation controls are found on some brands of rotary screw compressors. Some units utilize a turn valve which controls compressed air output. This device has the ability to close or open ports in the compressor housing. In essence, it has the effect of varying the size of the compression chamber.

The output volume depends on the position of the turn valve, which in turn depends on sensed system pressure. When system pressure is below desired level, the turn valve is fully closed and all output goes to the system. When system pressure is at or above desired level, the turn valve opens ports. This allows some unneeded air to return to the suction side of the compressor. The air output of the unit is reduced. Power need is also reduced. The VPER does not suffer as much with the turn valve control as it does with modulated inlet valve control.

Each compressor manufacturer has a similar device or approach to reduce energy consumption at part load.

4.5.3.3 Inlet Valve Control

This technique is applicable to reciprocating machines. Each cylinder in a reciprocating machine is equipped with one or more intake valves. These valves can be manipulated to allow full, part, or no net inflow of air. The compressor works at full load, part load, or no load, depending upon the amount of air drawn into and retained in the compression cylinder.

At full load, the intake valves are fully open during the intake stroke and are fully closed on the compression stroke. At no load, the inlet valves remain open on the compression stroke so no compression takes place. At part load, some, but not all valves open on the intake stroke. The cylinder does not fill with atmospheric air, it remains in partial vacuum at the beginning of the compression stroke. The compression stroke delivers less than a full measure of compressed air.

Each cylinder can be equipped with inlet valve control. Typically, inlet valve control will allow

a compressor to operate at 0%, 50% or 100% capacity. In double acting compressors, each cylinder can be controlled independently. This provides the ability to control compressor output at 0%, 25%, 50%, 75% and 100% delivery capacity. Each output level requires a different power input. The 0% output requires power input to overcome windage and friction. Depending upon the output level selected, increased output levels require more power.

Through the use of pressure transducers and programmable controllers, control systems can be designed for reciprocating machines that will closely match system demand to machine output in an energy-efficient fashion. This can be accomplished over a narrow system pressure range. Unfortunately, from an energy perspective, this efficient form of modulation is not being used because reciprocating compressors are losing market share to less costly machines.

Existing machines in good condition should be considered as candidates for energy efficient electronic control systems.

4.5.3.4 Speed Control Modulation

Positive displacement compressors are constant torque devices. Power required to drive the compressor is dependent upon the compression ratio and the speed at which the input shaft is driven. Air volume produced from the machine can be variable (within limits) if the speed of the machine is varied. With the use of an adjustable speed drive, power requirements will vary directly with volume.

Adjustable frequency, adjustable speed drives (ASDs) have gained wide use over the last several years. There may be a proper place for this type of control in industrial compressed air systems. By modulating machine speed the compressor can match system air demand while maintaining system pressure at a virtually constant level. This is a significant advantage over the "load/unload" control system which must rely on a system pressure differential of about 10 psig for normal operation.

Speed control modulation can be utilized on oil-flooded rotary screw designs, on dry rotary screw units and on reciprocating machines. Speed control is possible on centrifugal machines, but great care must be exercised because of the critical speeds associated with centrifugal operation. The user is encouraged to speak with a factory representative before attempting speed control on centrifugal compressors.

Reality suggests that the inherent higher cost of reciprocating machines will be driven even higher when the machine is equipped with variable speed drive controllers. This high initial cost further precludes that combination from serious industrial consideration. Rotary screw machines, therefore, are the most likely candidates for use with variable speed drives.

Modulated speed control has one major advantage over "low unloaded power" control. Speed modulation offers the ability to vary volume while maintaining pressure within very close tolerances. Through the use of a pressure transducer, system pressure can be precisely sensed with subsequent speed adjustments that produce a compressor output to exactly match most momentary requirements.

Figure 4 - 9 shows the relationship between full load power and compressor capacity with ASD control. Speed modulation on rotary screw compressors must be limited to a range from 100%

to 50% of capacity. Below 50% capacity, there are limitations on minimum speeds required by oil pumps and oil seals.

Where system demands require air output modulation below 50% capacity, the ASD control can be combined with another control form like inlet damper modulation for a complete range of capacity. However, the highest use benefit from a machine with ASD control will occur when the machine is called upon to deliver between 50% and 100% of rated capacity.

4.5.3.5 Combinations of Part Load Controls

With rotary screw compressors where a system is large enough and there is a wide variation in system demand, a combination of "modulated inlet damper control" and "load/unload" control may be appropriate. This method allows a system to take advantage of benefits of each control method over a limited range of capacity.

Because "low unload power" control is involved, there must be receiver tank storage in the system for proper operation.

Again, the capacity of the storage tank should be one gallon for each cfm of compressor capacity. The control must be carefully adjusted to match needs of the system. In order to take advantage of the combination control, an operator must be familiar with the system needs, the compressor characteristics and the control methods.

4.6 Sequencing Multiple Compressors

In industrial plants where there is more than one air compressor installed and used, there will be a best way, energy wise, for operating multiple units. The proper way of controlling the operation of the machines is referred to as sequencing.

4.6.1 Coordinating Resources and Matching to Load

Sequencing is defined as the most energy efficient method to match capacity to demand when using more than one compressor. The process of establishing a sequencing program begins with definition of demand.

On a gross basis, demand can be divided into segments by shift, by work day, or by weekend day. On a more refined basis, demand can be segmented by hour during the day. In more sophisticated systems, an intelligent system controller may be provided. The controller will have the ability to be programmed so that it will interact with the system by sensing and reacting to demand levels. Some units have the ability to learn and anticipate.

4.6.2 Sequencing Controllers

Basic sequencing begins with a two compressor system. The units may be started and stopped manually, according to a schedule as elementary as the shift time schedule. With a multiplicity of machines and a large diversity in demand, sequencing becomes complex and will probably involve the use of a microprocessor-based control system.

4.6.2.1 Manual

A simple system of manually sequencing compressors can be illustrated at a plant where there is a substantial air load during manufacturing hours but only a small load during overnight and weekends. The larger compressor is manually started at the beginning of each manufacturing shift.

It is common for industrial plants to be equipped with fire protection sprinkler systems. Many of these systems require that the piping system be charged with compressed air. Usually there is no air consumption other than system leaks. The off-shift load is full pressure, but low volume.

It often makes sense to use a second, smaller compressor to supply air for the sprinkler system. The larger unit(s) can be turned off during low need periods and the smaller unit can be operated with far less energy consumption.

There is nothing particularly complex about this sequencing system. A large number of industrial firms utilize this type of scheme by relying on operators to perform compressor switching. The sprinkler system pressure may be monitored by either on-site or off-site security services that notify appropriate plant personnel when there is a significant drop in system pressure. When an operator forgets to turn the smaller unit "on" there will eventually be a drop in system pressure and an alarm will be sounded.

The more critical situations will include immediate alarms when at least one compressor is not operating.

Manual sequencing can progress with more complex air systems. When demand is repetitive and can be predicted with confidence, a second, third and other compressors can be started and added to plant capacity by time of day. This practice of sequencing can save substantial energy over the practice of turning "on" all units at the same time.

Manual controlled sequencing systems have obvious limitations. Changes to a normal routine may cause a system to waste energy unless there is an operator involvement. Demands that vary from the norm may not be facilitated. All manual control is subject to human limitations and oversights.

4.6.2.2 Pneumatic

For many years compressors were sequenced through the use of pressure switches. These switches monitored system pressure and operated with changes in pressure. Typically, where reciprocating machines were involved, a switch would provide a signal that was used to control the position of the cylinder inlet valve. The signal would cause the valve to make the unit operate at a higher or lower capacity. A series of pressure switches would produce a series of signals that would control the output of multiple reciprocating compressors.

Pressure switches can also be used to turn a stand-by compressor "on". Combined with the use of time-delay relays, pressure switches have been used to interact with the system by sensing demand and causing the compressor capacity to increase and decrease with demand changes. Pressure switches operate over a pressure differential, i.e., they actuate at one pressure and deactivate as a second pressure that may be as little as 2 psi or as much as 10 psi from the original signal. When multiple compressors are involved in a system, the use of pressure switches soon becomes a serious limitation. The total system pressure differential may need to be 15 psi or 20 psi, to cover the range of "start"- "stop" and intermediate load levels for several units. There is a practical limitation to the number of compressors that can be successfully sequenced by way of pressure switch control.

4.6.2.3 Electronic

With the advent of micro processor technology, process control was provided with a more powerful and sophisticated capacity to perform automated functions. Not only can the micro processor perform logic, it can also utilize extensive memory capabilities.

Now an air compressor sequence control system can include the integration of time, demand, and compressor operating characteristics into a complex program that will insure the most energy-efficient operation at all times.

One of the keys to effective control is the use of a pressure transducer - a device used to monitor system pressure. A transducer can discriminate between very small changes in pressure, i.e., ± 0.1 psi. The transducer delivers an electronic signal to a controller on an instantaneous basis. Small changes in pressure can be used to initiate significant control changes. The limitation that is inherent with pressure switches is eliminated through the use of pressure transducers.

The sequencer control can be programmed with information about each compressor in the system. An energy-efficiency hierarchy can be established along with each machine's capacity and part load performance curve. The sequencer can determine the mix of machines that is most energy-efficient for the demand being experienced.

4.7 Air Demand Profile

An air demand profile is a record of the demand for compressed air by time of day. Most industrial plants operate on a regular schedule. Many processes require regular supplies of compressed air. Some processes are irregular in air use from hour to hour or from day to day, but in general there will be some degree of repetitiveness to the way a plant uses compressed air. When the pattern of use is well-defined, the plant engineer has a good opportunity to analyze use and look for effective energy conservation measures.

4.7.1 How to Develop a Demand Profile

An accurate demand profile may be developed in an indirect fashion. In section 4.3.2, there was a discussion of compressor performance curves. The performance curve can be a tool to develop a demand profile. This curve establishes a relationship between compressor air delivery and power used to drive the compressor.

(In Section 11 there are descriptions of processes to be followed when developing a compressor performance curve and demand profiles.)

By measuring and recording power to the compressor and using the performance curve, values of power may be converted to values of compressed air. A demand curve for air may be established by time of day.

In multiple compressor plants it is necessary to monitor the power used by each compressor. The most accurate results can be expected when monitoring occurs simultaneously. Acceptable results can be developed by monitoring each compressor individually for a length of time that is adequate to determine the repetitive use profile. Total combined air use is the sum of air delivered by each compressor.

4.7.2 How to Use a Demand Profile

Once the demand profile is established, the plant engineer can observe how the air resources (compressors) are being used to meet the system air needs. There may be opportunities for using multiple compressors in a more energy-efficient sequence. There may be opportunities for shutting down one or more compressors, part of the time.

It may be appropriate to obtain a smaller compressor which will operate part of the time at a high efficiency, so that a larger machine can be shut down. It may be cost effective to improve the air distribution system so that remotely located compressors can be used to feed the system.

A proper demand profile produced with the help of accurately developed compressor performance curves will allow a spreadsheet model of the compressed air system to be developed. The model can be used to test the effectiveness of various changes to the system.

5.0 Equipment for Conditioning Air

Most compressed air systems will contain a number of components of ancillary equipment. The purpose of the equipment will be to make the compressed air more suitable for the use for which it is intended. This section of the book presents a list of commonly found items and provides a brief explanation of why they are used and how they should be cared for.

5.1 Inlet Air Filter

Ambient air is subject to contamination from any number of sources in close proximity to an industrial plant. An air compressor uses ambient air as its source of matter to be compressed. Contaminated air may cause unnecessary wear on the compressor equipment and/or impose significant duty on air filters in the system.

Some examples of plants where airborne contamination is prevalent are: cement plants, sawmills, all types of secondary wood products manufacturing plants, food processing plants and any facility where the product or raw material is granular in form and subject to exposure to local winds.

Many problems are avoided when the user takes the time and makes the effort to see that incoming air is properly cleaned through air filters. Most compressors are equipped with air filters. It is incumbent upon the user to see that filters are changed/cleaned on a regular basis. Heavily contaminated intake air filters can cause undue pressure differentials to occur, which forces the compressor to operate at higher compression ratios. This causes energy waste and is an unnecessary expense.

5.2 Intercooler

An intercooler is a heat exchanger. The intercooler is used on multi stage compressors to absorb the heat of compression from the first stage. Heat transfer takes place through water that is circulated in a water jacket around the compression chamber.

An indication of the effectiveness of the intercooler is the temperature change of the cooling water from entering to leaving. When the intercooler is working well, there should be a large change in temperatures. Some compressor manufacturers publish a range of acceptable temperatures. The user must monitor those temperatures to be sure the intercooler is operating properly.

Malfunction of an intercooler will result in high compressor operating temperatures and a slight loss in compressor capacity.

5.3 Air/Oil Separator

Oil-flooded rotary screw compressors use lubricating oil to form an air seal between rotors and housing. Some other compressors also use oil as a lubricant. It is common, especially in rotary screw machines, for small droplets of oil to become entrained in the compressed air stream. Over time the amount of oil carried in the air is significant.

Oil-flooded rotary screw machines must be equipped with an air/oil separator to recover the

entrained oil and circulate it back into the system. Without such a device, oil consumption would take place at an unacceptable rate. As oil is removed, the air stream is cleaned-up. The degree of required cleanliness varies, depending upon the use for which the air is intended. Sometimes no other oil filtering is required. In other plants there may be one or more dedicated oil filters downstream.

5.4 Aftercooler

An aftercooler is also a heat exchanger. Aftercoolers are found on all types of compressors. The purpose of the aftercooler is to remove the heat of compression from the high pressure air. At the same time most of the entrained moisture is also removed. Depending upon the type of compressor and the desired system pressure, air temperatures may reach 200°F or higher upon exiting the compressor.

High temperature air is undesirable for a number of reasons. From a safety point of view, air piping should not be excessively hot to the touch. Much of the equipment that relies on compressed air is designed to operate at ambient temperature. High temperature air will often be detrimental to the equipment. It is therefore highly desirable, if not necessary, to cool compressed air as soon as it leaves the compressor.

Cooling with the aftercooler may be accomplished with air, oil or water. Many compressors are equipped with a radiator through which the cooling medium is passed. Rotary screw machines often use oil to not only provide an air seal between the rotors but also to carry the heat of compression to the radiator. Air is blown across the radiator tubes so as to carry the heat away from the machine.

Other compressors use water-jacketed housings. Water passed through the housing can be routed to a radiator where the heat is dissipated. Other water systems use a once-through process where fresh water passes through the machine and carries the heat away. That water is either sewerred, sent to a cooling tower or a cooling pond.

An important result of the aftercooling process is that the air is left either saturated or close to being saturated. This depends upon the amount of moisture in the ambient air as it enters the compressor and upon the amount of cooling that was accomplished in the aftercooler. There is almost always a water drain attached to or immediately following the aftercooler. Water that condenses in the aftercooler is collected and discharged through this drain.

In any event, users should be aware that additional cooling following the aftercooler will cause condensation of moisture to occur in the system. For this reason the next piece of equipment in the system is usually the air dryer.

5.5 Air Dryer

The purpose of an air dryer is to remove a substantial part of the water that exists in the compressed air after the air has passed through the aftercooler. There are three basic types of dryers. Each type is built with variations.

Deliquescent Dryer. This is probably the oldest and least expensive type of dryer. It works by filtering compressed air through material that absorbs moisture (deliquescent) out of the air.

For proper operation, this material has to be replenished regularly.

Deliquescent dryers have to be properly maintained or they will stop functioning, and in fact allow the air to carry the deliquescent material out of the dryer and into the system. Foreign material in the system could result in damage to the end use equipment. Moisture-absorbing material has to be changed regularly. The interval depends on the actual material used - the relative humidity of typical ambient air and the volume of air passing through the dryer.

The user should note the magnitude of pressure drop across the dryer after it has been thoroughly serviced. Over time as the dryer is used, the pressure drop will increase. Regular observation of the pressure drop will provide an indication of when maintenance is required. A large pressure drop across the dryer, compared to the original pressure drop, is an indication there may be a blockage due to saturated material.

It may be appropriate to provide downstream filters with these dryers to insure collection of airborne deliquescent material.

Refrigeration Dryers. The refrigeration-type dryer simply acts to further cool the air. This dryer works on the principal that when the air is cooled to a dew point temperature lower than the temperature at which it will be used, moisture will condense down to the dew point in the dryer. No moisture will be condensed in the distribution system because the ambient temperature of the system is expected to remain higher than dryer dew point temperature.

The refrigeration dryers typically work to a dew point of about 35°F. They have the advantage over the deliquescent dryer in that they require far less maintenance attention. When a refrigeration dryer is not working properly, water will soon be discovered in the system.

Desiccant dryer. This is the most expensive type of dryer and the most effective. Desiccant dryers can provide protection down to a dew point of -40°F. They normally operate with two towers that are each charged with an absorbent desiccant. Compressed air passes through either one or the other tower. One of the two towers is working (drying) at all times and the desiccant material in the other tower is being regenerated (dried).

Regeneration takes place by passing some dry compressed air from the tower doing the drying through an orifice into the wet tower. As dry air enters the wet tower, it is allowed to expand to atmospheric pressure, thus enabling it to pick up moisture from the saturated desiccant. This wet air is then released to atmosphere.

A variation of this regenerative cycle involves heating the dry compressed air as it enters the regenerating tower. The heated dry air can pick up more moisture as it passes over the desiccant bed. Thus, less air is required to do the drying. A major consideration is that compressed air is used to get rid of the moisture. In some cases, as much as 15% of the compressed air produced is used to purge a regenerating tower. This can be a major use of compressed air.

When a system is operated in this fashion, purge controls for varying the amount of drying air should be investigated as an energy savings measure. Frequently, desiccant dryers are also equipped with downstream filters to collect any airborne desiccant material.

5.6 Reheaters

Refrigerative dryers condense water out of the air by reducing air temperature and thus diminish the ability of air to hold water. Usually, air leaving the dryer is saturated. A reheater is another heat exchanger. As a reheater, this device uses some of the heat of compression to heat air as it leaves the dryer.

By heating the air, the air's ability to contain water vapor is increased. When air is heated adequately the downstream heat loss will be accommodated without letting the air temperature fall to a level which will allow condensation to take place. This will result in keeping the air system water-free. Many of the problems associated with the accumulation of water in the system can be avoided.

5.7 Filters

Coalescing filters. In many applications of rotary screw compressors, oil is mixed with the air in the compression cycle to seal the compressor and take away the heat of compression. As the air leaves the compressor it goes into a separator where the oil is removed and sent to an oil cooler and the air goes off to the system.

When oil is present in the distribution system, it will form a sludge when mixed with moisture. This slug will foul the application filters causing a pressure drop that will adversely affect equipment.

Filters should be installed downstream of the compressor to remove this residual oil. These filters coalesce, or cause the small oil particles to join together to a size where they can be removed from the air stream. These filters are called coalescing filters and they work for oil particles in the range of .01 to .8 microns. If the elements of the coalescing filter become clogged, there will be an excessive pressure drop across the filter. When this occurs, the elements have to be changed. Pressure-drop indicators should be installed to indicate when the elements should be changed.

Because the coalescing filters are easily plugged with particles, it is recommended that a general purpose pre-filter be installed in front of the coalescing filter.

In some cases, oil vapor particles too small for the coalescing filter must be removed. This can be accomplished with an oil vapor removal filter. This filter removes oil vapors that are in the sub-micron size. A plant should be sure that it needs this level of protection before it installs this type of filter. These filters should be installed downstream from the coalescing filter. The filters contain activated charcoal to remove the vapors by absorption.

The location of the filters is important. If deliquescent or desiccant dryers are used, both the general pre-filter and the coalescing filters should be upstream of the dryer. If a refrigerated dryer is used, the pre-filter should be ahead of the dryer and the coalescing filter should be behind the dryer. In all cases, when an oil-vapor removal filter is to be included, it should be located downstream from the dryer.

General purpose filters. The use of air filters has been suggested at a number of points in the air conditioning equipment system. The importance of using filters depends upon the end use

for which the air is intended.

There are some situations where all of the air is required to be extremely clean. In other instances it may be appropriate to branch-off with a part of the system and equip the branch line with an extensive set of filters, in order to deliver very high quality air to some discriminating uses.

In some rudimentary systems no filtering beyond that which occurs at the compressor is required. The need for filtering depends upon the usage requirements at the site.

5.8 Pressure Gauges

Although the use of pressure gauges will not directly affect the quality of the air, gauges should be used extensively to provide an indication of performance for air conditioning equipment. It is advisable to provide pressure gauges before and after each element of air conditioning equipment.

A more thorough discussion of the maintenance value of pressure gauges is presented in Section 8.2.

6.0 Other Compressed Air System Equipment

6.1 Drive Motors

Commonly, compressor drives are induction motors. The motor can be in an open drip-proof (ODP) housing or in a totally enclosed fan-cooled (TEFC) housing. Normally, a Nema¹ design B motor is used. The Nema rating refers to the starting torque provided by the drive. A B design provides 150% starting torque.

Motor speed is selected to match the compressor speed requirements, taking into account whether the unit is direct drive or driven through v-belts and sheaves.

Many compressor manufacturers design the drive/compressor package so that at full load the motor operates in the service factor range. For instance, a nominally rated 100 hp compressor may impose a load of 110 hp on the drive at full load. Typically, the drive will have a service factor of 1.15, meaning that it is rated for operation to a level that is 15% above the nominal nameplate rating.

The user should be aware that when the drive is operated in the service factor range for extended periods, there is a likelihood that the insulation life will deteriorate more rapidly than if the motor was always operated at or under nominal rated load. The motor will operate successfully, but the expected life will probably be shorter.

6.2 Motor Starter

The type of motor starter used with the compressor drive depends on the size of the drive, the nature of the plant electrical system and the regulations of the serving utility. Most utilities have regulations about the required use of reduced voltage motor starters. Required use is usually dependent only on the size of the motor, i.e. all drives 100 hp and above will be equipped with reduced voltage motor starters.

Some plants have electrical systems that cannot tolerate voltage dips such as may occur when large motors are started across-the-line. The decision to use a reduced voltage device may then be made on the basis of plant needs rather than on utility regulations.

There are significant benefits available when electronic soft-start, reduced voltage motor starters are used. In addition to the prevention of voltage dips on the system, the benefits are often related to reduced wear and tear on the equipment.

6.3 Pressure Switches and Pressure Transducers

In the discussion of part load controls there was reference to pressure switches and pressure transducers. It is appropriate to comment on each of these devices.

A pressure switch is a device that consists of a set of electrical contacts which are actuated by pressure. Below a given level of pressure the contacts may be in the "open" position and above

¹ National Electric Manufacturers Association

that level the contacts switch and go into a "closed" position. The pressure level at which action occurs can be fixed or adjustable, depending upon the type and cost of pressure switch obtained.

The pressure value required to return a switch to its initial position is different than the pressure required to activate it in the first place. An example would be to set the switch to operate at 110 psig. When system pressure reaches that level the contact closure takes place. In order to reverse the contact closure, the pressure may have to drop to 105 psig or lower. This provides a working "band" of pressure which prevents the switch from transferring back and forth over very short time intervals.

The differential is built into the switch as a means of providing a control range rather than strictly a control point. Pressure switches are relatively inexpensive - less than \$60 per unit. They are reliable and have been used to provide contact closures or openings in compressed air system for many years.

Pressure transducers are a more refined type of device. A pressure transducer provides a continuous electronic signal which is proportional to the value of the pressure that it senses. When detected through a controller, the sensed pressure can be displayed continuously. The transducer provides far more finite information than does a pressure switch.

The transducer signal is usually directed into a dedicated controller or programmable logic controller (PLC). Based on how the controller is programmed, the transducer signal can be used to trigger any number of events and activities based on very small pressure changes.

A transducer with control is considerably more expensive than a pressure switch. Each device has its appropriate use.

6.4 Relief Valves

Most compressors and receivers are equipped with pressure relief valves. Devices that are to be equipped with safety relief valves are usually specified in state industrial codes.

A relief valve is meant to provide a means for safely venting a system when air pressure becomes excessive. Plant engineers must be careful with the placement of relief valves. Released air needs to be directed in a safe direction. Upon operation of the valve, no one in the immediate area should be endangered from that operation.

Relief valves should never be blocked or purposely defeated in order to address other problems.

6.5 Traps and Drains

Despite efforts to remove water from the system, there will be instances where water accumulates at various points in the distribution system. This water can be discharged automatically with traps or manually through manipulation of hand valves.

Water traps operate in a manner similar to steam traps. When enough water accumulates in the trap a float is activated. The float causes a valve to open and water is discharged, often

under pressure. When enough water has passed, the float returns to its reset position and the valve is closed. The process repeats as often as necessary.

Manual traps can also be used effectively. Drop legs should be piped off of a main header at a number of locations. Water flows to and down the drop leg by gravity. On a regular basis, an operator should come-by and open a manual valve to discharge the water from the drop leg.

These simple procedures will be effective in riding the air system of fugitive water.

6.6 Heat Recovery Heat Exchangers

The operation of a large compressor or a battery of compressors results in production of a large amount of heat. The heat is usually dissipated to ambient surroundings. In some situations there may be a use for the heat.

In areas where the winters are cold, rejected energy can be used to provide space heat in the buildings. In order to take advantage of the available heat, some form of heat exchanger must be implemented.

There are some situations where the need for process heat can be fulfilled, at least in part, from rejected compressor heat. The advent of industrial heat pumps provides a means of increasing usability of heat by making it available at higher temperatures.

The ability to make use of rejected heat is dependent upon the local needs on a site- specific basis.

7.0 Compressor System Environment

The location of compressed air equipment often has a profound influence on how that equipment is operated and maintained. This section has some observations about the environment in which compressors should be located.

7.1 Space

Preferably, compressors should be located in a dedicated room. The room should be accessible to those individuals who have need to service, operate and maintain the equipment.

The compressor room should have an adequate amount of floor space for the compressors and ancillary equipment. Adequate amount of floor space means the machines should be located so that a maintenance person can access any part of the machine without difficulty. Piping between devices should be accomplished with reasonable distances between the devices.

Each device should be located so that it can be conveniently serviced. Enough free area should be allowed for removal and replacement of filter elements.

Drains and traps should be located so that discharge of water will be directed toward a proper drain.

Allowance for future additions should be made even if that is a remote possibility.

7.2 Access to Free Air

Compressors use ambient air as the source of air for compression. Cooler air is more dense than warmer air. Compressors will be more efficient when using cool air.

Logic suggests that a compressor room be located on an outside wall of a building where access to fresh air is readily available. Further, it seems appropriate that the outside wall be fitted with large areas of very low pressure drop screens or air filters. Only in the coldest climates is it necessary to be concerned with too low a temperature in the compressor room.

The outside wall should be one that is not subject to extensive sun exposure. The coolest sources of air are desired.

When it is not possible to locate the compressor room on an outside wall, the user should consider ducting ambient air through a roof or wall from outside to the compressor, through a roof or wall. Ducting cross sections should be generous so that the compressor is not compelled to draw a sizeable vacuum to induce outside air.

7.3 Light

The compressor room should have plenty of light. Maintenance checks and routines will be accomplished more quickly and more accurately if the service person has the ability to easily see what he/she is doing.

The room will be easier to keep clean when it is well lighted.

Gauges can be read more accurately in proper light.

Oil leaks and similar problems can be spotted easier when lighting is appropriate.

7.4 Proximity to Other Equipment

There is often a tendency to create an mechanical equipment room as opposed to a dedicated compressor room or boiler room. Many items of mechanical equipment are put into the room without regard for interactive effects between adjacent pieces of equipment.

Many plants have compressors located in the boiler room. Both of these devices need fresh air (burners on the boiler and air compressors) for proper operation. Both devices compete for the air that is brought into the room. When there is a generous supply of air, there is little problem. However, when air availability is limited, one or both machine will be forced to work harder (use more electricity) to draw-in air.

Both machines, compressor and boiler, give off heat as a by-product of the process. When the amount of heat given off by either machine is significant enough to drive up the ambient temperature in the room, that will have a detrimental effect on the other machines. More energy will be used because of the interaction.

When possible, it is advisable to dedicate a separate room for air compressor equipment.

8.0 Maintenance

Compressor equipment and air distribution systems are made up of mechanical devices. Each device has a limited useful life. The length of that useful life can be optimized with proper equipment and system maintenance. The energy cost of operating compressor equipment can be minimized when equipment is well maintained.

8.1 Tests for Standard Performance

There are two areas for establishing tests. One involves the performance of each compressor. The second addresses individual items in the air conditioning system and distribution systems.

8.1.1 Compressor Performance Tests

Those users who embrace the idea of establishing compressor performance curves set themselves up with standards against which the equipment can be compared for years into the future. When a performance curve for a compressor is established, that becomes a graphic representation of the capability of that compressor.

Any time thereafter a similar test can be conducted. When the results of the test are comparable to the standard results, the user is confident the compressor and its part load control system are functioning as originally intended. When test results deviate from the standard, there is cause for concern and it will be appropriate to investigate where the problem(s) exist.

The process for establishing compressor performance curves is described in Section 11.1.

8.1.2 Pressure Drop Tests

Pressure gauges should be installed before and after each item of significance in the air system. When the system is new or recently brought up to desired operating level, a record of pressure drops across each device should be made throughout the system. This is the baseline pressure profile. It is the standard against which future readings will be compared. The pressure drop across each device should be noted. Preferably, the pressure drop should be noted at a specific compressor output, i.e. 100% capacity or some easily repeated output.

A maintenance procedure for inspecting gauges should also be established. The procedure should first of all confirm that the gauge is functioning properly. After confirmation, the user needs to read gauge pressures and determine pressure drops across various devices. These pressure drops should then be compared to pressure drops which were established as a standard.

When a pressure drop changes (increases) significantly, there is reason to believe the device requires maintenance. Most pressure drops will be adversely affected by collections of airborne materials that collect at the device. Regular maintenance will minimize the occurrence of poor equipment performance. Sometimes, however, an event will cause a performance failure of a device and this should be manifest by a significant change in pressure drop across that device.

8.2 Regular Maintenance Program

There are a number of items to be addressed as regular maintenance practices.

8.2.1 Items to Include

Compressor-operating gauges should be read and recorded daily as part of a maintenance observation program.

A record of machine oil changes should be maintained. Frequency of oil changes needs to be published and performed regularly.

Dryers should be observed on a daily basis.

Filter changes should all be on a scheduled basis. Records of changes should be made in a log, as well as some record indication at the device itself.

The piping system should be checked for an excess accumulation of water at various points. When the accumulation increases or the frequency of accumulation increases, there is reason to suspect equipment malfunction.

Receiver tanks should be blown down on a regular basis. Safety valves need to be tested periodically.

Compressed air-consuming devices need to be inspected on a regular basis. Many leaks are found because of worn hoses or sticking air valves. Cylinder packing wears out. Cylinders begin to leak air and waste energy.

Many organizations make users of the equipment responsible for maintaining their devices in good repair. This activity should be a first line management responsibility.

Good maintenance practices will result in energy use and expense reductions. Poor maintenance practices will increase the cost of operations.

8.2.2 Frequency of Attention

Good maintenance practices are daily events. Every person involved with the use of a compressed air system has the responsibility of following good maintenance and conservation practices. This should become second nature to workers.

There should be a definite strategy on when testing should take place. Testing should be regarded as a preventive maintenance practice. Tests are to be carried out on a regularly scheduled basis. The exact interval for each type of test will vary with the type of plant, the type of air system and the cleanliness of the environment in which the compressed air system is located.

Compressor performance tests (Section 11.1) may require an hour or two for each machine. The piping system may have to be opened to allow for the connection of an air flow meter.

Unless there is some concern about a particular machine, this type of activity will likely be scheduled on an annual basis.

The task of reading air gauges throughout the system should be accomplished at least monthly, if not more frequently. Not only will the gauges need to be read, but the pressure drops across devices needs to be computed and compared to standards.

9.0 Compressed Air Distribution System

9.1 Purpose of the Distribution System

The air distribution system is meant to enable compressed air to travel to the sites where it is needed in sufficient quantity and at an arrival pressure that is adequate for the need. There are several implications in this purpose statement.

9.2 Allowable Pressure Drop

Power required to drive a compressor is proportional to the compression ratio. It is important to determine the output pressure of the compressor. That decision is driven by the needs of the system. There will be minimum pressure needs for air at various locations in the system. The designer must calculate pressure drops that are expected between the need point and the compressor. When those pressure drops are added to the pressure at the need point, the output pressure of the compressor is defined.

By making sure the needed pressure is as low as practical, the user creates the opportunity to operate the compressor at the lowest acceptable pressure ratio and therefore the lowest power demand.

An air stream flowing in a pipe suffers a friction-induced pressure drop. The value of pressure drop increases exponentially with increases in air velocity. Air velocity, pipe diameter and volume of flow are interrelated. In a given diameter pipe, volume of flow is a direct function of the velocity. However, when the engineer has the luxury of selecting the pipe size, then for a given volume of air flow, the velocity varies inversely with the square of the pipe diameter. A velocity in one pipe of diameter "x", will be four times greater than the velocity for the same flow in a pipe of diameter "2x".

Table 9 - 1 contains comparative data when three different diameter pipes are considered for the same duty.

Desired Air Flow is 200 cfm @ 100 psig, Pipe Length 1,000 ft		
Pipe Diameter	Resultant Velocity	Expected Pressure Drop
1-1/2 inch	1,813 ft/min	12 psi
2 inch	1,099 ft/min	2.5 psi
3 inch	499 ft/min	0.3 psi

Table 9 - 1

Since the pressure drop is a function of the velocity, the pressure drop is huge in the 1 inch pipe and small in the 3 inch pipe.

The system designer must make the piping system large enough to accommodate the required flow without excessive pressure drop, but not so large that the cost of building the piping system becomes excessive. Considerable investigation and experience are required to find the right balance.

There are several conditions in a distribution system that may affect the system performance. Friction from flow in the pipes lowers the pressure as air travels to the working device. The factors associated with friction are: diameter of pipe, length of pipe, number of bends and turns and volume of air flow.

Figure 9 - 1 Pressure Drops in Pipes

To overcome a low pressure situation in the system, a plant engineer will often consider putting an additional compressor to work. In many cases the real problem is in the distribution system. The system is not adequate to handle the volume of air. When the engineer adds compressor capacity and tries to force more air through the same system, the net effect can actually be a further decrease in pressure.

Figure 9 - 1 shows the friction pressure loss per hundred feet of pipe associated with 100 psi air flowing through different size pipes at different volumes.

9.3 Piping Designs

The layout of the system is crucial to efficient operation. The most effective system is a loop system. A loop arrangement allows air to reach the operating device from two directions. This is extremely beneficial in times of heavy instantaneous demand. Additional piping in the loop system acts as increased reservoir capacity.

Figure 9 - 2 illustrates the most effective and least effective types of piping layouts.

If parts of the plant or major machine centers are not used regularly, they should have valves to isolate them from the flow of air when not in use. This will prevent loss from leaks in the unused area.

Assuming the distribution header is located overhead, all application lines should tie into the

top of the header lines. This will prevent any accumulated moisture from draining out of the header into the application line. The drop lines should be located as close to the application as possible and be as short as possible. This will minimize pressure drop.

Figure 9 - 2 Air Distribution System Layouts

Air distribution headers should slope away from the compressor at a rate of about 1/8" per foot. This will prevent any moisture from flowing back to the compressor. The low spots in the system should be identified and have drain traps installed.

Elbows should be made with long radius fittings wherever possible. This minimizes the pressure loss due to friction in the joint.

If changes or additions to the distribution system need to be made, there are several choices in pipe. The best, but most expensive on a per foot basis, is copper pipe. It has adequate strength and will not corrode. That is important when high quality air supply is essential.

The most commonly used pipe is mild steel. A fault associated with its use is that it will rust when exposed to moisture. If the plant has a good drying system and the system is properly maintained, this is not a problem. Galvanized pipe is an option, but not a very good one. The galvanized material tends to wear and break loose, thus plugging the filters and wearing the application parts.

Plastic pipe has been used in the past because of its light weight and low cost. **However, PVC type plastic pipe is not recommended.** In many states its use with compressed air violates state safety codes. There is a non-PVC type of ABS plastic pipe specifically made for compressed air

systems. It is essential that state safety codes be consulted before recommending its use.

Energy saving measures, resulting from improvements to the air distribution system, will be unique discoveries in each system. The plant engineer will have to identify improvements and understand how the improvements will affect energy consumption. Putting a value on energy savings is discussed in Section 11.

9.4 Air Receivers

Air receivers are dedicated, pressure rated, compressed air storage tanks. There are a number of reasons for the use of receivers. Receivers at the compressor end of the system take pulsations out of an air system that is served by a reciprocating machine. Receivers at the far end of the system serve as reservoirs, which feed short term, high demand loads. Receivers are necessary when the compressor operates with a "low unloaded power", part load control. During the unloaded period, system air needs are satisfied through the storage capacity of receivers and the distribution system.

There are situations in an industrial plant where low pressure problems can be addressed through the use of air receivers. The compressor performance curve should be consulted to see whether or not the plant has adequate compressor capacity. It is possible for a compressor to be operating at part load with a good distribution system and the plant may still "run out of air". This occurs because there isn't enough storage capacity to meet momentary demands. Some loads are high volume, short term in nature. The compressor controls cannot react quickly enough to meet these short term demands.

A solution to this problem is not to add compressor capacity but to add system storage capacity in the form of receivers. In a looped system, a receiver should be located at the far end of the loop (away from the compressor). If this still does not solve the problem, a receiver should be located as near as possible to the device that has the high momentary demands.

Receivers are intended to be remedies for short term loads. In distribution systems where there is an inadequacy in piping size, a receiver located in the problem area can alleviate short term pressure drop problems. It is impractical to build receiver tanks large enough to substitute for compressor capacity for periods longer than 10 to 20 seconds. An expression¹ that relates time interval, system pressure, short term demand and receiver size is:

$$V = T \cdot C \cdot P_o / (P_1 - P_2)$$

Where:

V	=	Size of the receiver tank in cubic feet
T	=	Time in minutes
C	=	Demand in cubic feet per minute
P _o	=	Atmospheric pressure, psia
P ₁	=	Initial receiver pressure, psig
P ₂	=	Final pressure, psig

¹ Rollins, John P., Compressed Air and Gas Handbook, fifth edition, Compressed Air and Gas Institute, Prentice Hall, p 137.

As an example, the size of a receiver to sustain a demand of 200 cfm for 20 seconds with an initial receiver pressure of 110 psig and a final pressure of 90 psig would be:

$$\begin{array}{ll} T = 0.33 \text{ min (20 seconds)} & C = 200 \text{ cfm} \\ P_o = 14.7 \text{ psia} & P_1 = 110 \text{ psig} \\ P_2 = 90 \text{ psig} & \end{array}$$

$$V = (0.33 * 200 * 14.7) / (110 - 90) = 48.5 \text{ cubic feet or about 360 gallons.}$$

10.0 Air Consumption

The whole purpose of creating a compressed air system is to provide a utility at devices where it can be used beneficially in a process or task. It is essential that the properties of compressed air be delivered to the point of use at the desired levels. (Properties should be understood to include: pressure, volume, temperature and degree of cleanliness.)

Establishing realistic requirements is wise. Establishing excessive requirements is not only energy wasteful, but it is costly as well.

10.1 Sizing Devices, Design Volume and Pressure

When equipment is designed to be powered with compressed air, the manufacturer normally specifies an amount of air consumption needed for various types of duty. There is often a recommendation as to desired pressure level(s) for proper operation.

A plant engineer must take the consumption air requirements for all uses into account when designing a system.

Device	Peak Air Demand cfm	Avg Air Demand cfm	Demand Duration Time/Time%	Equivalent Demand cfm	Pressure psig
Air Cylinder No. 1	20	20	60%	12	90
Air Cylinder No. 2	20	20	40%	8	90
Air Cylinder No. 3	30	30	40%	12	90
Air Cylinder No. 4	30	30	40%	12	90
Air Cylinder No. 5	40	40	25%	10	90
Air Cylinder No. 6	40	40	25%	10	90
Tool Indexer No. 1	15	15	20%	3	75
Tool Indexer No. 2	15	15	20%	3	75
Blwdwn Wand No. 1	30	15	10%	1.5	80
Blwdwn Wand No. 2	30	15	10%	1.5	80
Blwdwn Wand No. 3	40	20	50%	10	80
Blwdwn Wand No. 4	40	20	50%	10	80
Air Knife	50	50	100%	50	80
Totals	400	330	-----	143	-----

Table 10 - 1

Table 10 - 1 contains data for a small system. Data in this table can be used to illustrate a designing/sizing technique.

First, it is important to recognize the various types of demands. Peak demand is the highest rate at which a device can consume air. Average demand is the rate at which air will be used, when it is used. There are other factors to consider, such as the time interval when the device will be using air at the average value and the frequency of that interval.

From data in Table 10 - 1, absolute peak load would be 400 cfm. This demand would occur when every device was being used at peak demand. The likelihood of this occurrence is remote.

The accumulation of average demands is 330 cfm. This demand would occur when every device is operating at average demand at the same time. There is a possibility this demand may occur, but given the duration diversity, the accumulated average demand requirement will be considerably less than the potential total most of the time.

Assuming constant operation, the equivalent demand is a smoothed demand for each device, This is the overall duty that the compressor will serve on-average.

How does the plant engineer size a compressor(s) to meet this need? There is no single method that constitutes an obvious answer. An analysis of data laid out in Table 10 - 1 is a good way to start. The need for a receiver in the system becomes apparent.

A compressor with a 200 cfm capacity and a 50 hp motor would deliver more than enough air to meet the equivalent demand. The machine would operate at an average volumetric load of 71.5% (143 cfm/200 cfm). With a modulated inlet damper part load control, the average power demand will be at about 90% of full load. With a low, unloaded power, part load control, the average power demand will be at about 76% of full load.

A receiver can be used to supplement compressor output to meet the momentary needs and to allow the use of a "low unloaded power" part load control. A receiver similar to the receiver sized at the end of the discussion in Section 9 would provide up to 200 cfm for 20 seconds to meet momentary demands.

The plant engineer needs to investigate the nature of air demands for all of the devices to determine the degree of diversity in the demands. The engineer must make a judgment as to the overall impact of combined demands. Based on the best data available and armed with the most likely assumed conditions, the engineer makes a decision on selecting one or more compressors to meet the plant needs.

Data in Table 10 - 1 shows a maximum pressure need of 90 psi. The system must meet the need of the highest pressure demand so the compressor will develop air at a pressure that will be at or above 90 psi at the point of use. An analysis to find expected pressure drops of the distribution system will be required to determine system pressure.

When it is possible to divide different pressure needs into conveniently served locations, it is possible to use regulators to reduce pressure in branch lines. This pressure reduction provides air at the required level, allowing the air to slightly expand and take better advantage of the compressed volume and reduce losses due to leaks.

Individual loads served from a main that can use air at lower pressures should be equipped with

pressure regulators to conserve air and energy. In some cases where there are substantial air needs at two distinct pressure levels, i.e. 110 psig and 90 psig, it may be appropriate to construct separate distribution systems, one for each pressure levels.

10.2 Intended Losses

Intended losses need to be differentiated from unintended losses. The easiest way to do that is to say unintended losses are leaks of all kinds. Intended losses are uses which were ill-conceived to begin with or which have gotten out of hand.

An example of an intended loss would be using a stream of compressed air to cool an overheating bearing. The obvious solution to the problem is to obtain a proper bearing so that it does not overheat.

Air is used to break some castings away from molds. There is always a tendency to use more rather than less air. Perhaps some more precisely controlled method is appropriate so that just the right amount is used.

In the food processing industry, air is used to provide a continuous sweep of belts, carriers or product. Attention to each use may result in less intended waste through the use of better or more sophisticated controls.

10.3 Unintended Losses

System leaks are the greatest source of unintended losses.

Leaks occur everywhere, from the compressor, through the piping system and at the points of use.

The struggle to control leaks begins the day the air system commences operation. That struggle will continue until the day the system is taken out of service.

Reports from utilities that have implemented conservation programs show that as much as 50% of compressor capacity is used to feed system leaks. Rarely is there an industrial plant that can claim leaks to be less than 20% of air load.

11.0 Calculating Performance and Valuing Savings

11.1 Establishing Compressor Performance Curves

A performance curve relates the amount of air delivered by a compressor to the power demand of the compressor drive. All compressors are at the highest level of energy efficiency when loaded to full capacity. It is when compressors must operate at less than full capacity (part load) that energy efficiency suffers.

When air consumption reductions are anticipated, those reductions should cause energy reductions. However, in order to accurately access the size of energy reductions, operating relationships between air flow and power must be known. Creating a performance curve establishes the desired relationship for the specific compressor in question.

Figure 11 - 1 is a test setup for recording compressor data from which a compressor performance curve can be developed. This set-up is for a compressor which has either on-off, low unloaded power, part load control, or is a reciprocating machine with inlet valve control. A receiver tank must be in the air circuit so that the pulsating flow from the reciprocating machine will be damped out and the intermittent flow from part load controls will be smoothed.

The air piping must be opened so the air flow meter can be inserted into the system. For test purposes the air passing through the meter will be discharged to the atmosphere. Obviously, the compressor must be taken out of service for the duration of the test.

Data used to construct a performance curve consists of developing a number of operating points by measuring power levels at several different air flow volumes.

Table 11 - 1, on the following page, contains simulated field test data. This type of data is required when dealing with a reciprocating machine or when the compressor control system dictates that the compressor operate at definite levels, i.e., low unloaded power control. (Data are collected according to Table 11 - 2 for compressors with true modulation control.)

In the Table 11 - 1, columns which have the word "Read" contain raw data that is measured and recorded in the field. Energy data in column "h" is calculated separately as explained below.

The data in the other columns are calculated according to the formula presented in the column.

(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)
Stop Time TOD	Start Time TOD	Elapsed Time Min	End Accum cf	Begin Accum scf	Total Accum scf	Average Flow Rate cfm	Energy Used kWh	Average Power KW
Read	Read	(a-b)	Read	Read	(d-e)	(f/c)	Calc	$h/(c*60)$
11:15	11:02	13	2600	0	2600	200	8.19	37.8
11:21	11:16	5	3500	2600	900	180	2.91	35.0
11:28	11:22	6	4460	3500	960	160	3.21	32.1
11:35	11:30	5	5110	4460	650	130	2.32	27.9
11:41	11:36	5	5610	5110	500	100	1.97	23.6
11:47	11:42	5	5985	5610	375	75	1.67	20.1
11:53	11:48	5	6185	5985	200	40	1.26	15.1
11:53	11:48	5	6185	5985	200	0	0.63	9.5

Table 11 - 1

In order to measure and collect these data, the user must obtain the use of some special instrumentation. A recording power meter is essential.¹ The power meter should contain the ability to measure true rms, three phase power and provide an integrated value in one minute intervals. Energy consumed (in kWh) over a one minute interval will be the value, determined by dividing the power reading, by 60 minutes.

Energy consumed over a longer period will be the sum of the energy consumed during the individual minutes. This is the energy value that is placed in column "h".

An air flow meter² is also essential to gathering proper data.

Figure 11 - 2 is a graphic plot of the data points. The plot is the compressor performance curve.

This curve represents a rotary screw type compressor with "low unloaded power" part load controls. If it were a reciprocating machine, the data points would be at 0%, 50% and 100% capacity, if the compressor was double-acting and

Figure 11 - 2

¹ A Dranetz Model 808 Demand Analyzer or similar recording multimeter is the type of instrumentation that is required.

² Flow meters for measuring compressed air flow are unique instruments. Contact Carroll, Hatch & Associates, Inc. for information about such devices.

had simple intake valve control. If the reciprocating compressor were equipped with more complex inlet valve controls, points would also be at 25% and 75%.

Figure 11 - 3 is a test set-up for full modulation-type part load controlled compressor testing. It is simpler to record data points for a compressor equipped with inlet damper modulation or an adjustable speed drive part load control. In those cases the compressor output adjusts to the level of demand. Once the flow reaches a steady state the power and flow can be read directly. There is no need to accumulate values and do calculations.

Table 11 - 2 is a data table for a fully modulating-type compressor.

(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)
Stop Time TOD	Start Time TOD	Elapsed Time Min	End Accmu scf	Begin Accum scf	Total Accum scf	Average Flow Rate cfm	Energy Used kWh	Average Power KW
N/A	N/A	N/A	N/A	N/A	N/A	Read	N/A	Read
						200		37.8
						180		36.7
						160		35.5
						130		33.8
						100		32.1
						75		30.7
						40		28.3
						0		26.5

Table 11 - 2

A performance curve, for the compressor from which data in Table 11 - 2 was gathered, is

presented in Figure 11 - 4. Observation of the curve shows that the power does not drop off nearly as much as it does in the graph for Figure 11 - 1. This is a visual presentation of the difference between the two types of part load controls.

The relationship between power and air flow appears to be a straight line in Figure 11 - 4. The linear relationship is described in the expression " $Q = m_Q P + b_Q$ " when power is the independent variable and flow is the dependent variable. Conversely, the expression is " $P = m_P Q + b_P$ " when flow is the independent variable and power is the dependent variable.

Where:

Q = Compressed Air Flow, cfm
m = Slope of line
P = Required Power
b = Constant

A regression analysis may be used to establish the values of m and b to be 15 and -1050 respectively, when flow (Q) is the dependent variable and 0.0667 and 70 respectively, when power (P) is the dependent variable. (These values are used in the next section to develop demand curves.)

In summary, the expressions that relate power and air flow for this compressor operating under the existing part load controls are:

$$Q = 15P - 1050 \text{ (When flow is the dependent variable.)}$$

$$P = 0.0667Q + 70 \text{ (When power is the dependent variable.)}$$

There are occasions when the relationship may be curvilinear. In these cases it will be necessary to find an expression that does a better job of relating power and air flow than the straight line that comes from a regression analysis.³

When data are gathered, the process must be well planned. Critical to the entire exercise is the selection of a control pressure value. The control pressure should be a value that is the usual system pressure. Frequently it is the pressure at the output of the compressor. However, when a receiver is in the test set-up, the pressure should be sensed on the down stream side of the receiver.

³ The user is encouraged to obtain a copy of a program named "Curvefit". This program uses data points to establish best fit curves. That program is a "Shareware" program, \$10.00. Copies may be obtained from Thomas A. Cox, 102 Evergreen Street, Easley, SC 29640.

A typical test procedure will begin with the compressor running at full air output at the desired pressure. The throttle valve on the test unit is opened until the compressor is not able to hold desired pressure. The valve is then slowly closed until the desired pressure is reached. This is the first data point. Power and flow readings are taken. When using the first type of set-up (Figure 11 - 1) the accumulated figures must be recorded.

After noting the first data point information, the throttle valve is slightly closed. This simulates a small drop in system demand. The compressor reacts to the drop in demand through whatever part load control is in command. Upon reaching steady state conditions, all readings are taken again.

The system pressure value may increase during this process. The amount of increase is dependent upon the part load control valve. Some air valve controls require as much as a 10 psi differential to drive the inlet damper from full open to full closed. The smallest range will be about 2 psi.

Data points readings continue in the same fashion. The throttle valve is inched toward a closed position. Finally, a reading is taken with the valve fully closed. This is the zero flow power.

It is advisable to take at least two sets of readings per machine to be sure of the data. With a larger amount of data the results should be more reliable.

11.2 Establish A Demand Profile

In order to adequately describe compressor duty, the demand to each compressor drive must be monitored for a representative period of time. Demand refers to the amount of power being used by the machine. With a performance curve (described in the previous section), power demands can be readily converted into air demands. Thus, by measuring and recording power used by the compressor drive, a profile of compressed air use may be generated.

Figure 11 - 5

A representative period of time means an adequate amount of time to describe the day-to-day demands on the compressors. Where loads are uniform, this may be only a period of two or three days. Typically, where loads are variable, the period may be one or two weeks. The data is to be entered into a spreadsheet. A plot of demand versus time of day for each compressor is to be produced in graphic form.

The demand profile assumes the compressor will operate at either a constant system pressure (modulating part load control) or within a narrow band of pressures (low unloaded

horsepower control). It may be necessary to monitor system pressure during the demand recording in order to be sure the compressor is operating against the typical system pressure during the recording period.

The curve in Figure 11 - 5 is a demand profile for a typical air compressor in an industrial plant. The curve follows a time profile which would indicate maximum demand during the day shift and significant reduction at other times. There is still a substantial power requirement in the off-shift hours.

Area under the demand curve represents energy in kiloWatt-hours. Through the use of a spreadsheet, the daily energy use is calculated at 2,371 kWh.

11.3 Establish Air Demand Curve

Figure 11 - 6 is the air flow demand profile for the plant. The relationship between power demand and air flow was developed in the prior section. Air flow was determined by using the expression that makes air flow dependent on power.

Area under the curve represents total air delivered. In a day the amount of air is 621,772 cubic feet.

When implementation of energy conservation measures (ECMs) are proposed and the volume of compressed air to be saved can be estimated, the profile of the air demand curve should be altered to reflect the change.

Figure 11 - 6

A new power curve can be forecast through the use of the performance curve. However, now the air flow is the independent variable and power will be computed as a function of air flow. Energy savings can be forecast from a reduction in area under the demand curve.

11.4 Illustration of Savings

The plant engineer studies his/her system and determines that the following air savings are possible:

- Leak reduction - 100 cfm around the clock.
- Elimination - 100 cfm on day shift.

These savings are imposed on the air flow curve. The result is a diminished demand for compressed air. Figure 11 - 7 shows the new air flow curve as well as the original curve. Now the total amount of air used in a day is 477,772 cubic feet. The difference is a savings of

Figure 11 - 7

144,000 cfm, or a reduction of 23.2%.

Through the use of the expression developed as part of the compressor performance curve exercise, air demand is translated into power demand. The new and old power curves are presented in Figure 11 - 8.

Daily energy use under the new curve is calculated as 2,211 Kwh. This is 160 kWh less than current practice. The energy savings is 6.7% (160 kWh/2,371 kWh).

The air use reduction of 23.2% results in an electrical energy savings of only 6.7%. This is testimony to the fact that the inlet damper part load control is extremely energy inefficient.

The plant engineer is well advised to evaluate the possibility of changing part load controls on the compressor.

11.5 Value Savings

Daily energy savings are 160 kWh. If the plant operates 250 days per year, the annual savings would be 40,000 kWh per year (160 kWh/Day * 250 days/Yr). At a value of \$0.04/kWh, the expense reduction would be \$1,600/Yr (40,000 kWh/Yr * \$0.04/kWh).

Figure 11 - 8

11.6 Section Summary

The relatively small amount of savings that has been achieved in the example should not discourage the plant engineer from embarking upon an air conservation program. The process that was described is the essence of the instruction. An understanding of how to proceed with the process will provide the engineer with tools to intelligently access and value savings potential.

12.0 Investment Justification

Energy conservation measures in compressed air systems take effort to accomplish. In the world of business, effort is measured in terms of expense and/or capital investment. Justification of effort requires a cost benefit comparison. There are many methods for making that comparison. Each company or organization usually adopts a specific method that they like.

One of the most serious mistakes a plant engineer can make is to assume that because a project is an energy project, it will receive special funding consideration. It is far more realistic to assume an energy project must compete with all other projects for funding. As such, the engineer should state a case for the project in the traditional sense. If there are special incentives offered by a utility or energy service company, those incentives need to be integrated into the evaluation. A well documented proposal that addresses the situation in the prescribed company format is the best method for seeking project authorization and funding.

12.1 Cost of Improvement

As part of any analysis, the cost of making an improvement must be determined. That cost must be found through the traditional process of describing the change that is desired. The description must include quantification of labor costs and material costs. Company policy will dictate whether or not to include plant engineer time costs or any other overhead costs. The more accurate the cost estimate is, the better the financial analysis.

12.2 Value of Improvement

In the prior section of the book there is an explanation of a method for valuing energy savings. That method, or some other suitable method may be used to establish a reasonable value for the energy saved.

The valuing method should be sophisticated enough to provide the type of information used in the desired analysis. For instance, a one year savings value is adequate when the simple payback period method is used. However, when a discounted cash flow method is used the annual savings must be projected over several years. The projections should take into account inflation, energy cost changes and any other cost changes that are foreseeable.

12.3 Method of Evaluation

As a first cut, many firms use a simple payback period determination. The payback period is calculated by making a ratio of the value of capital cost required to affect the conservation

Air Compressor Project Cost - A	\$8,000
Annual Energy Saved, kWh per year	120,000
Value of Energy, \$/kWh	\$0.040
Value of Energy Saved - B	\$4,800
Simple Payback Period - Years (A ÷ B)	1.7

Table 12 - 1 Simple Payback Period Determination

measure, to the value of the annual expense reduction resulting from the measure. This is an easily accomplished calculation and provides a quick measure of effectiveness.

The simple payback method has limitations. That method does not take into account the time value of money. Secondly, the method assumes static energy costs. There are occasions when these limitations make the simple payback method unsuitable.

A discounted cash flow analysis is more sophisticated and does take into account the time value of money. This type of analysis also allows a company to include the effects of income taxes and the cost of borrowing money to be taken into account in the analysis. In order to use a discounted cash flow analysis, there are a number of financial assumptions that need to be taken into account. Tables 12 - 2 and 12 - 3 set the conditions for a company to take a loan to finance the project.

Table 12 - 2 provides the conditions that are included in a loan the company makes to finance the project. The assumptions begin with the company funding 20% of the cost (\$1,600) and borrowing 80%.

Company Out of the Pocket Contribution	\$1,600
Loan @ 80%	\$6,400
Interest Rate	10.0%
Term, Years	5

Table 12 - 2 Loan Terms

The loan payments in Table 12 - 3 are projected over five years. The payments are divided into interest and principal.

Year	0	1	2	3	4	5
Annual Payment	\$0	\$1,688.30	\$1,688.30	\$1,688.30	\$1,688.30	\$1,688.30
Interest Expense	\$0.00	\$640.00	\$535.17	\$419.86	\$293.01	\$153.48
Principal Payment	\$1,600	\$1,048.30	\$1,153.13	\$1,268.45	\$1,395.29	\$1,534.82
Unpaid Balance	\$6,400	\$5,351.70	\$4,198.56	\$2,930.11	\$1,534.82	\$0.00

Table 12 - 3 Five Year Loan Schedule

Some additional assumptions go into the cash flow analysis. The company is assumed to be in a federal tax bracket where income tax rates are 33% of profit from operations. The additional maintenance expense included for the energy conservation measure is \$500 per year, with the cost inflating at 4% per year. The cost of electricity is projected to increase at the rate of 5% per year. Table 12 - 4 presents all of the information for a five year period.

Year	0	1	2	3	4	5
Annual Energy Saved, kWh per year	0	120,000	120,000	120,000	120,000	120,000
Value of Energy, \$/kWh	\$0.040	\$0.040	\$0.042	\$0.044	\$0.046	\$0.049
Value of Energy Saved	\$0	\$4,800	\$5,040	\$5,292	\$5,557	\$5,834
Additional Maintenance Expense	\$0	(\$500)	(\$520)	(\$541)	(\$562)	(\$585)
Interest Expense	\$0.00	(\$640)	(\$535)	(\$420)	(\$293)	(\$153)
Net Profit Contribution	\$0	\$3,660	\$3,985	\$4,331	\$4,701	\$5,096
Income Taxes	\$0	(\$1,208)	(\$1,315)	(\$1,429)	(\$1,551)	(\$1,682)
Principal Payment	(\$1,600)	(\$1,048)	(\$1,153)	(\$1,268)	(\$1,395)	(\$1,535)
Net Cash Flow	(\$1,600)	\$1,404	\$1,517	\$1,634	\$1,754	\$1,880

Table 12 - 4 Five Year Cash Flow Projection

The internal rate of return to the company is 90.5%. This is the equivalent of investing money in the bank and obtaining an annual return of 90.5%. It is a very good investment.

The choice of method(s) to be used to accomplish a financial analysis is up to the plant engineer and his/her management to decide. The important consideration is that a financial analysis is accomplished so that the project can compete with other projects for funding.

13.0 Generic Energy Conservation Measures

A number of compressor system energy conservation measures have been identified. A list of those ECMs is presented in this section as a means of assisting energy analysts in discovering opportunities.

13.1 Repair System Leaks

In poorly maintained compressed air systems, more than 30% of the air produced can be wasted through leaks. During plant shutdowns, leakage rate (cfm) can be measured and the energy savings from leak repairs can be estimated. Major leak points should be marked for repair and efforts to minimize the number of leaks should be included as part of the regularly scheduled maintenance program. Knowing the location and extent of leakage, as well as the dollars wasted, is the first step in encouraging repairs.

13.2 Sequencing of Multiple Compressors

At sites with multiple individually-controlled compressors, it is common to have periods of low demand, during which time the compressors operate at minimal loads. Since the performance of compressors is reduced at part loads, there is an opportunity to save energy by operating fewer compressors at full load to meet the demand and completely unload or shut off others.

Sequencing control systems accomplish this by matching the compressor output to changing demand. In general, this involves all but one compressor operating at full load and any compressors idling for extended periods be shut off.

Control systems can range from simple sequencers to complex computer-based designs which control and monitor many operating parameters. For sites where sequencing is used, a brief description of how the system operates should be recorded. Multiple compressor sites that do not use sequencing, but have variable compressed air demand, should schedule a follow-up study to determine the potential for energy savings and the feasibility of installing sequencing controls.

13.3 Proper Selection of Compressor Part Load Control Options

Manufacturers supply some models with a choice of capacity control modes, such as five-step part load for reciprocating machines and different methods of inlet throttling for rotary screw machines. When operating these types of compressors at low loads for extended periods, energy can be saved by selecting a control mode that maximizes overall performance. Selection is usually accomplished manually, however, some compressors are able to choose control modes automatically in response to changes in demand. Compressor control options available at each site should be assessed to estimate the potential for energy savings.

13.4 Compressor Performance Improvement

During a compressed Air System Study, full load output flow, full load power input can be measured for each compressor to get an indication of its performance (i.e. ratio of full load cfm output per KW input). If these parameters are re-measured after compressor overhaul or maintenance, performance improvement and energy savings can be calculated. Note that this

ratio should not be compared with the compressor's rated performance provided by the manufacturer. Manufacturer's data is based on brake hp while the field test will be based on power provided to the compressor drive.

13.5 Optimization of End-Uses of Compressed Air

For many industrial tasks, electric or hydraulic power can prove to be more cost-effective than compressed air and potential conversion opportunities should be examined. Where compressed air must be used, properly sized, energy-efficient components (i.e. amplifiers) should be included in the design. Often, the pressure drop across fittings and connections can be as large as the losses through main supply piping. Open tubes, long lengths of hose, undersized parts, quick disconnects, etc., should be avoided wherever possible and on/off solenoid valves should replace continuous blow-offs on processes with intermittent air needs. Recognizing the cost of compressed air as a utility is the key to the design of efficient end-uses.

13.6 Switch End Use Device to Another Utility

There may be situations where an electrically-driven tool can be used to replace an air-driven tool. Energy requirements for that tool may drop by 80% or more. Hydraulic systems are more energy-efficient than are air systems. Where possible, consider the use of hydraulics to replace pneumatics.

13.7 Reduction of Piping and Distribution System Losses

Inadequate pipe diameter and inefficient distribution layout contribute to large drops in system pressure and increased energy costs. A pressure drop of 10% across the entire system is usually tolerable, however, long runs of small diameter piping, excessive use of elbows and complaints of low pressure are signs that piping losses may be much greater. These sites should be recommended for a more detailed follow-up study to determine the feasibility of replacing undersized piping or redesigning the layout.

13.8 Reduction of System Discharge Pressure

Frequently, the compressor discharge pressure is set higher than necessary, in order to overcome piping losses and maintain pressure level or to serve a small high pressure requirement. Typically, these systems step-down the pressure to lower levels of various end-use locations. Wherever possible, energy can be saved by setting the compressor discharge pressure to the minimum acceptable level and avoiding step-downs in pressure. A reduction in discharge pressure from 110 psig to 100 psig can reduce energy consumption by about 4%. If there is a relatively small requirement for high pressure air, consideration should be given to a smaller dedicated compressor and separate piping. Altering the pressure setting is a simple procedure on most compressors.

13.9 Compressor Lubrication

To maximize performance and reduce energy consumption, proper compressor lubrication must be an on-going routine. Levels should be checked regularly and the lubricant should be replaced at regular intervals or as required by the operating conditions. It is good practice to take periodic samples to monitor the lubricant condition and schedule maintenance.

Proper compressor lubrication (feed rate on reciprocating machines) ensures safe, reliable operation and minimizes energy and maintenance costs. Aside from reducing friction losses, compressor lubrication has several functions, including cooling, sealing (screw type machines), cleansing and corrosion protection. Lubricants are generally petroleum-based oils, however, high performance additive packages and synthetic lubricants are also readily available.

13.10 Proper Selection and Operation of Air Dryers and Purge Air Controls

Dryers should be selected and located to provide the minimum required degree of dryness (i.e. maximum allowable dew point) with the least overall energy consumption and operating cost. The three main types of dryers are deliquescent, refrigerative and regenerative. Each type has its own dew point capabilities and energy requirements. If a dryer system is undersized or the wrong type, pressure drop and energy consumption may be excessive and an acceptable dew point may not be reached.

13.11 Relocation of Air Intake to Outdoors

Compressor performance can be improved by installing intakes at locations providing the cleanest, coolest, driest air possible - usually outdoors. In addition to the energy savings, less moisture and contaminants entering the compressor also help to reduce maintenance costs. The feasibility of extending indoor intakes to the outside should be investigated for all existing sites and new compressor installations should be designed with outdoor air intakes whenever possible.

13.12 Improvement of Filter Condition

Undersized and/or dirty filters restrict air flow, resulting in excessive pressure loss, reduced compressor capacity, poor tool efficiency and increased energy consumption. Filter conditions can be checked by inspecting the element visually or by noting the pressure drop under flow conditions. Proper selection and regular inspection and maintenance are required to minimize filter losses.

13.13 Improvement of Intercooler, Oilcooler and Aftercooler Condition

Intercoolers (on multi-stage compressors) and aftercoolers are basically air or water-cooled heat exchangers designed to lower the temperature of air after removing moisture. Similarly, oilcoolers are heat exchangers designed to remove the heat of compression absorbed by the compressor oil. Proper sizing, installation and maintenance of this equipment is required to ensure effective cooling with minimum pressure drop and energy consumption.

Most air-cooled heat exchangers can be inspected visually. Blocked cooling passages, inadequate draining of condensate, lack of maintenance, etc., are signs of reduced cooling effectiveness and should be corrected. If pressure gauges are in place at the inlet and outlet, a measurement of pressure drop can be made.

13.14 Installation of Dew Point Monitoring

Heatless regenerative dryers waste energy by consuming large amounts of dried compressed air for regeneration of the desiccant beads. Purge air can consume as much as 18% of the air

delivered to the dryer when regular (timed) purge cycles are used to control the dew point. Dew point monitoring and other more efficient control techniques are available to minimize purge air consumption.

13.15 New Compressor

When field tests produce a volume/power effectiveness ratio of less than 4.0, the use of a new, higher efficiency machine should be considered. Justification of a new machine will depend partially on the efficiency improvement and partially on the capital cost. Careful analysis of efficiency, values and economic incentives will be required before a purchase decision is made.

Case Study No. 1 - 25 hp AS Demonstration

Demonstrate the use of an adjustable speed drive controller for part load control on a rotary screw air compressor.

A laboratory-type demonstration was conducted to compare the power requirements of a 25 hp rotary screw compressor equipped with inlet valve modulation control, with the power requirements of the same compressor when modulation control was provided through a variable speed drive controller (AS).

Power and flow measurements were taken at several levels of compressor output volume and at three system pressure levels. The results showed substantial energy saving possibilities with AS control versus modulating inlet valve control.

Background

Rogers Machinery Company of Tigard, Oregon, is a supplier of compressed air equipment and air system services. The company provides goods and services to users of compressed air throughout the Pacific Northwest. Rogers Machinery sells, assembles, repairs and tests air compressors at its plant in Tigard, Oregon.

The company is energy-conscious and provided the consultants a great deal of assistance during the conduct of the Industrial Compressed Air Study project. Rogers was willing to host a demonstration at its plant where careful and detailed tests could be conducted. The Rogers site offered an opportunity to work in an industrial operating setting without the stress of interfering with an active production process.

Throughout the Industrial Compressed Air Study, the consultants observed a number of compressors with "load/unload" and "modulating inlet valve" part load controls. Many of the "load/unload" controls were not being used correctly. The principal complaints of the users concerned the amount of time required for the compressor to cycle through the control sequence and the attendant pressure-drop associated with the "load/unload" control process.

Many users found that by the time the compressor had "unloaded", air demand was such that it was necessary for the unit to "load" again. Often, the cycling process took so much time that the system pressure dropped to an undesirable level. (It should be understood that the too-frequent cycling problem could also be addressed through the use of one or more air receivers.)

It was common practice for maintenance personnel to adjust the pressure sensor controls so that the machine would not go into the unload mode. This action defeated the "load/unload" part load control strategy, but served to keep the system pressure close to constant. Compressor part load control reverted to modulated inlet valve control. Many companies consider the modulating inlet valve control to be preferable because it maintains a more constant system pressure even though the technique is far less energy-efficient than the "load/unload" control.

These situations seemed prevalent when the system demand caused a swing load compressor to

operate in the 50% to 100% capacity range. This is also the range where the use of a rotary screw design compressor with an ASD appeared to be most effective.

The use of an ASD on the compressor drive was expected to provide a modulated control method which would allow the system pressure to remain relatively constant while significantly reducing the drive load. The objective of the demonstration was to illustrate this strategy.

Test Procedure

The procedure was developed to determine the power requirements for several levels of compressor output and system pressure. The output of the compressor was directed through an air flow meter equipped with a throttling valve. System pressure drop occurred across the throttling valve. Discharge from the throttling valve was to atmosphere. The valve was used to simulate a plant load. As the valve was opened, the air demand increased. Closing the valve caused a decrease in air demand.

The compressor used in the test was a Quincy Northwest Model No. QNWF25. The compressor was driven via a 25 hp induction motor. The motor has a full load speed of 1760 RPM. Figure 14 - 1 provides a one-line-diagram of the controls. The motor was operated on an across-the-line basis for testing with inlet valve modulation part load control. The identical compressor set-up was controlled via the ASD by throwing a transfer switch.

Figure 14 - 1 Electrical One Line Diagram

The variable speed drive was controlled through a Saftronics Model G3U4018, constant torque, adjustable speed drive. The drive controller has both manual and automatic control features. Manual control is keyed into the unit from a control pad on the face of the device enclosure. When the automatic feature is utilized, a signal from a pressure transducer was used to control the adjustment of machine speed.

The air flow and pressure was measured with the Carroll, Hatch air flow meter (See Figure 14 -

2 on the following page). The components consist of a Schlumberger vortex shedding flow meter, a Kessler and Ellis Masstrol flow computer, a 100 OHM RTD and a 4-20 milliamp pressure transducer.

Figure 14 - 2 Test Set-Up

The power was monitored with a Dranetz Model 808 Demand/Analyzer power meter.

Power input in KW and the corresponding air flow output in SCFM were measured simultaneously at several air flow values. Tests were conducted with the compressor using first, modulated inlet valve part load control and then adjustable speed drive part load control. Performance curves were developed.

Under both methods of part load control the throttling valve position was set. Flow was allowed to stabilize. Stability was defined as allowing the compressor to maintain a desired system pressure as measured at the test set-up. When the pressure stabilized, power and flow readings were taken. The throttling valve position was then changed and the process was repeated.

The procedure began with the throttle valve open to the degree that the compressor was putting out maximum delivery while scarcely maintaining the desired pressure. The valve was then progressively closed in steps. Measurements were recorded at each step. This process was repeated beginning at the zero flow and progressing stepping up to full flow.

Test Results

When the compressor was under modulating inlet valve control, the ability to sustain the desired pressure occurred most satisfactorily in the flow values range of 20% to 90%. Below that range the pressure rose to significantly higher levels and below the range the pressure dropped to significantly lower levels. Conversely, when the compressor was under the control of the ASD, the pressure held very steady from 50% flow through 97% flow. Above 97% the pressure began to drop as the compressor capacity was taxed.

The demonstration illustrated the ability to maintain a far more constant system pressure with the use of an ASD for part load control as compared to the modulated inlet valve method.

The curve in Figure 14 - 3 is the compressor performance curve for the compressor operating with inlet modulation control.

Figure 14 - 3 Performance Curve for Modulating Inlet Valve Control

The slope of the curve shows this is not an efficient mode of operation. There is only a 22% drop in power associated with a 100% change in air flow. Thus, operations below full load suffer poor efficiencies. The lower the flow, the worse the efficiency.

Figure 14 - 4 shows curves for the same compressor operating under the control of the ASD for three different pressure settings. The ASD was equipped with a control loop. A desired pressure set point was selected. A pressure transducer provided an input which was compared to the desired set point. The ASD hunted for the proper speed so that the system pressure was the same as the set point pressure. When stability occurred, the measurements were made.

Figure 14 - 4 25 HP Compressor with ASD Control

The curves cover compressor operation between 50% and 100% of capacity. Operation below 50% of normal machine speed is outside the safe machine operating limitations.

Each curve shows a substantial drop in power required as the capacity is reduced. That power drop occurs because power is a function of torque and speed. While the torque remains constant when the speed is reduced, the power is reduced. The relationship is linear.

For operation at below the 50% capacity level, the compressor control reverts to modulating inlet valve control. This is achieved by setting the modulation control pressure at a slightly higher pressure setting than the set point on the controller.

The modulating control must be allowed its full range above the controller's set point. Thus, if the controller is set at 115 psi and the compressor's modulation control has a 2 psi range, the pressure setting on the compressor must be at least 117 psi. Failure to set the compressor in this fashion will result in the compressor control working against the ASD control.

Figure 14 - 5 presents a graph with performance curves from both methods of part load control. The area between the curves represents energy savings associated with the use of an ASD drive control.

Figure 14 - 5 Modulation Power Comparison - ASD Versus Inlet Valve

Economics

The objective of the demonstration was to investigate, in a controlled setting, the potential energy savings benefits from the use of an ASD. Once the potential benefits were determined, it was necessary to consider anticipated cost to determine the degree of justification for this part load control approach.

Table 14 - 1 shows the cost of the drive.

Safronix 616G3 Drive	\$3,900.00
Line Reactor for Harmonic control	\$235.00
Installation & Misc. Supplies	\$1,000.00
Total	\$5,135.00

Table 14 - 1 Cost of the 25 HP ASD Assembly

Given an energy cost of \$.04 per Kwh, 128,375 kWh of energy must be saved to pay for the drive control.

Using the performance curves for the part load control and the ASD the controller established in the previous section, energy consumption can be estimated for various conditions. Table 14 - 2 shows the time in years required to pay for the drive, on a simple payback basis. The table presents various levels of compressor use.

Hours/day 365 days/year	Load Levels % of Full Air Capacity				
	60%	70%	80%	90%	100%
	No. of years required to recover investment				
8	7.8	9.0	10.7	13.2	17.1
12	5.2	6.0	7.2	8.8	11.4
16	3.9	4.5	5.4	6.6	8.6
20	3.1	3.6	4.3	5.3	6.8
24	2.6	3.0	3.6	4.4	5.7

Table 14 - 2 Comparisons of Simple Payback

The payback is more rapid when the compressor is used many hours per day at lower levels; i.e., 60% of capacity. As the need for part load operation diminishes (the compressor is more heavily loaded), the payback period becomes significantly longer.

Conclusions

The conclusions to be drawn from the demonstration are presented below.

Consider the use of an ASD for rotary screw air compressor part load control when:

- There is the expectation that the air load will be highly variable, particularly in the range between 50% to 100% of compressor capacity.
- The need for constant system pressure is important.
- The compressor will operate annually for a high number of hours.
- Electrical energy costs are high.

By-and-large, although the use of an ASD has many advantages, the cost of the control system

is quite high compared to other approaches. Each potential compressor part load situation should be examined for the optimum approach. The ASD is only one of many possible methods of providing effective part load control. There are situations where benefits from the use of the ASD approach will be well worth the expense. Because of high costs, those situations are expected to be limited in the immediate future.

Case Study No. 2 - Add Air Dryer and Receiver to System

Demonstrate energy savings that result from the use of an air dryer and receiver in the compressed air system.

An industrial chemical plant, Neste Resins, operated a compressed air system at the plant. The system did not include an air dryer or a receiver tank.

Power to drive the compressor was monitored before and after the additions to the system were made. The results were obscured because of significant changes in the use of air from beginning to end of the demonstration.

Background

Neste Resin is located in Springfield, Oregon. The company manufactures melamine resins, urea resins, phenolic resins and phenol resins. These resins are used by the wood products industry in the manufacture of plywood, particle board, waferboard, and as an adhesive in finger-jointing and edge gluing. These products are made in batch processes. The batches range in size from 1,000 to 20,000 gallons.

There are two steps in the process. The first step is to make formaldehyde. Formaldehyde is the principal component of all the final resin products. There are three formaldehyde manufacturing plants at the site. Once the formaldehyde is made it is stored in tank farms located on the site. When the resins are being made the formaldehyde is transported via pipeline to the mixing area where it is combined with the other products necessary to make whatever resin is being produced.

Compressed Air System

There are three forms of compressed air used at the site. The first is a 12 psi blower system that is used in the formaldehyde manufacturing process. The second use of compressed air is for the instrument system.

The third use of compressed air at the site is for plant utility air. A single, 75 hp, rotary screw compressor provides this air. This compressor is backed up with a similar diesel machine. The system of interest was the plant air system. The compressor is a Sullair Model 16-b75H ACAC. System pressure is maintained at 110 psig. The compressor delivers about 340 cfm. The compressor was equipped with a modulating inlet valve control.

Compressor output was piped directly into the air distribution system. There was no conditioning equipment in the air system.

Moisture was purged from the air lines through devices located at the low spots in the air distribution system. When water condensed in the air lines it collected at the low points. Devices called "drain airs" would sense the presence of water, and when there was an adequate amount the device would blow off and discharge water and air. When the water was eliminated the discharge stopped.

Unfortunately, these devices were very sensitive and many were not working properly. When

malfunctioning they often continually blew off air. It was not uncommon to walk around the plant and find several of them blowing air. This was, in essence, a planned air leak.

Plant Air Load

The distribution system was rudimentary. There was no receiver tank or dryer. A 2" diameter pipe serves as the main header to both the west and south. Air was piped about 200 feet south and west of the compressor. Branch lines were 1/2" or 3/4".

There were several uses for compressed air at the plant:

- When small amounts of chemicals need to be introduced into the resin manufacturing process, a device called a "blow pot" is used. Chemicals are placed into the device and it is secured. Compressed air is introduced, forcing the materials into the large mixing tank.
- Tank cars carrying raw materials are unloaded with compressed air. Air eductors are used to start the flow of methanal out of a tank car into the piping system. Once in the system, the system pumps take over.
- Lines that carry chemicals are blown clean with compressed air.
- There are air-driven chemical pumps.
- There are many air-operated valves on-site.

Duty Cycle

The plant operates around the clock, seven days a week. Formaldehyde is manufactured in a continuous process. There is an operator on-site to monitor the process controls at all times.

Resin is manufactured in batches. The resin plant operates three shifts per day, seven days a week. Maintenance takes place five days a week on the day shift.

The compressor operates continuously, supplying air for various needs. When the compressor is forced to shut down the diesel back-up is used.

When system pressure is sustained at the desired level the compressor goes into a modulated part load operation.

Energy Requirements

A Dranetz model 808 meter was used to monitor power at the compressor drive. Power readings were recorded in half hour increments for a two week period. Because of the way the plant operates, the air load was steady. The load fluctuated between 44 and 50 KW.

On a daily basis the plant used about 1,150 kiloWatt-hours for utility compressed air.

Demonstration Equipment

Energy saving strategy called for a number of changes. The first was to equip the compressor with a "load/unload" part load controller. However, in order for that method of part load control to work properly, the system needed to be equipped with a receiver tank. Secondly, the addition of an air dryer would preclude the need for the "drain air" units. All of the air discharged by those devices would be saved.

The following equipment was procured and put into service:

1. Sullair model 250025-722 control kit. This controller would allow the compressor to operate with a "load/unload" part load control.
2. Sullair model MPF35 oil removal coalescing filter. This filter is necessarily placed in front of the refrigerated dryer to remove liquid water or oil from the air before it goes into the dryer. The filter is capable of removing particles down to 1 micron and liquid down to .4 PPM.
3. Zeks model 400 HBS refrigerated dryer with a low ambient package. This dryer is rated at 400 cfm at 100 psi to a dew point of 38° F with an inlet temperature of 100° F. The low ambient package is an insulation package that will keep the dryer from freezing if the ambient temperature falls below freezing. This dryer is equipped with an automatic drain device.
4. Brunner 400 gallon, 36" X 96" vertical receiver. This receiver is equipped with a pressure gauge and pressure relief valve as required by law.
5. Zeks EZDB1 timed drain valve. This is on the receiver tank.

Figure 14 - 6 shows the arrangement of the equipment. Typically, the dryer and receiver are arranged in response to the needs of the plant. If the capacity of the compressor is always adequate to meet the needs of the plant and there are no sudden and large demands for air, the receiver can be placed in front of the dryer; i.e., between the compressor and the dryer. Air is dried at a steady rate just prior to being sent to the plant.

When there are sudden demands for air, those demands are met out of the receiver. When that occurs, air passes through the dryer too quickly to be thoroughly dried. In those circumstances it is better to locate the receiver behind or downstream from the dryer. In this arrangement air is dried and put into storage prior to being used. The system demand is not limited to the capacity of the dryer.

At Neste Resin there are rarely any long term peak demands. However, the receiver was placed in front of the dryer for a slightly different reason. The pipe run from the compressor to the dryer will be out-of-doors. The airstream leaving the aftercooler is at 100° F, saturated. During most of the year the ambient conditions at the receiver and dryer will be less than 100°F. Thus, air on the way to and in the receiver will be cooling and moisture will be condensing. With the receiver tank placed in front of the dryer, air and moisture will enter and be stored in the receiver. Moisture will be collected and discharged via a water trap system. A timed valve will purge water from the receiver on a regular basis.

Figure 14 - 6 Equipment Arrangement

From an energy saving standpoint, this arrangement will help the dryer save energy. The more moisture that condenses before air reaches the dryer, the less work the dryer has to do.

The dryer is a cycling-type refrigerated unit. The dryer has a three horsepower compressor motor. The unit uses its cooling capacity to cool the mass of a cold storage heat sink reservoir. Once the heat sink reaches its minimum cooling temperature, the refrigerant compressor turns off. Compressed air is cooled as it passes through the heat sink. As air cools, water vapor condenses to a liquid. The liquid is collected and discharged from the dryer.

When the temperature of the heat sink rises to an upper limit, the refrigerant compressor switches back on and cools the heat sink again. The cycle continually repeats. This cycling of the dryer is an energy-saving feature.

Neste has done the installation with its own personnel. The equipment was housed in a covered three-sided shed, 30 feet from the compressor. The area offers protection from rain and wind but not from cold.

After the receiver and dryer were installed the maintenance personnel also disabled the drain air devices.

Results

A Dranetz 808 meter was again used to monitor power to the compressor drive. The average load on the drive diminished, but only slightly - about 0.5 KW. This was far less power drop than had been expected. It was necessary to look deeper at the situation to find out why the drop was so minimal.

An eighteen month period elapsed between the initial monitoring and the final monitoring. The peak demand during the "after" monitoring period went up 2.4 KW from "before" period. It was clear that the facility added some load. The additional load was confirmed in conversations with Neste engineers. They had added a bag house. The bag house demands air for various periods of time throughout the week. This can be seen by the increased high demand.

With the drain air devices off-line, the company has replaced wasteful load with useful load. If the drain air devices were not removed the average load would be much higher and there is the probability that a larger compressor or an additional compressor would have been added.

Economics/Finance

The delivered cost of the equipment was \$8,790. Neste purchased piping, valves and miscellaneous electrical equipment for the dryer motor. Table 14 - 3 shows the cost for all of the various items.

Item	Cost
Equipment	\$8,790
Piping	\$675
Misc. Electrical	\$825
Labor	\$1,575
Total	\$11,865

Table 14- 3 Project Costs

The plant compressor operates 8,400 hours a year. The energy savings is really minimal over past use levels. However, the savings would be substantial when compared to the new load, plus the "drain air" load that was eliminated.

A complete comparison needs to include the load in _____ by the new air dryer. The dryer utilizes a 3 hp refrigerant compressor. It is a cycling dryer, which means it operates intermittently. When running it has about a 2 KW load. This is an additional system load.

Neste Resins is served by Springfield Utility Board, a municipal utility and a Bonneville Power Administration customer. Neste pays \$0.017 per kWh for energy. Even with substantial net energy savings, the Neste situation would fail to justify this investment on economics alone.

While it appears that the payback is negligible, it really tells only part of the story. The compressor is doing about the same work, but it is doing useful work instead of feeding leaks. The energy/economic benefit is avoiding the imposition of new load by making the system more efficient.

In addition, Neste will find that clean, dry air will provide a more maintenance-free operation for many items of equipment that are actuated via compressed air. There will be savings in future maintenance costs.

The demonstration successfully showed how an existing compressor was made more effective through the use of better part load controls and through the use of a basic air drying system.

Project Problems

The compressor was so old that there was no standard Sullair unloader kit available. It was necessary for the factory to develop a kit. This custom kit was not as effective as the more contemporary models. The compressor requires a 20 psi pressure differential to work effectively; i.e., "load/unload" trigger points. This is much greater than most facilities consider acceptable. The differential did not, however, present any operating problems for Neste.

Case Study No. 3 - Synthetic Lubricants

Compare the energy levels required to operate a rotary screw air compressor when the machine is lubricated using several different fluids.

A laboratory test was conducted at Roger's Machinery facility in Tigard, Oregon. Test the theory that synthetic oils save energy when used in a rotary screw air compressor. Several synthetic oils were compared to ATF¹.

Test results showed there was no significant energy difference between any of the lubricants. The ATF actually tested at a slightly lower power level than any of the synthetics.

Background

Since the inception of the Industrial Compressed Air System Study various people have been encountered who claim that synthetic oil, when used in a compressor, will save modest amounts of energy. Although there were numerous testimonials from users of the oils with claims of savings ranging from 3% to 8%, there was no hard data were available to substantiated these claims. The synthetic oil industry has a reputation for high quality products, thus these claims seemed plausible.

Some compressor manufacturers questioned the validity of the oil manufacturers claim and others supported the claims. Many people had opinions, but no one had any test data. It was decided to include an oil test as part of the demonstration work. In the interest of including the best information available, it was necessary to develop evidence supporting or refuting these claims.

Test Procedure

The following test procedure was devised. The testing would be done under controlled conditions. The only element that was allowed to vary was the oil in the compressor. The compressor output would be metered through an air flow meter which monitored system pressure, temperature and flow. At the same time, power to the compressor would be monitored with a Dranetz 808 demand meter.

The oils would be tested in the same compressor, under the same operating conditions, one at a time. A purge cycle between each test was used to remove all residual lubricant from the previous test. Once purged of the lubricant, a new one would be put in and tested.

Some manufacturers claimed their lubricant had the ability to cling to the metal and affect the tests of the remaining oils. This claim was tested.

The first oil tested was the standard ATF. Then each of the synthetics were tested. The test concluded with a second test of ATF. The second test of the ATF was intended to prove or disprove the claim that some lubricants left residual which would effect the performance of other lubricants.

¹ ATF - This is the petroleum based lubricant that is customarily furnished for use with air compressors. This lubricant was originally developed for use in automatic transmissions.

The lubricant test was conducted December 7th and 8th, 1992. It was conducted at Roger's Machinery facility in Tigard, Oregon. The test was conducted inside the manufacturing facility. The test area was heated to a temperature of 62° F.

The lubricants tested were provided by various manufactures. The test procedure was explained to them and they were asked to provide five gallons of the lubricant that they recommend as being a substitute for ATF. ATF was chosen as a base lubricant for a comparison, because it is acceptable to all compressor manufacturers. In the past, it was the lubricant that would come standard in a new compressor. Since the oil embargo of the 1970's, many manufacturers have switched to other less expensive lubricants and some offer synthetics as an option.

The compressor used for the test was a Quincy Northwest Model F30J, rotary screw compressor. The compressor was a new rental machine. The compressor had a modulating inlet valve control which was disabled for the test. The lubricant capacity for the compressor was about three gallons.

The drive motor was a Lincoln, Lincguard 30 HP, 1800 RPM, 460 volt motor with a service factor of 1.15. The compressor was belt-driven.

All testing was done at full load conditions. In order for the compressor to be working under the same load conditions, the air system pressure was maintained at a constant value for all the lubricants.

Air flow and pressure was measured with the Carroll, Hatch air flow meter. The components include: Schlumberger vortex shedding flow meter, a Kessler and Ellis masstrol flow computer, a 100 OHM RTD and a 4-20 milliamp pressure transducer.

The flow meter was attached to the compressor outlet via a 15 foot long, 2 inch diameter hose. The hose was large enough that there was no pressure drop associated with it.

The testing commenced with ATF. Standard procedure called for the compressor to be warmed up for about 5 minutes, allowing the lubricant to achieve operating temperature. At that time, data were recorded. The drive power, air flow volume in scfm, pressure at the air flow meter and compressor temperature were recorded.

The ATF was then drained from the compressor and the first synthetic lubricant was tested. It was important to fully drain the fluid so there would be no mixing of lubricants. Procedure called for the drain plug to be pulled and the lubricant drained until it stopped dripping. Residual lubricant left after the lubricant was drained, was purged from the compressor by adding 2 gallons of the next fluid, starting and running the compressor for a short time. This lubricant was then drained and the compressor refilled with the remaining 3 gallons of the next test lubricant.

Once refilled with test lubricant, the compressor was started and operated until the lubricant had heated to normal operating temperature. At that time, the flow at the meter was adjusted so that the system was at the same pressure as it was in the previous tests. The system air pressure, air flow, power and temperature were then recorded for this lubricant.

The purging, filling and testing procedure was repeated for each of the synthetics and then

repeated at the end for the second ATF test.

Test Results

Table 14 - 4 shows the results of the test.

Lubricant	Machine Temp	Air Pressure	Air Volume	Power KW
ATF	143	107.8	97.06	24.40
Schaeffer 158 Molly Pure	141	107.9	97.06	24.57
Mobile SHC 924 150 VG 32	142	107.9	95.28	24.44
Ultrachem Chem Lube 228	142	107.8	96.13	24.62
CPI Engineering CP4601-46	141	107.8	95.53	24.62
Lubrication Engineers 84-20 GFS 20	142	107.8	96.16	24.58
Royal Purple Synfilm GT 150-46 M0912	142	107.8	95.37	24.46
ATF	141	107.9	95.56	24.40

Table 14 - 4 Results of Synthetic Lubricant Tests

It was the intention of the test procedure to capture all of the readings at the exact same pressures as measured at the pressure transducer. It was possible to set the meter so as to stabilize system pressure ± 0.1 psi. The throttling valve on the meter is a two inch gate valve. Valve adjustments are too coarse to get the pressure settings exactly the same.

The ATF was tested at two different pressures equal to the high of 107.9 psi and to the low of 107.8 psi. The power requirements were the same. Thus, it would appear that maintaining the pressure to one tenth of a psi was adequate.

Before drawing any conclusions from the table, it is important to recognize that there is really no significant difference between power levels required by any of the lubricants. While the ATF showed the best energy saving characteristics, there is less than 1% difference between the largest and smallest power user. Putting this difference in perspective, if this compressor operated 4,000 hours a year under these conditions there would be a difference of only 880 kWh per year. At \$.04 per kWh this amounts to a difference of \$35.20 per year.

The fact that the ATF power requirements tested the same prior to and at the conclusion of the synthetic lubricant tests, sheds doubt over the claim that there are additives in the synthetics that cling to the metal and affect the power requirements after the lubricant has been replaced.

While the test data indicates that there is no significant difference in power requirements between the synthetics and the standard ATF, it does not make any observations about quality or life of lubricants. Most of the synthetics have lives much longer than the ATF.

Case Study No. 4 - Equip a 150 hp Plant Air Compressor with an ASD

Demonstrate energy savings resulting from equipping a 150 hp rotary screw air compressor with an adjustable speed drive controller for part load control modulation.

A sawmill was the site for this demonstration. The operating environment is typically industrial and the working conditions are subject to production demands.

Power to the compressor drive was monitored before and after the ASD was used. A compressor performance curve was developed and provides the basis for expectation of significant energy savings using this technique for part load control.

Control problems were encountered and proved difficult to solve.

Background

Avison Lumber Company is located in Molalla, Oregon. The company manufactures various sawmill lumber products.

There are several buildings on the site. Three buildings house different sawmills. Each mill serves a dedicated purpose; i.e., small log mill, large log mill and export mill. There are two sets of dry kilns and a boiler plant. There is also a planing mill and a remanufacturing plant.

The focus of this study was the large log mill. The mill consists of log debarking and infeed system, primary and secondary breakdown equipment and sorting and stacking equipment. The drying and finishing operations take place at other locations.

Portland General Electric Company provides the electrical service to the plant site.

A sawmill consists of machine centers connected by conveyors. Compressed air has historically been used for controlling log and lumber position as the material moves either endwise or crosswise on the conveyors.

Avison's large log mill includes two 150 hp rotary screw air compressors. These compressors are equipped with modulating inlet valve, part load controls. These controls are responsive and do maintain a constant air system pressure. However, this method of part load control is not energy-efficient.

Avison Lumber did not want to give up the benefit of constant air pressure associated with the modulating inlet valve control. Thus, a "load/unload" control which operates over a ten psi differential was not acceptable. Instead, a variable speed drive control was installed on the compressor drive to match air output to system demands while maintaining a constant system pressure.

Compressed Air System

The compressors feed a distribution system. Both compressors are Quincy Model 650-II, rotary screw compressors. Each is driven by a 150 hp electric motor. The nameplates on the

compressors indicate a capacity of 750 cfm. Outlet pressure from the compressors to the system is 120 psig.

One compressor is designated as the base load machine while the other is the swing load machine. Avison's control strategy calls for one machine to operate at full capacity, i.e. maximum efficiency, and the second machine to modulate as needed to meet the demand.

The strategy is accomplished by setting the pressure sensing control valve of the swing load machine at a lower pressure level than that of the base load machine. The swing load machine is intended to fully modulate before the base load machine begins to modulate. This requires about a 5 psi differential in sensing pressure setting.

The compressors are located in an open basement in the mill. There is no dedicated compressor room. Ambient air comes into the area from immediately outside the mill building. Temperature of the ambient air will be whatever the temperature of the air is in the area around the building.

Capacity meters are present on each compressor. These indicators display momentary air compressing capacity. Casual observation of the indicators show that one was always at full capacity and the other vacillates between 50% and full capacity.

The pressure of the compressed air system in the plant is 100 psig.

The compressors each feed Wilkensen Air Dryers, Model A 13-III-POO. Each dryer has a capacity commensurate with the capacity of the compressor.

The distribution system in this mill is a single line header, extending from one end of the mill basement (ground floor) to the other end in an east-west orientation. The compressors are located at the approximate middle of the distribution header. Distribution lines branch away from the header and proceed up through the mill floor to the locations of the using devices.

The main header begins as a 4" diameter pipe. As air is drawn off along the way, the header size reduces to 3" and finally down to 2" toward the end of the system. Branch lines range in diameter from 1-1/2" down to 3/4".

The first receiver tank has a 70 cubic foot capacity. The pressure at that tank is 100 psig. Toward the east there is a second receiver tank with a 25 cubic foot capacity. The pressure at this tank is also 100 psig. Toward the west end of the mill there is a third tank with a 70 cubic foot capacity.

There is also a 1-1/2 inch line that feeds air to another mill. Avison Lumber realizes the severe capacity limitation of this line. Air flow in the line supplements the output of a dedicated compressor in the other building. While this line is primarily a stand-by line, the company has been using it steadily in the recent past.

Duty Cycle

(Market forces and raw material supplies which drive the wood products industry are very

cyclical. When the project started, the facility was operating two shifts a day. As of this writing the availability of raw materials has forced the mill to curtail the second shift. Even though the mill is now being operated one shift a day, the analysis was done for the two shift a day scenario and will appear as such.)

The mill operates five days per week, two shifts per day. The first shift starts at 7:00 a.m. and continues until 3:45 p.m. There is a 1/2 hour partial shutdown at lunch, around 11:00 a.m.. The second shift begins at 4:30 p.m. and continues to 1:00 a.m.. A half hour partial shutdown occurs about 8:30 p.m., the dinner break. As a rule, one of the compressors is left on overnight. One unit also runs part of the day on Saturdays. (Demand records show that both compressors may operate overnight.) The units are turned off on Sundays.

Plant Air Load

There are several major types of air usage. The first is associated with operation of the mill and such air cylinders that operate stop fences, jump chains, movable fences, log turners and log and lumber positioners. The second type of usage comes from the maintenance in the mill which includes the use of air wrenches and tools associated with changing saws. In addition, on frequent occasions large quantities of air are used to blow sawdust away from machine centers.

Circular saws in mills are usually guided. Guides are often made of babbitt material or some other low friction material. The guide contacts the saw plate to keep the saw from varying dimensionally while it is cutting. It is common to use a combination of air, oil and water as a lubricant between the saw plate and the saw guide. Compressed air use in this application is a constant demand.

Energy Requirements

Prior to any equipment changes, recording demand meters were attached to the motor starters of both compressors. The meters recorded demand for a period of 16 days.

Recording demand for the compressor drives reveals that compressor No. 1 typically operates at a level of 120 KW. This level is within the service factor (115%) range of the drive. Demand level of this magnitude indicates that the compressor is working at full capacity when the mill operates.

Overnight the load drops to around 100 KW. This indicates a lowering of air demand, but it is still a high demand level for equipment that is servicing a "shut-down" facility.

Demand curves for Compressor No. 2 show a slightly lower load pattern. During normal working hours at the mill the load on the drive fluctuates between 90 KW and 120 KW. This unit is the swing load compressor.

When operating, the swing load compressor drive is typically loaded at 80% of drive capacity, or 96 KW. During high demands it is often required to go into the service factor range of the drive (above 100% load).

When the mill is not operating overnight during the work week, the combined power level

drops to about 180 KW. On Saturday, the level drops to about 120 KW (about half normal full load) and further drops to zero when the maintenance crew is finished about 1:00 p.m. Typically the compressors do not operate on Sundays.

Energy Savings Technology

The energy-saving strategy used for the compressors at Avison envisioned the use of a variable frequency, constant torque, adjustable speed drive controller (ASD) for one of the compressor drives. With the use of the ASD the compressor could be made to function as a variable air delivery device through speed control. It was believed this would allow the compressor to deliver compressed air at the desired pressure, but with a substantial reduction in power requirements when operating at less than full capacity.

An ASD was installed with switchgear that allowed either compressor drive to be the variable speed drive while the other unit was the base load machine drive.

Project Demonstration

The variable speed drive was controlled through a Saftronics Model H110B, constant torque, adjustable speed drive. The drive controller has both manual and automatic control features. ASD was manually controlled during the performance tests. Manual control is keyed into the unit from a control pad on the face of the device enclosure.

Figure 14 - 7 Compressor Control Layout

The sketch in Figure 14 - 7 provides a graphic description of the compressor system.

Test procedure required that power to the drive be measured before and after the ASD was utilized. The air flows and pressures were measured with the Carroll, Hatch air flow meter.

Power was monitored with a Dranetz Model 808 Demand/Analyzer power meter.

Power input in KW and the corresponding air flow output in scfm were measured simultaneously at several air flow values. Tests were conducted with the compressor using first, modulated inlet valve part load control and then adjustable speed drive part load control. The test set-up is shown in Figure 14 - 8. Performance curves were developed.

Figure 14 - 8 Test Apparatus Set-up

Under both methods of part load control the throttling valve position was set. Flow was allowed to stabilize. Stability was defined as allowing the compressor to maintain a desired system pressure as measured at the test set-up. When the pressure stabilized power and flow readings were taken. The throttling valve position was then changed and the process was repeated.

The procedure began with the throttle valve open to the extent that the compressor was putting out maximum delivery while scarcely maintaining the desired pressure. The valve was then progressively closed in steps. Measurements were recorded at each step. This process was repeated, beginning at the zero flow and progressing in steps, up to full flow.

The data generated in this test enabled the development of two compressor performance curves, one for each method of modulation. Compressor performance curves are presented in Figure 14 - 9.

Compressor minimum speed was limited to 50% of normal. Below this speed the compressor manufacturer is concerned about the ability of the compressor to pump enough oil for proper air sealing, heat dissipation and unit lubrication.

Energy savings can be projected by using the difference in demand as illustrated by the two curves. Typically, under inlet valve control, the swing load compressor operated at 96 KW. Under ASD control for the upper range and inlet valve control at the lower range, the compressor would deliver the typical air volume at a demand of 63 KW. The power reduction is 33 KW.

Figure 14 - 9 Compressor Performance Curves

The performance curves enable an accurate prediction of energy savings or requirements, due to changes in plant load.

A recording power meter was used to monitor the power for two weeks before and after the ASD was installed. The graph on the following page shows the total energy requirements to operate the two compressors during a production shift. The top line shows the energy before the compressor retrofit and the lower line shows the energy required after the retrofit. As can be seen, there is a 578 kWh reduction in energy over the course of the shift when the ASD is used.

The ASD does indeed, provide a method for maintaining desired air flow and system pressure while offering a significant reduction in power and energy.

Economics/Finance

Avison Lumber Company pays almost \$0.04 per kWh for electrical energy. The plant has a typical monthly peak compressor demand of 238 KW and energy requirement of 114,217 kiloWatt-hours (kWh). The annual cost of providing electric power and energy for the compressors is approximately \$61,550.

Given the facility operates 50 weeks a year, with two shifts a day, five days a week, then the facility operates 500 shifts a year. At 578 kWh savings per shift the yearly energy savings would

be 289,000 kWh. The value of these savings at \$.04 per kWh is \$11,560.

While it appears there is a reduction in demand, there probably is not. When observing the frequency of the power during production, the drive runs at about 35 hz for 3 to 10 minutes. In response to a demand for air, the drive control goes to 60 hz for about 5 seconds and then slowly drops back to 35 where it dwells for another 3 to 10 minutes. When it cycles up to 100% speed, it is demanding the same or slightly more power than it does at high load without the ASD. The recording power meter records the average demand over the specified period not the maximum demand.

The cost of the project, including equipment, installation and engineering was \$33,000. Future applications could be accomplished for less than that value because of the leaning that took place on this project.

A simple payback period for this investment is 2.9 years based on two production shifts a day. The serving utility has an incentive of \$25 per hp for installing an ASD. That reduces the project cost to \$29,250 and the payback period becomes 2.5 years.

As the cost of energy increases, economic justification will improve.

Figure 14 - 10 Energy Comparison Before and After ASD

Problems

Control Mode Conflict

Modulation control problems were encountered. When the automatic speed control feature was utilized, a signal from a pressure transducer was used to control the adjustment of machine speed. In testing, the engineers noticed a tendency for the inlet valve control to modulate at the same time the ASD was controlling the compressor output via speed modulation. The two methods of modulation were found to be in conflict.

Recognizing that it will be necessary to utilize the inlet valve control below the 50% delivery level, it was necessary to find a way to use both modulating methods without conflict. The problem was solved with a relay that sent a signal out to the compressor when the drive reached its minimum (50%) speed. The drive utilized a 0 to 10 volt signal for its operation. When the voltage was greater than 5.5 volts, there was no signal sent out to the compressor.

A Solenoid was added at the compressor that allowed air to go to the modulation controls only when the signal from the compressor indicated that it had dropped to 50% speed. As long as the drive stayed below 55% speed, air was allowed to go to the controls and they worked normally. When the drive advanced above 55% speed, the airflow to the compressor controls was cut off, forcing the modulation valve to stay in the fully opened position.

Speed Change Delay

There were other unanticipated problems associated with the operation of the ASD. The consultants envisioned the compressor going immediately to a specific speed and staying at that speed until demand changed, at which time the control would change the motor speed to the next appropriate speed. This did not happen. The control had a tendency to do some "searching" (going slowly up and down searching for its set point).

A similar problem was encountered and corrected at Roger's demonstration (Case Study No. 1). At Roger's, the response time on the PID loop was set very high. This had the effect of immediately changing motor speed with small changes in demand. The controller was able to adjust to changes rather quickly with a minimal amount of over-shoot. At Avison the response time for increasing speed was set very high and the decrease speed was somewhat slower. The drive advanced quickly in the face of lowering pressure and then slowly backed down to its desired speed.

Compressor Oil Pressure

There has been a rise of about 20 psi in oil pressure associated with the compressor operating under the drive control. While this was still within the safe operating range, it represented a changed condition and was investigated. This appears to be an anomaly associated with the specific compressor model. These compressors have external oil pumps. Newer rotary screw compressors do not have oil pumps, since the compressor itself acts as the oil pump. The manufacturer assures us that the increased pressure is normal when this compressor is slowed. The manufacturer stated that at one time these compressors were available as a portable diesel-

driven unit. Under those conditions, the motor could be throttled back, thus slowing the compressor. When this occurred the oil pressure in the compressor experienced the same pressure increase.

The first priority of the oil is to serve as a lubricant to the compressor. This is done before any oil goes to remove the heat of compression. The breakdown of lubricant function is about 10% of the fluid needed for lubrication and 90% for cooling. Therefore, the gauge can fluctuate in the wide safety band without adversely affecting the compressor.

Of more concern to the manufacturer would be a rise in air temperature. This would indicate that the oil is not able to remove the heat of compression. The compressors are equipped with a safety device to shut them down if the air temperature is too high. This would indicate a problem with the cooling, and thus the oil flow and potentially the lubrication. No such rise in air temperature occurred. Thus, the manufacturer was not concerned about the oil pressure.

There were, however, a few minor problems associated with the increased oil pressure. The first problem was a seal failure in the compressor. The seal was old and would have failed soon. It was replaced with a new seal of the same type and has operated without a problem. There are several high pressure hydraulic hoses with press-on fittings carrying the compressor's oil. The increased pressure caused some of the fittings to leak. These hoses were replaced.

Motor Trips

Nuisance motor trips were a problem at start-up. The ASD has over-current protection devices that varied the maximum acceptable current, depending on motor speed. The over-current was set to trip the circuit between 138 and 175 amps. At slow speeds the motor was drawing more than the 138 amps and was frequently tripping. The control manufacturer's representative suggested the over-current device be reset to 175 amps, regardless of motor speed. This cured the problem.

Control Cabinet Cooling

The control had some cooling problems. The location of the drive is in the basement of the sawmill. By sawmill standards, the basement of this particular mill offers a very clean environment. However, from the standpoint of the drive control the environment is hot and dusty. This presented some problems for the drive. It was overheating and tripping off the line. Additional cooling vents and fans were added to the control. This provided more air, but the sawdust in the air acted as a contaminate to the drive. Filtering cloth was put over the sources of air to prevent contamination, but this restricted the cooling. Thus, more venting was necessary. Eventually, enough cooling was provided.

2

of

2

15.0 Bibliography

1. B.C. Hydro. Efficient Compressed Air Systems, A Reference Guide for Positive Displacement Air Compressors, Capacity Controls and Dryers. 1991. Vancouver, BC, Canada.
2. B.C. Hydro. Energy-Efficient Compressed Air Systems. GEM Series. 1989. Vancouver, BC, Canada.
3. Baughn, E.S. Cutting The Cost of Compressed Air. *Plant Engineering*. January 1992. Pg 85-87.
4. Cooper Industries. *Engineering Data Handbook*. Gardner-Denver Industrial Machinery Division. 1991. Quincy, Ill.
5. Cooper Industries. Your Guide to Reciprocating Compressors: A Brief Review of Theory & Practice. Quincy, Ill.
6. Galus, T. How Much Air Drying Equipment is Necessary. *Hydraulics & Pneumatics*. April 1989. Pg 91-92.
7. Georgia-Pacific Corporation. Reduction in Mill Air Compressor Power Requirements through Installation of a Variable Frequency Speed Control. Bonneville Power Administration Document DE-MS79-86BP92222. March 1988.
8. Rogers Machinery Company. *Handbook of Compressed Air Data*. Portland, Oregon.
9. Rollins, J.P. *Compressed Air and Gas Handbook (Fifth Edition)*. Compressed Air and Gas Institute. 1985. Cleveland, Ohio.
10. Scales, W. Air Compressor Energy Audit. Written for New Jersey Department of Energy.
11. Staff. Application Profile: Synthetic Gear Lubricants. *Fluid & Lubricant Ideas*. September/October 1981. Pg 18-21.
12. Stofen, K. Energy Evaluation in Compressed-Air Systems. *Plant Engineering*. Reprint by: Technical Publishing. July 1992.
13. Talbot, E. M. *Compressed Air Systems: A Guidebook on Energy and Cost Savings*. The Fairmont Press. 1986.
14. Van Ormer, H.P. Is Your Air Distribution System Stealing Profits and Output, *Power*. February 1988. Pg. 35-36.
15. Van Ormer, H.P. Trim Compressed Air Costs with Synthetic Lubricants. *Power*. February 1987. Pg. 43-45.

16. Varigas Research Inc. *Compressed Air Systems: A guidebook on Energy and Cost Savings*. U.S. Department of Energy, Office of Industrial Programs. DOE/CS/40520--T2, DE 84 008802. 1984. Washington, D.C.
17. Wells, L., *Energy Savings Opportunities in Compressed Air Systems*. Presentation at Northwest Plant Engineering Show. 1992, Portland, Oregon.

DATE

FILMED

3/21/94

END