AN ABSTRACT OF THE THESIS OF

Jonathan B. Maxwell for the degree of Master of Science in Mechanical Engineering presented on November 16, 1992.

Title: Improving Part Load Efficiency of Screw Air Compressors

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George Wheeler

Air compressor systems are inefficient energy transfer devices even under the best of conditions, at full load. When only part load is required, efficiency drops further. This thesis attempts to improve part load efficiency of twin rotor screw air compressors in three ways.

First, a guidebook was written to help educate compressor users and purchasers about the significance of part load efficiency and to aid in selecting the most efficient controls for a given application.

Second, a spreadsheet-based model was developed to analyze the performance of cycling control strategies by performing a detailed simulation of one complete compressor cycle. Model calculations demonstrated that cycling losses can significantly increase average power as cycle time decreases, and that low-unload controls may be more efficient at low loads than is traditionally assumed.

Third, a microprocessor-based controller was designed and built to enhance part load performance of combined modulating and unloading type control systems. The "smart" controller is presented in this thesis. Case study results showed energy savings of 4% to 32% over conventional controllers.

Improving Part Load Efficiency of Screw Air Compressors

by

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Redacted for privacy

Head of department of Mechanical Engineering

Redacted for privacy

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IMPROVING PART LOAD EFFICIENCY OF SCREW AIR COMPRESSORS

I. INTRODUCTION

A. Background

Over the last thirty years, rotary-screw air compressors have gradually permeated the plant air systems (90-140 psig) market. Screw compressors now share the market about equally with reciprocating compressors and are still growing in popularity. They are beginning to dominate sales of large compressors. The success of screw compressors can be attributed to lower first costs compared to double-acting reciprocating compressors, more dependability than single-acting compressors, low maintenance requirements, ease of installation, and in the case of very sensitive equipment, less pulsation than reciprocating compressors.

In general, the full load efficiency of screw compressors is similar to that of reciprocating compressors. Full load efficiency is only part of the story, however. In practice, very few compressors operate at full load all of the time. One manufacturer estimates the average load to be 60-70% of full capacity.[1] Traditionally, part load efficiency has been the Achilles heel of screw air compressors. In one common design, 70% of the full load power is required when no air is delivered. Thus, the issue of part load control has become an important issue for those interested in energy conservation.

B. Thesis Contents

This thesis focuses on improving the part load efficiency of industrial twin rotor screw air compressors.

First, a guidebook on existing screw compressor controls was developed to help compressor users and purchasers understand how screw compressor controls work, how much energy can be saved by employing more efficient part load control strategies, and what operating sacrifices (if any) must be made to use them. The complete guidebook is contained in this thesis as an appendix. Basic concepts and terms are introduced in the guidebook. Also included is a list of potential air compressor conservation recommendations. They represent a compilation of personal experience and opportunities uncovered during research.

The second section of this thesis presents in detail a spreadsheet model designed to accurately predict the part load performance of screw compressor with throttling and unloading controls. This section also includes the results of actual compressor testing and analysis that were used to determine performance characteristics of screw air compressors.

Third, a new control design invented by the author is introduced. Hopefully, the new control strategy will make efficient unloading controls more appealing to otherwise uninterested compressor users.

The thesis closes with a summary, conclusions, and general recommendations to improve efficiency in industrial air compression.

Eight Appendices support the body of the thesis. They include detailed data acquisition and smart controller equipment descriptions, software listings, compressor control hardware descriptions, two equation analyses, and a prototype design cost listing.

II. AIR COMPRESSOR CONTROL GUIDEBOOK

A complete reproduction of A Guidebook for Screw Air Compressor Controls: Operating Principles and Selection for Minimum Energy Use is included in the thesis as Appendix A. First, the guidebook introduces the different types of air compressors. Next, a simple screw compressor system is described. After this orientation, the guidebook focuses on control strategies. Eight different control strategies are reviewed from the standpoint of energy use, mechanics, and applicability to different environments. The final section compiles Energy Conservation Opportunities for compressed air systems.

III. AIR COMPRESSOR DATA ACQUISITION, ANALYSIS, AND MODELING

A. Introduction

Considerable effort was invested in the study and operation of a 15 hp Quincy Northwest B-15 screw air compressor. The compressor has slide valve throttling and a low-unloading mechanism that can be adjusted or disabled. Much of the education and most of the analytical conclusions developed for this thesis were derived from a continuous clash of wills with this compressor. A description of the compressor pneumatic proportional control system is described in Appendix B. A schematic of the existing low-unload controls for this compressor and their operating sequence can be found in Appendix C.

This section provides the results of compressor performance analysis and describes a spreadsheet model designed to estimate the results of actual compressor behavior.

B. Equipment

A personal computer-based data acquisition system was used to measure, convert, and store pressure, time, and power variables. The hardware and software is described in detail in Appendix D.

C. Compressor Performance

1. Power

a. Measurement - Since a typical ammeter measures total apparent power, power factor behavior was the first characteristic to be analyzed. Based on previous experience, it was clear that power factor for any motor is primarily a function of load and, to a much lesser degree, line voltage.[2]

The compressor motor was subjected to a wide range of loads. A Dranetz power meter was used to measure voltage, current, power factor, and real (kW) and apparent (kVA) power. Based on this data, the power factor curve, shown in Figure 1, evolved.

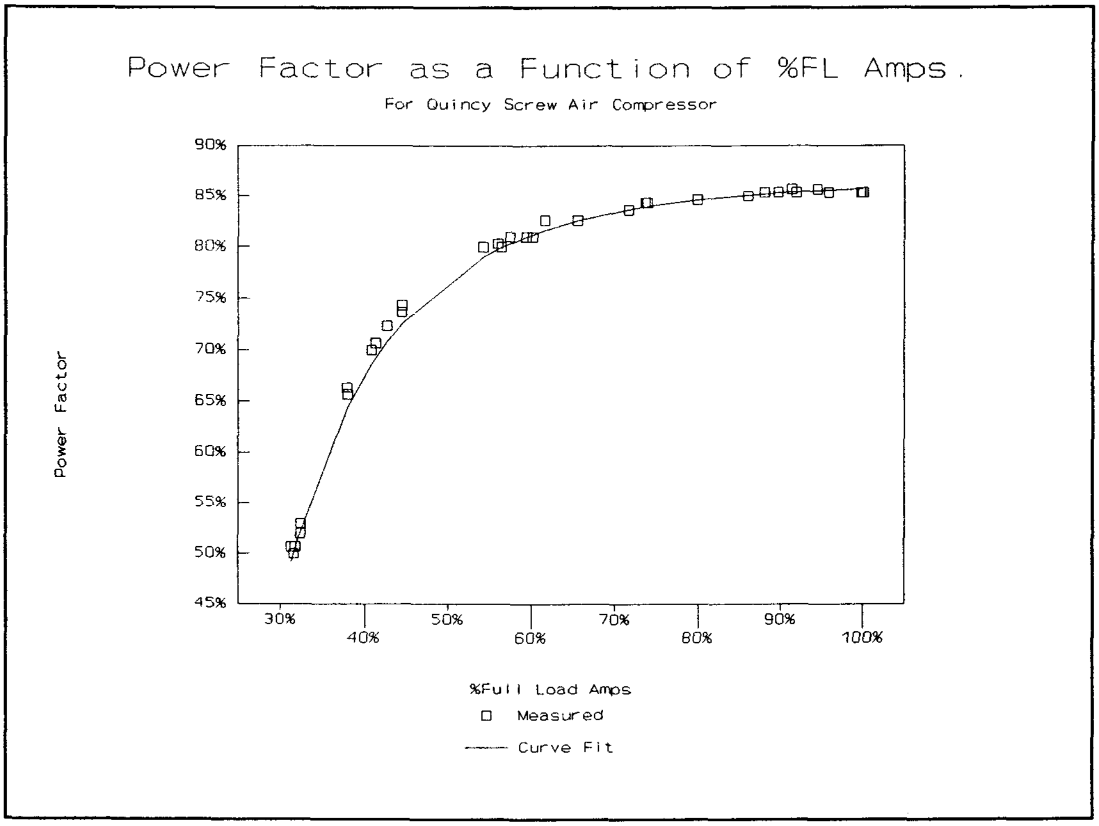

Figure 1: Power Factor Curve

A plot of the mathematically modeled curve is also shown on the graph. Its formula is:

$$pf = -0.816 + \frac{0.1056}{\%FLA} - \frac{0.06493}{\%FLA^2}$$

where,

$$\%FLA = \frac{measured\ current}{maximum\ rated\ current}$$

Standard deviation of error for the curve fit was 1.0% power factor.

Initial power measurements also indicated that the line voltage difference dropped modestly during high load periods. Since actual voltage was not measured during regular data collection, the following voltage correction factor was used:

$$V = 209.8 - \frac{5.1}{56.5}A$$

where A is measured amps. The equation is based on data shown in Figure 2.

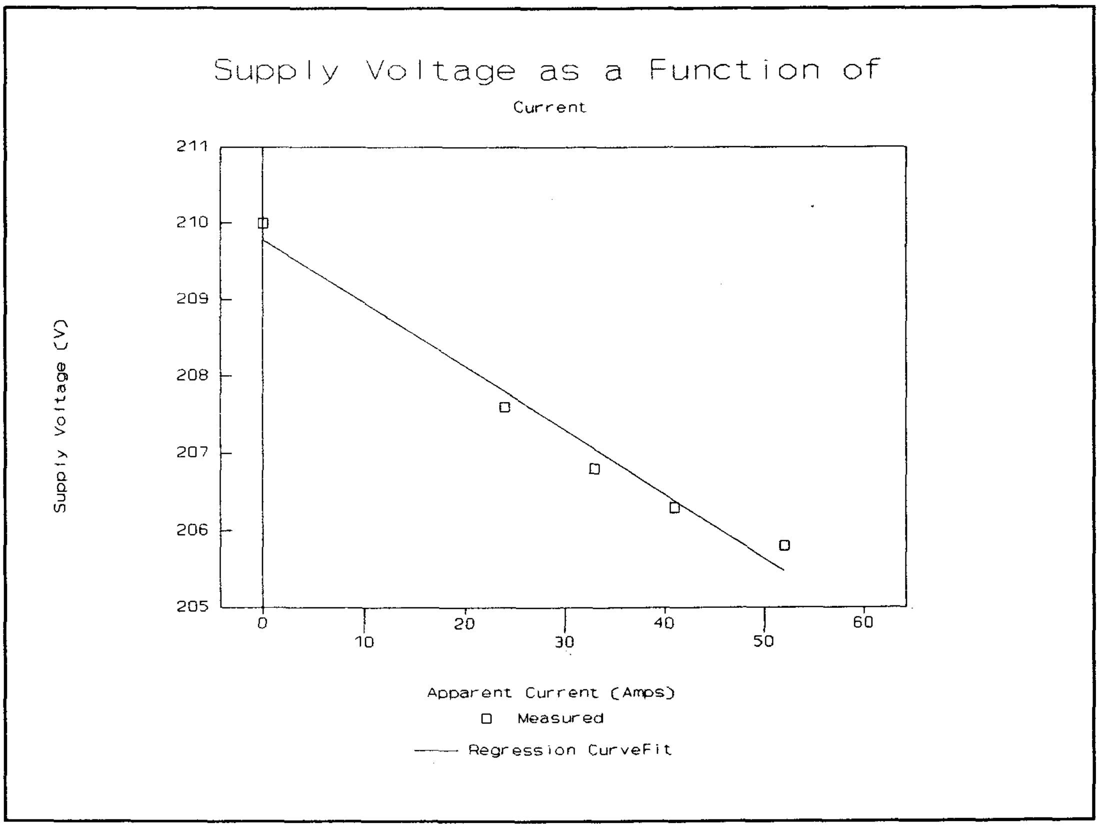

Figure 2: Volts vs. Input Amps.

Standard deviation between modeled and measured behavior was 0.9% of the measured voltage.

Using these two formulas, real power was calculated based solely on measured current.

The formula for power in kilowatts is

$$P = \sqrt{3}\frac{V\,A\,pf}{1000}$$

Mathematical approximation error was normally well within the erratic deviation of measured current, where a typical range during constant load was 7.5%.

Once power was specified as a function of measured current, only three other variables were considered crucial to define compressor behavior: suction pressure, discharge pressure, and compressed air demand.

NOTE: According to the manufacturer, this particular compressor model should require approximately 110% of nameplate power at full air capacity. The measured power was higher, 118%. While the exact cause of the discrepancy was not identified, the variation was attributed to the age of the compressor (built in 1976 and rebuilt in 1989), lower operating voltage (the motor was designed for 230/460V instead of the actual 208V), and the high number of starts required per hour of use in a laboratory environment. Further, the motor has been operated by inexperienced students and until recently, the compressor was regularly operated at 120 psig while only rated to 100 psig. A traditional culprit in high power operation is a clogged oil separator. This possibility was investigated, but the pressure drop across the separator was only 2 psig, within the normal range. The deviation did not adversely affect evaluations as long as it was properly accounted for in the analysis.

b. Analysis. In the past, it has been assumed that there was a linear relationship between discharge pressure and power and between suction pressure and power for compressors. Before proceeding further, these correlations were confirmed. As shown by the graphs in Figures 3 and 4, the two relationships are in fact linear.

The two graphs show slopes of 0.60 %FLP/psi discharge and 0.26 %FLP/%Cap, values close to rules of thumb commonly used in making conservation recommendations (%FLP= Percent Full Load Power, %Cap = Percent Capacity).[3] The relationships are also independent. Thus, these are the only two variables necessary to define power requirements for a given air compressor. The next section addresses the capacity-suction pressure relationship.

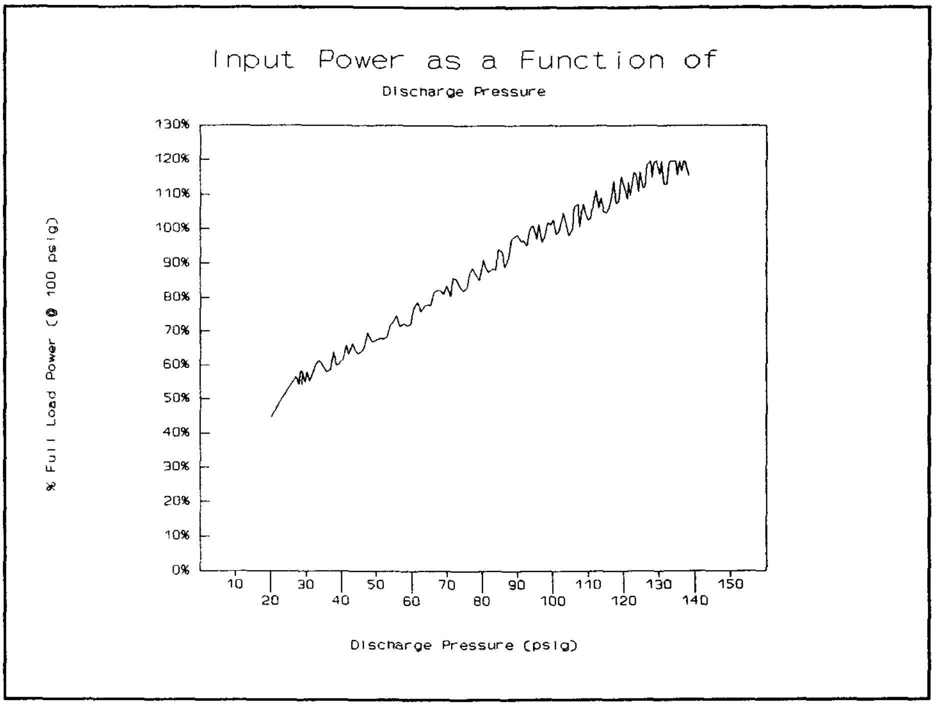

Figure 3: Power vs. Discharge Pressure

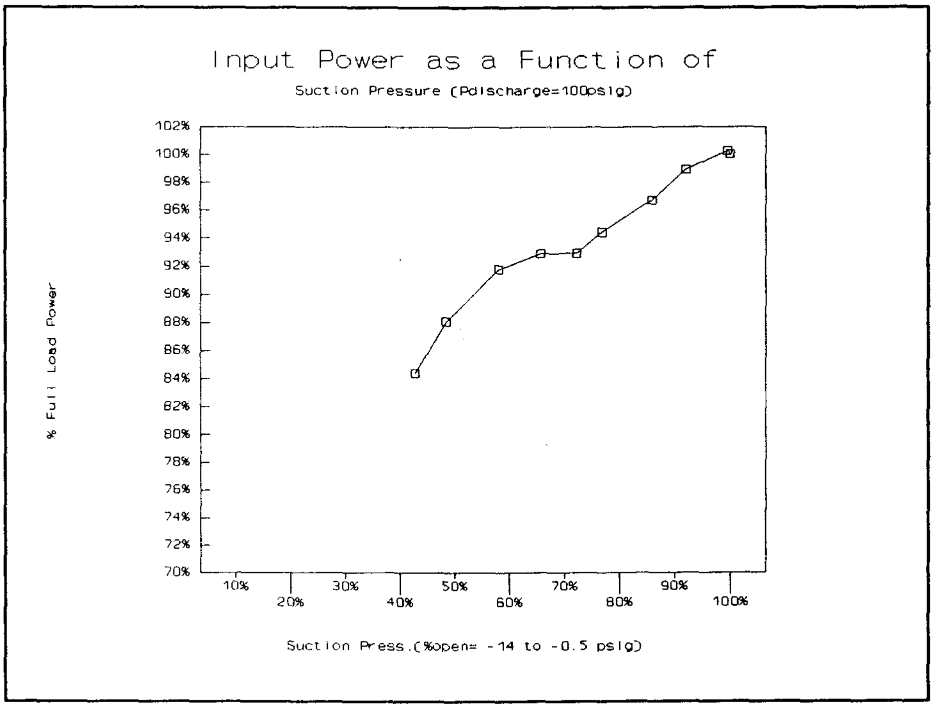

Figure 4: Power vs. Suction Pressure

c. Modeling. Variables such as screw design, manufacturing tolerance, operating altitude, oil separator condition, reheat equipment, lubricant viscosity, or any number of other variables will affect performance, to be sure. They may reduce the compressor's efficiency, increase the required discharge pressure, or decrease the suction pressure. But once a compressor's performance is defined by measuring the suction and discharge pressure under three different load conditions, the power requirements under any other conditions may be accurately estimated. This is easiest when the measured conditions are fully unloaded, fully throttled, and fully loaded, but the demand formula can be extrapolated from any three points as long as both pressures vary at least once. A fourth intermediate point would be useful to verify proper operation.

To help explain this, an example using "typical" performance data may help. Given:

Condition	Power	Suction Pressure	Discharge Pressure
Fully Unloaded	20%	-14 psig	2 psig
Fully throttled	70%	-14 psig	100 psig
Fully loaded	100%	-0.5 psig	100 psig

Power under any conditions (%FLP) would be defined as

$$\%P = [70\% + (100\% - 70\%)(1 - \frac{P_s + 0.5}{-14 + 0.5})] \ x \ [\frac{20\%}{70\%} + (100\% - \frac{20\%}{70\%})(\frac{P_d - 2}{100 - 2})]$$

Figure 5 shows compressor power modeled using the above equation compared with actual measured power during unloading. Unloading is a good time for comparison because both suction pressure and discharge pressure change. In this example, the power formula was

$$\%P = (68\% + 32\% \ (\%p_{suct})) \ x \ (25\% + 75\% \ (\%p_{dis}))$$

For this graph, full load power was normalized to avoid error relative to this variable. $\%p_{dis}$ and $\%p_{suct}$ represent the fraction of the possible pressure ranges.

Discharge pressure range was 0 to 100 psig; suction pressure range was 0 to -13.6 psig.

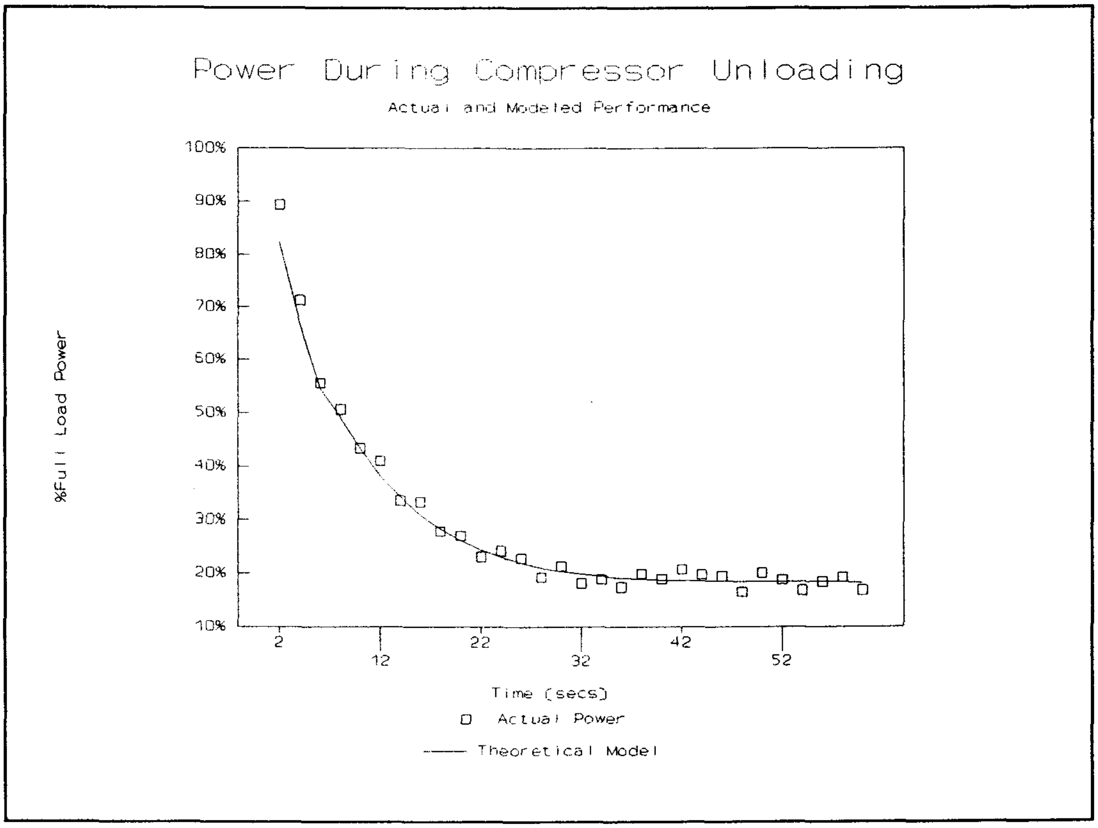

Figure 5: Actual vs. Modeled Power

A simple formula for calculating full load power based on motor nameplate data and known operating pressure only is included as Appendix F.

2. Throttling Modulation

a. Analysis. Air flow was shown to be proportional to inlet pressure in Figure 6. Therefore, suction pressure measurements could be used to measure air flow. In fact, one manufacturer uses a vacuum gauge at the compressor inlet to indicate capacity.

All air compressors that modulate to control capacity, whether they use butterfly, slide, turn, spiral, or poppet valves, employ simple first order proportional control. Consequently, the compressor never responded with any overshoot in response to changing conditions unless the conditions themselves

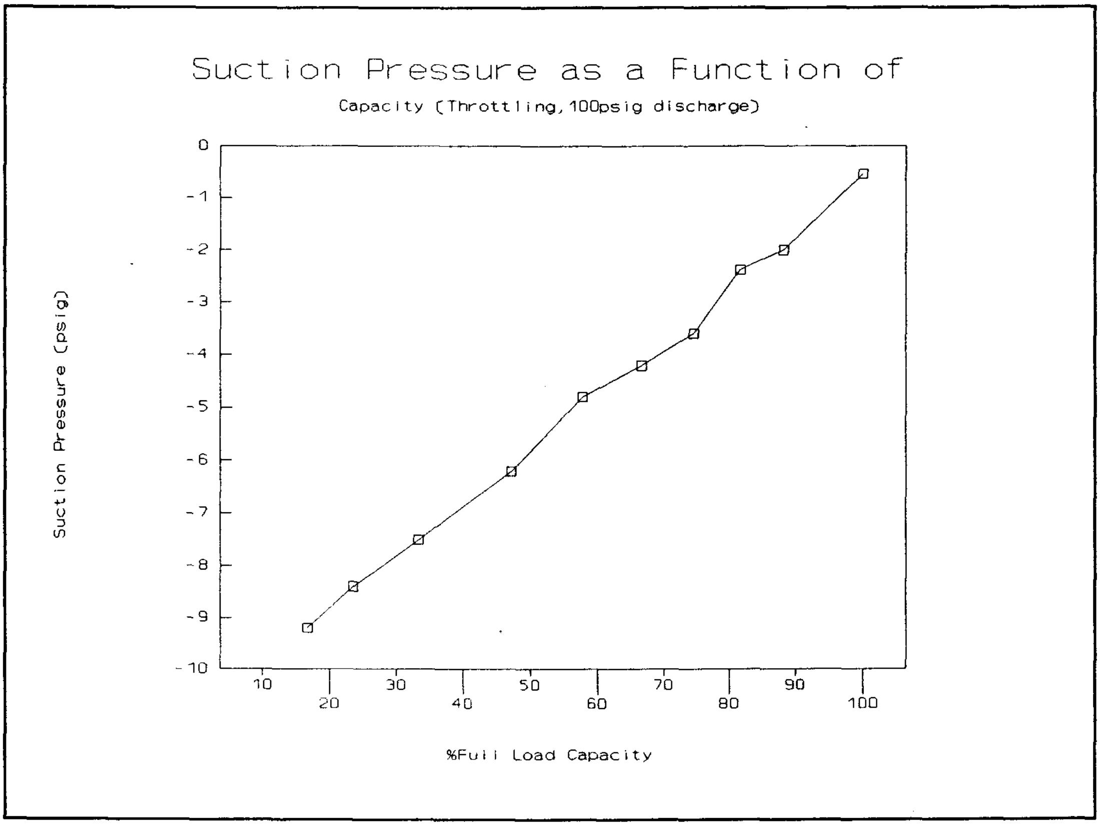

Figure 6: Air Flow vs. Suction Pressure

overshot and receded. Figures 7 and 8 show measured suction and discharge pressures with the compressor operating at 15% capacity. They suggest that the proportional control range was approximately 6 psig. For example, the throttle was 100% open up to 94 psig, 50% open at 97 psig, and 15% open at 99 psig. It would have been 0% open at 100 psig. Linearity was maintained in that discharge pressure was directly proportional to suction pressure in the throttling range.

Note 1: When in modulating only mode, the compressor suction pressure did not drop to its minimum possible pressure (near -14 psig) when demand was reduced to 0% capacity. Though a detrimental factor to unit efficiency under modulating conditions, it did not cause problems when modeling. According to the manufacturer, this anomaly was unique to the smallest compressors and should not be considered representative of typical screw compressor performance.

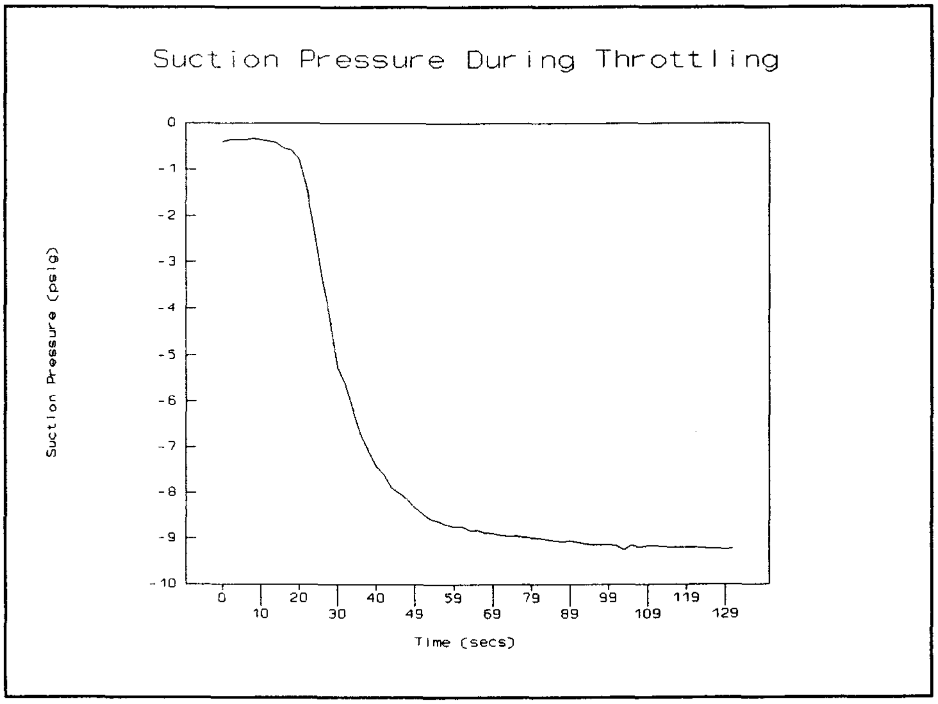

Figure 7: Suction Pressure during Throttling

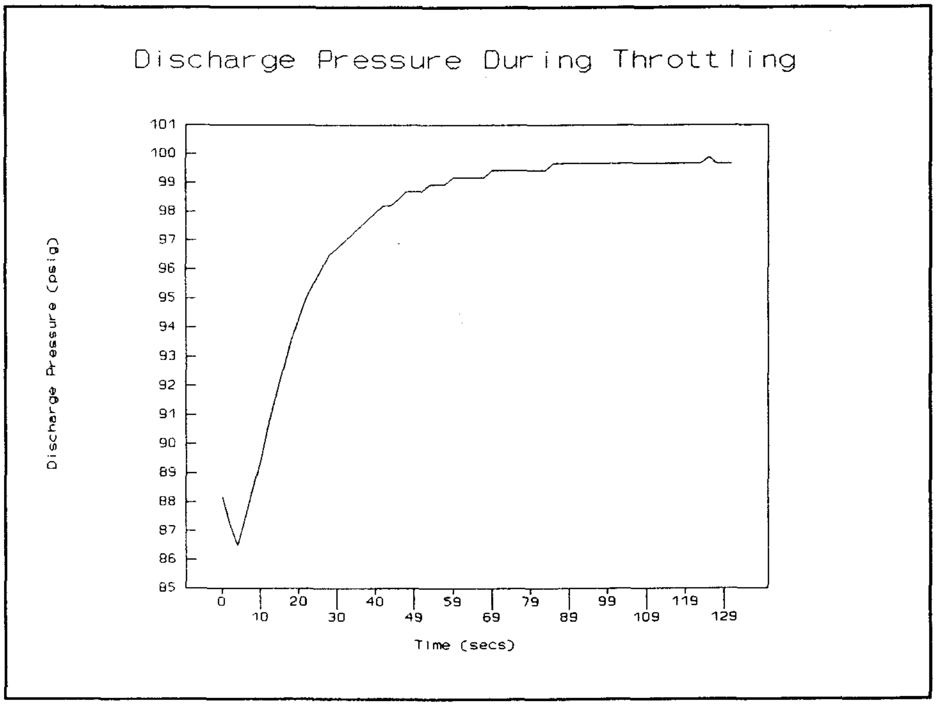

Figure 8: Dis. Press. during Throttling

Other manufacturer's representatives and the author's field experience agree with this statement.

Note 2: According to manufacturer's literature for the compressor being studied, the throttle should range from fully open to fully closed over a 4 psi range. This discrepancy was not a problem, it simply needed recognition during analysis.

b. Modeling. The spreadsheet model simulates modulation for part of the compressor cycle. It uses the proportional control relationship strictly and simply. When the discharge pressure is in the proportional control range, the throttle position for each calculation is base on the discharge pressure from the previous iteration, the proportional control range, and the maximum pressure setting. Actual and modeled behavior are shown in Figures 9 and 10. A single formula was used for all six curves.

Standard deviation between calculated and actual discharge pressure was only 0.23 psi, or 2.8% of the proportional control range. This is impressive because the measured parameter in this example is not the controlled variable itself, suction pressure, but a downstream variable. It indicates a pilot valve that is working well as much as it indicates good mathematical modeling.

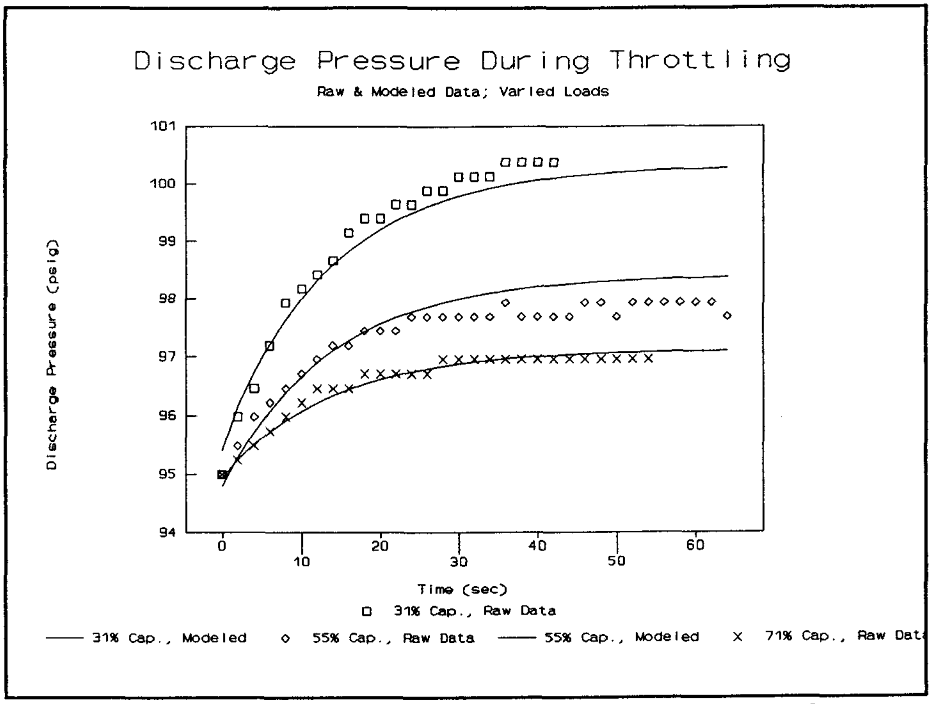

Figure 9: Raw and Modeled Data During Throttling, 1 of 2

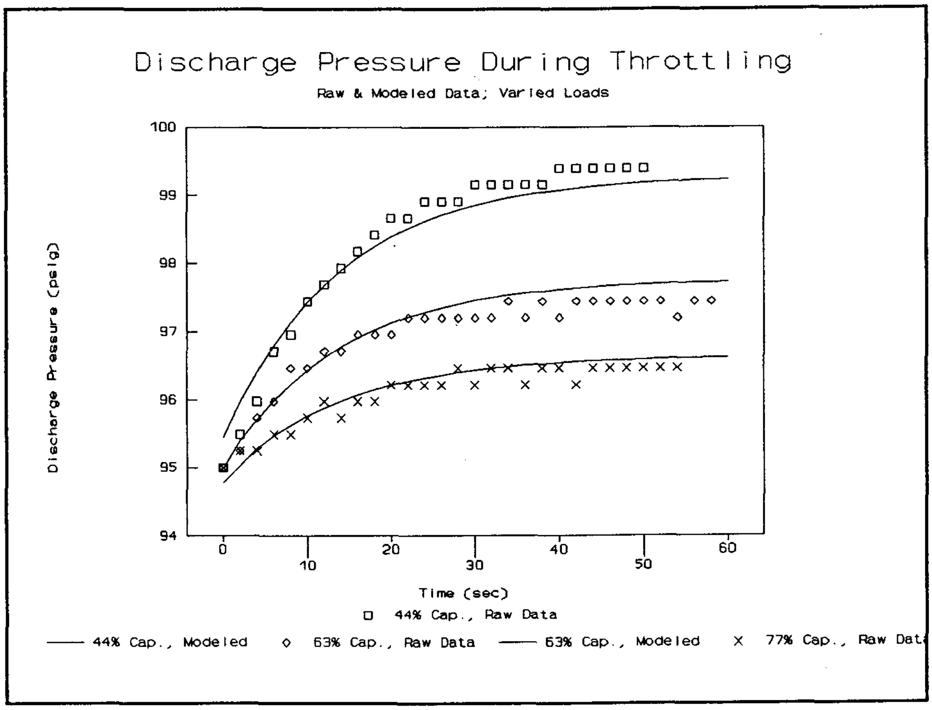

Figure 10: Raw and Modeled Data During Throttling, 2 of 2

3. **Cycling**

a. **Summary** - Compressor behavior during cycling was much more interesting than when the unloading controls were disabled and the compressor simply modulated. (Perhaps this hints at why unloading controls are often disabled). An operating cycle can be broken down into five distinct conditions: fully loaded, throttling, unloading, idling/unloaded, and reloading. Each element was analyzed separately.

b. **Fully loaded** - By definition there is no control action while the compressor is fully loaded. The throttle is wide open and compressor power is solely a function of gradually increasing discharge pressure.

c. **Throttling** - Throttling is only slightly more complicated. Throttling begins as soon as the discharge pressure increases past the beginning of the proportional control region. Behavior is identical to modulation-only operation as if the desired pressure has just been increased. The rate of change of discharge pressure depends on throttle position (which also affects inlet pressure). It will continue to increase as it has while fully loaded, but the pressure rise will slow as the throttle closes. The spreadsheet models the throttle position as described in the Throttling Modulation section.

d. **Unloading.** Unloading begins as soon as the throttle closes past the set minimum throttling position and the vacuum switch opens (refer to Appendix B, if necessary). Unloading is unique compared to the other operating modes in that suction and discharge pressure are dynamic yet modeled independently. This is appropriate because the hardware itself is separate.

1) **Unloading, discharge pressure** - After the vacuum switch opens, it takes time for the air between the compressor discharge and the main check valve to vent out. This includes air in the oil separator. Since the change in pressure depends on actual pressure and slows as the pressure drops, pressure can be modeled as an exponential decay. However, testing indicated that exponential decay was not accurate during initial unloading. Further investigation revealed that the exhaust flow rate was limited until the discharge pressure

dropped below a certain pressure, probably due to small blowdown control line size. An example of actual and modeled performance is graphed in Figure 11.

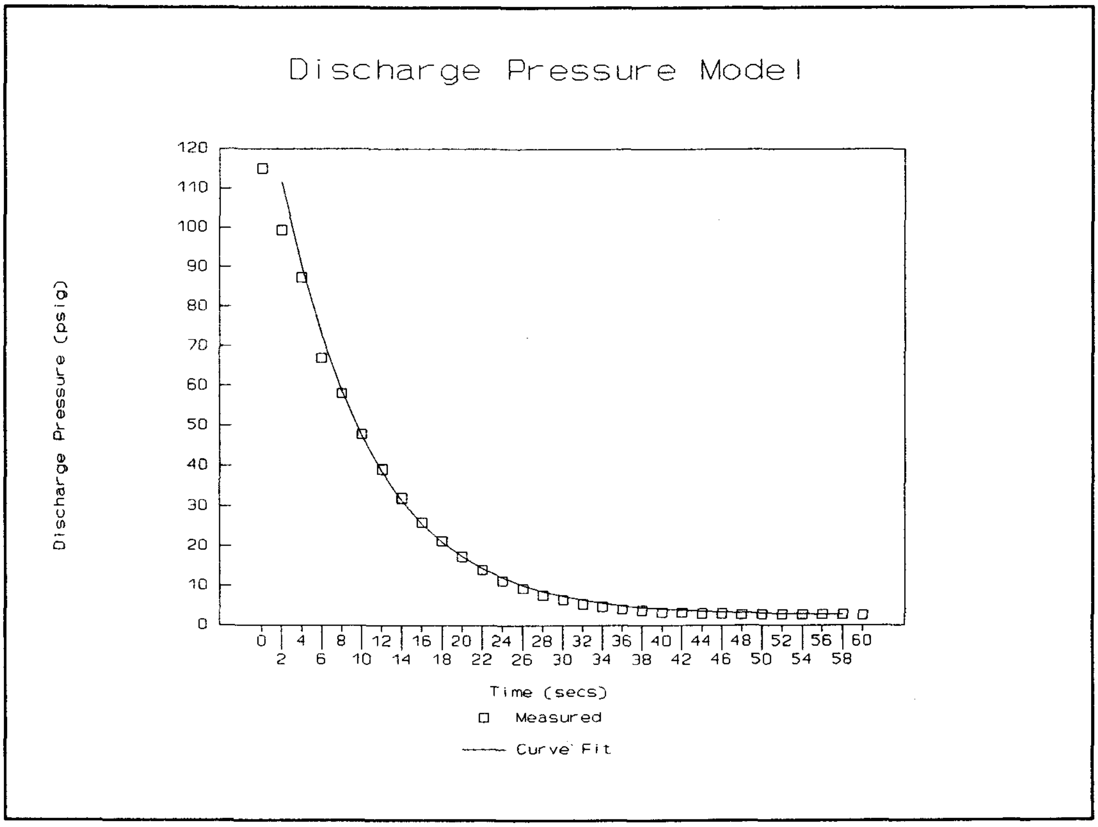

Figure 11: Discharge Pressure Model

As can be seen by the departure of measured data from the exponential curve in the graph, pressure decays more linearly than exponentially above 90 psig (p_{max}), at about 8 psi/sec. Below 90 psig, the decay followed the exponential formula:

$$p_{disch} = p_{\min} + (p_{\max} - p_{\min}) e^{-\frac{t}{\tau_d}}$$

where,

p_{max} = maximum pressure for exponential decay

τ_d = discharge time constant.

p_{min} = minimum discharge pressure

The minimum desired pressure is a separate issue and is more likely to vary among manufacturers than with other variables. Some compressor designs maintain a minimum oil separator sump pressure of approximately 30 psig and use the pressure differential between the sump and the compressor internal oil injection point to circulate oil through the screws when compressed air is not being produced. Other designs use an oil pump that allows the sump pressure to drop down to less than 5 psig. The compressor used for this project is of the latter variety. For the project compressor, $\tau_d = 9.0$ sec.

Overall, the standard deviation of error between measured data and the exponential curve over the last 56 seconds shown in the graph was 1.1 psi. Error between the data and the linear approximation in during initial unloading stages was equally small.

2) **Unloading, Suction Pressure** - On the suction side of the compressor, the throttling valve closes very quickly once the vacuum switch opens. However, there is an approximately exponential decay of the suction pressure after the throttle closes. A vacuum gradually forms as the compressor draws the air from the suction intake. The formula is of the same form as the discharge pressure unloading. The example shown in Figure 12 has an unload point of about 50% capacity and a suction pressure time constant $\tau_s = 5.5$ seconds for the model. The standard deviation between actual and approximated throttle position was 3.4%.

e. **Unloaded or Idling.** This refers to the condition when the throttle is fully closed and the discharge pressure has reached the minimum sump pressure. Idling continues until the minimum system pressure switch is triggered.

f. **Reloading.** For throttle behavior, this is again a case of air gradually being released to atmospheric pressure, albeit control line air and not process air. Thus, an exponential fit might again be expected. However, the air supplied to hold the slide valve closed when unloaded must pass through a shuttle, backwards through an adjustable pressure reducer, and out of a calibrated orifice. Some of the air will be at system pressure; it is exhausted first. The rest of the

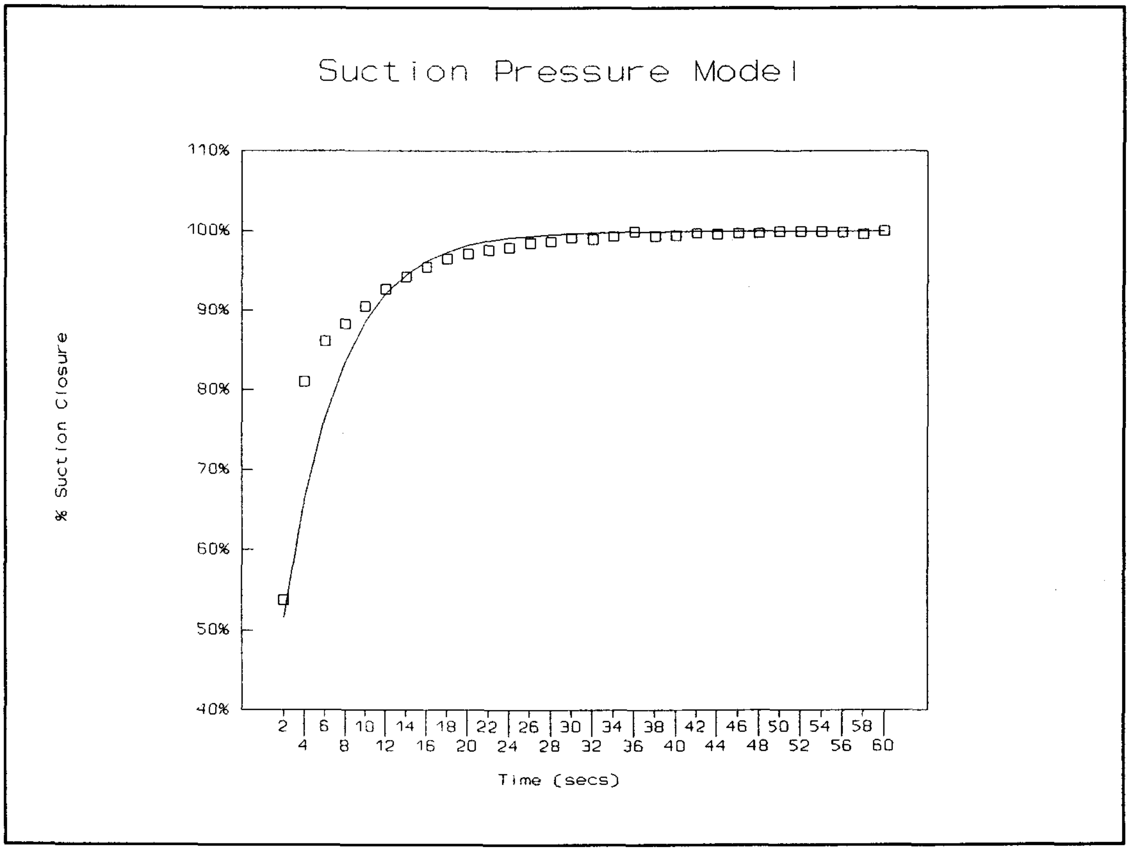

Figure 12: Suction Pressure Model

air will be at a reduced pressure (3:1 reduction is normal) when reloading begins. What all of this means is that it takes time for the control pressure to drop initially, and the change in pressure felt by the piston that controls the throttling valve starts out slow before joining a more exponential-type behavior. Towards the end of the reloading cycle, the control line pressure is low and therefore its rate of change slows correspondingly.

An arctangent curve reflects the measured suction pressure behavior well and is shown in Figure 13. Standard deviation was low, 0.3 psi, for the case used to define the arctangent curve. Since the arctangent represents an empirical curve fit of measured data, the mathematical formula should and does model the suction behavior well. Reloading time varies with control line size, pressure reducer setting, piston stroke and diameter, and by orifice hole size. For the model, reloading time is variable. The values on the time axis shown in Figure 14

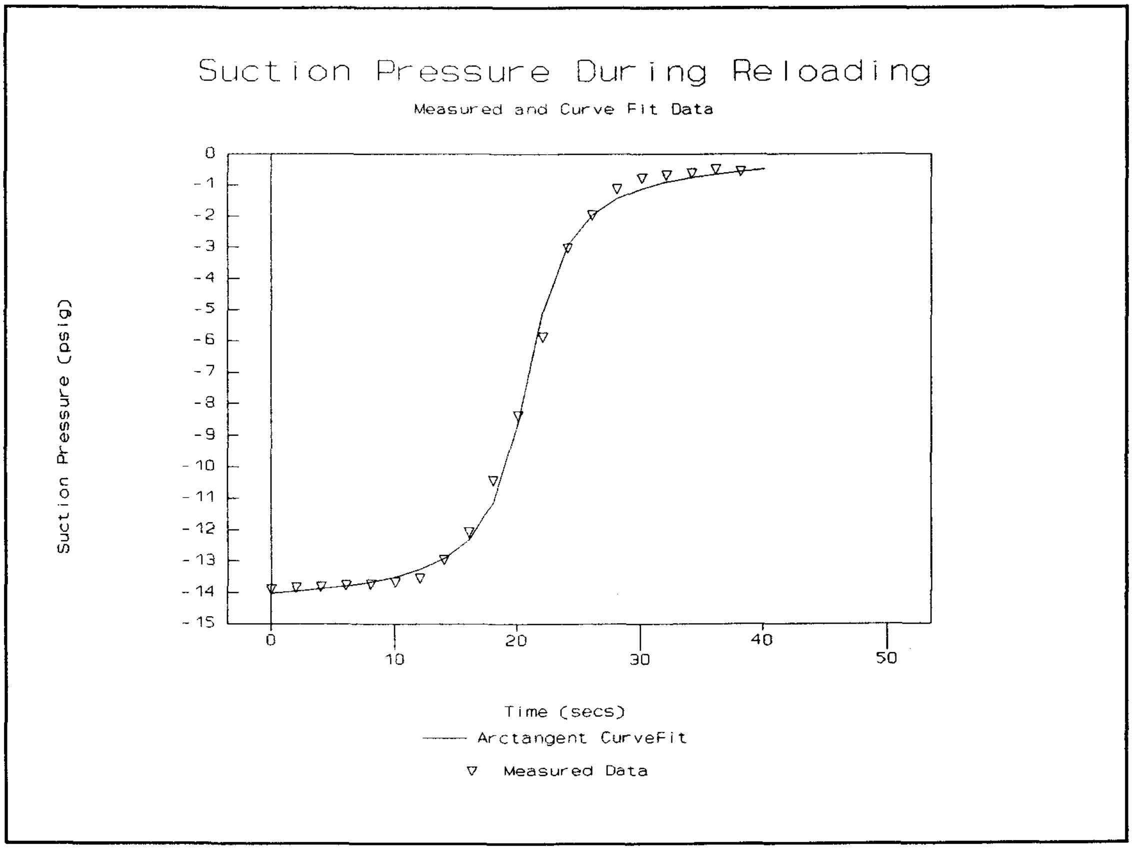

Figure 13: Reloading Suction Behavior

are simply increased or decreased to meet variable specifications. Other variables may also be adjusted, but would not be part of standard operating procedure. This is hardly "elegant" programming but it works quite well.

Note: This model simulates behavior for compressors that use a sliding throttle. Butterfly throttles may behave less linearly with respect to the piston travel-suction pressure relationship.

Discharge pressure during reloading is simply a function of amount of air delivered by the compressor. First the compressor must fill the oil separator with compressed air until system pressure is reached. Then, the volume being compressed becomes the whole receiver and piping network.

4. Receiver. Pressure in the receiver depends on the air that leaves due to plant use and air supplied by the compressor. The volume of air injected into the receiver from the compressor (V_c) over a small time interval ▲t will be

$$V_c = Q_c \Delta t$$

where Q_c is the flow of air delivered by the compressor (compressed cfm, not scfm). Similarly, the volume of air released from the receiver and used by plant equipment and leaks (V_p) will be:

$$V_p = Q_p \Delta t$$

If the addition and removal of air from the receiver is considered an ideal isothermal process, the receiver conditions at the end of the time interval can be approximated by the following relationship.

$$p_2 V_r = p_1 V_r - p_1 V_p + p_1 V_c$$

where

p_1 = receiver pressure at time 1

p_2 = receiver pressure at time 2

V_r = receiver volume

This isothermal relationship should not be confused with the polytropic compression relationship. Rearranging the equation to solve for the receiver pressure at time 2 and substituting,

$$p_2 = p_1 + \frac{\Delta t}{V_r}(Q_c - Q_p)$$

There will be some error due to the assumption that air supplied and released are at pressure p_1 during the time interval, but the error will be small if the change in pressure during the time interval is small.

This equation is the heart of the model. All other calculations provide the variables so that p_2 can be calculated from this equation.

D. SPREADSHEET SECTION

A spreadsheet-based empirical model was created to aid in the analysis of screw compressors that unload. Based on variable inputs, the spreadsheet can model on-off, load-unload, or throttling-unload (low-unload) control strategies. The unload point in low-unload control is also variable. The model does not simulate other types of modulation such as turn valves. The program goal is to simulate a single operating cycle of an air compressor at a given load. Average power for the cycle is the ultimate calculation, but other intermediate values such as cycle time and pressure during the cycle are also evaluated. Detailed variable and formula descriptions can be found in Appendix G.

E. Overall Compressor Cycle Performance

In addition to calculating average power for a compressor cycle under entered conditions, the model produces graphs that show modeled behavior over the course of a single cycle. An example of the graphs is shown in Figure 14.

1. Modeling. The true test of spreadsheet performance is to enter known operating conditions for all of the variables and compare the results with measured data. Based on a measured full load power of 15.1 kW and 45% capacity, the spreadsheet calculated the average power over a single cycle to be 9.91 kW. Measured data averaged 9.73 kW. Error between modeled and measured data was 1.8%.

While a 1.8% error is certainly laudable, it should not be construed as spreadsheet accuracy in normal use. First, the combined error of measuring devices, iterative approximations, and curve fits exceeded 2%. Figure 15 shows a comparison of measured and modeled data. Power measurements, for example, were erratic, and the oil separator built up pressure faster than was modeled. Also, each variable used for input data had been meticulously analyzed for this thesis. In most realistic situations, many of these variables (such as reloading time or unloading time constant) will be estimated.

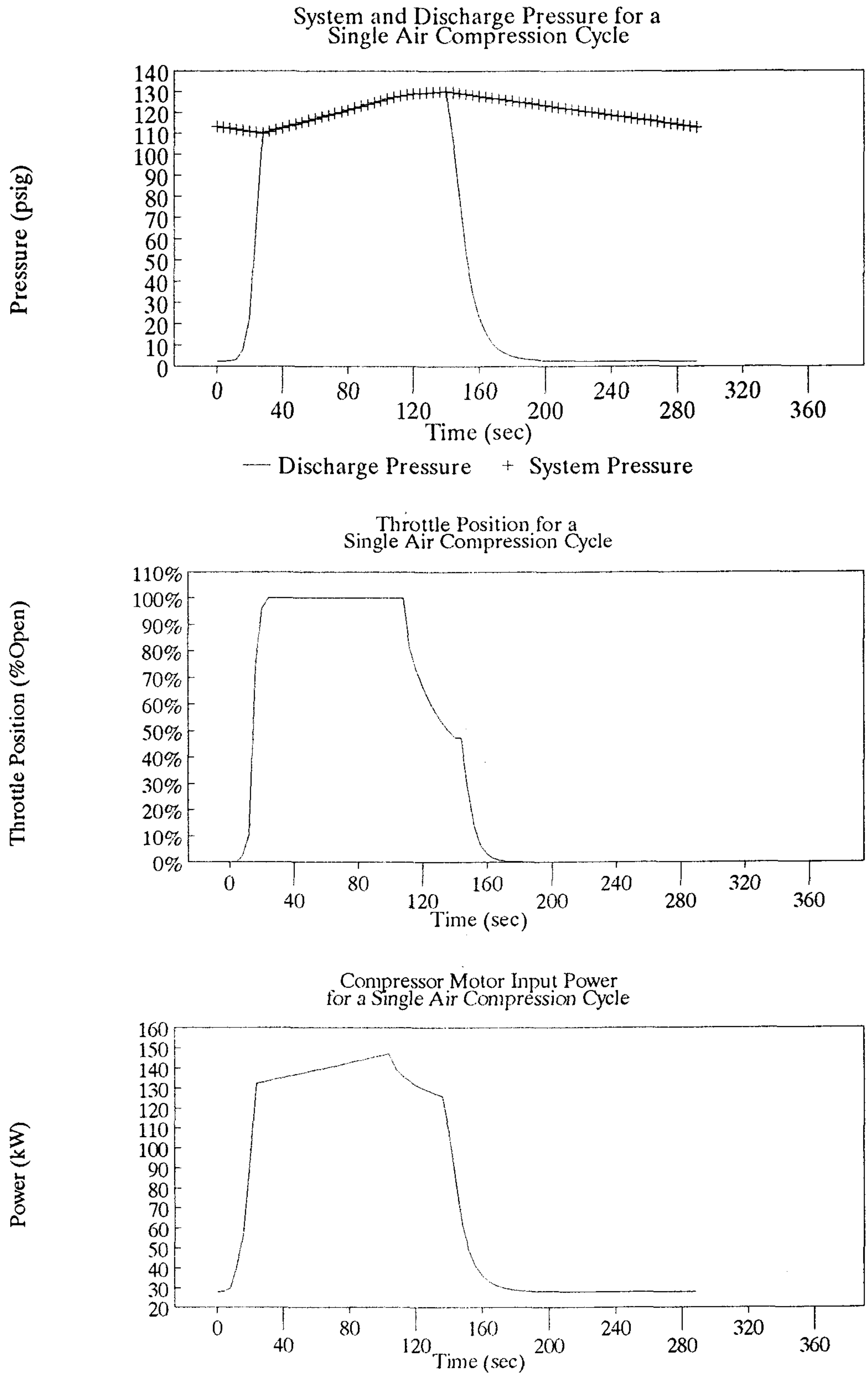

Figure 14: Spreadsheet Modeled Single Cycle Performance

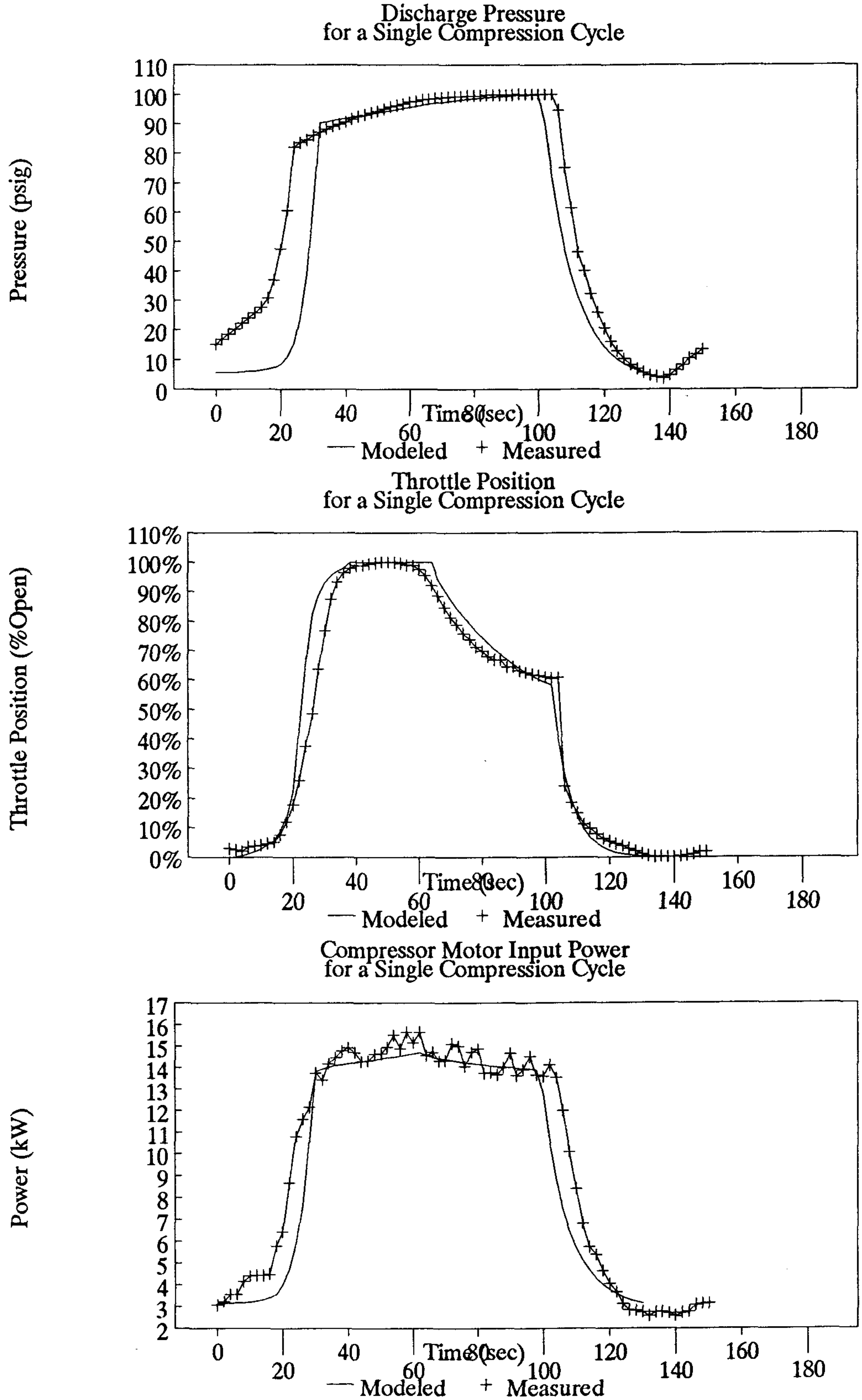

Figure 15: Modeled and Measured Power for One Complete Cycle

Nonetheless, it was rewarding to find that the model does in fact calculate accurately. The spreadsheet will be valuable and accurate in comparative analysis, where first one condition is evaluated and then another.

2. Evaluation. Several interesting patterns evolved from using the spreadsheet model repeatedly. Traditionally, part load power when cycling and using low-unload control has been modeled as shown in the controls guidebook (Appendix A). Specifically, the average power during cycling (between 0% and UP% capacity) has been shown as a straight line between the minimum power and the power when throttling at the unload point. All manufacturers researched for this thesis use this form to approximate average power.

In contrast, the model indicates that part load efficiency in this region is substantially better than is normally assumed when either the unload point is low or when receiving capacity is large. Figure 16 plots average power calculated by the model for various loads. Conditions for the curve are 757 scfm rated, 200 ft^3 of receiver capacity, and a 20 psi range between maximum and minimum pressure. The curve suggests that behavior approaches that of load-unload control faster than traditionally assumed. The trend is accelerated as receiver size increases and cycling losses are less significant. Figure 17 shows an exaggerated example, where the receiver size was increased to 4000 ft^3 and other conditions were unchanged. Average power may be lower by up to 10% of full load power.

One reference was found that showed similar part load behavior with low-unload control, but no analysis of this characteristic was offered.[4]

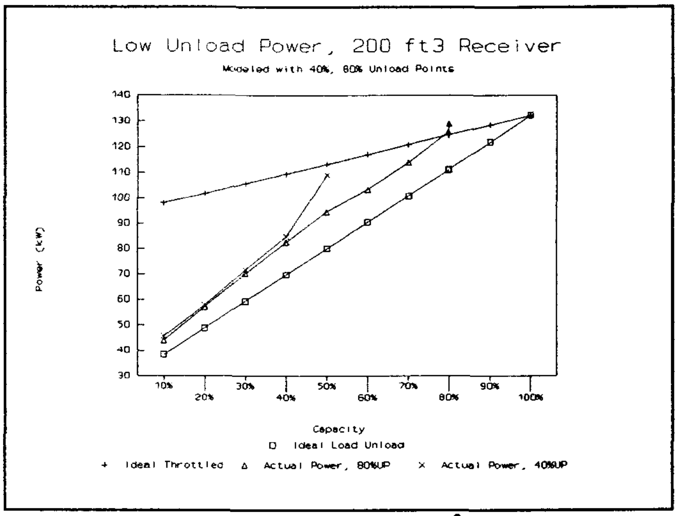

Figure 16: Actual Power, 200 ft^3 Receiver

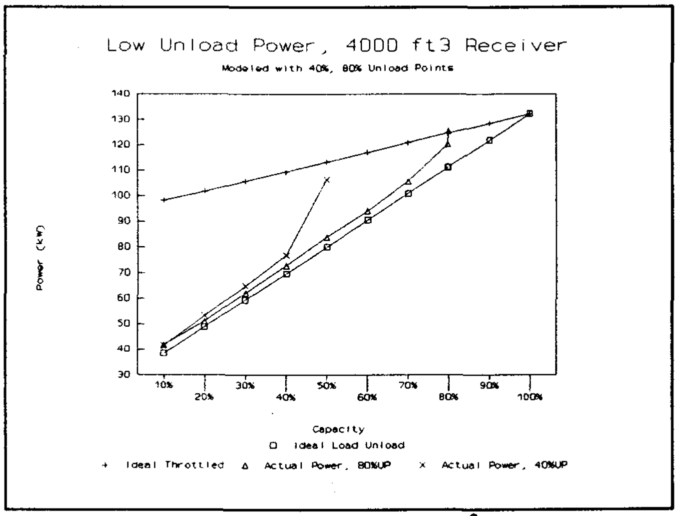

Figure 17: Actual Power, 4000 ft^3 Receiver

The relative behavior of the power in these figures suggests another important trend that was further developed. For analytical purposes, cycling losses are commonly assumed to be relatively small, and thus neglected. The model provided an easy way to vary operating conditions and monitor the effect on these losses. One of the model calculations is ideal load-unload power. In order to evaluate the effect of increased cycle time on losses, the configured for load-unload control and then executed at a variety of receiver sizes. Plant air demand and all other variable were left constant. Ideal and actual power were compared, and the difference between the two was expressed as a percent of full load power. Figure 18 shows the effect of increased cycle time on unloading losses.

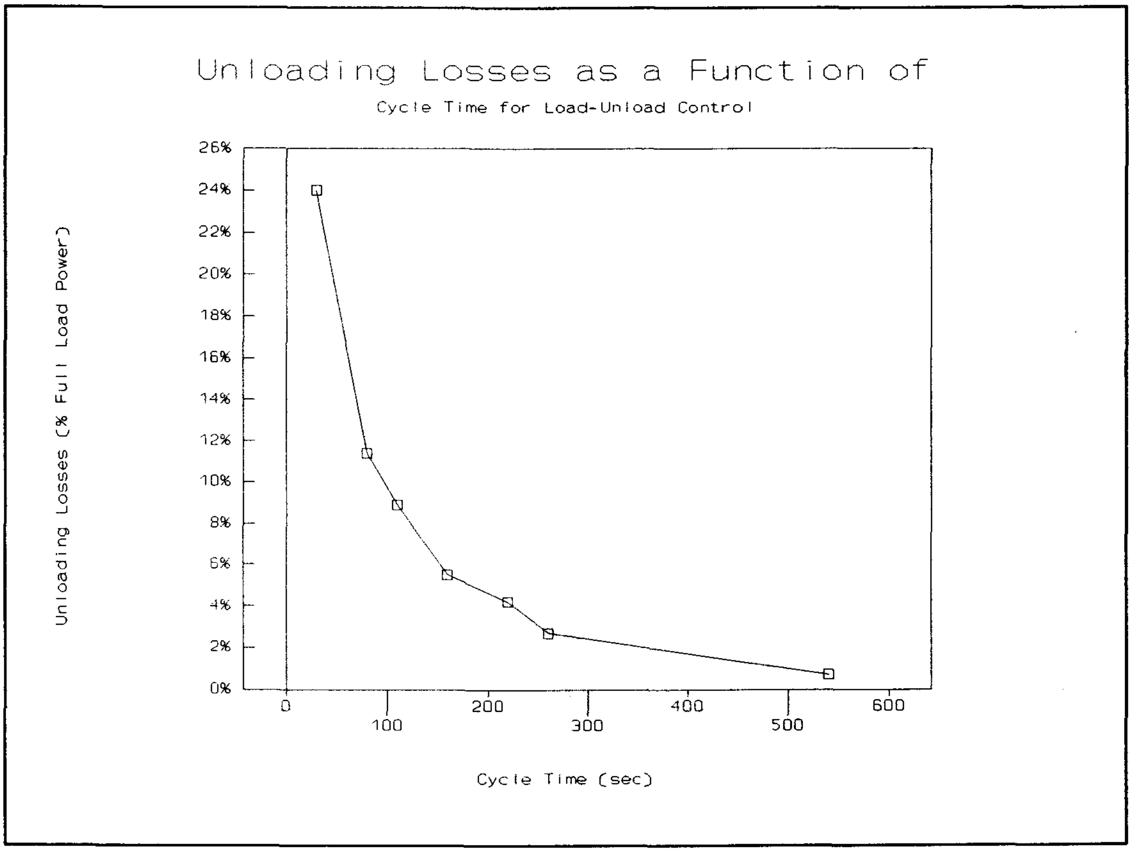

Figure 18: Unloading Losses

IV. SMART CONTROLLER

A. Introduction

In addition to writing a guidebook and developing a compressor model, the third major thesis goal was to improve upon the state of the art in part load control of rotary screw air compressors by designing a microprocessor-based controller. Several manufacturers currently offer computerized controllers. However, none of them are "smart." All operating parameters are fixed. The smart controller designed here does not change either the mechanical compression process or the control actuation. Therefore, energy savings potential for this design does not provide a significant improvement in operating efficiency over existing control designs. The smart controller simply optimizes the compromise between load-unload and throttling modes. However, this is not to say that savings are always small. A case study showed savings of 4% over low-unload control with a fixed 70% unload point and 12% over low-unload control with a fixed 40% unload point.

Fundamentally, the smart controller's strength is that its smart logic may make the option of an unloading controller possible for applications that would normally be restricted to modulation-only control. When compared to throttling modulation control in the case study, average savings were 32%.

B. Existing Control Technology

At least five manufacturers now offer some level of electronic control for individual compressor control. Ingersoll-Rand and Atlas-Copco have more advanced systems while Gardner-Denver, LeRoi, and Sullair electronics are oriented to actuation, monitoring, and display. Though electronic control can be found on all of these compressors, none of them are actually "smart." That is, none of them gain experience based on monitored compressor operation and adjust operating parameters dynamically. Nor do they attempt to measure or calculate energy use.

Several manufacturers incorporate timers that will turn the compressor off after a set period of idling. This is not necessarily clever design for three reasons. First, a plant with small air storage capacity and some leaks will never realize savings from this feature. Second, if the compressor is going to shut off during the unloaded part of the cycle, then it wastes energy to idle for five or ten minutes before doing so. Third and perhaps most significantly, these designs do not help the timer make decisions. For example, if idle time is 5 minutes and 5 seconds and the timer is set at 5 minutes, the timer will tell the motor to shut off and it will have to restart again five seconds after shutdown. The disadvantages associated with on-off control are added to operation while savings are reduced. Atlas-Copco's Electronikon is intelligent enough to avoid this problem. The controller chooses on-off or load-unload operation based on the required number of starts per hour.

Ingersoll-Rand offers perhaps the most versatile microprocessor-based controller commercially available on the market at this point. Their controller allows the compressor to throttle in the higher demand region, but when demand is low the compressor cycles in load-unload mode, rather than in low-unload mode. However, this controller does not make the next step, of optimization.

C. Design and Development

1. Program Objective: A "smart" controller was designed to switch a compressor between throttling only and load-unload operation depending on average energy use, cycle time, unloaded time, and instantaneous required capacity. The maximum capacity for cycling is adjusted and optimized continuously.

Though the written program (included in Appendix G) is several hundred lines long, there are only a few basic rules applied to guide decision-making. The rules are as follows:

1) In throttling mode. The controls will continue throttling until three criteria are met. If met, the controller will switch to load-unload cycling. The

criteria are:

a) Demand drops below a previously determined minimum capacity for throttling, or Unload Point (UP).

b) Demand must remain below this minimum continuously for at least 60 seconds (all number used in this section are variables that can be adjusted).

c) Average demand for the 60 second period must be more than 2% below the UP as well.

2) Cycling mode. The compressor will stay in the more efficient cycling mode as long as three different criteria are met. If any of the three criteria are not met, the compressor returns to throttling mode. The criteria are:

a) The average power per cycle must remain below the power that the compressor would require if throttling at the UP.

b) The unloaded time must exceed the minimum acceptable unloaded time.

c) The cycle time must exceed the minimum acceptable cycle time.

3) Optimization. One other characteristic is evaluated: UP. This variable is adjusted and optimized dynamically during compressor operation. UP adjustment is based on the ability of the compressor system to meet the minimum unloaded time, minimum cycle time, and average power criteria. For example, if the current UP is set at 80%, the compressor idles at 75% capacity before unloading, and all criteria are significantly exceeded, the UP may be increased to 85%. Ultimately, the optimization is primarily a function of receiver size and system pressure range, which is unknown as far as the controller is concerned. The detailed logic structure for this decision can be found in the program flowchart and commented program listing.

2. Hardware and Software. Hardware and software for the controller are described in detail in Appendix G. A commercially available compact single board computer/controller provided the microprocessor, EPROM, RAM, ROM, and parallel and serial interface. The input/output board used to send and receive data was constructed specifically for this project. Two pressure sensors were used as input devices for suction and discharge pressure. A single relay and three LEDs were used for output.

D. Results

The controller was successfully installed and programmed. By hooking up the data acquisition hardware simultaneously (and in parallel) with the smart controller, performance was recorded and compared with data taken under throttling and low-unload operating conditions. In all cases, compression was more efficient using the smart controller, because it was able to initiate load-unload cycling more often. The smart controller also was successful in determining when throttling-only operation was most appropriate.

One of the most significant conclusions made by the smart controller was that there were few conditions under which modulation was more efficient than cycling. When cycling, the minimum cycle time and minimum unloaded time were more often the parameters that triggered the return to throttling. Whenever the power calculations requested a return to throttling, it was because the load increased well above the UP rather than because throttling was more efficient than cycling at the same capacity.

In addition to saving energy, the controller performed well in avoiding improper unloading. For example, suppose that 70% capacity is the ideal unload point for a certain facility. If average demand for a period is 95% capacity, it would be desirable if the compressor never unloaded. However, if the duty cycle was such that demand temporarily dropped to 60% for thirty seconds and normal low-unload control was used, the compressor throttling valve would close. This is because the controller has no means of measuring average air demand instead of instantaneous demand, though having a large receiver will help smooth out some irregularities. The smart controller measures both average and maximum demand over a time period to ensure that the controller reacts to average conditions rather than transient conditions.

The value of properly avoided unloading is worth emphasizing because it addresses one of the main reasons low-unload controls are not purchased and why they are so frequently disabled once installed.

The data suggests that any facility with large storage capacity and a tolerance for pressure variation should use load-unload control. There is no reason not to take advantage of the efficiency offered by load-unload controls. Smart control is normally the best option when load-unload control is not possible, because it offers maximum savings without sacrificing performance. Each facility is unique, and the smart controller makes it easy to determine the optimum operating mode for maximum energy savings without sacrificing performance.

Smart control will not always be applicable. A modulating controller is the only acceptable alternative if pressure variation tolerance is minimal. Also, modulation will best meet load requirements if little storage capacity is available and air demand is erratic. There is no benefit to investing in smart control when it modulation is always the best operating mode.

E. Case Study

In order to compare energy use and behavior of throttling, low-unload, and smart controls, a 24 hour day was simulated. Because air flow variations were manually controlled with a globe valve, each 24 hour day was compressed to about 4 hours for the case study simulation. Figure 19 shows the sample plant demand. Allowances were made for breaks, shift changes, leaks, and gradual variation in equipment demand. Average demand for the entire 24 hour cycle was 50%. The time periods between 12:50 pm and 2:10 pm and between 7:50 pm and 9:30 pm were used to simulate erratic loading. Average power for the 24 hour day for each control strategy is summarized below for a 15 hp Quincy screw compressor.

Control Strategy	%Time Unloaded	Average Power
Throttling Control	0.0%	12.40 kW
Low-unload, 40% Unload Point	34.5%	9.62
Low-unload, 70% Unload Point	36.5%	8.83
Smart Control	53.0%	8.50

Measured power is shown in Figures 20 through 23. The power profiles show graphically how the compressor was able to unload and save energy more often with the smart controller and higher unload point. Figure 24 also shows how the 70% low-unload controller had trouble dealing with the erratic high load profile during first shift. The compressor unloaded while average demand was still high. Consequently, the compressor took a long time to raise the discharge pressure back up to the desired level. At 9:15 am, 1:15 pm, and 3:45 pm for the smart controller, data shows the compressor measuring average conditions when throttling, before switching to load-unload mode.

Note: One interesting control strategy was not possible to measure. The vacuum switch on the compressor used for testing was unable to open and close properly between 0 and -3 psig. This meant that a low-unload control strategy with a 90% or so unload point was not simulated. Because this condition was not possible to simulate with hardware, the spreadsheet model was used. After normalizing the data, the projected average power for low-unload operation over the 24 hour period was 8.94 kW. This value is statistically no different than the power measured by the smart controller's case study performance. The smart controller ultimately settled on about a 82% maximum capacity for cycling. The 90% low-unload would be expected to use more energy than the smart controller when cycling because the smart controller does not modulate a significant amount.

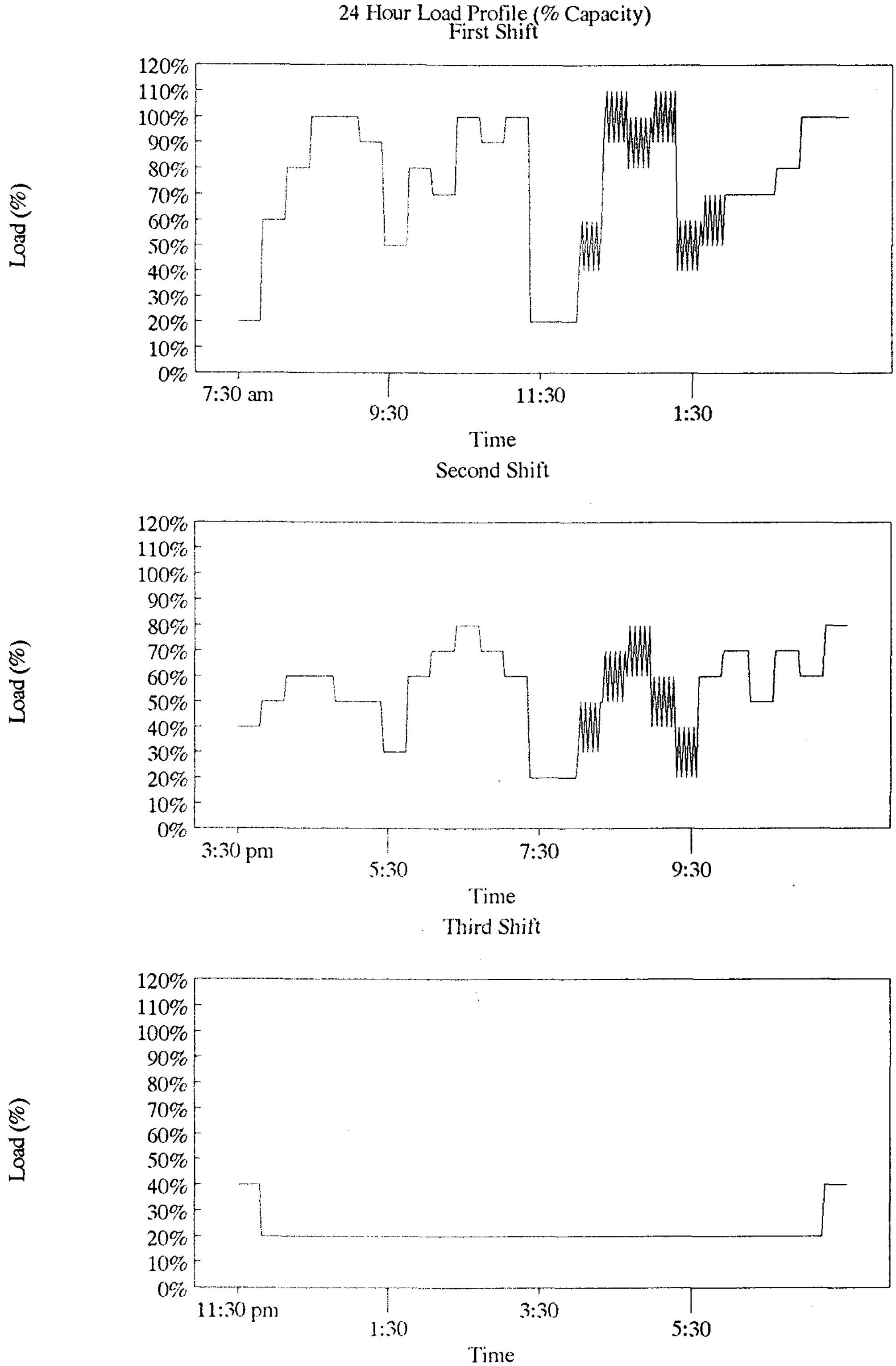

Figure 19: Case Study, Load Profile

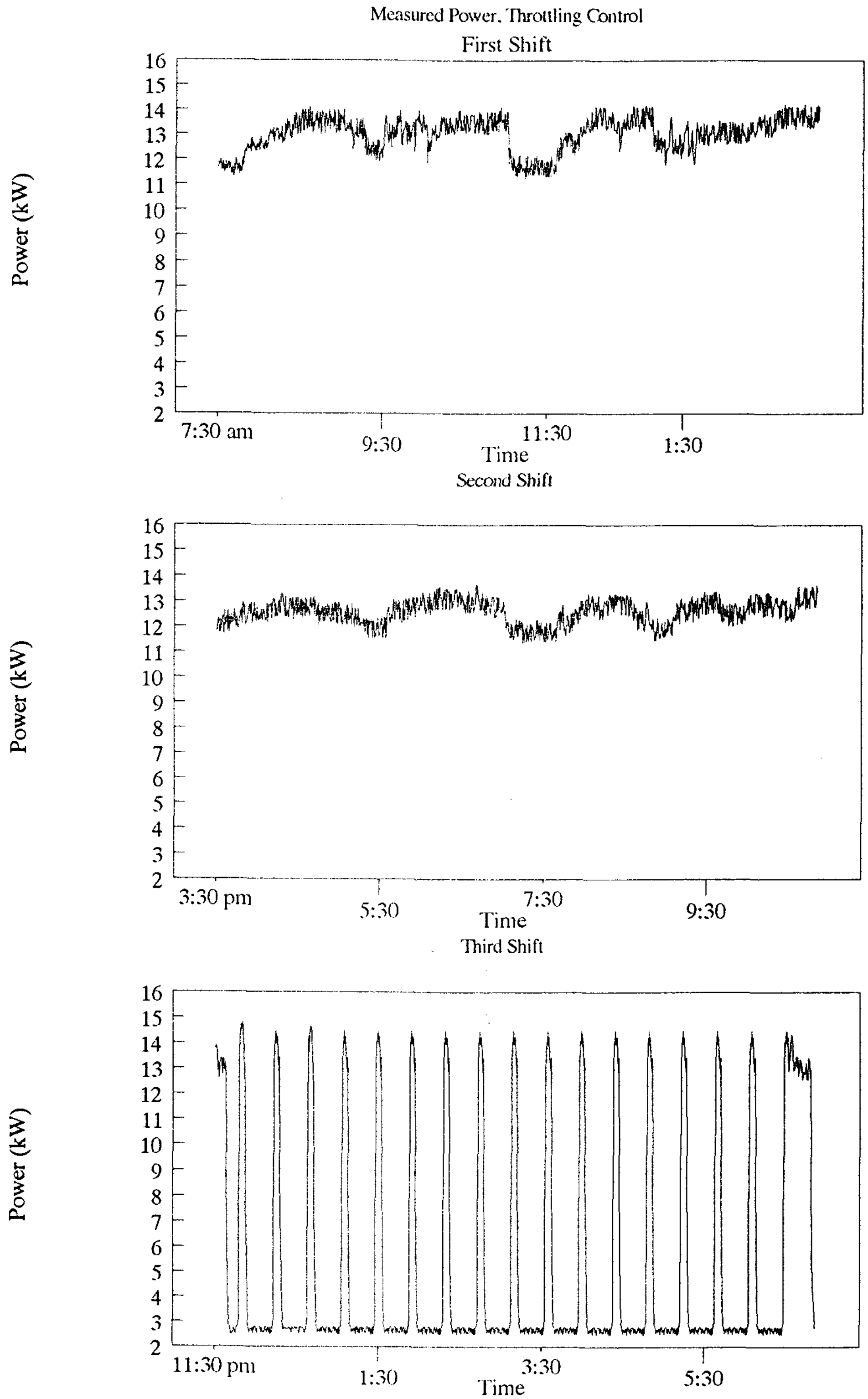

Figure 20: Case Study, Throttling Power

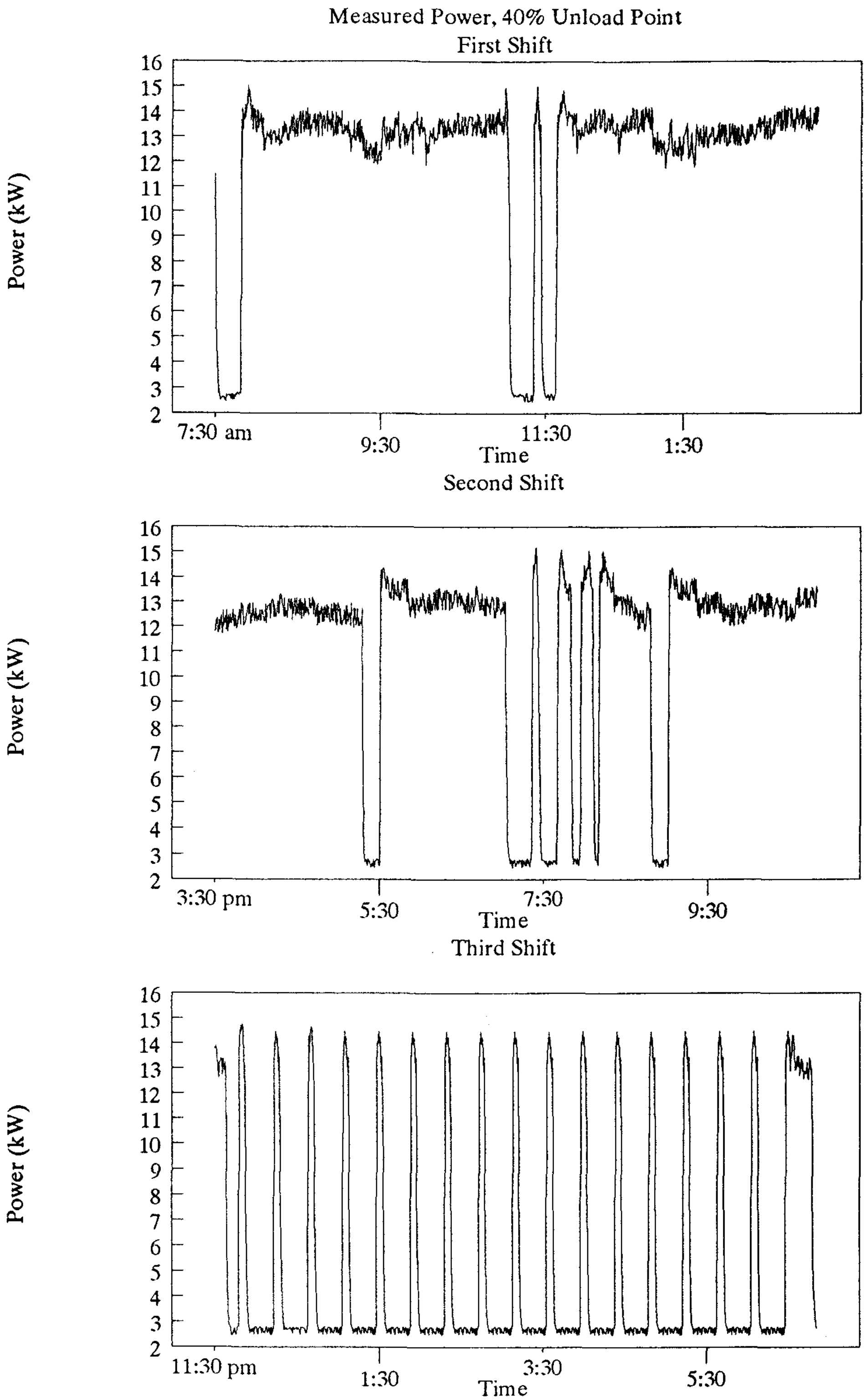

Figure 21: Case Study, 40% Unload Point

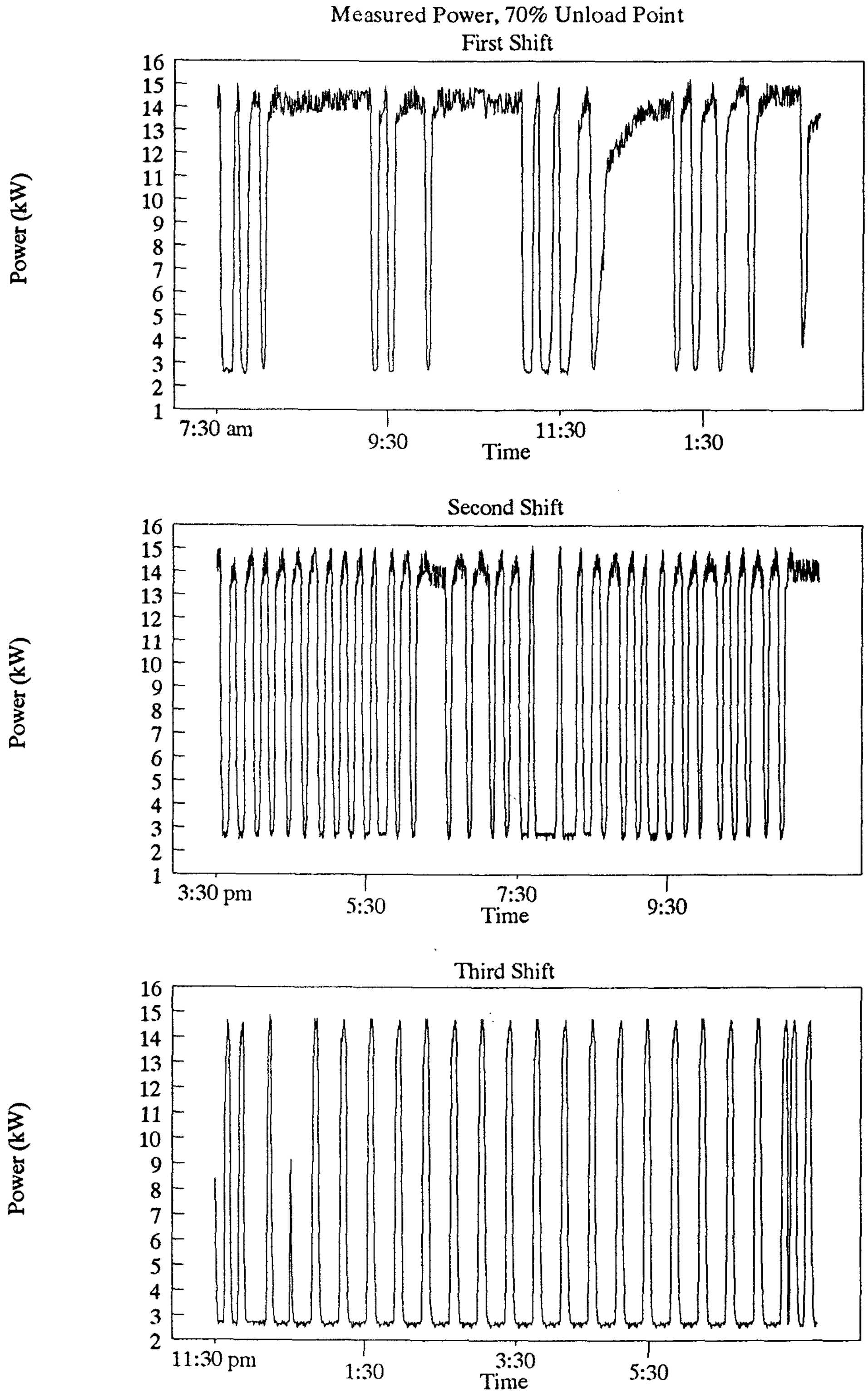

Figure 22: Case Study, 70% Unload Point

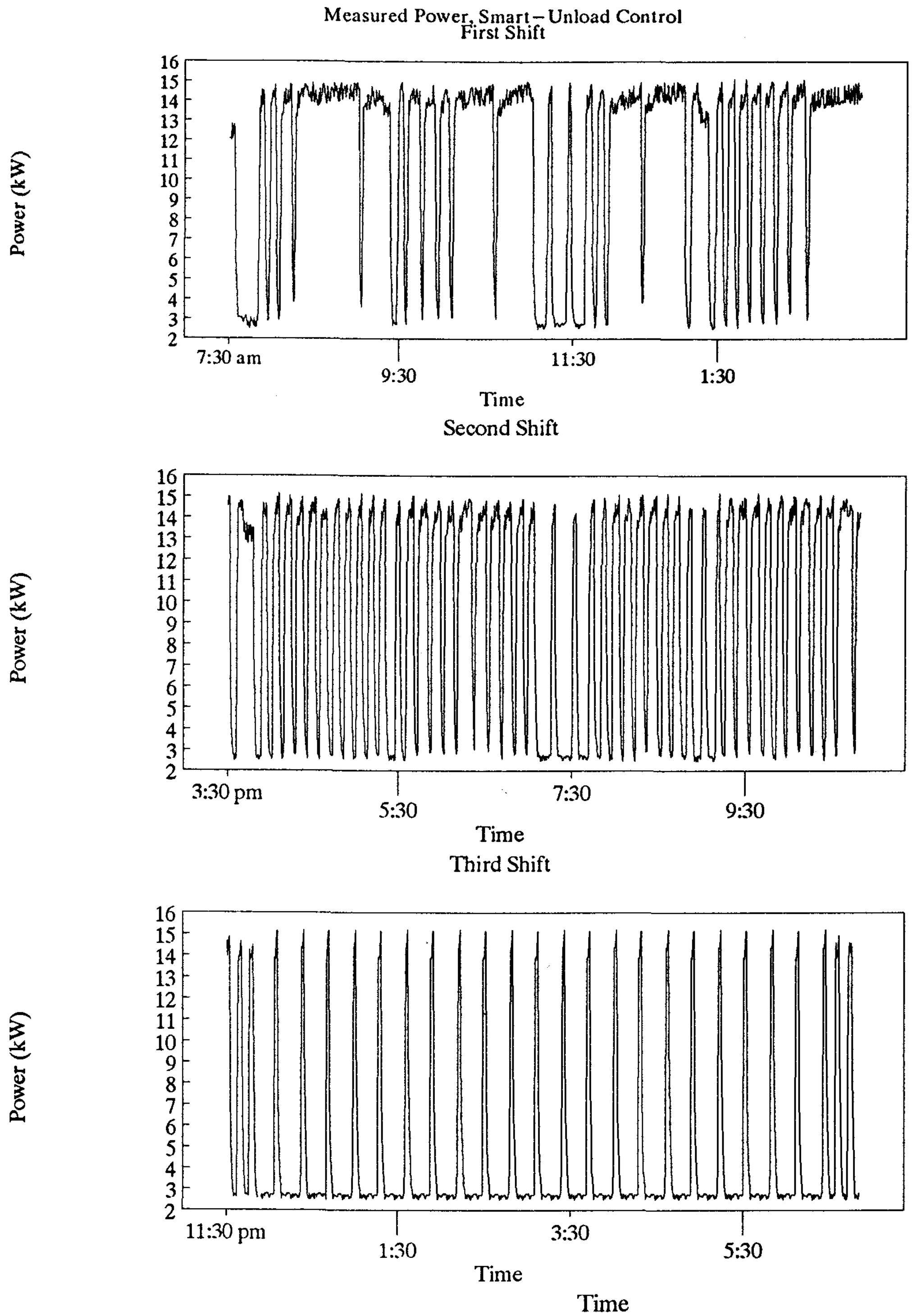

Figure 23: Case Study, Smart Control

However, the low-unload mechanism would cycle when plant demand was between 82% and 90%, possibly using less energy than the modulating smart controller. Thus, it is plausible if not conclusive that the 90% low-unload average power would match the smart-controller average power. Smart controller behavior should be preferable however, for the same energy consumption, because of the operating logic described above (lower average power when cycling, modulation when unload or cycle time is unacceptably low, or average power would increase).

F. Cost

Total hardware cost for the prototype smart controller was $583. A detailed price list can be found in Appendix I. Two pressure sensors represents 50% of the total cost. For a production unit, the parts would cost substantially less.

G. Installation

1. Quincy Compressors. This design attempts to improve the efficiency of low-unload type controllers without changing any hardware or actuators. It is specifically designed for use on Quincy air compressors. The smart controller could be offered with new air compressors or retrofitted on existing compressors within an hour. Since low-unload controllers are also retrofitable, the system could be installed on an existing compressor with throttling part load control. The installation process is as follows, assuming low-unload controls are already installed.

1. Turn the compressor off.

2. Install two 1/4" tees in the air lines that lead to the inlet capacity (vacuum) gauge and the oil separator pressure gauge. Screw in the two pressure sensors to the tees.

3. Disconnect the three wires that lead to the vacuum switch (C, NC, NO), and connect them to the similarly labeled terminals attached to the smart controller.

The compressor can now be turned back on. No data input is required during installation. If the smart-control unit is not plugged in (to a standard 120 VAC socket), the compressor will operate in throttling-only mode. Once the smart control is energized, with the compressor either on or off, it will switch from throttling-only control to smart-control. Two LEDs indicate activation of the system.

For best results, the controller should remain energized even when the compressor is off, because all dynamic parameters will be reset to their default values whenever power is removed from the smart controller.

2. Non-Quincy Compressors. The smart controller was designed for use with a Quincy compressor. The electronics, sensors, and hardware could be transferred directly to any compressor design that employs throttling and cycling control.

The basic optimization and mode switching concept, and probably the electronics, could also be applied to any compressor that offered both modulation and cycling control, including those with inlet volume modulation. For example, the discharge pressure gauge could also be used to calculate operating capacity on compressors with inlet volume modulation. Alternately, a turn valve position sensor could be used, or poppet valve position could be taken from existing electronic controls. The output switch would change its logical location in the electrical schematic, but would remain a simple two position switch. Both sensing and actuating devices would likely require significant change. Some software subroutines would be subject to revision as well.

3. Maintenance. Little additional maintenance is required. If the power supply receives a surge, the controller may need to be disconnected for 30 seconds and then reconnected because it shuts itself off for protection. The pressure sensors are calibrated within the program.

4. Later Revisions. The logical decision-making structure can be updated at any time with little hardware cost by replacing the removable EPROM with a reprogrammed chip. If a UV light, an additional 12VDC power supply, and a computer with an RS-232 serial port are available, the program can be upgraded without a new EPROM.

5. Data Output. There are no provisions for long term storage of performance data. Instantaneous operating conditions can be monitored by attaching a personal computer with terminal emulation software (or a terminal) to the RS-232 port on the BCC-52 board. Output includes throttle position, discharge pressure, maximum capacity for cycling (UP), and instantaneous power. Because BASIC-52 only delivers line output, the data scrolls and is more functional than convenient.

V. GENERAL RECOMMENDATIONS

The following recommendations represent a collection of observations made while researching part-load control of screw air compressors:

- Large receivers are a good investment. They are relatively low cost items (about $3 /gallon) that can reduce cycling losses by 5 to 20%, reduce compressor wear, and stabilize air supply. This benefits the compressed air user as well as the compressor and its control system.

- Load-unload controlled compressors with large receivers should be used more often, particularly if full 0 psig unloading is possible. When cycling is an appropriate control strategy, it makes sense to operate in the most efficient operating mode possible.

- Based on personal experience, field interviews and discussion with manufacturer's representatives, the reliability of unloading controls is questionable. Even when originally installed on compressors, they are often disabled by maintenance personnel. Regardless of whether difficulties are real or perceived, the end result is that inefficient modulating controls are used in many situations where unloading controls are applicable and more efficient. For this reason, variable speed drives are a good alternative. They offer modulating control, which appeals to operators, while offering relatively efficient performance at the same time.

  Alternately, a recommended approach for pneumatically actuated cycling controls is that they be designed with larger control lines and valves, and better ways of removing water and oil from existing configurations.

- Better training of maintenance personnel and plant engineers is important. They often do not realize the financial value of maintaining air compressor controls to their design condition.

- Electronic controls allow more control options. For example, the proportional control range could be reduced to two psi or less. Another possibility for low-unload type, all-electronic control systems (no pilot valve, stepper motor actuation of throttle position) is throttling at the low end of the acceptable pressure range instead of the high end. This would save 5 to 10% of energy use.

VI. SUMMARY

This has been a wide ranging project. The summary is equally diverse.

- **Pressure Modeling.** Based on research results, the following relationships summarize compressor suction and discharge pressure during different times in the low-unload operating cycle. The relationships were combined in a spreadsheet to model behavior for a single operating cycle.

Mode	Relationship
Throttling	Proportional control of capacity/suction pressure.
Unloading	Exponential decay of suction, discharge pressure.
Reloading	Arctangent curve fit of suction pressure.
All	Isothermal change in receiver pressure due to compressor contributions and plant demand.

- **Power Modeling.** Compressor power (P) was shown to vary linearly with suction pressure (which is directly related to capacity) and also with discharge pressure. The formula for power under any operating conditions can be expressed as:

$$P = [FTP\% + (1-FTP\%)\ x\ C\%]\ x\ [\frac{FUP\%}{FTP\%} + (1-\frac{FUP\%}{FTP\%})\ x\ p_{d_{max}}\%]$$

where, FTP% is fully throttled power expressed as a percent of the power at rated pressure and capacity (typically 65 to 75%), FUP% is fully unloaded power (typically 15 to 30%), C% is compressor operating capacity, and $Pd_{max}\%$ is the percent of rated compressor pressure.

The spreadsheet model was built around these relationships. The model can estimate the average power for an operating cycling with a 5% accuracy if all variables are known. Figure 24 shows an example of modeled power over the course of one complete operating cycle.

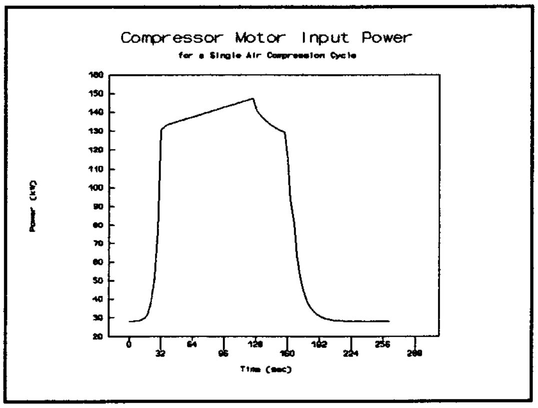

Figure 24: Modeled Power for One Cycle

- **Unloading Losses.** The model was used to estimate cycling losses under different conditions. By using the model as a tool, some interesting conclusions were made, particularly relating to the issue of cycling losses. Figure 25 illustrates the effect of cycle time on unloading losses.

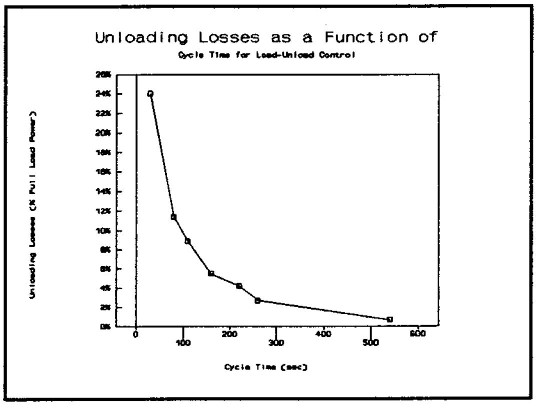

Figure 25: Cycling Losses

- **Low-Unload Average Power.** Low-unload average power during cycling may be lower than is traditionally assumed by up to 10% of full load power. Figure 26 shows an example of traditional and modeled low-unload Power for a system with a large cycle time (10 minutes). When cycle time drops below 1 minute (not shown), the traditional curve reflects average power consumption fairly well, because unloading losses increase and average power approximately follows the traditional low-unload line.

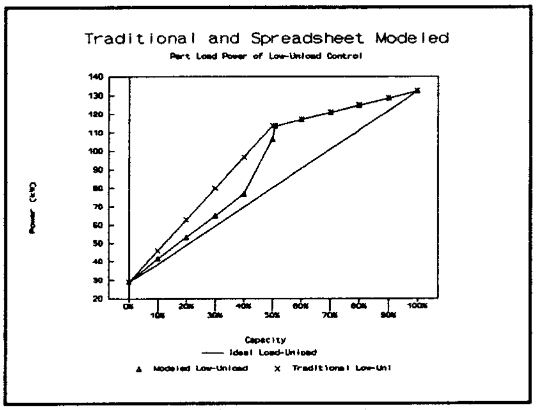

Figure 26: Traditional/Modeled Low-Unload Power

- **Receivers.** The two preceding results highlight the value of large receivers. Not only do they smooth out air demand and slow pressure changes, but they can also reduce cycling losses and energy consumption.

- **Smart Control.** The smart controller designed for this project was successful in reducing energy use, optimizing mode switching between throttling and load-unload control, and in responding to average instead of instantaneous load conditions. The results are summarized below.

Control Strategy	Energy Savings with Smart Control*
Throttling	32%
Low-Unload, 40% UP	12%
Low-Unload, 70% UP	4%
Low-Unload, 90% UP	5%

*Measured savings based on case study results, except 90% unload point (UP), which is based on modeled behavior.

Though only modest energy savings potential exists compared to a low-unload controller with a properly set unload point (in part due to the Low-Unload results), the smart controller showed substantial savings over an incorrectly set low unload controller. Smart controls also contribute to the industry by making unloading control more feasible in environments where they otherwise would not be considered a viable option. In this situation, the savings potential is most significant, because savings is relative to modulating control.

VII. CONCLUSIONS

This thesis focuses on improving part load efficiency of screw air compressors through education, improved performance modeling, and a new "smart" controller design.

- A guidebook was written to help educate compressed air users of the cost and importance of good part load efficiency. The guidebook is intended to increase the use of efficient control strategies.

- A spreadsheet model was developed to simulate compressor operation when cycling. Average power can be estimated with better than 5% accuracy if all variables are known. The model also works well as a comparative tool when some variables must be assumed.

- Cycling losses are not always negligible, and they depend primarily on cycle time. Based on model calculations, losses ranged from 23% to less than 1% as cycle time increased from 30 seconds to 8 minutes.

- Low-unload type control may operate as much as 10% more efficiently than is traditionally assumed when cycling. This is especially true for compressor systems with high (>5 minute) cycle times. As cycle time decreases to one minute or below, unloading losses increase and the traditional power model becomes more accurate.

- A "smart" microprocessor-based controller was successfully designed to improve compressor part load efficiency. Compared to conventional controllers, energy savings ranged from 4% to 32%. The controller switches the compressor between load-unload and throttling control strategies, optimizes the air flow capacity at which switching occurs, and makes decisions based on average demand characteristics rather than instantaneous behavior. A prototype unit was designed, programmed, built and installed on a compressor. The controller should make efficient unloading controls a feasible option for some users that would otherwise be limited to modulating control. Hardware cost was approximately $600.

1. Gardner-Denver-Cooper Industries, Electra-Saver Rotary Screw Air Compressor, 100-500 hp, 13-10-200, (1989), p. 3, and LeRoi Division of Dresser Industries, WE Series Rotary Screw Compressors, (Sidney, OH: 1986), p.6.

2. McCoy, G.A., et.al., Energy Efficient Electric Motor Selection Handbook, prepared for the Bonneville Power Administration, U.S.D.O.E., September 1991, pp. 27-28.

3. 0.5 %FLP/psi is frequently assumed for conservation recommendations. In particular, the Sullair Technical Manual shows this correlation for many of their compressors. 0.3 %FLP/%Cap is standard part load behavior for throttled compressors. See the guidebook in Appendix A for more information.

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7. Sullair Corporation, Engineering Manual, (Michigan City, IN: Sullair, 1991), performance curves. Also taken from Charles W. Gibbs, ed., Compressed Air and Gas Data, (New York: Ingersoll-Rand, 1971), p.16-4.

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10. O'Keefe, p. S-7.

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14. Lyle Wells, Compressed Air Systems, presentation at The Electric Ideas Workshop: Compressed Air Systems; Opportunities for Energy and Cost Savings, December 12, 1991.

15. Walter M. Novak, Some Basic Measures for Conserving Energy in Compressed Air Systems, reprinted from Plant Engineering, February 19, 1976, p.4.

16. According to Mark D. Oviatt and Richard K. Miller, Industrial Pneumatic Systems: Noise Control and Energy Conservation, (Atlanta: Fairmont Press, 1981), p. 49, the savings will be about 1% power per 5°F. This estimate is based solely on the increasing inlet air density. However, compressor must work harder to compress the denser air. Therefore, the net effect on power is closer to 1% savings per 10°F. Interview with Lyle Wells, Rogers Machinery Co., November 12, 1992.

17. B.C Hydro, Efficient Compressed Air Systems: A reference guide for positive displacement air compressors, capacity controls, and dryers, Vancouver, BC, p. 18.

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19. Frank P. Incropera and David P. DeWitt, Fundamentals of Heat and Mass Transfer, John Wiley, (New York: 1985), p. 767.

20. Incropera & DeWitt, p. 767.

21. J.P.Holman, W.J.Gajda, Jr., Experimental Methods for Engineers, McGraw-Hill, (New York: 1989), figure 7-13, p. 239.

22. Holman, figure 7-15, p. 241.

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APPENDICES

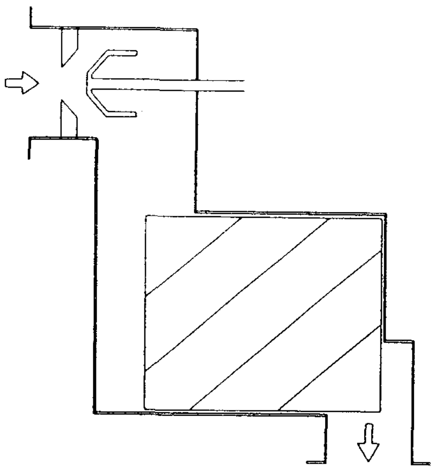
APPENDIX A: SCREW COMPRESSOR CONTROLS GUIDEBOOK

Notes: Part of the introductory section of the guidebook was used for the background section of this thesis. No other elements are replicated.

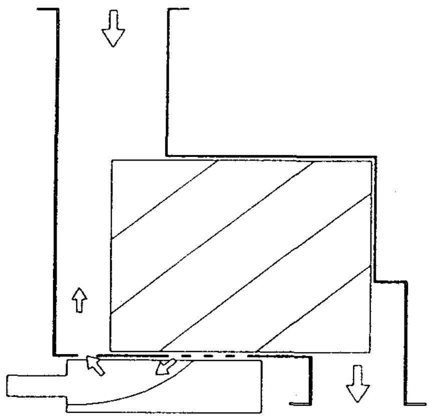
References to Figure numbers in the text of this appendix are not correct because the Figures were renumbered for inclusion in the thesis.

A Guidebook for Screw Air Compressor Controls:

Operating Principles and Selection for Minimum Energy Use

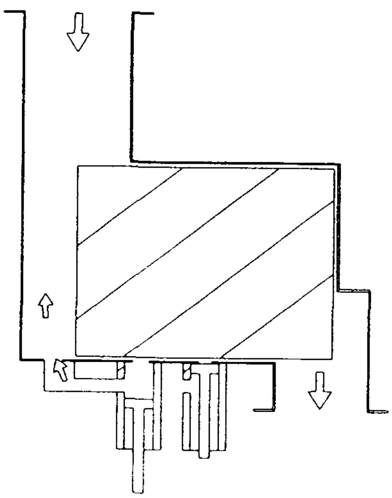


Throttle

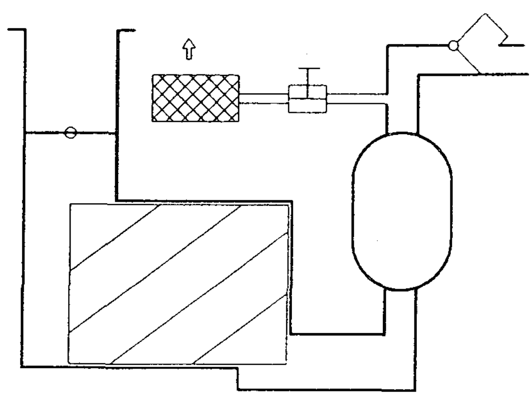


Turn Valve

?



Poppet Valve



Unloading

Bonneville Power Administration November 1992

DISCLAIMER

ACKNOWLEDGEMENTS

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WHO SHOULD USE THIS GUIDEBOOK:

This guidebook was written primarily for two audiences. The first intended group is composed of people who want to learn about the various types of control strategies produced in industry today for part load control of rotary screw air compressors. The second group includes maintenance personnel, facilities engineers, and other purchasers who are actively shopping for a compressor system and would like an objective presentation of the different approaches to improved energy efficiency at part load.

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INTRODUCTION

> "It is stated that the entire power cost for running the air compressors to supply the whole shop is not more than ten cents per day... However the air may be used, and however profitable it may be to use it, it will always be in order to get it as cheaply as possible, and economy in air-compression must always be a clear gain." (from Practical Information Upon Air Compression and the Transmission and Application of Compressed Air, 1895)[5]

In 1895, energy efficiency in air compression was a significant issue, because electricity was gradually winning the battle between the two methods as the better large scale energy transfer mechanism. In modern times, we are motivated to be thrifty instead by rising energy rates induced by environmental concerns and shrinking natural resources. However, the goal of economy in air-compression still rings true 97 years later.

Compressed air remains a wonderfully convenient form of energy for the end user. Pneumatic tools are fast. Pneumatic cylinders are cleaner than hydraulic ones, especially if leaks occur. Compressed air is great for cleaning off equipment. Compressed air is safer to use than an electric motor in an explosive environment. And to the user, compressed air is as easy to use as electricity. Unfortunately, the energy provided by compressed air is not cheap. The amount of electric energy purchased to drill a hole in a piece of wood may be 20 times higher for an air-powered drill compared to an electric one. Nonetheless, the unique qualities of compressed air make it an indispensable resource in the industrial processing environment, and will continue to provide a convenient form of energy in the foreseeable future.

Over the last thirty years, rotary-screw air compressors have gradually permeated the plant air systems (90-140 psig) market. Screw compressors now share the market about equally with reciprocating compressors and are still growing in popularity. They are beginning to dominate sales of large compressors. The success of screw compressors can be attributed to lower first costs compared to double-acting reciprocating compressors, more dependability than single-acting compressors, low maintenance requirements, ease of installation, and in the case of very sensitive equipment, less pulsation than reciprocating compressors.

In general, the full load efficiency of screw compressors is similar to that of reciprocating compressors. Full load efficiency is only part of the story, however.

In practice, very few compressors operate at full load all of the time. One manufacturer estimates the average load to be 60-70% of full capacity.[6] Traditionally, part load efficiency has been the Achilles heel of screw air compressors. This is where controls become important.

There are many different methods to match the amount of air provided by a compressor with plant air requirements. Each approach has strengths and weaknesses. This guidebook first presents the different types of air compressors, and then introduces a basic screw compressor-based system. The report next explains existing approaches to part load control, and evaluates them for meeting different types of demand and for energy efficiency.

AIR COMPRESSOR TYPES

There are many ways to compress air. The chart below shows a typical method of grouping mechanical air compressors. The most fundamental division between compressor types is that between positive-displacement compressors and dynamic compressors. Positive-displacement compressors draw in a fixed volume of air, reduce the volume occupied by air, and then eject it (see Figure 1). In contrast, dynamic compressors introduce energy to the air through transfer of momentum. Dynamic compressors do not compress discrete volumes of air. As the chart indicates, screw, reciprocating and rotary vane compressors are all positive-displacement compressors. These are by far the most common type of compressors used to supply plant air. Different types of air compressors are shown in Figure 2, below. Before exploring the details of compressed air systems, each general type of compressor will be introduced.

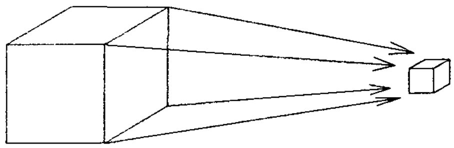

Figure 27: Positive Displacement Compression

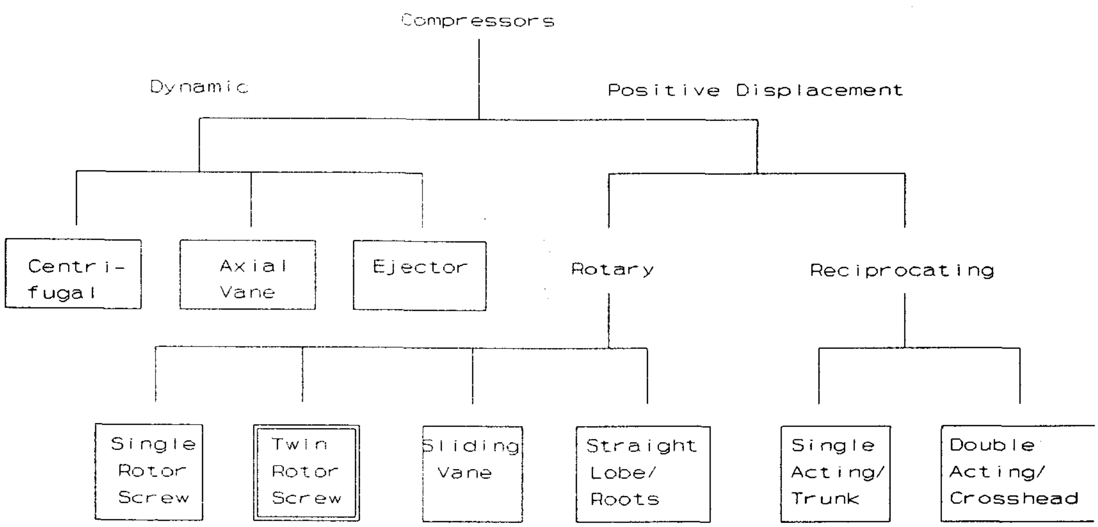

Figure 28: Basic Compressor Types

Screw compressors: There are both single screw and twin screw compressors, as well as multistage screw compressors. This guidebook focuses on the most common version, single stage twin screw compressors.

A screw compressor is shown in Figure 3. In operation, filtered air is drawn in at the "front" of the compressor through the inlet port as shown in the top diagram. As the screws rotate, the male and female lobes intermesh. A portion of air is drawn into the compressor, and is trapped between the two lobes and the casing. The trapped air is shown as the shaded region in the middle diagram. Since the rotors are spiraled, the air gradually moves "backward" as the rotors spin. The air is compressed because the space between the lobes gets smaller as the air approaches the discharge port of the compressor. After compression is complete, typically about 3/4 of a rotation of the male rotor, the portion of air is released to the system through the discharge port. This compression process repeats as often as 25,000 times/minute.

The male rotor commonly will have between three and five lobes and is driven by the motor. The female rotor is most often driven by the male rotor and typically will have one or two more lobes than the male rotor.

Oil normally provides the seal necessary to prevent air leakage, but water and other oil-free lubricants also can be used. Dry compressors are also marketed. They require external cooling and usually will compress the air in two lower ratio stages in order to reach plant air pressure levels without leakage.

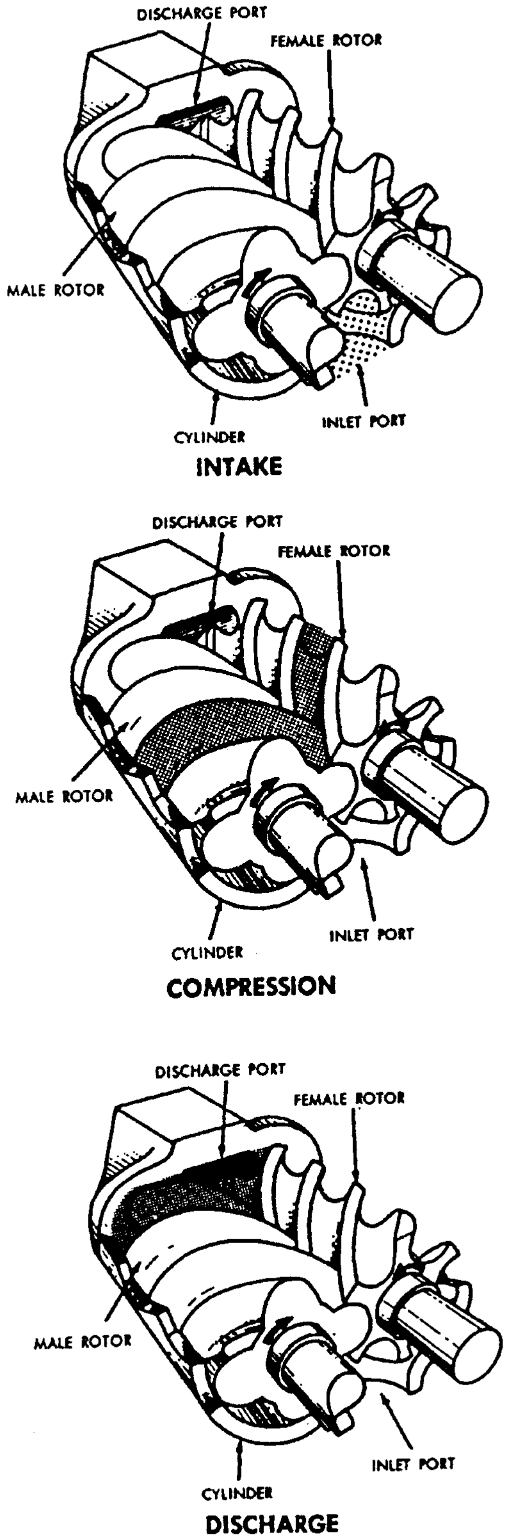

Figure 29: Screw Compressor

Two stage units use two pairs of screws and are more expensive than single stage compressors. Since the overall compression ratio is the same as for single stage units but the number of stages is doubled, each stage performs less compression. The reduced compression ratio for each stage reduces the pressure difference across the screw threads during compression. This in turn reduces internal leakage, called slip. Efficiency at full load for two stage units is likely to be about 10-13% higher than single stage unit efficiency.[7] Two stage compressors usually have the same controls and part load performance characteristics as single stage units.

Reciprocating Compressors: In the past, reciprocating compressors were the workhorses of the plant air market, and some partisans would argue that they still are preferable to screw compressors. Single acting, or trunk, compressors operate like an automotive piston. Air is compressed in a chamber on one side of the piston head (see Figure 4). Most small reciprocating compressors are single acting models. Double acting, or crosshead, compressors compress air on both sides of the piston head. They are more expensive, more efficient, and have a longer average life than single acting units. Most new large reciprocating compressors are double acting.

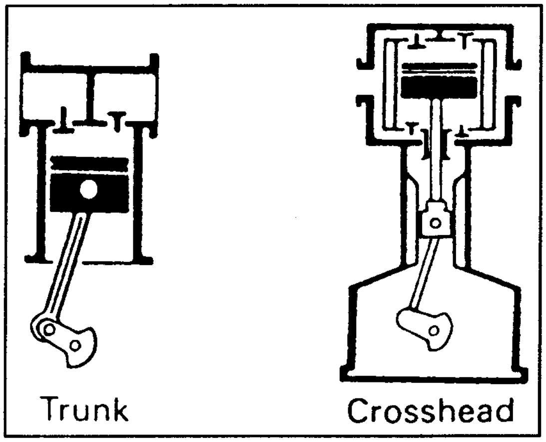

Figure 30: Reciprocating Compressors

Reciprocating compressors are made as both single stage and multiple stage machines. Multi-stage machines compress the air in two or more stages and are required for high pressure applications. They are also often used in lower pressure applications because they are more efficient than single stage units. The additional machinery does make multi-stage compressors more expensive than single stage compressors. The part load efficiency of reciprocating compressors is very good. Compared to screw compressors, they offer similar full load efficiency (water cooled crossheads may be up to 15% higher) and as good or better part load efficiency. However, capital and installation costs are likely to be higher. Depending on whom you ask, operating & maintenance costs may also be higher.

Rotary vane compressors: Vane compressors appear to be vanishing breed in this country's plant air market. A cylinder with free sliding radial vanes rotates off-center in a larger cylinder (see Figure 5). The vanes slide in and out to follow the inner surface of the outer cylinder. Air is drawn in at the point where the pocket between the two vanes is largest. The air is trapped between the vanes and the cylinder walls, compressed, and then discharged as the rotor turns. Vane compressors are often less expensive and only require moderate maintenance, but they are generally less efficient than rotary screw compressors.

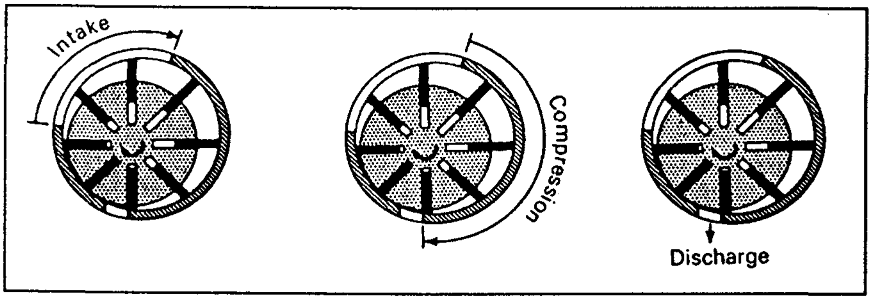

Figure 31: Sliding Vane Compressor

Straight Lobe Blower: As indicated by the term 'blower,' this type of machine only supplies compressed air up to about 15 psig. An argument could be made that lobe blowers should not be classified as true compressors. Two intermeshing rotating lobes trap segments of air one at a time, and force the air through the discharge (see 'Roots' in Figure 6). They don't actually compress air, but force air from the inlet to the discharge without allowing reverse flow. Lobe blowers are more efficient at providing low pressure air than reducing 100 psig plant air to low pressure (see The Compressed Air Energy Conservation Opportunities section for more details).

Liquid Ring compressors: Liquid ring or liquid piston compressors look like rotary vane models but use a compressant liquid that moves in and out like a radial piston. Liquid ring compressors are less efficient than rotary vane compressors, but are extremely reliable. They are also used in vacuum service.

Centrifugal compressors: Centrifugal, or radial, compressors work by slinging air radially out through an impeller and increasing its momentum. The energy added by momentum transfer is converted to pressurized air partly by the end of the impeller, and by using a diffuser. Single stage centrifugal compressors usually have a pressure ratio of less than three, and normally several stages are used together. Centrifugal compressors are quite dependable, but inefficient at the lower flow rates normally required for plant air. They are likely to be used when large flow rates (up to 150,000 cfm) are required.

Axial vane compressors: Turbines are the most glorified type of axial vane compressors. Like centrifugal compressors, axial compressors add energy by momentum transfer, but air flows parallel to the rotor shaft instead of perpendicular to it. Axial compressors are basically multistage fans. Almost always multi-staged, the vast majority of vane compressors are found in aircraft applications and as gas turbine compressors.

Ejector compressors: For the sake of completeness, ejector compressors deserve mention. In operation, a jet of steam or other vapor is ejected from a nozzle into an air stream. The gas is entrained in the steam jet and is compressed as the mixture passes through another nozzle, where energy changes state, from velocity to pressure. Ejector compressors are most often used to compress gases from a partial vacuum up to atmospheric pressure.

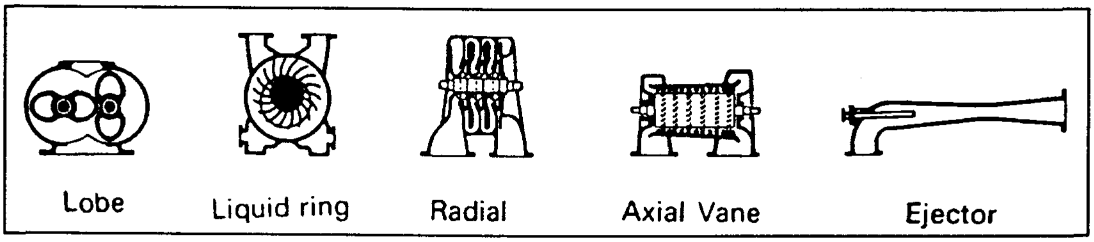

Figure 32: Other Compressor Types

COMPRESSED AIR EQUIPMENT

This section introduces the basic components of a compressed air system.

COMPRESSOR - The heart of the system is the compressor itself. From this point on, the guidebook focuses on rotary screw compressors.

AIR FILTER - Just like automotive engines, all compressors will have an air filter preceding the intake. A simple schematic below, Figure 7, introduces the motor, screw compressor, and air filter. Other components will be added to the diagram as they are described.

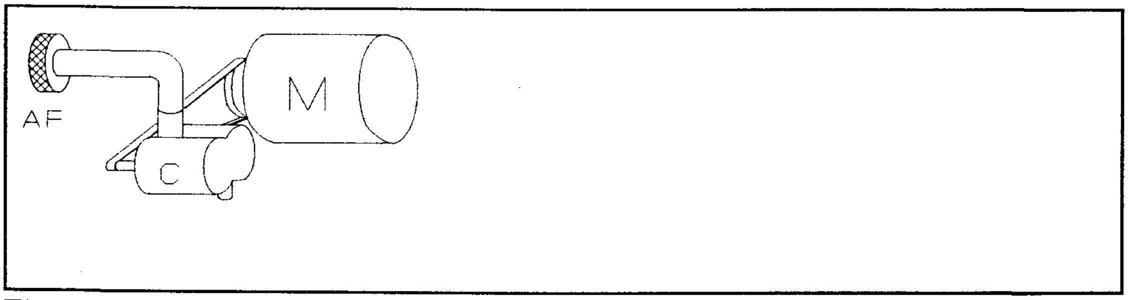

Figure 33: Air Filter (AF), Compressor (C), and Motor (M)

OIL SEPARATOR - The majority of screw compressors use oil, transmission fluid, or other lubricant for cooling, lubrication, and as a seal between moving parts. Recently, synthetic lubricants have been promoted as a route to increased efficiency for air compressors as well as automobiles. Most of the oil is injected into the compressor at the beginning of the compression zone, see Figure 8. A small amount of oil may be introduced into the air to be compressed prior to the intake. The air leaving through the compressor discharge port is therefore a mixture of air and oil. A properly working oil separator will remove virtually all of the oil in the air. One test found that the air leaving the oil separator actually had less suspended oil in it than the ambient air drawn into the compressor intake port![8] Most separators do not work quite this well. The element itself is normally composed of a coalescing filter supported by a perforated metal baffle. As the mixture rushes up through the separator, gravity causes the larger oil droplets to fall into a sump at the bottom of the separator. The filter causes the smaller droplets to collide and collect into larger into larger droplets (coalesce) that then fall into the sump.

OIL COOLING - The compression process generates considerable heat energy. Some of the generated heat is due to friction, but the majority of the heat is due to the fact the air gets hot as it is compressed. A lot of this energy is absorbed by the oil. Consequently the oil needs to be cooled before it is injected back into the compressor. This can be done with either an oil-air heat exchanger (radiator, OC) as shown in the diagram, or with a oil-water heat exchanger. The diagram also includes an oil filter (OF).

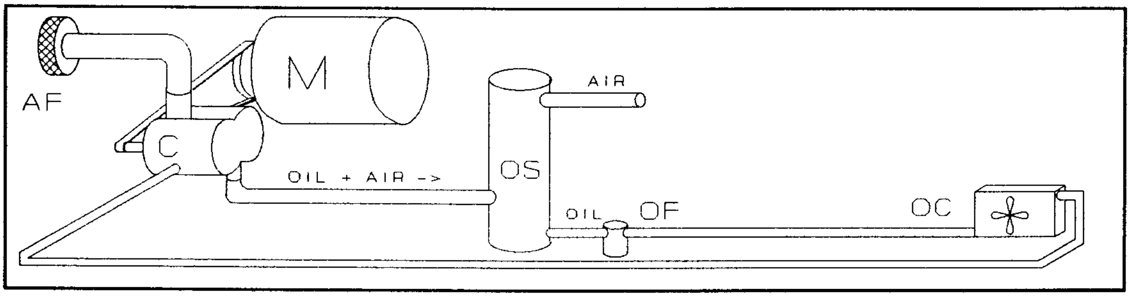

Figure 34: Oil Separator (OS), Oil Filter (OF), and Oil Cooling (OC)

AIR DRYING - Unless your compressor operates in a desert, moisture will need to be removed from the hot compressed air before it can be delivered to equipment for use. The most common method is to cool the air, like the oil, with either an air or a water heat exchanger, see Figure 9. This causes the water vapor in the compressed air to condense and fall out of the air. A moisture trap collects the water until it is drained from the system. Refrigerated, heated, and desiccant dryers can be used to supplement the air/water heat exchanger.

REHEAT (optional) - Since the delivery of compressed air is fundamentally an energy transfer process, it is normally economical to reheat the cooled air with either the hot oil or the hot pre-dried air. The oil-air heat exchanger shown below adds energy back to the compressed air.

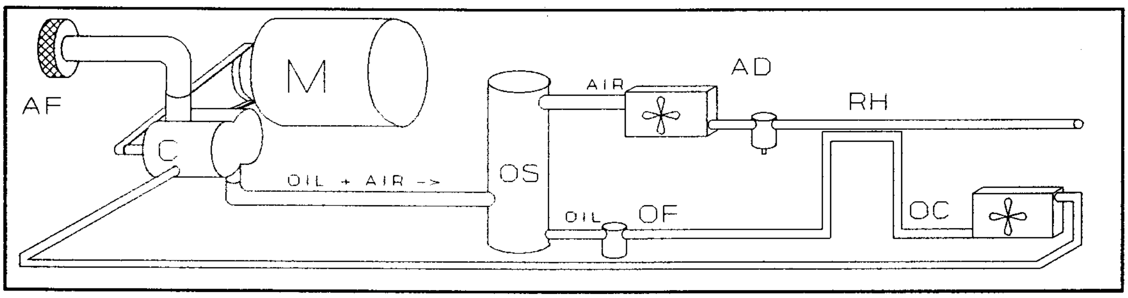

Figure 35: Air Drying (AD) and Reheat Option (RH)

RECEIVER - Finally, the compressed air arrives at the receiver. From the point of view of the air, the volume of space available for storage includes not only the receiver, but the miles of air lines that may be scattered throughout the plant. The main job of the receiver is to smooth out the erratic starts and stops of equipment into a manageable amount of pressure fluctuation. For a compressor with only modulating controls, a huge receiver may not be necessary. However, any compressor that unloads will benefit from a large receiver. With regards to cycling, bigger is unconditionally better. In fact, any application that can tolerate a 10 psi range for system pressure can use the efficient load-unload or even on-off control strategies if enough storage capacity is available. This is a low-tech. solution for major energy savings and will be re-emphasized as the guidebook continues. One manufacturer recommends a minimum of 1 gallon capacity per scfm output. If air use is intermittent, a larger receiver should be considered.

AIR COMPRESSOR CONTROLS

For some types of machinery, people can treat controls conveniently as magic black boxes with little penalty. Unfortunately, air compressor controls play a major role in the performance of the compressor. A bad choice may silently absorb thousands of dollars in unnecessary energy costs, or even render the equipment incapable of meeting plant air requirements. The good news is that compressor controls are understandable if not altogether simple, and selecting the proper control strategy can be a rational and financially rewarding process.

Full load conditions do not require intricate and confusing control strategies. All control systems leave the compressor running the same way - flat out. But few compressors operate at full capacity at all times. New compressors are bought oversized in anticipation of future growth; cutbacks in production reduce air demand, a compressor sized for peak air demand is twice as large as necessary for normal demand. In practical terms, a compressor is not usually going to be fully loaded unless perhaps it is one in a series of sequenced compressors. The objective of a good control strategy is to balance less-than-ideal part load efficiency with adequate capacity to reliably provide compressed air to users.

Performance Table: For each control strategy that is presented, a summary of part load performance characteristics is included in the header. They are:

Minimum Storage Capacity. This indicates the minimum receiver size required for proper operation. Control strategies employ cycling require more storage capacity than those that modulate, particularly if demand is high.

Low Demand Efficiency. This rating represents the relative efficiency of the control strategy based on average power required for a compressor operating uniformly between 0% and 50% capacity.

High Demand Efficiency. This indicates relative efficiency between 50% and 100% capacity.

Controlled Variable(s). Controlled variables are those conditions that are directly manipulated by the controller.

Efficiency ratings are based on the following table:

Rating	Part Load Efficiency (Percent Full Load cfm/hp)
Excellent	85-100%
Good	70-85%
Average	55-70%
Fair	40-55%
Poor	25-40%

Thermodynamics. This section describe the effect of changes made in the controlled variable(s) cited in the table. Ignoring mechanical inefficiencies, the following equation applies to air compression up to a few hundred pounds.[9]

$$p_i V_i^n = p_o V_o^n$$

where,

p_i = Inlet pressure
p_o = Outlet pressure
V_i = Inlet volume
V_o = Outlet volume
n = Polytropic constant

The polytropic constant is fixed for a given gas (such as air) and compression process. For any screw compressor control strategy, the other four parameters can be physically adjusted to reduce compressor capacity. The discharge pressure (p_d) may also be manipulated. The discharge pressure is the pressure of the air at the discharge port and in the oil separator, while the outlet pressure represents the pressure of the small compressed volume of air immediately **before** it is released to the discharge port. Therefore, the outlet pressure may be affected by changes in inlet conditions even when the discharge pressure is not affected.
Finally, the duty cycle and average compression rate can also be reduced. In this guidebook, the term pressure ratio (PR) is used to describe the air pressure after it has been expelled from the screws (p_d) divided by the air pressure just prior to the intake port of the compressor (p_i). Thus,

$$PR = p_d / p_i$$

Mechanics. The Mechanics section explains how the controlled variable(s) are physically manipulated.

Energy Use. There will also be a section on energy use for each control strategy.

The principal instrument in this section is the Part Load Power curve. The graph indicates possible operating points as a solid line and with discrete points. Overall average power is indicated by a dotted line when it departs from actual operating points. A dashed line indicates the control mode's efficiency. Air flow rate out per horsepower input (cfm/hp) is a measure of efficiency. There will also be at least one formula presented in the section. The formula defines the power curve shown on the graph. In most cases, these formulas model specific curves or data provided by equipment manufacturers. Actual performance will, of course, vary.

Applications. The final section describes appropriate operating environments for the control strategy.

THROTTLING CONTROL			
Minimum Storage Capacity	Low Load Efficiency	High Load Efficiency	Controlled Variable
Small	Poor	Good	Inlet Pressure, p_i

Thermodynamics: Throttling controls reduce the inlet pressure below atmospheric pressure (0 psig) by causing a partial vacuum to form at the compressor inlet. The outlet pressure will be affected, the inlet and outlet volumes and the discharge pressure will not change. In thermodynamic jargon, the pressure ratio increases and the volumetric compression ratio remains constant. The introduction to this section describes all of these terms.

Mechanics: In general, modulating controls are a treat for the finicky user because the compressor tries to maintain a constant discharge pressure instead of bouncing between a high and low set pressure. Throttling modulation works by starving the compressor of air. The mechanism itself is a butterfly valve or a slide valve that is mounted upstream of the compressor inlet, as shown in Figures 10 and 11. If the system pressure sensor perceives that the pressure is increasing (because plant demand has decreased below the system capacity), the modulating valve starts to close. This creates a partial vacuum at the compressor inlet. Consequently, the air entering the compressor is less dense and less air mass enters the compression chamber between the screws. The compressor delivers less air to match demand. The discharge port is always at system pressure. The location of the discharge port on the compressor casing defines the volume ratio, and different ratios are available for different pressure ranges. However, once it is machined, it cannot be changed.

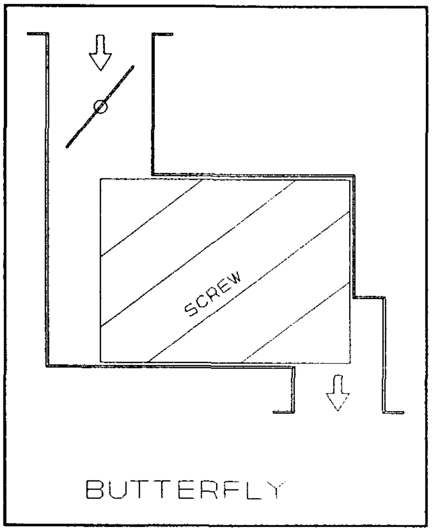

Figure 36: Butterfly Valve

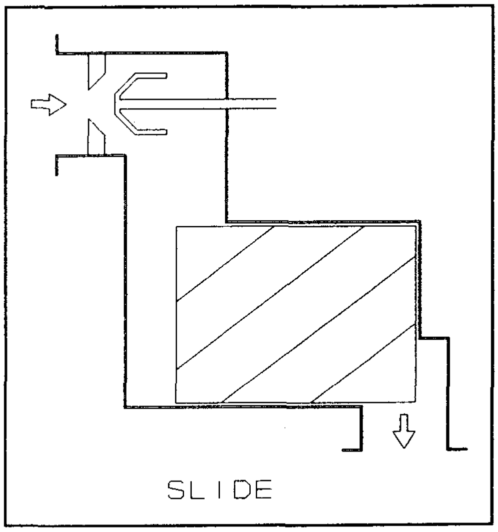

Figure 37: Slide Valve

In summary, a fixed volume of air is drawn into the compressor and the volume is always compressed by the same amount (V_1/V_2 = constant). The number of compression cycles per minute does not decrease. But the pressure ratio does change, and the mass flow rate of delivered air also changes.

Energy Use: Life would be easy if throttling controls were also efficient. Unfortunately, this mode is often not efficient. Because the compressor constantly works against system pressure at the discharge port, the motor never really gets a chance to relax. Part load efficiency is particularly poor during low demand periods. Sixty-eight percent of the full-load power is required at 0% capacity! The part load performance curve in Figure 12 shows the a linear relationship between power and capacity. The equation for part-load power is:

$$\%P = 68\% + 0.32 \times \%C$$

This mode of control can be efficient for high loads, because excessive cycling is avoided (see unloading control strategies described later).

The Part Load Performance curve indicates that the compressor can operate continuously at any point between 0% and 100% capacity.

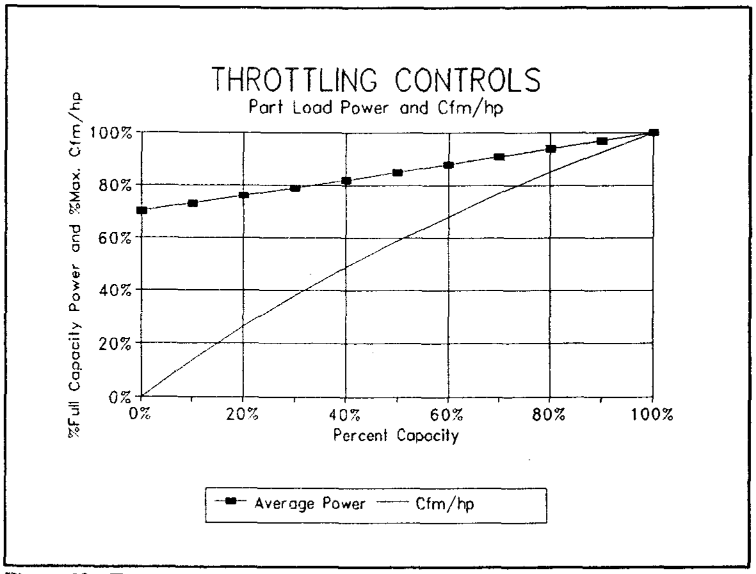

Figure 38: Throttle Energy Use (data from Rogers Machinery Company)

Applications: Throttling is desirable when over-all plant demand is high or erratic, when receiver size is small, or when the acceptable range for system pressure is small. Modulation-only control is a low-risk option because of mechanical simplicity, small pressure variation, and cycling is avoided. Consequently, it is a common control strategy. Throttling is not desirable if extended low load periods are expected.

TURN VALVE CONTROL			
Minimum Storage Capacity	Low Load Efficiency	High Load Efficiency	Controlled Variable
Small	Poor	Excellent	Inlet Volume, V_i

Thermodynamics: The **inlet volume** is the controlled variable. Turn valves control the supply of compressed air by changing the volumetric compression ratio. The inlet pressure remains at atmospheric pressure and the pressure at the discharge port remains at system pressure, so the pressure ratio does not change. Since the compressor undercompresses the air, it essentially becomes a combination compressor-blower. The introduction to this section describes all of these terms.

Mechanics: Turn (or spiral) valve control is another form of modulation. Consequently, no minimum or maximum pressure settings are required. However, the mechanical similarity to throttling ends here. As shown in Figure 13, the "turn valve" itself is composed of a spirally threaded shaft and four or more discrete ports in the compression chamber wall. The shaft lies parallel to the rotors. When plant demand drops below full capacity, the shaft gradually rotates. This opens up some of the ports to atmospheric pressure. Air compression cannot begin until the section of air being rotated through the screws has past the last open port and there are no more escape routes.

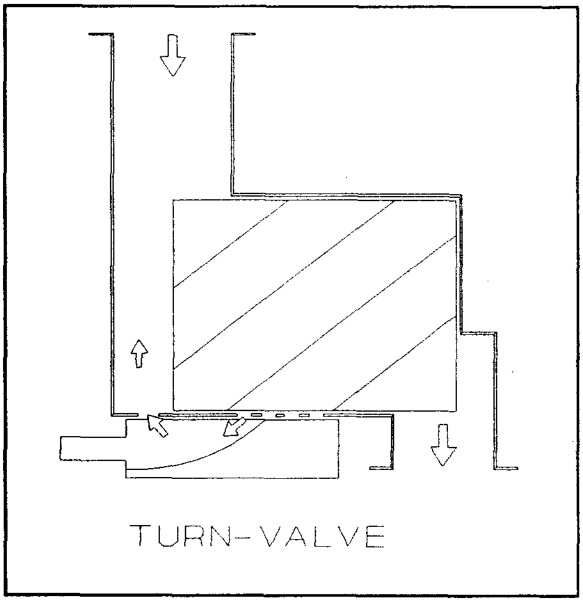

Figure 39: Turn Valve

Since part of the full load compression range is eliminated, the volumetric compression ratio is effectively reduced in order to reduce the air flow rate. Even though the outlet pressure of the trapped air leaving the screws actually is reduced, the "external" discharge pressure at the system port remains at system pressure. Consequently the effective pressure ratio remains constant.

In summary, the pressure ratio and the number of compression cycles per minute remain constant, but the volumetric compression ratio and the mass flow rate change.

Energy Use: This method is more efficient than throttling. However, since the compressor works against system pressure at all times, this is still a relatively energy-intensive control strategy at lower loads. The part-load power curve in Figure 14 shows good performance at high loads, but about 57% of full load power is still required at 0% capacity. If average capacity is known, the part-load power can be calculated from the following parabolic equation:

$$\%P = 57\% + .43 \times \%C^2$$

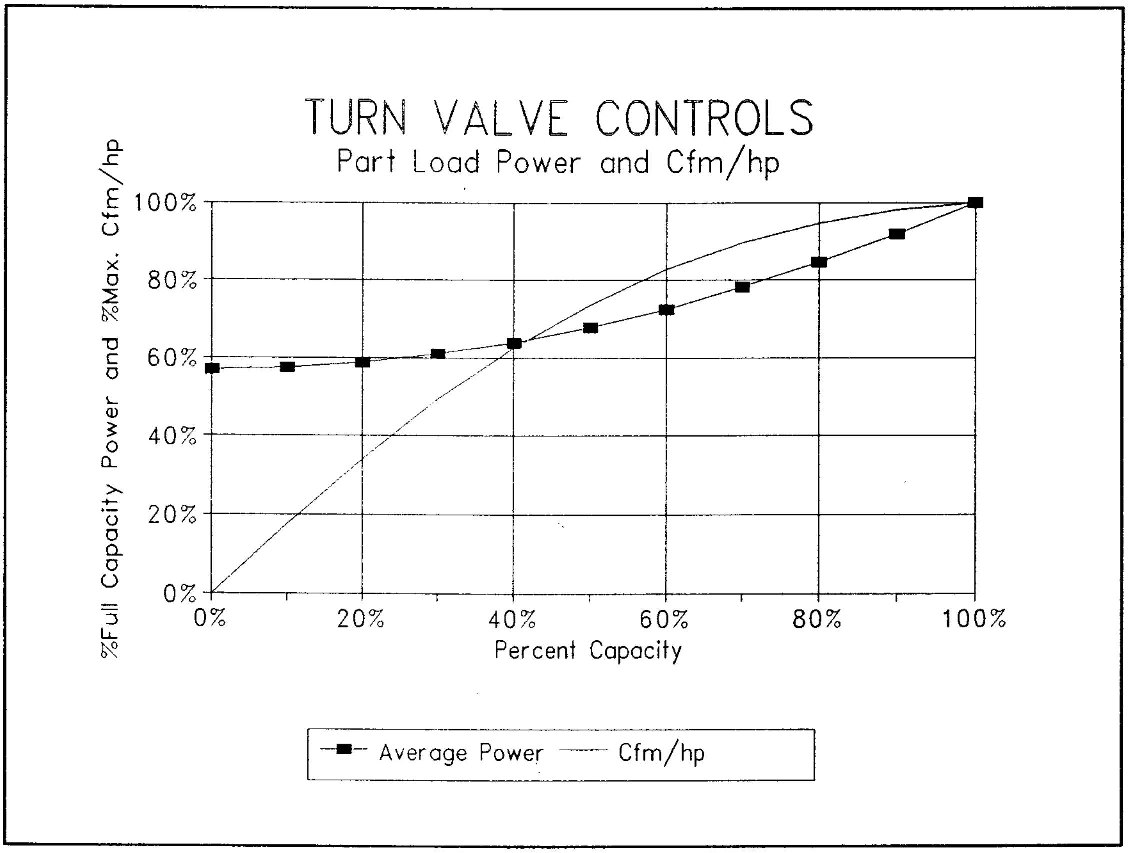

Figure 40: Turn Valve Energy Use, Approximated from Gardner Denver data

Applications: Turn valves are an effective control strategy when over-all plant demand is high or erratic, when receiver size must be small, or when the acceptable range for system pressure is small. Modulation is a low-risk option and consequently, a common control strategy. Though more efficient than throttling, turn valve control is not desirable if extended low load periods are expected.

POPPET VALVE CONTROL			
Minimum Storage Capacity	Low Load Efficiency	High Load Efficiency	Controlled Variable
Small	Poor	Excellent	Inlet Volume, V_i

Thermodynamics: Inlet volume changes. Poppet valves control the supply of compressed air by changing the volumetric compression ratio. The inlet pressure remains at atmospheric pressure and the discharge pressure remains at system pressure, so the pressure ratio does not change. Since the compressor undercompresses the air, it essentially becomes a combination compressor-blower. The introduction to this section describes all of these terms.

Mechanics: Poppet valves certainly sound like fun. Functionally, they operate using the same principle as the turn valve: the volumetric compression ratio is reduced by opening discrete ports in the compression chamber walls, (see Figure 15). But instead of using a single rotating shaft, four or five pneumatic valves open and close to expose the ports and vent the air to the inlet which is at atmospheric pressure.

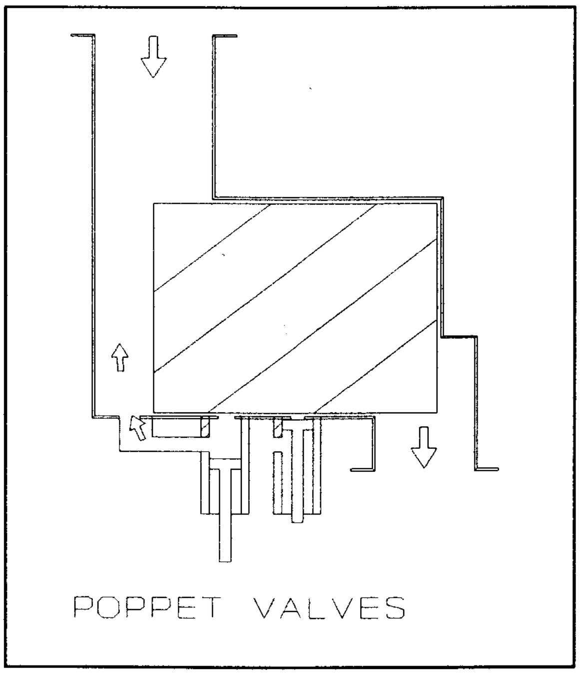

Figure 41: Poppet Valve

Energy Use: Unsuprisingly, the energy use for poppet valves is similar to turn valve control. The thermodynamics are the same; only the mechanics of implementation vary. Poppet valves are more efficient than throttling but the compressor works against system pressure. Consequently, poppet valves are a relatively energy-intensive control strategy. The part-load power curve in Figure 16 shows very good performance at high loads, but about 61% of the full load power is still required at 0% capacity. The power-capacity relationship can be approximated by the following parabolic equation:

$$\%P = 61\% + .39 \times \%C^2$$

Differences in modeled performance between poppet valves and turn valves should not be considered significant.

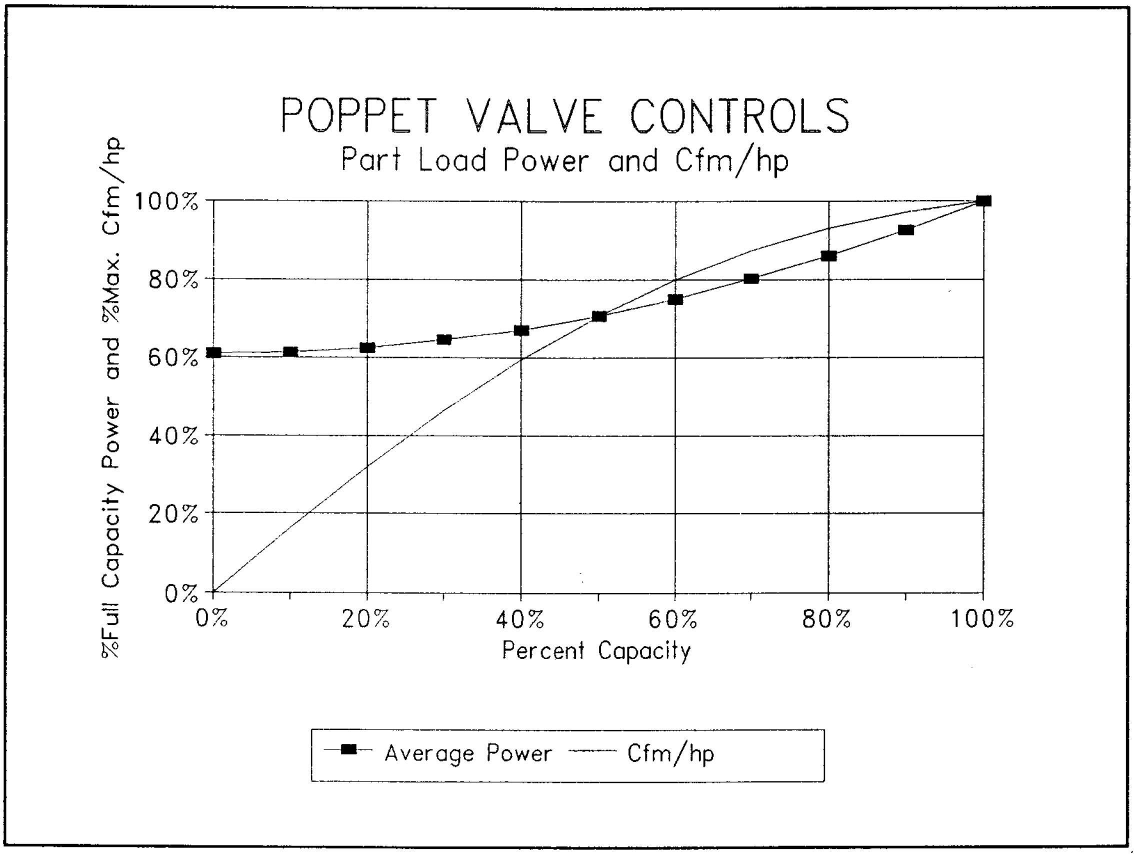

Figure 42: Poppet Energy Use, Approximated from LeRoi data

Applications: It must start to sound redundant, but suitable applications for poppet valve control are the same as those for other previously described modulation only controls.

ON-OFF CONTROL			
Minimum Storage Capacity	Low Load Efficiency	High Load Efficiency	Controlled Variable
Large	Excellent	Excellent	Compression Rate

Thermodynamics: On-Off controls run the compressor at 100% capacity, or shut off it completely. Only the duty cycle changes.

Mechanics: On-off controls are typified by the gas-station compressor (finally, a different strategy than modulation). An air compressor with this type of control requires that a range of acceptable system pressure be tolerable. The compressor will run at 100% capacity, as shown in Figures 17 and 18, until the system pressure reaches the pre-set maximum. Then both the compressor and motor completely shut off. A check valve (CV) prevents the flow of air back through the compressor. At the time of shutdown, an unloading valve (UV) opens so that air at the discharge port is released to atmospheric pressure. The discharge pressure reduction makes it easier for the compressor to restart.

Once the compressor turns off, Figure 17 shows that high pressure air between the discharge port and the check valve will exhaust through a blowdown filter (BF). The filter acts as a muffler. Some control strategies instead will vent the air to atmospheric pressure at the inlet pipe, as shown in Figure 18, and use the inlet air filter (not shown) as a blowdown filter also. In either case, the reverse flow of air through the compressor is avoided. Once enough air is used that the system pressure downstream of the check valve drops to a set minimum pressure, the compressor and motor restart and run at full load.

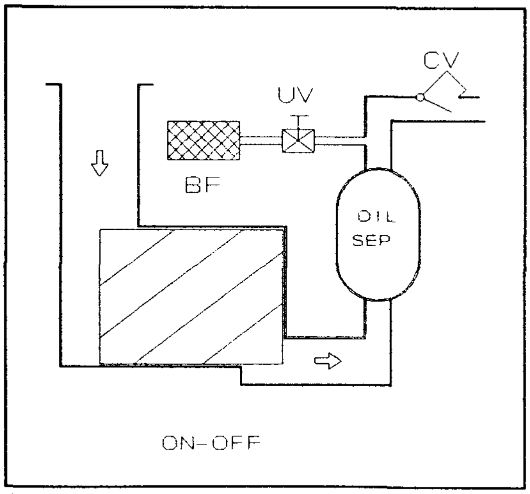

Figure 43: On-Off with Unloading Filter

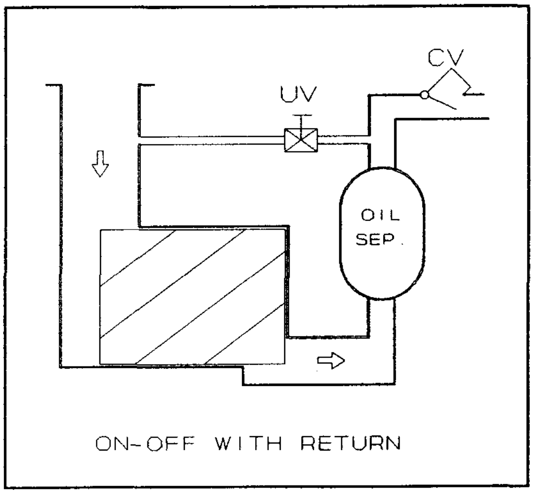

Figure 44: On-off with Return Line

Timers can be incorporated to reduce the number of starts, but this can adversely affect the operating pressure and should be used with caution.

Energy Use: This control strategy is actually the most efficient mode. Since a compressor operating in this mode only produces air while running at 100% capacity and never idles, performance approaches the "ideal," as shown in Figure 19. There will be modest losses because any compressed air that is upstream of the system pressure check valve will be lost once the compressor is shut down. For example, a system with that runs six minutes per cycle will likely have losses of less than three percent. If start-up losses are neglected, power consumption can be modeled by the simple relationship:

$$\%P = \%C$$

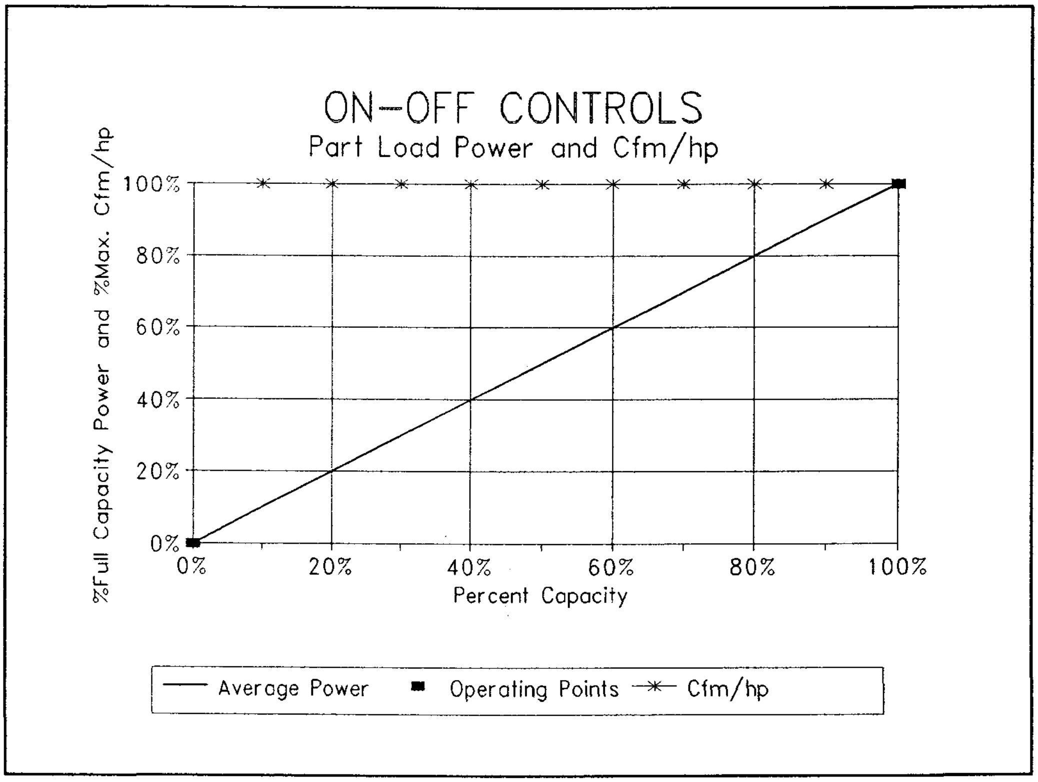

Figure 45: On-Off Energy Use

Applications: This type of control strategy works best when the user is confident that there will be long periods of either very high or very low use, and when the maximum and minimum pressures are not close together. Large receivers are important. Like automobile engine lubrication advertisements say, machinery experiences the most wear and tear during start-up. This method of control is not recommended if the compressor cycles more than once every six minutes.[10] A small plant with an occasionally used sandblaster would be an appropriate application for this control strategy.

LOAD-UNLOAD CONTROL			
Minimum Storage Capacity	Low Load Efficiency	High Load Efficiency	Controlled Variables
Large	Average	Excellent	Inlet, Discharge Pressure

Thermodynamics: Both inlet and discharge pressure are reduced when the compressor idles. The introduction to this section describes these terms.

Mechanics: Load-Unload controls on screw compressors allow the compressor to operate at only two points: fully loaded at 100% capacity and unloaded at 0% capacity. This strategy is similar to On-Off controls except that the motor and compressor never completely shut off. The compressor runs at full power until the system pressure increases to the maximum pressure. Then, an unloading valve at the compressor discharge side opens, and the air leaving the compressor is vented to a lower pressure. Figure 20 shows the controls in an unloaded mode. The most efficient controls use a small oil pump and vent the discharge all the way to atmospheric pressure. However, some manufacturers maintain a pressure of about 30 psig to circulate oil through the compressor while it is unloaded. In either case, a check valve prevents the back flow of air at system pressure. Simultaneously, a valve in front of the intake closes, preventing air from being drawn into the suction port. The butterfly valve shown does NOT modulate in this strategy; it is either fully open or fully closed. In the unloaded condition, the compressor does little work, because it is starved of air at the inlet (near -14 psig) and only working against reduced (0 or 30 psig) pressure at the outlet.

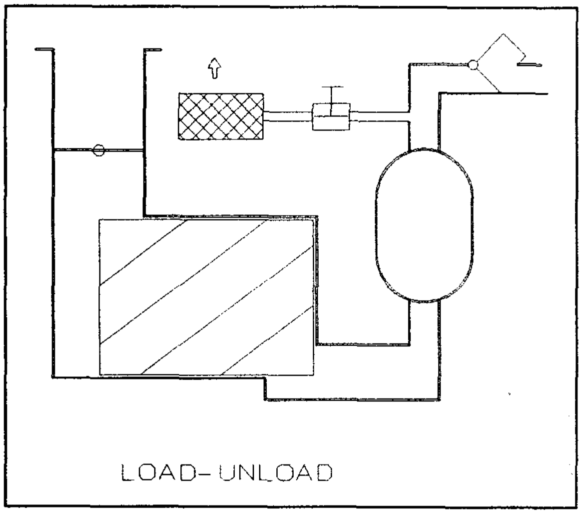

Figure 46: Load-Unload Control

Energy Use: When appropriate, this method of control has very good average energy use characteristics since it only produces air at 100% capacity and idles with low energy use at other times. There will be a small loss of energy each time the outlet blows down, because any compressed air preceding the check valve will be vented to attain a lower pressure. The simplest way to estimate energy use is to ignore these losses and those that may accrue as the intake valve opens and closes. This sounds rash, but the volume of air lost will be less than 2 ft^3 per cycle with most oil separators. If the compressor discharge pressure drops all the way to atmospheric pressure, the Average Percent Full Load Power consumption will be approximately:

$$\%P = 16\% + 0.84 \times \%C$$

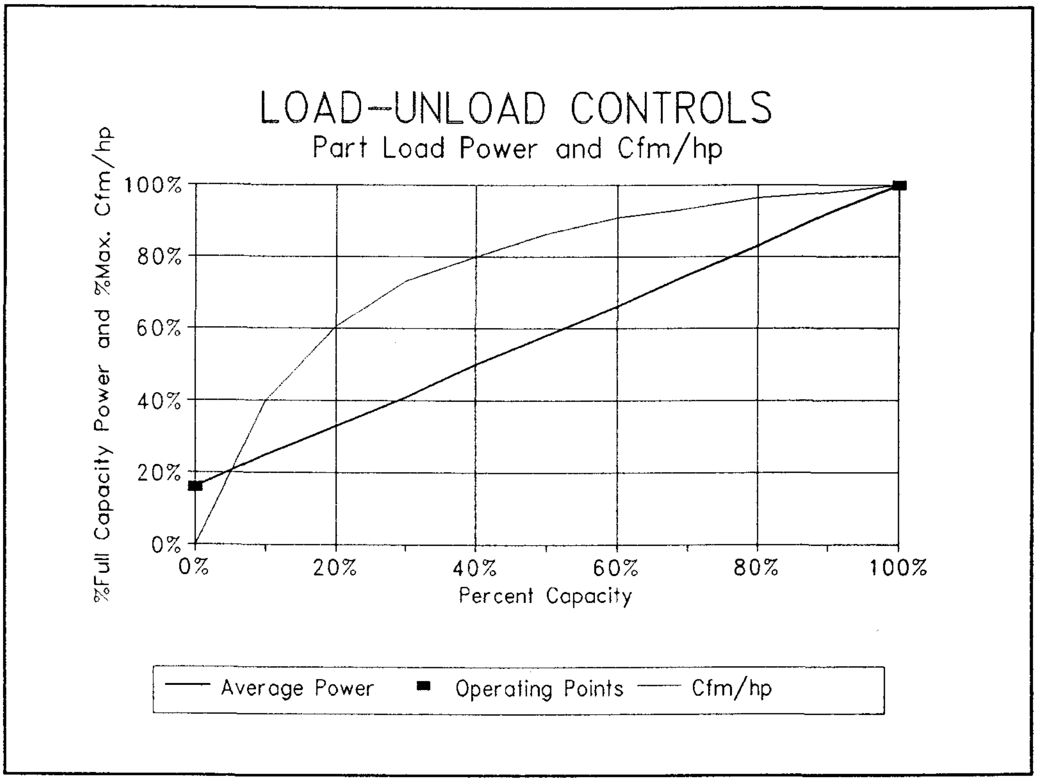

Figure 47: Load-Unload, Data from Rogers Machinery Company

The graph is for a compressor that is completely vented to atmospheric pressure. A partially vented compressor at 0% capacity will normally be near 25% of full load power. The capacity-power relationship will still be linear. Thus, the equation for a 30 psig sump pressure would be:

$$\%P = 25\% + 0.75 \times \%C$$

Applications: Load-unload control is most appropriate when conditions will not cause unloading too often, though it can operate with more unloading cycles than On-off controls can. An on-off controlled compressor would not be suited to restart every 2 minutes, for example, but a load-unload controlled compressor and motor could handle the cycling. A plant with a large air storage capacity and equipment without exacting pressure requirements is ideally suited for load-unload control.

LOW-UNLOAD CONTROL			
Minimum Storage Capacity	Low Load Efficiency	High Load Efficiency	Controlled Variables
Medium	Fair	Good* Excellent**	p_i, p_d V_i, p_i, p_d

*Throttle + Unloading **Turn/Poppet + Unloading

Thermodynamics: During modulation, either the inlet pressure or the inlet volume ratio changes, depending on whether throttling or turn/poppet valves are used. During unloading periods, the inlet and outlet pressures are reduced, like with load-unload controls.

Mechanics: Low-Unload control represents a combination of Load-Unload control and modulation. The modulation may be a throttling valve, a turn valve or a poppet valve. It is recommended that those descriptions be read and understood before proceeding further.

Low-unload controls are designed to modulate during periods of high demand and unload if demand drops below a certain percentage of full load capacity. The unload point may be permanently pre-set by the manufacturer, (40% and 50% settings exist), or it may be manually adjustable, depending on the compressor manufacturer.

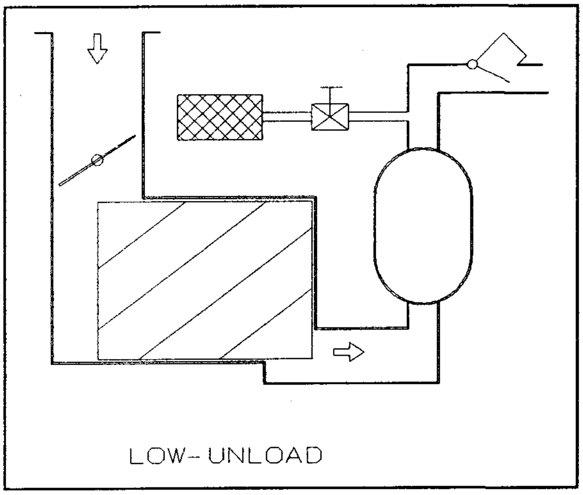

Figure 48: Low-Unload Control

Low-unload hardware looks similar to the load-unload control in the schematic diagram, but modulation is included with the low-unloading. With low-unloading there will also be a maximum and a minimum pressure. The compressor runs at 100% capacity and gradually increases the system pressure (assuming demand is less than 100% of capacity). However, before the system reaches the maximum desired pressure the inlet control starts to modulate. Modulation reduces capacity until it either balances compressed air demand with supply, or until the capacity drops to the set unloading point (% capacity), whichever comes first. If the unloading point is reached, the compressor drops to an unloaded idle condition, as described in the load-unload section, and waits there until system pressure drops to the minimum. At this point, the modulating valve fully opens, the blowdown valve closes, and the compressor returns to full capacity.

Energy Use: Unsuprisingly, the energy efficiency of low-unload controls falls between that of load-unload control and modulation-only control. Since the operating mode varies depending on the magnitude and regularity of the plant air demand, receiver size, pressure range, and unload point, it is not easy to make a simple mathematical model. However, as an example the following simplified form can be used. Assuming inlet throttling modulation is used, the unload point is set at 50%, the compressor completely unloads to atmospheric pressure, and unloading losses are ignored, energy use can be approximated as:

$$\%P = 16\% + 1.36 \times \%C \text{ (Ave. capacity < 50\%)}$$
$$\%P = 68\% + 0.32 \times \%C \text{ (Ave. capacity >= 50\%)}$$

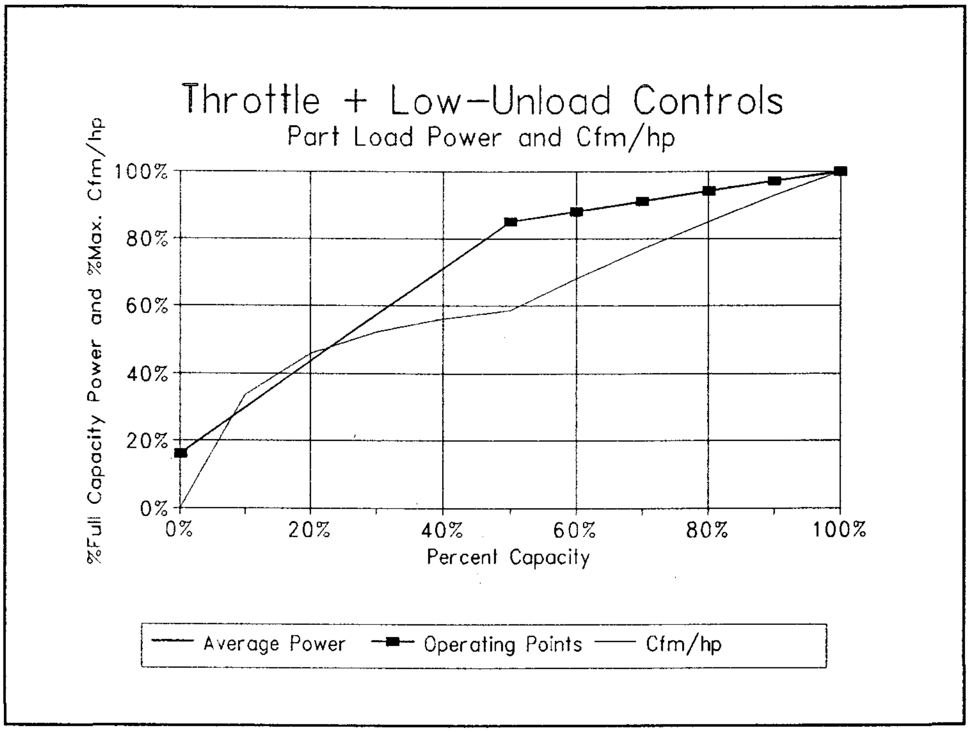

Figure 49: Throttle + Low-Unload Energy Use, Full Unloading

Or, with a turn valve, a 40% unload point, and 30 psig minimum sump pressure:

$$\%P = 25\% + 0.972 \times \%C \text{ (Ave. capacity < 40\%)}$$
$$\%P = 57\% + 0.43 \times \%C^2 \text{ (Ave. capacity >= 40\%)}$$

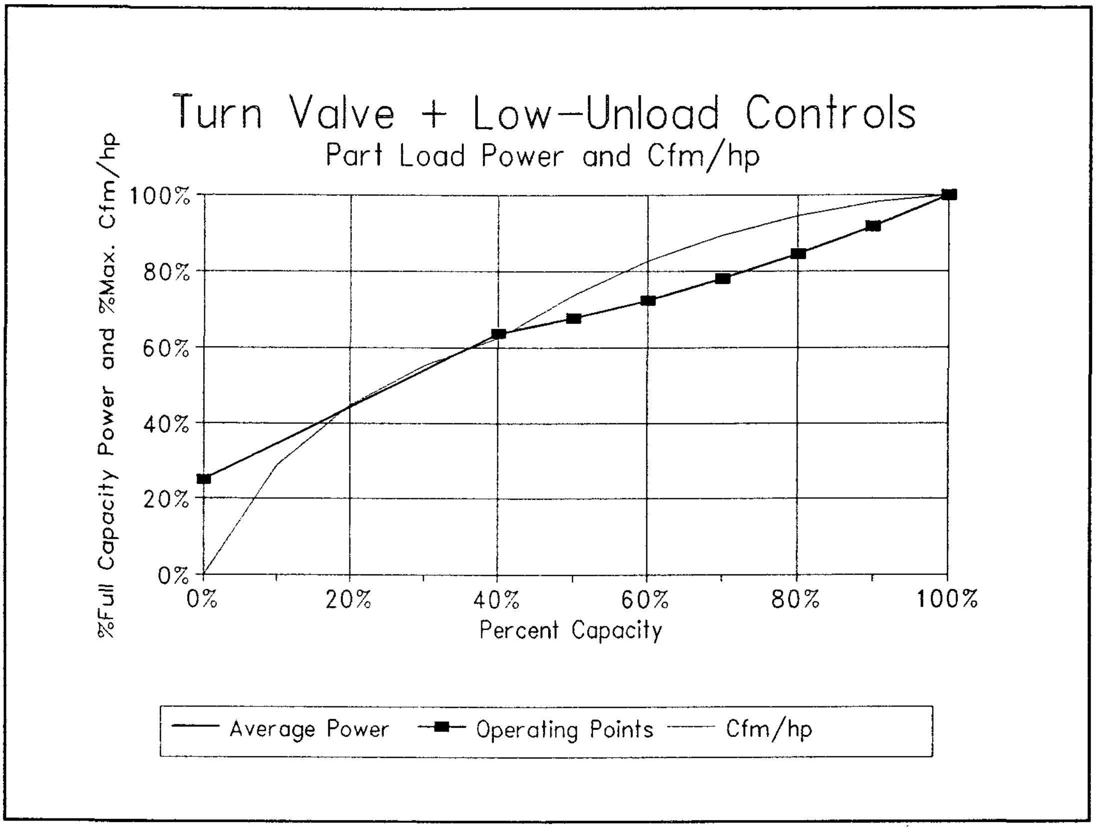

Figure 50: Turn Valve + Low-Unload Energy Use, Partial Unloading

Note: In cases where the unload point is adjustable, it is recommended that the unload point be increased as high as possible, 80% capacity for example, until the cycle time is a short as deemed acceptable for operation. A minimum unloaded time of 30 seconds under normal plant demand conditions is suggested. The energy savings can be seen graphically by looking at Figure 23 and then moving the unload point to the right. The total area under the curve is reduced, and additional savings are possible. Since spiral/turn/poppet valves are already efficient in the higher load region, available savings for this adjustment are less significant for compressors with this type of control.

Applications: Low-Unload control is a good compromise between modulation and load-unload. It does not outperform either of the other two modes if the operating conditions can be clearly defined as matching requirements of modulation or load-unloading. Low-unload control excels when load conditions vary over the course of a day. An example of appropriate application of low-unload control would be a plant where there is a steady high load during first shift, intermittent demand on second, and holding pressure for a fire system at night. In this case, the compressor would mostly modulate during the day, load-unload at night, and mix it up during the evening shift. Since this type of situation is common, this control strategy is frequently the preferred choice.

VARIABLE SPEED DRIVE + MODULATING CONTROLS			
Minimum Storage Capacity	Low Load Efficiency	High Load Efficiency	Controlled Variables
Small	Fair	Excellent	Compression Rate and Inlet Pressure

Thermodynamics: Only the compression rate changes until the minimum compressor speed is reached. At this point throttling begins.

Mechanics: A compressor system with a variable speed drive (VSD) controls the compressed air production rate by changing the rotational speed of the screws. When demand is high, the screws turn faster and displace more air per minute than when demand is low and rotation is slower. The minimum capacity (and speed) has a lower limit of between 25% and 50% capacity.[11] Once the screw drops below the critical speed, seals between the screws and either the cylinder walls or each other will begin to fail and the resulting leakage will prevent proper operation. Therefore a low load strategy must be included with VSDs. Either modulation or unloading can be used to drop capacity without further slowing the screws. This guidebook only considers modulation, because the biggest advantage VSDs seem to offer is that they offer modulation down to low loads while maintaining efficiency.

Energy Use: Like low-unload controls, there are two different operating modes. During periods of high demand, the VSD controls the modulation. If the average load drops below the minimum level that can be compensated for with the VSD, the compressor will stop reducing speed and reduce capacity further with either a butterfly, slide, turn, or poppet valve arrangement. An example situation would be a plant that has scaled back operation and has an oversized compressor, but has tight pressure requirements.

Using throttling and assuming 35% minimum VSD functional capacity as an example, the energy use equations are:

$$\%P = 35\% + 0.46 \times \%C \text{ (Ave. capacity} < 35\%)$$
$$\%P = 24\% + 0.76 \times \%C \text{ (Ave. capacity} >= 35\%)$$

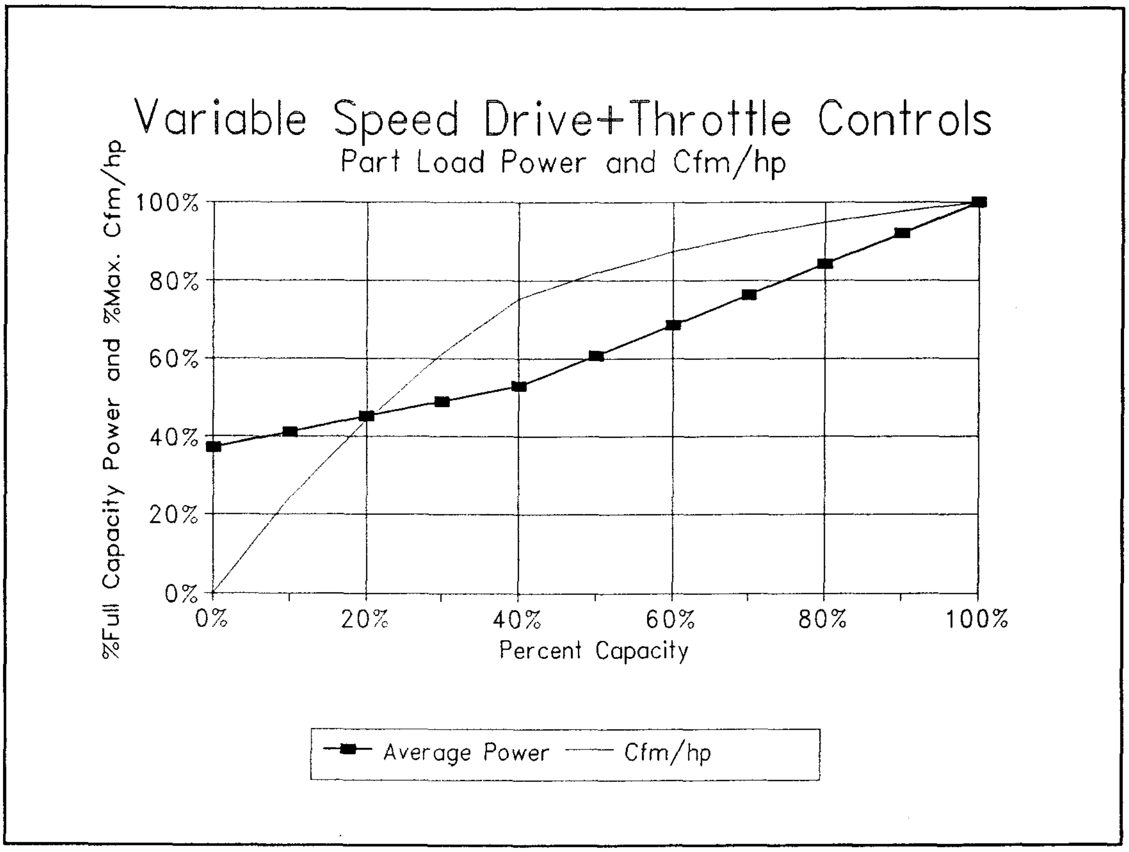

Figure 51: VSD with Throttle, modified from DOE bulletin CS/40520-T2

Applications: VSDs are the most expensive of all of the control options and are not typically offered by compressor manufacturers as a standard configuration. When combined with modulation, however, VSDs have the singular advantage of offering relatively high efficiency across the full capacity range with the convenience of full modulating control. VSDs would be appropriate when extended low demand periods are expected and a range of supply pressures is unacceptable.

CONTROL STRATEGY SUMMARY

The following table summarizes headers for each control strategy. Efficiency values are provided to help quantify relative performance for part load conditions. They represent the percent of maximum possible cfm/hp required to compress air if demand operated uniformly between 0 and 50% load (low), and between 50 and 100% load (high).

Control Strategy	Minimum Receiver Size	Low Load Efficiency	High Load Efficiency	C. V.+
Throttling	Small	Poor 31%	Good 80%	p_i
Turn Valve	Small	Poor 39%	Excellent 90%	V_i
Poppet Valve	Small	Poor 38%	Excellent 88%	V_i
On-off	Large	Excellent 100%	Excellent 100%	rate
Load-Unload	Large	Average 57%	Excellent 94%	p_i, p_o
Low-Unload (Throttle)	Medium	Fair 44%	Good 80%	p_i* p_i, p_d**
Low-Unload (Turn/Poppet)	Medium	Fair 41%	Excellent 90%	V_i* p_i, p_d**
VSD+Throttle	Small	Fair 48%	Excellent 92%	rate p_i

*when modulating
*when unloading
+Controlled Variable(s)

Each control strategy has strengths and weaknesses. The job of the equipment purchaser is to determine which ones are most appropriate for the application being considered. Figure 26 shows the relative efficiency for most of the control strategies described above, combined on one graph.

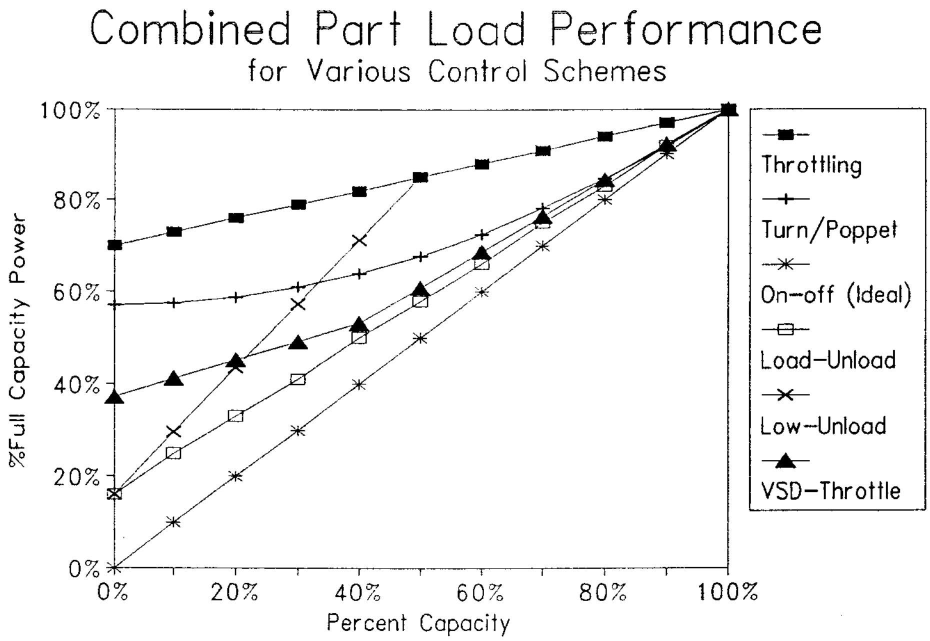

Figure 52: Combined Part Load Performance

If efficiency were the only issue, it would be simple to look at the graph, determine the most frequent load condition expected, and purchase a compressor with the most efficient controls available at that condition. Of course, extenuating circumstances affect every decision. Receiver capacity is important. If the maintenance department is understaffed, modulating compressors may merit extra consideration based on their simplicity. Or, if the staff is not highly trained, the issue of microprocessor-based controls becomes more important. Sometimes the local manufacturer's representative for a particular brand may have earned brand loyalty due to exceptional support in the past. Obviously, these issues do not show up in mathematical formulas.

Nonetheless, a compressor system will cost much more to operate over its lifetime than to buy. And it won't always run at full load, so part load efficiency must be considered an important decision-making criteria.

The decision-making tree shown in Figure 27 may help in the decision-making process. It is based on the assumption that energy efficiency is the most important issue. When using the tree, keep in mind that adding receiving capacity will almost always pay for its cost when part load conditions are expected and modulating control would otherwise be the only option. It also may be a worthwhile investment to buy a receiver to make load-unload operation possible when low-unload operation would otherwise be chosen.

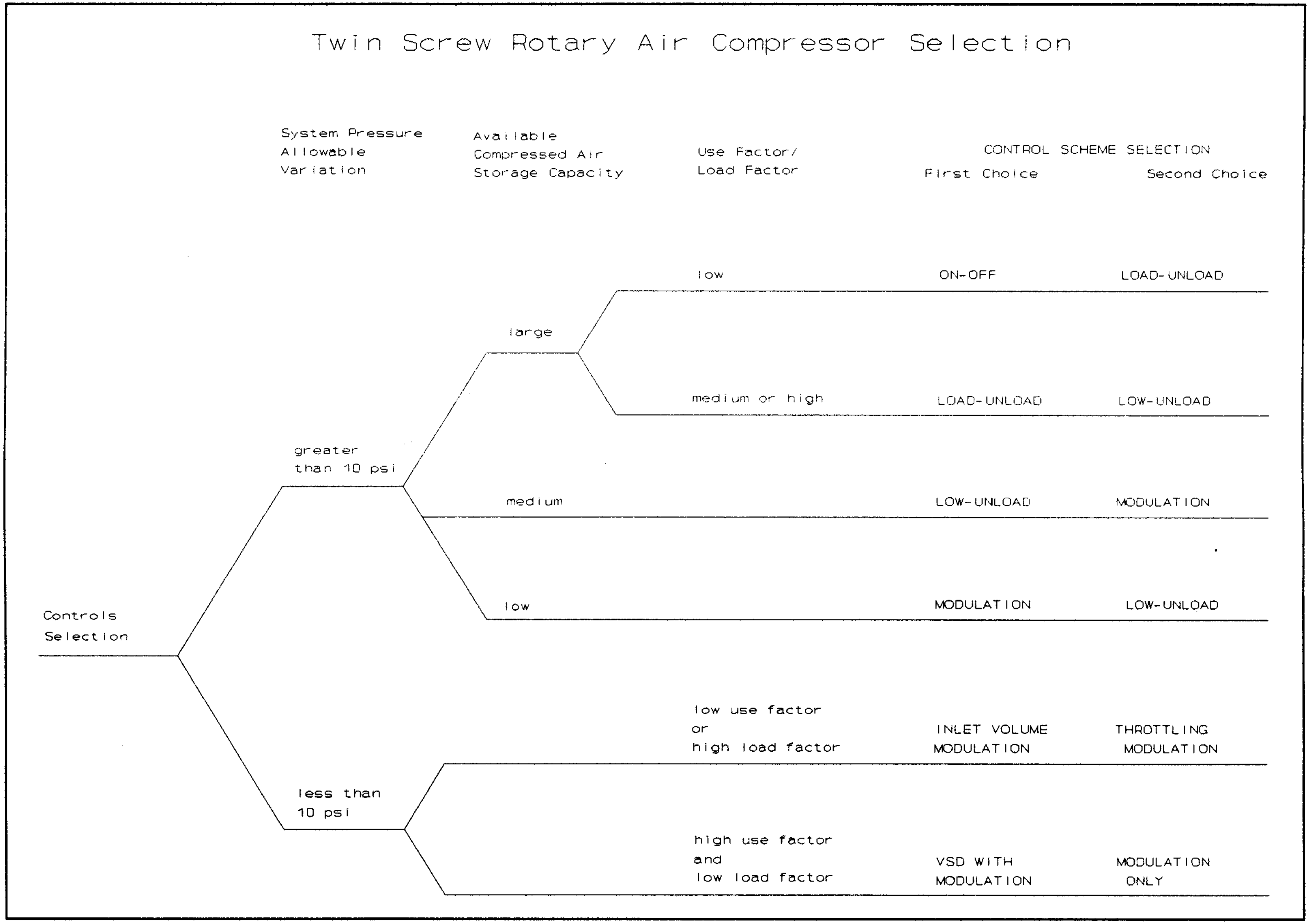

Figure 53: Compressor Control Selection Guideline

A WORD ABOUT SEQUENCING

Compressor sequencing control is a whole subject in its own right, and several manufacturers offer elaborate microprocessor-based controls to operate multiple compressors together. The primary objective of sequencing is to deliver the proper quantity of air at the desired pressure. The secondary objective is to do this in an efficient manner. If you are not fortunate enough to have a magic black box to make decisions for you, the following tips can serve as guidelines:

- Use the compressor with the best part load efficiency characteristics as the "top" or "swing" compressor. For example, arrange the compressors so that modulation controlled compressors always run at full load or are off, and a reciprocating compressor or screw with unloading runs at part load to meet demand.

- Use compressors with the best full load cfm/hp as base compressors. Obviously, this rule will interact with the previous one if the compressor with the best part load control is also the most efficient compressor at full load.

- Meet demand requirements as closely as possible. For example, if you have a 50 hp,a 150 hp, and a 300 hp compressor with a load that can be met by running the smaller compressors together, use them rather than the 300 hp compressor alone, because the smaller compressors will run closer to full load and avoid poor part load performance by the large compressor.

- Sequence compressor use so that the fewest number of compressors actually operate at any given time. The previous recommendation gets priority to this one, but does not necessarily contradict it. For example, if you have 175 hp of plant air demand and have one 200 hp, and two 100 hp compressors available, energy consumption will normally be less if the single 200 hp compressor is used. This is because the volumetric efficiency (cfm/hp) is frequently better for larger compressors. This type of decision-making can be challenging as the number of available compressors increases.

- All of the above being stated, it is also healthy to ensure that all compressors are rotated for use regularly. Just like cars, compressors seem to be happiest when they are used at least occasionally.

- Do not operate more than one compressor at part load. Setting the maximum pressure on one compressor to be slightly lower than the others is an easy way to meet this objective. Two compressors operating at part load may disguise the fact that one of them could be turned off. Also, multiple compressors with unloading controls probably will not operate properly if their set pressure ranges overlap.

- Avoid starting all of the compressors at one time. It is hard on the distribution equipment.

A WORD ABOUT MICROPROCESSOR-BASED CONTROL

Computerized controllers currently do not include actuating devices other than those already described. They do offer options such as performance monitoring and maintenance scheduling. More significant for this book, electronic devices offer logical options for part load control that are not otherwise available.

One manufacturer offers a controller that can switch from load-unload mode to throttling mode depending on demand. Another design calculates demand and directs the compressor to switch between load-unload and on-off control. Timers are also more likely to be incorporated when microprocessor-based control is used.

Compressor sequencers are often computer controlled when the desired sequencing logic cannot be simulated with pneumatic controls. While electronic controllers are not yet installed on all single compressor systems, their popularity is growing. More manufacturers are offering microprocessor controls as standard equipment, especially on large (>200 hp) horsepower systems.

COMPRESSED AIR ENERGY CONSERVATION OPPORTUNITIES

There are basically three parameters that can be adjusted to reduce the average power required by an air compressor: required system pressure, amount of air required, and equipment efficiency. First, the effects of each parameter are described, and then a list of ideas to reduce energy use are offered.

Reduce Air Demand

Compressed air is an expensive commodity. Any action taken to reduce the overall plant demand will save money. As a rule of thumb, screw compressors commonly require between 0.2 and 0.3 hp/cfm at 100-125 psig and full loaded.

Measures

Fix leaks: Air leaks are the pneumatic equivalent of leaving lights on. Leak repairs are generally a cost-effective use of time. Leaks are most easily found during breaks or when the shop is otherwise quiet. Ultrasonic detectors that can find leaks based on sound are available for less than $2,000. Plumbing connections, hose fittings, pressure regulators, line and control valves, valve packings, hoses, and pneumatic cylinders are all common criminals. For example: a 1/8" hole in a 100 psig air line will release 26 cfm and cost $763 /yr, based on a $0.02 /kWh energy cost, a $4.00 /kW-mo demand cost, two shift operation, and a 0.3 hp/cfm compressor efficiency. Other leak rates are shown below.[12]

Hole Diameter	1/64"	1/32"	1/16"	1/8"	1/4"	1/2"
Leak Rate (cfm)	0.406	1.62	6.49	26.0	104	415

Engineered nozzles: Perhaps the most common use of compressed air is the blowdown of dirty equipment or people. Engineered nozzles offer two main benefits: First, they can provide the same air velocity as an open hose with a significantly lower volumetric air flow rate, much like a water-saver shower head. Second, they often have quick-release grips, which means that air is only delivered for the instants it is needed. Washing your car certainly requires less water when a squeeze grip handle is used. The same applies for air use.

Reduce branch system pressure: If all equipment and processes that draw air from a branch of the main air supply line have minimum pressure requirements lower than the minimum system pressure, the air pressure can be dropped for the entire branch. This saves money in two ways: first, any leaks in that particular branch will lose less air compared to a higher line pressure; and second, the equipment uses no more air than is needed.

Capacity-controlled regenerative dryers: Regenerative dryers employ two tanks that contain desiccants, chemicals used to attract moisture from the air. The tanks alternate, with one tank drying air while the other is being reactivated. Reactivation is performed by sending clean dry air through the off-line tank and ejecting it. Normally regenerative dryers operate on a timer. However, since the regeneration process can use a considerable quantity of compressed air, it is beneficial to perform the regeneration only as often as necessary. Installing a moisture-sensing device instead of a timer to prevent unnecessary regeneration cycles and effectively reduce plant air demand.

Isolate fire system: If your fire system requires compressed air, then setting up a dedicated air line can save money by avoiding leaks that exist in the rest of the air system. Since leaks won't be fed during shutdown periods, substantial savings are possible.

Reduce Required Discharge Pressure

As a rule of thumb, 1/2 % of full load power can be saved for every psi the discharge pressure can be decreased. Savings will be slightly less, about 0.4%, for two stage screw compressors because they are more efficient to begin with.[13]

Measures

Reduce *minimum* system pressure: Set the minimum system pressure so that it delivers air at the minimum acceptable pressure level to the user. If the pulley ratio between the motor and compressor is also adjusted, this has a potential benefit of increasing compressor capacity as well. For example, a 150 hp compressor may be rated to deliver 620 scfm at 125 psig but it can deliver an additional 140 scfm by decreasing the discharge pressure to 100 psig and increasing the compressor speed.

Reduce pressure drop across oil separator/dryer/filter: Every obstruction to a straight path for compressed air leaving the compressor will drop the system pressure relative to the compressor discharge. A corroded separator element, clogged drying coils, a dirty drying filter, plugged or faulty automatic drain valves, or ignored manual liquid drain valves will cause an unnecessary pressure drop before the air ever gets to the receiver. To avoid this, check equipment and replace expendable parts according to scheduled maintenance. If there is a significant amount of equipment downstream of the compressor, you can install a differential pressure gage between the receiver and the discharge port of the compressor to make it easy to monitor the pressure drop. In most cases, the pressure should not drop by more than 5 psi (unless unloaded, of course).

Install larger pipes: There will always be a pressure drop between the compressor and the end user because of pipe friction. If the air lines are too small because of increased air use in an area (or perhaps low-bid contracting), the compressor will have to discharge air at an unnecessarily high pressure to deliver the desired pressure at the demand source. Large pipes also improve operations as described in other recommendations on local manifolds and increasing receiver capacity.

Looped piping: Air lines throughout the plant should form a closed loop with take-offs for individual areas. Looping effectively doubles pipe capacity in terms of response to demand. Consequently, the pressure drop due to pipe friction can decrease significantly. Looping has the additional benefits of balancing air lines and improving the efficiency contribution of remote receivers.[14]

Local manifolds: A local manifold in a work area acts as a mini-receiver for that area. This is beneficial because it helps minimize the effect of one user's air demand on other user's supply pressure if the two users are close together. This can save energy as well as provide more desirable working conditions because the first action the users will take to remedy low pressure problems is to march over to the maintenance shop and demand higher pressure air so he or she can get the job done.

Reduce pressure at night: Since high pressure air is more expensive than low pressure air, its is beneficial to drop the system pressure whenever there is no reason to maintain a high pressure. For example, if a day-shift-only piece of equipment requires 120 psig air and the machinery used for all three shifts only needs 90 psig air, then 10% (2/3 shifts x 30 psi x 0.5 %/psi) savings can be realized by dropping the system pressure when the day-shift equipment goes off-line. This is not necessarily the most practically implemented course of action, but it can be done.

Equipment Related Opportunities

A diverse collection of actions can be taken to improve equipment efficiency and performance.

Measures

Install low-unload controls: Since modulating-only controls can be so much less efficient than methods that employ unloading when capacity is low, there is often significant available savings from installing unloading controls on the existing compressor. For one manufacturer, low-unload controls can be installed for less than $1,500. As an added option on new equipment, the additional cost will likely be less. When appropriate and used as designed, payback time is commonly measured in months.

Increase receiver capacity: All compressors that unload or turn off experience cycling losses, as alluded to repeatedly in previous sections. Losses are primarily caused by the release of compressed air in the oil separator, compressor discharge pipe, and any other space upstream of the main check valve that precedes the aftercooler. Reduced efficiency during unloading and reloading also occurs, because the compressor is likely to operate briefly in a part load condition. On-off controls will endure potential life reduction and modest energy losses every time the motor restarts due to temporary high input current. Perhaps most importantly, increasing the air storage capacity can significantly help improve the applicability of the first measure described above. Adding air storage capacity can improve equipment performance and product quality by reducing the rate of change of supply pressure. For example, an airbrushing operation, can be very sensitive to quick pressure changes.

Reheat air: The delivery of compressed air is fundamentally an energy transfer process, little different than electricity. Thus, adding heat to the air increases its energy value. When considered from this perspective, conservation measures such as capturing heat energy from the compression or drying processes becomes quite understandable. There are many potential heat sources; pre-cooled oil from the oil separator sump and compressed air are the most commonly captured heat sources. However, waste heat from industrial processes other than the compressor system also can be considered as a source of energy for the compressed air. In a typical 100 psig system, 15%-20% of the rejected heat can be recovered as compressed air energy.[15] An additional benefit is that the relative humidity of the compressed air will be significantly lower - 8% compared to 100% relative humidity if the temperature is raised from 70°F to 160°F. Reheating is appropriate for many applications, but may be less cost-effective when air demand is very low compared to storage capacity, or if the receiver is located outside in a cool climate and the reheater must precede the receiver.

Heat recovery: If the heat energy from the air and oil cooling operations cannot be returned to the compressed air, you may be able to use the rejected heat to warm ambient plant air in the winter. One company makes a heat recovery unit that will thermostatically regulate the temperature of exhaust sent air to the plant at a comfortable level in the winter, and reject the hot air outdoors during warm weather conditions. Claimed savings is about 5.5 MMBtu/hp/year based on three shift operation.

Install Air intake in cool location: In seeming contrast to the above opportunities, it is beneficial to draw air into the compressor from the coolest possible location. Under atmospheric conditions, air becomes denser as temperature drops. Since the required pressure does not change, the compression ratio can be reduced without reducing the flow rate or discharge pressure of the compressed air. As a rule of thumb, you can save 1% of compressor demand for every 10°F that the inlet air temperature is reduced.[16]

Efficient motors: This is an easy one to understand. Since any 50 hp motor will deliver 50 hp to the shaft, it is obvious that you will save money if less energy is required by the motor to deliver that 50 hp. Many compressor manufacturers offer the option of specifying an energy efficient motor instead of a standard efficiency motor when purchasing a new compressor. The economic decision is easy if payback is considered relative to the life of the motor - always buy an energy-efficient motor - but the payback issue is not nearly as straightforward when payback time is measured in years. A simplified rule of thumb for a user with a $0.04 /kWh average energy cost is that the additional cost of a high-efficiency motor to pay for itself within three years if the compressor is used an average of 10 hours/day or more. The same logic can be used when replacing or rewinding a burnt out motor, but the numbers will vary. Replacing a working standard efficiency motor with a high efficiency motor is rarely a good investment. However, an easy way to save compressor energy savings with an efficient motor is to find an efficient motor of the same frame size and horsepower elsewhere in the plant that is not used as often as the compressor motor. Then, swap the motors. You have to be pretty lucky to find a match, but if possible, this action requires no capital expenditure.

Efficient drive belts: Forget this if the motor is directly coupled to the compressor. No belt can transfer the energy of the motor shaft to the compressor shaft with 100% efficiency. Standard V-belts lose most of their energy by slipping around the sheave. The friction causes heat generation and energy loss. There are two alternatives to standard V-belts. Notched V-belts fit on a regular pulley but are approximately 2% more efficient than standard V-belts (source). The notches help the belt bend around the pulley more easily and reduce slippage. They cost slightly more than regular belts, but the reduced slip, reduced heat generation, and better dissipation of heat that does get generated help notched belts last longer than standard belts. They are therefore a smart choice in many applications, including compressors.

The second alternative is installing a high torque drive. This is a more expensive option because high torque drives require special matched pulleys. They work more like a chain than a conventional V-belt, and slippage is eliminated. A high torque drive will be 4-8% more efficient than a standard V-belt and can be economical for compressors that have high operating hours.

Clean intake filter: It is always desirable to maintain a clean air filter at the compressor intake. An oily or clogged filter can reduce compressor capacity by causing a partial vacuum to form between the filter and the compressor intake. This could cause the unnecessary purchase of additional air compressors (see maintain throttle below). Some manufacturers include a vacuum gauge that indicates the pressure drop across the filter. You can also install a gauge yourself.

Combine air lines: If the plant currently uses two or more independently operating compressed air circuits operating at approximately the same system pressure, it is frequently beneficial to combine them into a single system with multiple sequenced compressors. Savings are especially high when the involved compressors typically operate at part load. Savings will be low if one of the loops requires near full capacity from its compressor.

Increase unload point: Unless the receiver is too small, load-unload type controls are typically more efficient than low-unload controls. This is because the power required for modulation is higher than the average power required when the compressor loads and unloads. Load-unload controls don't modulate at all while low-unload controls do. Therefore, if adjustable low-unload controls are set with a higher unload point, the compressor will modulate less often during each cycle. Furthermore, a higher unload point will cause the compressor operate in the more efficient cycling mode up to a higher percentage of full load capacity. The only notes of warning necessary here are that undersized receivers can cause the compressor to reload too soon at high loads, and that some vacuum switches used to measure capacity can be unpredictable if the unload point is set above 80%.

Maintain modulating valve performance: Screw air compressors are low maintenance, dependable pieces of machinery, but the modulating valves can gradually lose their accuracy without any obvious indication of deterioration. One consultant estimated that over 80% of all throttling valves fail to maintain their designed behavior.[17] If your compressor has a capacity gauge, you can judge throttle performance easily, by first opening a relief valve so that maximum compressor capacity is required and then intentionally valving off the compressor discharge completely. If the gauge does not range from 100% to 0% capacity, then the valve is not working properly (or the gauge has failed). The most disastrous energy cost from this problem occurs when plant demand appears to exceed the capacity of a single compressor so a second one is unnecessarily added. The capital cost isn't particularly pleasant either.

Use blowers for low pressure applications: Up to about 10 psig, blowers are a more efficient means of providing compressed air than by reducing plant air pressure down from 90+ psig to the desired pressure. The unnecessary compression and decompression costs money. This situation commonly occurs when a liquid requires non-mechanical agitation. It is much easier to tap a line off of an existing air line than to go to the trouble of purchasing a blower. However, using a blower is frequently more economical.

Downsize at night: Often, the main plant air compressor is used to maintain pressure in a dry fire system. If manufacturing stops after one or two shifts, a much smaller compressor can maintain pressure for the fire system without all the inefficiencies of low part load use. The same logic can apply if third shift demands are much lighter than first shift. For maximum savings, be sure to eliminate as many leaks as possible before sizing the small compressor. If the leak load is a large percentage of the total load, this opportunity won't work.

APPENDIX B: COMPRESSOR PROPORTIONAL CONTROL DEVICE

This appendix explains the operation of a pilot valve-based throttle modulation system. Figure 54 is a schematic representation of the control equipment that is involved in throttling modulation.

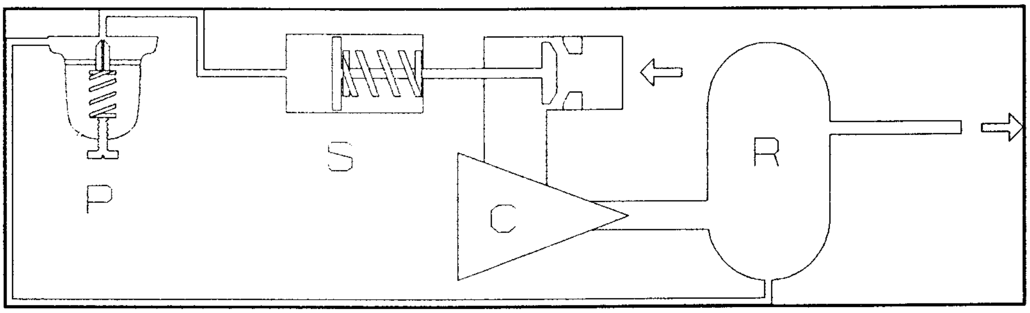

Figure 54: Proportional Control Schematic

The main components are a bell-shaped pilot valve shown on the left (P), a slide valve (S), the compressor (C), and the receiver (R). The controller works as follows:

The desired receiver pressure is set using the screw on the pilot valve. Depending on the screw position, the pilot valve spring will exert more or less force against the diaphragm in the pilot valve. The small air line running from the receiver to the pilot valve provides pressurized air that supplies a force acting downward on the diaphragm. As these two forces act in opposite directions, a needle valve opens or closes and controls a supply of pressurized air to the slide valve. For example, when pressure from the receiver is high, it compresses the pilot spring. The result is high pressure supplied to the slide valve.

The slide valve opens and closes depending on the pressure supplied by the pilot valve. Continuing with the previous example, high pressure is supplied to the slide valve. The piston moves the plunger to the right and closes off the supply of air to the compressor. The throttling action causes a partial vacuum to form between the plunger and the actual compressor inlet because the compressor is trying to suck more air in than is available. Once the supply is reduced, the output is reduced, and the pressure in the tank will slow and eventually stop increasing.

Conversely, if air demand increases, the pressure in the receiver will drop, the pilot valve will close, the pressure supplied to the slide valve will drop (air bleeds out through the needle valve and out a small calibrated orifice), the plunger will open, and the compressor will provide more compressed air. There is no reason that the controller should overcompensate for a change in demand.

APPENDIX C: OPERATION OF COMPRESSOR LOW-UNLOAD CONTROLS

This appendix chronologically describes the sequence of operation for a QNW B-15 air compressor from the time the compressor is first turned on until it completes one low-unload cycle of operation. This is the type of compressor used during testing and analysis for this project. Most analysis and modeling in this thesis is based on the premise that controls of other manufacturers will behave similarly, even if hardware details vary. **In the control diagrams, numbers are used to indicate the sequence of operations. Duplicate numbers imply simultaneous actuation.**

0 psig everywhere; power off. Figure 51 shows the controls as they are positioned when no power supplied to the controls. The compressor intake, discharge, and the oil separator are all at atmospheric pressure (0 psig). Though the receiver could maintain pressure while the compressor is turned off, the receiver shown in the diagram also is unpressurized.

$P_{sys} < P_{min}+3$ and rising; full load. As soon as the compressor is turned on, the control power supply is energized (1). Due to the normally closed position of the vacuum switch and minimum pressure switch, the blowdown valve and unloading valve are simultaneously energized (2). The blowdown valve prevents flow of air from the oil separator back to the inlet. The inlet valve prevents pressurized air in the receiver from being supplied to the pressure regulator and slide valve. The check valve is opens while pressure builds in both the oil separator and the receiver (3). Figure 52 shows the state of all components from the time the compressor is turned on until the pressure exceeds the minimum pressure (P_{min}) + about 3 psi.

P_{min} + 3 is significant because the minimum pressure switch has a make-break span of approximately 3 psi. This means that the switch will be "closed" any time the receiver pressure is below p_{min}, and when pressure has dropped below

p_{min} and exceeds p_{min} + 3. Conversely, the switch will be open whenever pressure is above p_{min} + 3 and when pressure has risen above p_{min} + 3 and is still above p_{min}.

P_{min}+3 < P_{sys} < P_{max}-2 and rising; full load. Once the receiver pressure exceeds p_{min} + 3, the switch opens (4) and the indicator light is lit (5), as shown in Figure 53. Nothing further happens at this time because the unloading and blowdown valves are still energized by the vacuum switch line, which is parallel to the minimum pressure switch.

P_{max}-2 < P_{sys} < P_{max} and rising; throttling. The controls remain static until throttling begins. As shown by Figure 54, throttling begins somewhere around p_{max} - 2. At this point, force on the pilot valve diaphragm due to air pressure from the oil separator control line begins to exceed the opposite force set by the spring and screw on the pilot valve (6). This gradually opens a needle valve that supplies reduced pressure through an free sliding shuttle valve (7) to the slide valve (8). As the needle opens further, more pressure is supplied to the slide valve. As the slide valve closes, a partial vacuum forms between the slide valve plunger and the compressor inlet. This reduces the mass of air drawn into the compressor and reduces the compressor's discharge flow rate.

P_{min} < P_{sys} < P_{max} and dropping; unloading. If plant demand is high, the controls will stabilize in this mode and the compressor flow rate will match demand. If demand is low, however, the pressure will rise and close the slide valve further until the partial vacuum exceeds the value set on the vacuum switch. This is shown in Figure 55. The vacuum switch opens (9) and the vacuum indicator light turns on (10). Both the vacuum and minimum pressure switches are now open and power is not available for either the unloading or the blowdown valve. The blowdown and unloading valves open (10), and the compressor unloads. Once the blowdown valve opens, all of the air upstream of the check valve is released through a control line and the air filter to the atmosphere. Sometimes a dedicated air filter is used (not shown). The pilot valve closes (11) because the control pressure drops. Since it is desirable to close the slide valve when air from the compressor discharge is vented, the unloading valve opens and allows air from the receiver to pass through a pressure regulator. The air in this line forces the shuttle valve over (12) so that the a supply of pressurized receiver air can be used to keep the slide valve closed (13). With the slide valve closed and the blowdown valve open, oil separator pressure drops below that of the receiver, and the check valve closes to prevent reverse flow (14).

$P_{min} < P_{sys} < P_{min}+3$ and rising; reloading. With the compressor unloaded, receiver pressure will gradually drop due to plant demand until the minimum pressure switch closes (15), (see Figure 56). Once this switch closes, the minimum pressure indicator light goes off (16) and the blowdown and unloading valves are energized (16). Once the unloading valve closes the pressurized air previously used to hold the slide valve closed is released to the atmosphere. The slide valve opens (17), and the shuttle valve relaxes after the pressure in all lines balances out to 0 psig (17). As the slide valve opens, the vacuum at the intake is reduced, and the vacuum switch closes (18). This causes the vacuum indicator light to turn off (19).

The controls have now prepared the compressor to supply air to the receiver again. First, pressure in the separator builds up to the receiver pressure. Finally, the check valve opens (20) as air is delivered to the receiver. At this point the cycle repeats, starting with Figure 52.

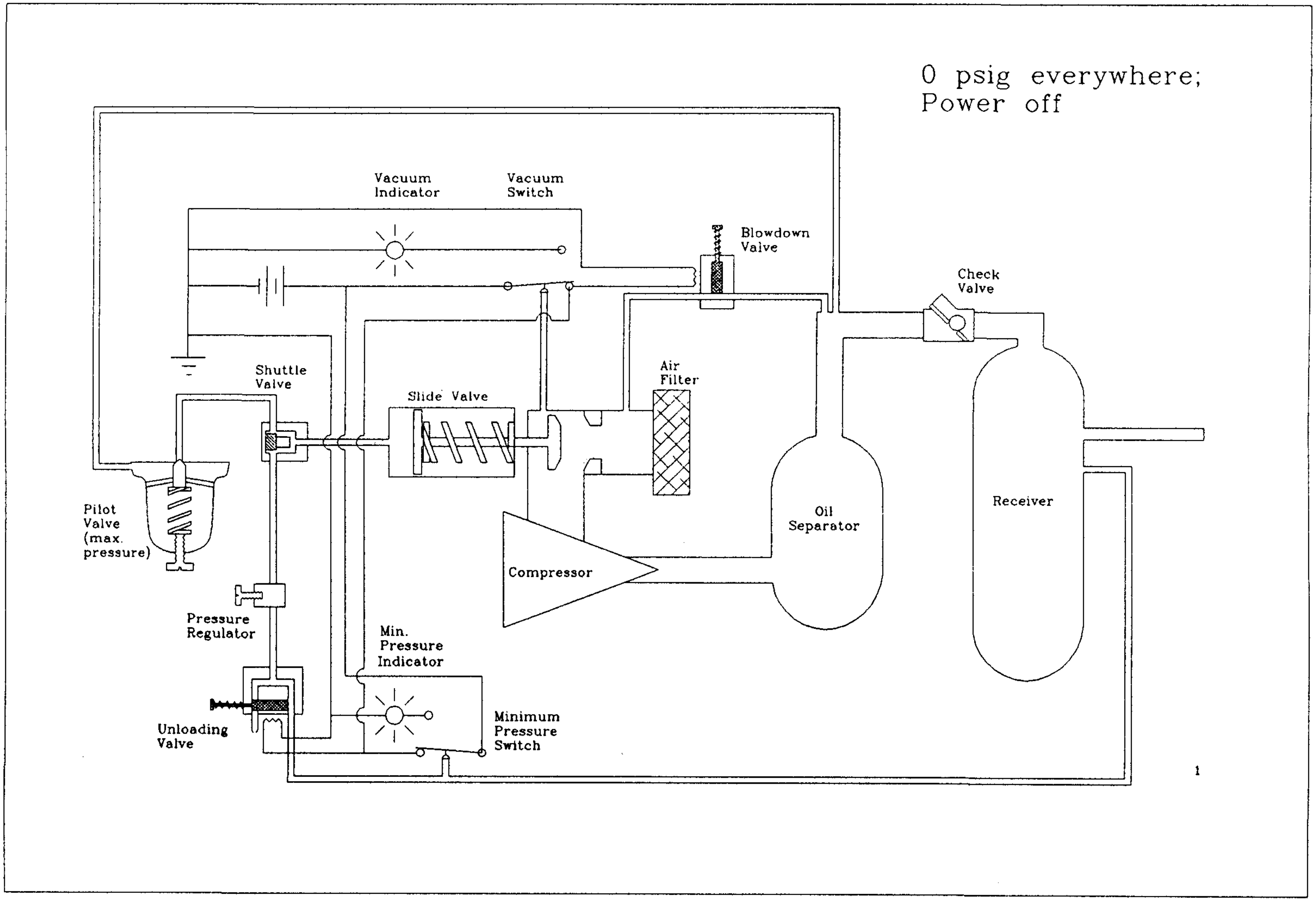

Figure 55: Low–Unload Control Operation, 1 of 6

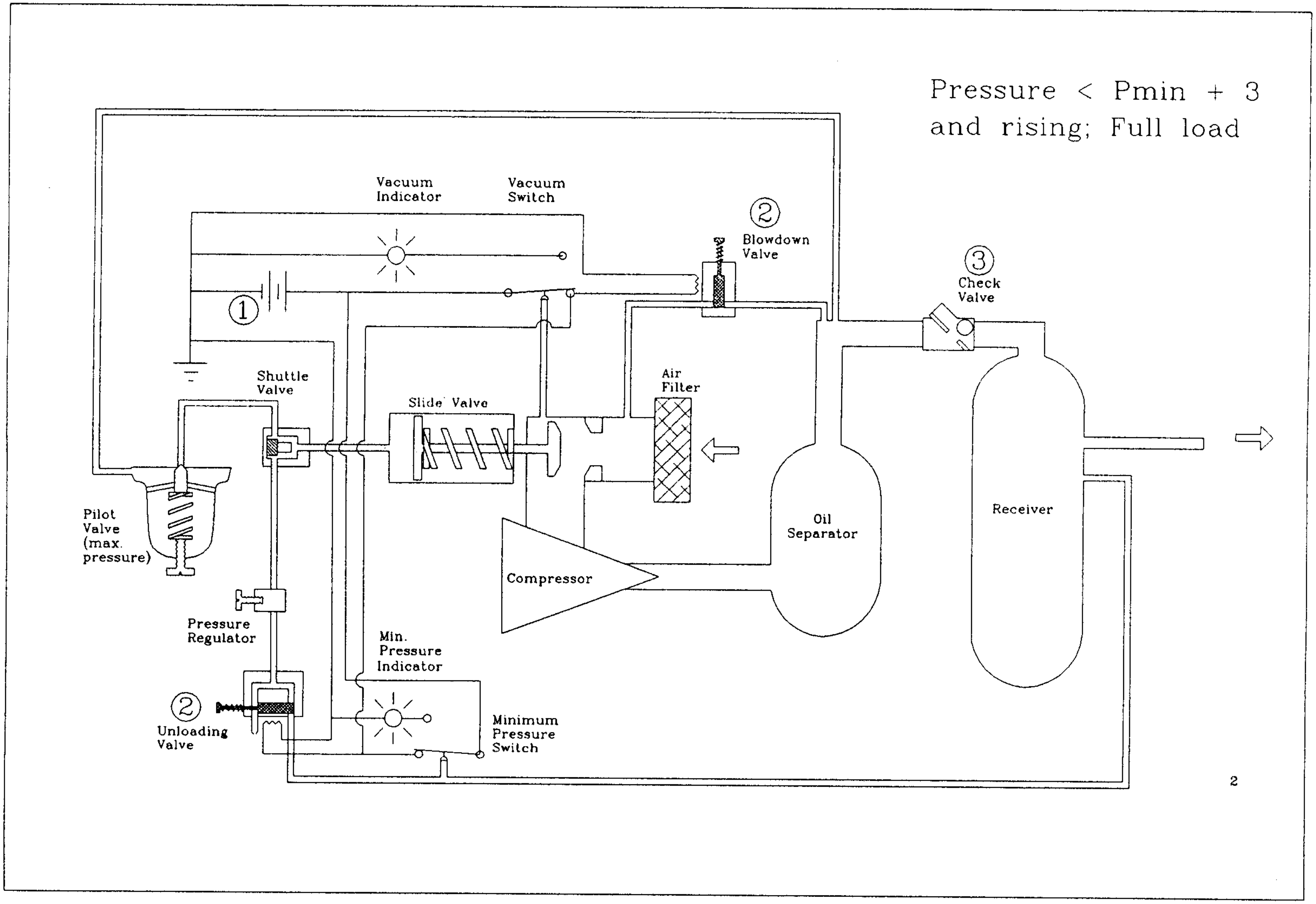

Figure 56: Low-Unload Control Operation, 2 of 6

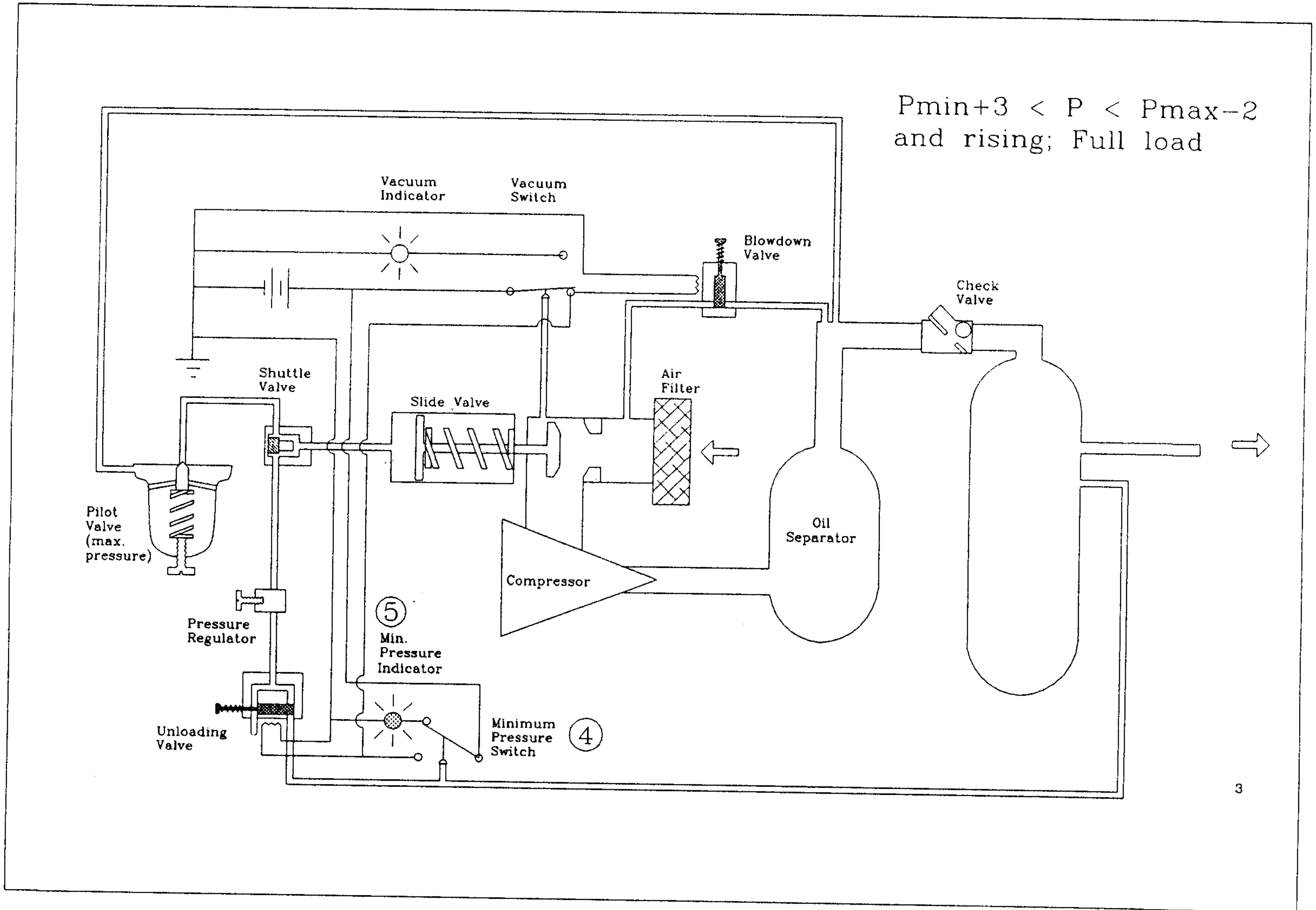

Figure 57: Low–Unload Control Operation, 3 of 6

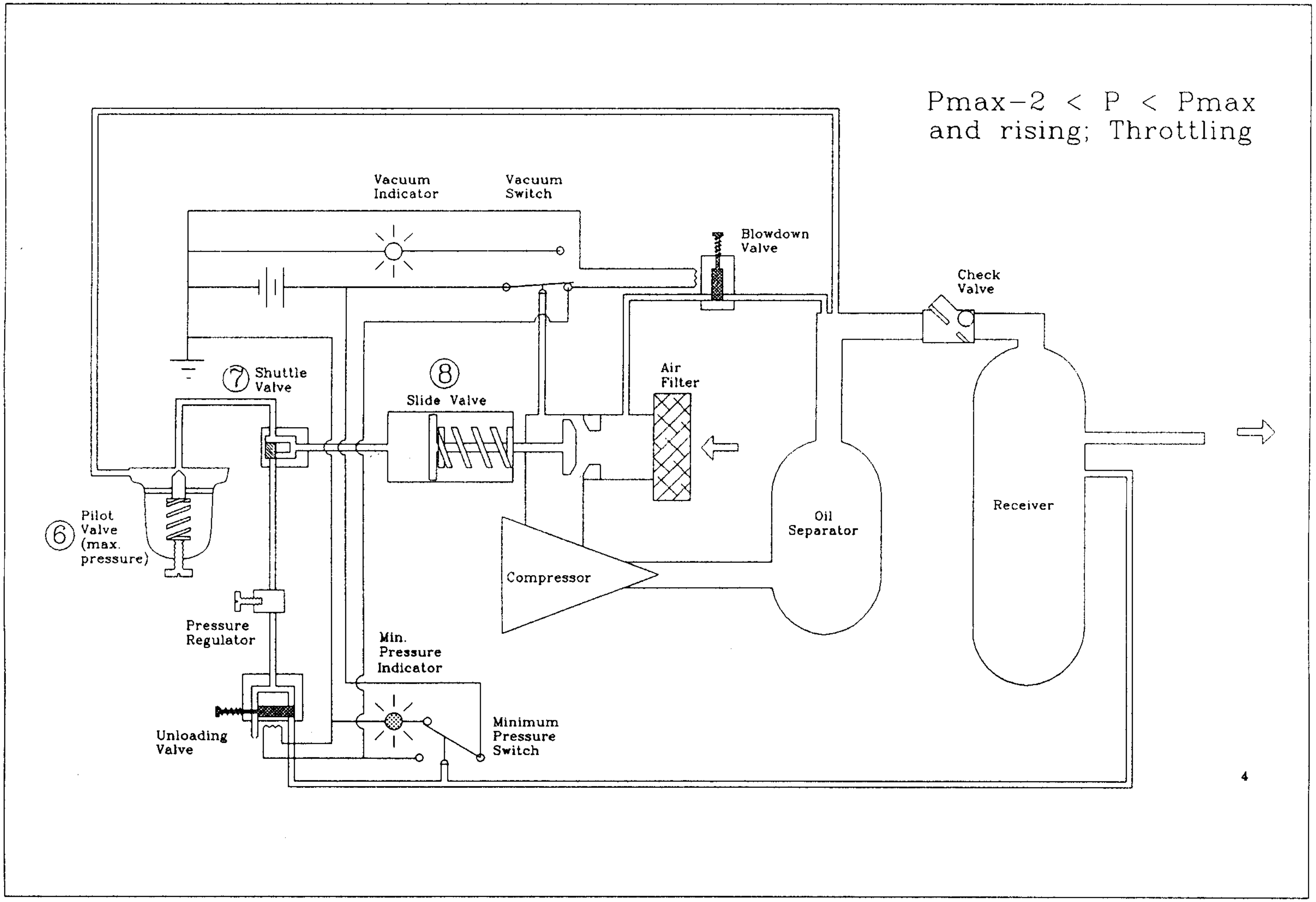

Figure 58: Low−Unload Control Operation, 4 of 6

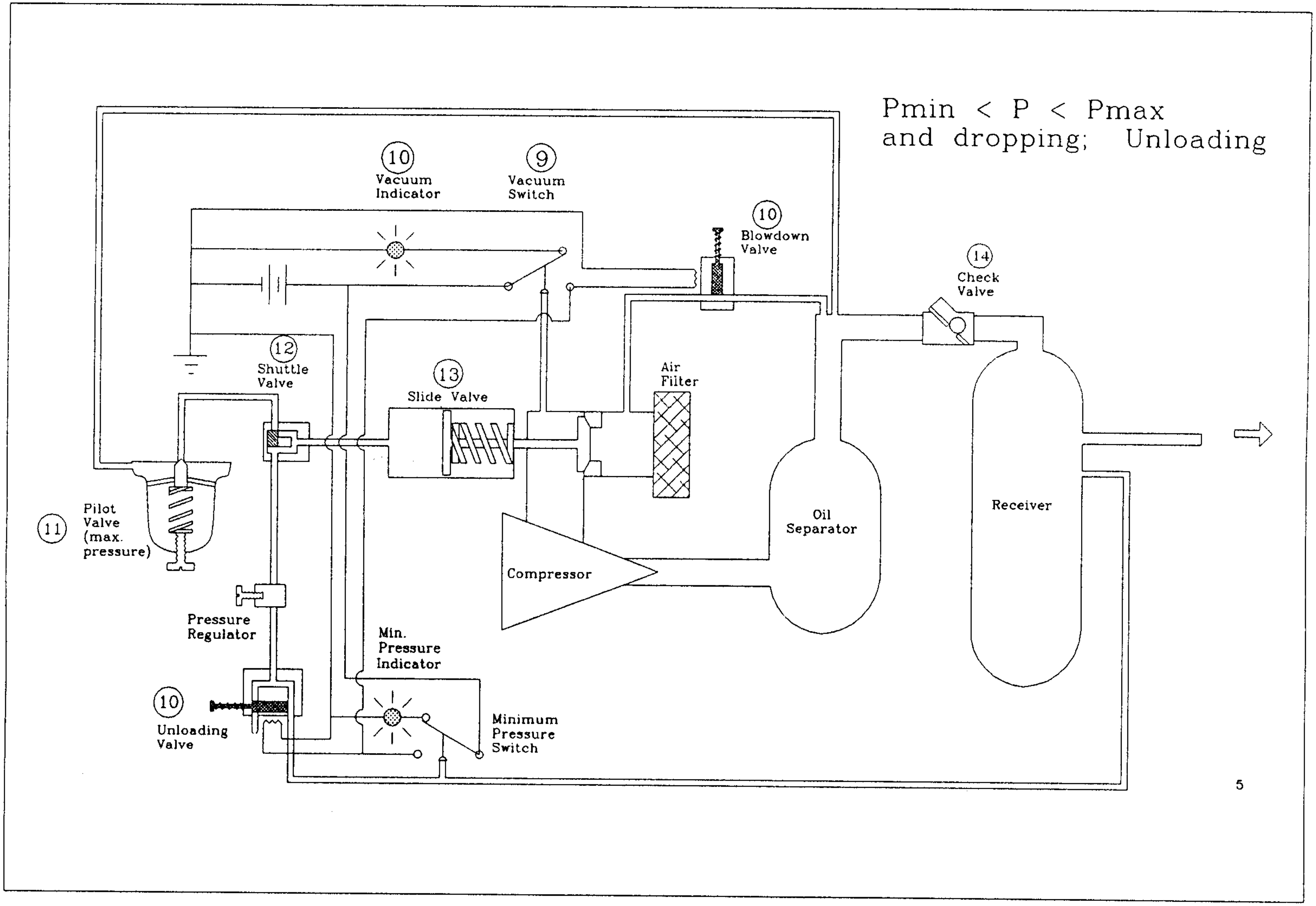

Figure 59: Low–Unload Control Operation, 5 of 6

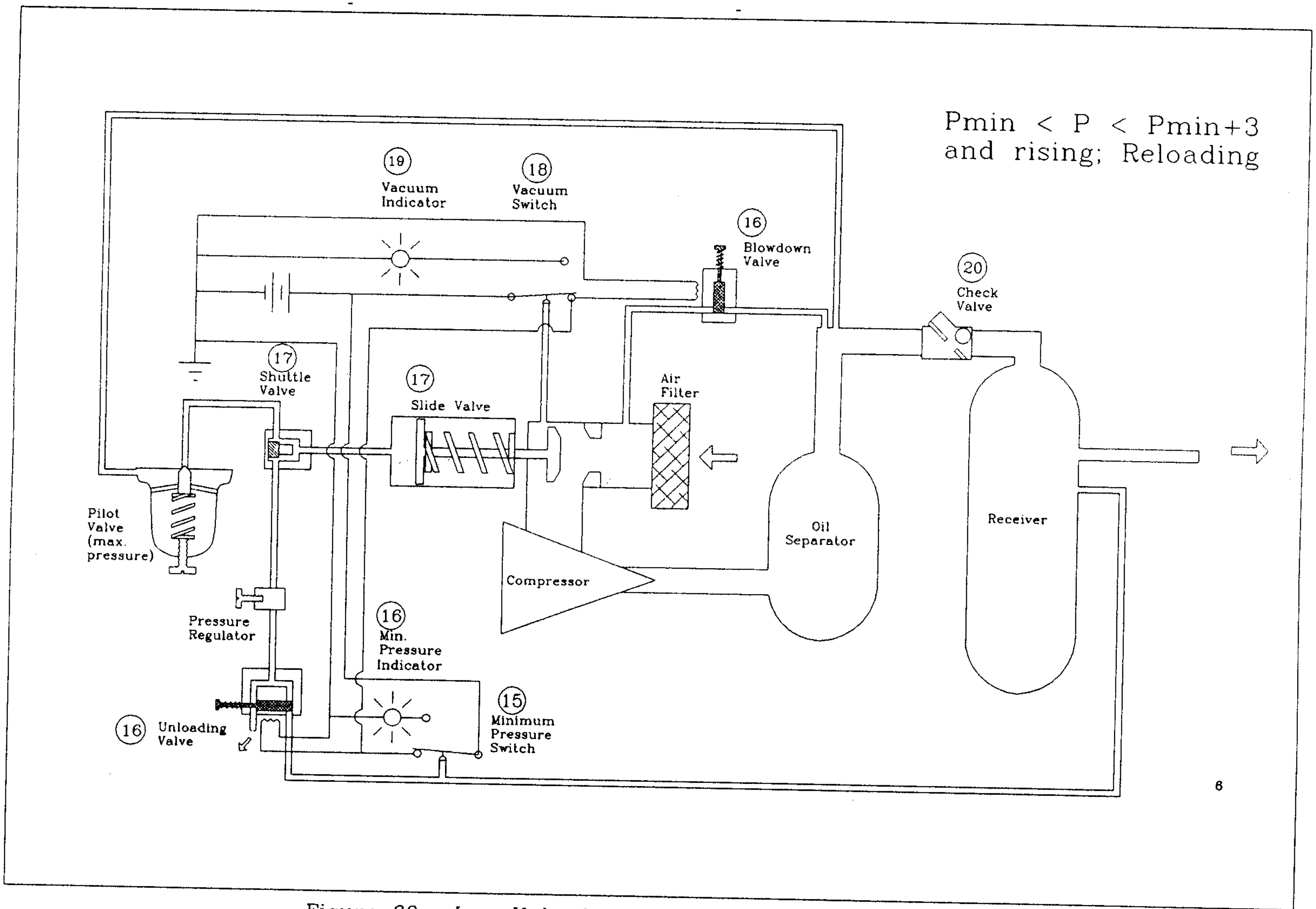

Figure 60: Low–Unload Control Operation, 6 of 6

APPENDIX D: DATA ACQUISITION SYSTEM

This appendix describes the hardware and software used for data collection and analysis of the screw compressor used for testing and evaluation.

1. Hardware

a. Sensors

1) **Input Current** - Current was measured with a 50:5 current transformer and converted to a variable voltage output with an EIL CTM-510 current-voltage transducer. A 100 ohm resistor was connected across the transducer output terminals to maximize output resolution in the 0 - +10V range, the acceptable range for the data acquisition equipment.

2) **Pressure** - Suction pressure was measured using an Omega 241PC15M 0 to -15 psig transducer. Based on a +8V reference supply voltage, output ranged from +1 to +6V. Similarly, discharge pressure was measured with an Omega 242PC250G 0 to 250 psig transducer. Supply and output voltages were +8V, and +1 to +6V, respectively.

3) **Air flow rate** - Unlike the other parameters, demand measurements were never automated. Instead, an orifice and a differential pressure gauge were used. Ultimately, the formula defining air flow rate (Q) became

$$Q = 13.99 \Delta p^{1/2}$$

Flow rate was controlled using a globe valve well upstream of the orifice. This formula is derived in Appendix D.

b. Data Acquisition Equipment. The equipment and software used to convert voltage signals from the transducers was an unfortunately colorful assembly. The heart of the system was a Data Translation DT2801-A input/output conversion board that plugged into an expansion slot of an IBM-PC/XT. The principal function of the board was to convert the analog input signal into a binary output signal. A Hewlett-Packard HP82993B thermocouple/general terminal module was used as an analog data bus for the I/O board. Once the binary signal was delivered to the computer, PCLAB V3.00 packaged software routines converted the binary data into a decimal number representing the input voltage. This data in turn was accessed using an interpreted BASIC program. The program converted the analog voltage values into pressure, power, and time and sent the data to an ASCII format output file and to a formatted screen display. Intermediate values such as power factor were also manipulated. Finally, LOTUS 1-2-3 V2.3 was used to import the ASCII data and analyze it.

The chief characteristics of this setup were speed (slow to run, slow to learn) and cost (none - already owned by OSU). The hardware did provide good resolution (+/- 1/2%) and ultimately resulted in good data for analysis.

2. Software. The following program was used to convert the raw data into meaningful values and store the output in an ASCII format file.

DATA ACQUISITION PROGRAM LISTING

```
100 '                              Jon Maxwell
110 '                              Fall-Spring 91-92
115 '
120 ' Program Goal: This program will monitor power consumption of a 15 hp
121 '   Quincy screw air compressor with throttling and low-unload controls.
122 '   Data translation equipment is used to translate CURRENT, TEMPERATURE,
123 '   and AIR PRESSURE. Hopefully, the program will also ultimately include
124 '   output controls to alter either the low-unload setting or the blowdown
126 '
130 ' Lines 200-300 represent names of commands or subroutines that can be called
131 ' from this program. To make these commands accessible, from DOS type
132 ' PCLLDR PCLDEFL. This will load PCLLDR which loads PCLIBAS.EXE and the
133 ' BASIC interpreter. (In case you are starting a new program, the PCLab
134 ' Definitions Long (PCLDEFL.BAS) program is also automatically loaded. Use
135 ' merge" to add a new program below line 300.) If you've already built a
136 ' file with these lines in it, simply load" it as you normally would.
140 '
145 ' PCLI.BAS writes the proper execution information for each subroutine to
150 ' pre-determined memory addresses, based on the DEF SEG, which matches the
155 ' base address selected in the SETUP PROGRAM.
160 '
200 ' PCLAB BASIC definition file (long names)
202 ' This file may be MERGEd with a user program
204 ' to define routine offsets and establish the PCLAB
206 ' segment.
207 '
208 ADC.VALUE=3                      : ADC.ON.TRIGGER=6
210 SETUP.ADC=9                      : ADC.SERIES=12
212 BEGIN.ADC.DMA=15                 : TEST.ADC.DMA=18
214 WAIT.ADC.DMA=21                  : DAC.VALUE=24
216 DAC.ON.TRIGGER=27                : SETUP.DAC=30
218 DAC.SERIES=33                    : BEGIN.DAC.DMA=36
220 TEST.DAC.DMA=39                  : WAIT.DAC.DMA=42
222 SET.CLOCK.DIVIDER=45             : SET.SLOW.CLOCK=48
224 SET.CLOCK.FREQUENCY=51           : SET.CLOCK.PERIOD=54
226 ENABLE.FOR.INPUT=57              : ENABLE.FOR.OUTPUT=60
228 INPUT.DIGITAL.VALUE=63           : OUTPUT.DIGITAL.VALUE=66
230 INPUT.DIGITAL.ON.TRIGGER=6       : OUTPUT.DIGITAL.ON.TRIGGER=72
232 SET.ERROR.CONTROL.WORD=75        : GET.ERROR.CODE=78
234 SELECT.BOARD=81                  : SET     .BASE.ADDRESS=84
236 SET.DMA.CHANNEL=87               : SET.ADC.RANGE=90
238 SET.ADC.CHANNELS=93              : SET.DAC.RANGE=96
240 SET.LINE.FREQUENCY=99            : SET.TOP.GAIN=102
242 SET.TIMEOUT=105                  : GET.DT.ERROR=108
244 RESET.DT=111                     : GET.DT.STATUS=114
246 CALL.WFC=117                     : CALL.WFO=120
248 CALL.WFI=123                     : STOP.ADC.DMA=126
250 STOP.DAC.DMA=129                 : CONTINUOUS.ADC.DMA=132
252 CONTINUOUS.DAC.DMA=135           : MEASURE.VOLTS=138
254 MEASURE.THERMOCOUPLE=141         : MEASURE.COMPENSATION=144
256 ANALOG.TO.VOLTS=147              : VOLTS.TO.DEGREES=150
258 DEGREES.TO.VOLTS=153             : DELAY=156
260 STROBE=159                       : WAIT.ON.DELAY=162
262 GENERATE.CLOCK=165               : COUNT.EVENTS=168
264 READ.EVENTS=171                  : GET.FREQUENCY=174
266 STOP.CLOCK=177                   : INITIALIZE=180
268 TERMINATE=183                    : ISBX.READ=186
270 ISBX.WRITE=189                   : FIND.DMA.LENGTH=192
272 ENABLE.SYSTEM.CLOCK=195          : DISABLE.SYSTEM.CLOCK=198
```

```
274 '
290 '------------------------------------------------------------------------
300 '                              Set-up Parameters
310 DEF SEG=&H0    'assign a base address for the PEEK statement to reference
320 PCLSEG = PEEK (&H4FE) + 256*PEEK (&H4FF)
330              'peek returns the number from the specified relative address
340 DEF SEG=PCLSEG 'Base address for the PCLAB segment is assigned based on the
350              'contents of the above addresses.
360 ERROR.VALUE%=0 'XSECW sets the error control word to 0 (no error, I think)
370 CALL SET.ERROR.CONTROL.WORD(ERROR.VALUE%)
380 CJCHAN% = 0    'Measure cold junction temp. on terminal board on channel 0
390 TYPE% = 107    'Set thermocouple as type K
395 '------------------------------------------------------------------------
400 '
460  SCRNCNT = 1
480  OPEN "da.dat" FOR OUTPUT AS #1
490  NOPRNT = 1 'set # of data intervals between display lines (multiplier)
500  GOSUB 4500 'set parameters necessary for A->D conversion
1000 '
1035 COUNTER=1
1038 PCOUNT =0
1040 TIMEINC=5    'set the timer interval for repeated data retrieval, in secs.
1045             'less than 2 seconds is not recommended
1050 STARTIME=TIMER 'zero value for timer, which shows seconds past midnight
1051 IF (TIMER - STARTIME) < TIMEINC - .029 THEN 1051 'wait until time to go
1053 ADDTIME= TIMER-STARTIME 'time increment for this particular iteration
1055 STARTIME=TIMER          'reset zero for next iteration
1060 '
1070 GOSUB 1090              'do the work, baby
1075 GOTO 1051
1080 END                d of main program here
1082 '
1085 '------------Main subroutine that calls each channel-------------------"
1090 '
1110 COUNTER=COUNTER + 1        'time counter
1130 TOTTIME =TOTTIME + ADDTIME 'TOTTIME=COUNTER*TIMEINC
1140 IF PCOUNT/17 < > INT(PCOUNT/17) THEN 1160
1150 CLS : GOSUB 3000 : LOCATE 7,1
1160 BOARD% = 1
1170   CALL SELECT.BOARD(BOARD%)
1200     ' Set channel to receive signal
1201         'Set signal ampl.: 1=1,2=10,4=100,8=500 (match # in setup)
1205         CHANNEL% = 1
1210         GAIN% = 8
1220         GOSUB 2500
1230         TEMPF = 32.2 + 9/5 * TEMP + 9 '***** 9 is a BS corr. factor*
1240         '
1250         CHANNEL% = 2
1255         GAIN% = 1
1260         GOSUB 5000
1270         CURRENT = ABS(44!/7.41 * VOLTAGE!)          'Conversion
1280         '
1290         CHANNEL% = 3
1300         GOSUB 5000
1310       ' PRESSUR1 = -3*(VOLTAGE!-1) 'Ideal Conversion
1320         PRESSUR1 = -3*(VOLTAGE!-1.07)*13/15 'Actual Conversion
1325 'PRINT "inlet volts:",VOLTAGE,"   ",
1330         '
1340         CHANNEL% = 4
1350         GOSUB 5000
1356    '    PRESSUR2 = 50! * (VOLTAGE! -1)       'ideal Conversion
1360         PRESSUR2 = 50!* (VOLTAGE! - 1.09)*16/13 'actual Conversion
```

```
1361 'PRINT "outlet volts:",VOLTAGE
1365       '
1370       ' MAXAMPS = 40!*230/240          ' This 15hp motor @ 240V
1375       MAXAMPS = 51  ' Based on Dranetz test that PF curve was fit to
1380       PFLA = CURRENT / MAXAMPS       '% of full load amps
1385       IF PFLA < .3 THEN PFLA = .32  'set minimum operating condition
1390       PF = .816 + .1056/PFLA - .06493/PFLA^2 'curve fit power factor
1395       IF PFLA<.3 THEN PF=.5
1410       '
1420       LINEVOLT = 208               'phase to phase voltage
1430       KW = LINEVOLT * CURRENT * PF * 3^.5 / 1000 '3 phase demand calc
1440       '
1450       'POWMAX = 15.28              average measured full load power
1455       POWMAX = 15 * .746 / .875 '100%Nameplate full load inpt power
1460       PARTLD = KW/POWMAX * 100
1470       '
1480  'WRITE #1, TOTTIME, TEMPF, CURRENT, PRESSUR1, PRESSUR2, PF, KW, PARTLD
1485   WRITE #1,TOTTIME,CURRENT,PRESSUR1,PRESSUR2,KW
1490   '
1500   IF COUNTER/NOPRNT <> INT(COUNTER/NOPRNT) THEN 2480
1530       PCOUNT = PCOUNT + 1
1540       PRINT USING "####.##";TOTTIME;
1550       PRINT USING "#####.#";TEMPF;
1560       PRINT USING "########.## ";CURRENT;
1570       PRINT USING "#########.## ";PRESSUR1;
1580       PRINT USING "########.# ";PRESSUR2;
1590       PRINT USING "#######.### ";PF;
1600  '    PRINT "     ";          'skip efficiency for now
1610       PRINT USING "####.##"; KW;
1620       PRINT USING "######.#"; PARTLD
2480 RETURN
2490 '
2499 '-------Subroutine to acquire data for Temperature ----------------------
2500   '
2501   CJTOTAL = 0 : TOTAL = 0 : SAMPLES = 5
2502   '
2503   '  Samples= will set the # of data samples to average/value out
2504   '  ADC.VALUE ROUTINE is important!
2505   '  This routine is built into the PCLAB software.  It performs
2506   '  Analog to Digital Conversion of a voltage sent to the
2507   '  specified channel.  The returned value is the 3rd variable
2508   '  in the call.  It is an integer ranging from 0 to 4096(2^12)
2510   FOR I = 1 TO SAMPLES
2520       CALL ADC.VALUE(CJCHAN%, GAIN%, CJADATA%)
2530       CJTOTAL = CJADATA% + CJTOTAL
2540       CALL ADC.VALUE(CHANNEL%, GAIN%, ADATA%)
2550       TOTAL = ADATA% + TOTAL
2560   NEXT I
2590   '
2730   CJADATA% = CINT(CJTOTAL / SAMPLES)
2740   ADATA% = CINT(TOTAL / SAMPLES)
2745   '  find an average digital value for cold junction temperature and
2746   '  temperature in question
2750   CJTEMP = (((CJADATA% * .04) / 4096) - .02) * 2000
2755   '  calculate cold junction temperature in °C "by brute force"
2760   CALL DEGREES.TO.VOLTS(TYPE%, CJTEMP, CJVOLTS)
2765   '  subroutine to determine the compensation voltage based on CJ temp
2770   VOLTS = (((ADATA% * .04) / 4096) - .02) + CJVOLTS
2775   '  convert digital value for temperature in question into a
2776   '  representative voltage, and then add the previously determined
2777   '  cold junction voltage/compensation to it
2780   CALL VOLTS.TO.DEGREES(TYPE%, VOLTS, TEMP)
```

```
2785   '  finally, call routine to convert compensated voltage to °C
2790 RETURN
2980 '-----------------------Subroutine to print header----------------------
2990 '
3000 CLS
3010 LOCATE 1,25 : PRINT "Compressor Operating Conditions"
3020 LOCATE 3,1
3030 PRINT "                            Inlet     Outlet              Percent"
3040 PRINT " Time    Temp    Current    Pressure    Power    Full Ld"
3050 PRINT " (sec)   (F)    (Amps)    (psig)    (psig)   Factor  (kW)    Power"
3060 PRINT "--------------------------------------------------------------------------"
3070 RETURN
4490 '
4500 '----Subroutine to set parameters for analog to digital conversion------
4510 ANALOG.VALUE% = 0
4520 HIGH.V! = +10!           ' Highest voltage in range.
4530 LOW.V!  = -10!           ' Lowest voltage in range.
4540 RANGE! = HIGH.V! - LOW.V!  ' Total voltage range.
4550                          ' These are not variable for each run
4560                          ' Rather, they match the # in setup program
4570 NOC! = 4096!              ' 12 bit conversion
4580 '
4590 LSB! = RANGE!/NOC!        ' Voltage of Least Significant Bit/each bit
4600 '
4630 '
4640 RETURN
4990 '
5000 '---------Subroutine for normal analog to digital conversion------------
5010 '
5013 SCALED.LSB! = LSB! / GAIN%        ' Calculate scaled LSB.
5016 SCALED.LOW! = LOW.V! / GAIN%      ' Calculate scaled low voltage.
5020 ANALOG.VALUE% = 0
5030 CALL ADC.VALUE (CHANNEL%, GAIN%, ANALOG.VALUE%)
5040 CALL GET.ERROR.CODE(ERROR.VALUE%)  ' Check for errors
5050 GOTO 5080                    'IF ERROR.VALUE% = 0 GOTO 5080
5060   PRINT "Idiot warning!  Error #:  ",ERROR.VALUE%," has been committed."
5070   STOP
5080 '
5090 'Calculate the effective voltage.
5100 '
5120 VOLTAGE! = (ANALOG.VALUE% * SCALED.LSB!) + SCALED.LOW!
5130 '
5140 '  PRINT " Analog Data Value is ";ANALOG.VALUE%;" decimal,";
5150 '  PRINT " computed voltage is ";VOLTAGE!
5160 RETURN
5170 '-----------------------------------------------------------------
```

APPENDIX E: ORIFICE MEASURED AIR FLOW RATE CALCULATION

Average flow rate through a venturi tube or a sharp-edged, accurately machined orifice can be approximated by the following equation:[18]

$$Q = A_1 \beta^2 C_D Y \left[\frac{2g_c(p_1 - p_2)}{(1-\beta^4)\rho}\right]^{\frac{1}{2}}$$

where

Q = flow rate (ft^3/min)

A_1 = pipe area = $\pi\, d_p^2 /4$ (in^2)

ß = ratio between orifice and pipe diameters (d_o/d_p)

C_D = discharge coefficient

Y = compressibility factor

rho = air density

g_c = gravitational constant

p_1 = upstream static pressure

p_2 = downstream static pressure

For this application,

Q = unknown

A_1 = $\pi(2.067)^2 /4$ = 3.356 in^2 = 0.0233 ft^2

ß = 1.000/2.067 = 0.4837

rho = air density at 100°F = 0.0710 lb_m/ft^3 [19]

g_c = 32.17 lb_m-ft/lb_f-s^2

To estimate the discharge coefficient, we need to know the Reynold's number, pipe size and orifice ratio.

$$Re = \frac{\rho V d}{\mu}$$

Assuming an air flow (Q) of 49 scfm, for the Reynold's number calculation

V = velocity

= Q/A

= 49 scfm / 0.0233 ft^2

$= 2{,}092$ ft/min

$d \quad = 2.067$ in $= 0.1723$ ft

$\mu \quad = 1.28 \times 10^5\ lb_m$/ft-sec @ 100°F [20]

then

$$Re = \frac{0.071\frac{lb_m}{ft^3}\ 2{,}092\frac{ft}{min}\ 0.1723ft}{1.28x10^5\frac{lb_m}{ft\text{-}sec}\ 60\frac{sec}{min}} = 34{,}700$$

Therefore, the discharge coefficient will be

$C_D \ = f(Re, \beta) = 0.625$ [21]

To estimate the compressibility factor, we need to know the pipe and orifice areas and assume pressure conditions. If

$p_1\text{-}p_2 = 3$ inches water $= 0.108\ lb_f/in^2$

$\tau \quad = 1.4$

$A_2 \quad = \pi(1.000\ in)^2/4 = 0.785\ in^2$

Then,

$Y \quad = f(\tau,\ p_1\text{-}p_2,\ A_1,\ A_2)$ [22]

$\quad = 0.969$

Substituting all of this into the main equation,

$$Q = (3.356inch^2)\ (0.4837)^2\ (0.625)\ (0.969)\ [\frac{(2)\ 32.17\frac{lb_m ft}{lb_f s^2}}{(1-0.4837^4)0.071\frac{lb_m}{ft^3}}]^{\frac{1}{2}}\ (\Delta p\frac{lb_f}{ft^2})^{\frac{1}{2}}$$

where

$\Delta p \ = p_1 - p_2$

Then, to convert to convenient units,

$$Q = 14.71inch^2\ (\Delta p\frac{lb_f}{ft^2}(\frac{1}{144}\frac{ft^2}{inch^2})(27.70\frac{inw\text{-}inch^2}{lb_f}))^{\frac{1}{2}}\ (\frac{1}{144}\frac{ft^2}{inch^2})\ (60\frac{sec}{min}$$

$$Q = 13.99(\Delta p)^{\frac{1}{2}}$$

where Δp is in inches of water and Q is in scfm.

APPENDIX F: FULL LOAD POWER CALCULATION

The term "full load power' permeates much of the guidebook and this thesis. Full load power for a compressor can be measured directly with a three phase power meter. However, if measured power is unknown the following formula can be used to estimate full load power for compressors:

$$FLP = \frac{hp}{\eta} \; x \; 0.746 \frac{kW}{hp} \; x \; [0.5\% \; x \; (p_a - p_r) + 100\%] \; x \; 110\%$$

where,

hp	=	rated motor output horsepower
η	=	rated motor efficiency at full load
p_a	=	actual operating pressure
p_r	=	maximum rated compressor pressure
110%	=	percent nameplate power required at maximum rated compressor pressure and full capacity (may vary modestly among manufacturers)
0.5%	=	percent change in power per psi deviation from maximum pressure (may vary modestly among manufacturers)

APPENDIX G: SPREADSHEET MODEL OPERATION

This Appendix describes the operation of the spreadsheet model in detail.

1. Variables: The following information must be supplied in order to perform the calculations:

minimum desired operating pressure (pd_n)
maximum desired operating pressure (pd_x)
receiver + plant piping volume (V_r)
oil separator + compressor-to-separator pipe volume (V_s)
compressor motor nameplate horsepower (hp_n)
motor efficiency (^{n})
compressor rated air delivery (cfm_r)
pressure at rated delivery (p_r)
percent of rated motor input power when fully loaded at rated pressure (flp)
percent of fully loaded power when fully throttled ($\%flp_t$)
percent of fully loaded power when fully unloaded ($\%flp_u$)
unload point (UP, in percent capacity).

All pressure values are in psig. All volumes are in cubic feet. The above parameters are likely to change for each application.

There are other variables that can be adjusted in the spreadsheet but will theoretically remain constant for a specified manufacturer's control equipment:

Proportional control range ($\blacktriangle$p). This is the pressure range across which the compressor reduces the throttle from 100% to 0% open (typically 4 to 10 psi).

Throttle effectiveness. This indicates how much of a vacuum exists when the throttle is intended to be fully open minus what the vacuum is when fully closed. Calculated as a percent of design vacuum range, it is typically close to 100%.

Minimum sump pressure ($pdis_{min}$). For compressors with full blowdown, the modulator cannot induce the compressor to form a perfect vacuum at the inlet. Therefore, the discharge pressure will never drop completely to atmospheric pressure; a typical value would be 2.5 psig. For partial blowdown designs that use the sump pressure to circulate oil, 30 psig is common.

Total reloading time. This is the time it takes for the throttle to fully open from being fully closed, once the minimum set pressure is reached. The delay is due to the time it takes for the pressurized control line air that holds the throttling valve closed to release to the atmosphere. The orifice through which the air escapes can get clogged with water or dirt and cause a long reloading time (normally 10-20 seconds if not clogged).

Unloading time constant for suction pressure (τ_s). This is used for the exponential decay calculation that simulates the suction pressure drop after the compressor unloads (for example $\tau_s = 4$ seconds, or 16 seconds until suction pressure has dropped 98% of the maximum).

Unloading time constant for discharge pressure (τ_d). This exponential decay time constant defines discharge pressure drop rate, and lags the suction pressure drop (example value: $\tau_d = 8$ seconds).

Linear unloading pressure drop rate (K_{unl}). Before exponential decay begins, the control hardware may limit the discharge pressure change to a linear decay rate. A typical value would be 8 psi/sec.

Limiting pressure ($Pdis_{max}$) indicates the end of the linear decay region and beginning of the exponential decay region.

2. Overall Spreadsheet Design

a. Layout: The spreadsheet was developed using Lotus 1-2-3, V2.3. A compressed printout of the spreadsheet is shown in Figure 61 to help with orientation. The spreadsheet is laid out in the following manner: variables that are likely to change with each application are in the upper left corner of the spreadsheet, section 1. Compressor and motor specifications, receiver capacity, and operating parameters such as average demand, unload point, and

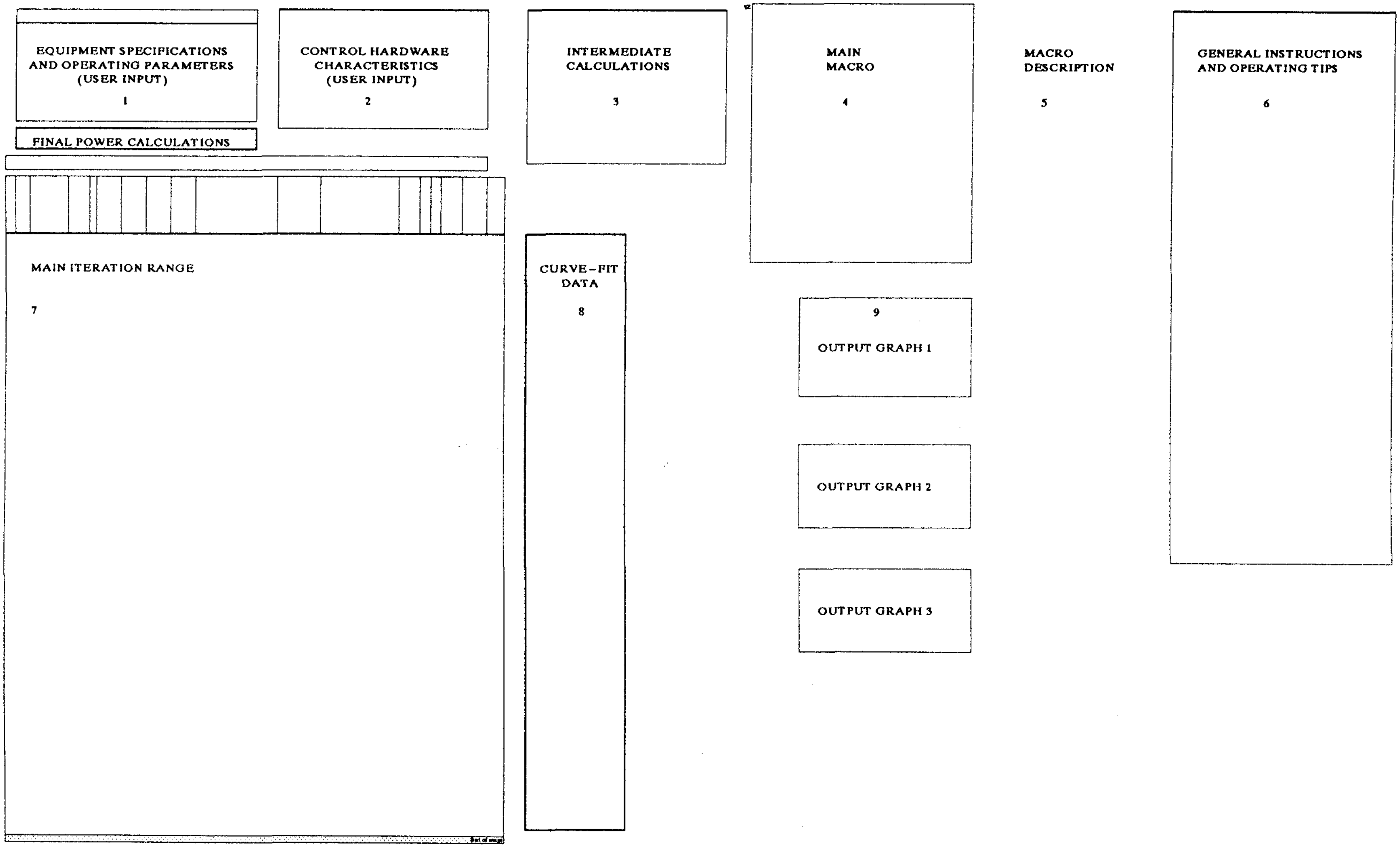

Figure 61: Spreadsheet Model Layout

maximum/minimum desired pressure are all defined here. The ultimate output, the average power per cycle, is in the lower part of this region. Other variables that can be manipulated are one tab to the right of this range, in section 2. These variables affect the mathematical modeling of the control system and are not likely to change often. Further to the right in section 3, the intermediate calculation variables are displayed. These cells are all formulas and should not be altered.

The main macro can be found to the right of this range in section 4. The macro is annotated another tab to the right in section 5. Finally, the instructions and other tips can be found to the far right in section 6. They can be reached by hitting Alt-M at any time. The tips are included as Figure 62. On the far left below the main variable list, a small hidden and protected region is reserved for a query operation that is nested in the main macro. Below that, section 7 contains the main iteration formulas and the iteration range itself. This range is about 90 rows by 25 columns. To the right of this range, section 8 holds a column of numbers used for curve fit calculations. The variables in this range probably will not need adjustment during normal use of the spreadsheet. The graph output range is located further to the right in section 9, below the main macro.

b. General Operation: There are only three steps to follow in order to operate the spreadsheet.

Step 1: Enter the mechanical parameters in section 1. It is suggested that the time interval variable, DT, be set to a low value (such as 1/2 sec) but this is not required. If DT is too low, the spreadsheet will take a long time to execute. If high, modeling resolution will suffer.

Step 2: Review the control hardware values in section 2 to verify that the behavior is as desired.

Step 3: Press the keys, Alt-Z to run the calculating macro. All calculations will be performed, graphs generated, and final results tallied.

Since the macro is described line by line on the spreadsheet, this text offers a general overview. After Alt-Z is hit, all iteration formulas are copied down to

GENERAL INSTRUCTIONS AND OPERATING TIPS:

Macros:

\M {goto}bl – Sends the display to this page.

\Z The main macro that runs all of the iterative calculations
Re-run after any change in data.

Tips:

The spreadsheet will not detect or properly model conditions when the compressor throttles and never unload(i.e. when the unld.pt. is lower than plant demand).

If your graphs turn out screwy, one likely culprit is that the unload point is close to the average capacity; combined with a pmax,desired greater than than the pressure at which plant demand was estimated. For example 38% capacity rated @ 120 psig will never reach a 40% unload point at 130 psig. The compressor will modulate (and the spreadsheet will blow up) because demand exceeds the unload point.

If you are in doubt about a good value to choose for DT, the calculation interval, it is better to choose a small value such as 1 second rather than an excessively large value. The spreadsheet macro automatically will increase DT as necessary until the cycle is completed. The only cost to this approach is that the calculation process may take longer if extra calc. cycles are required.

If reloading is very slow, the unload pt. is very high, and demand is very low, it is possible that the throttle position will still be <unld pt. when the system pressure increases past pmin + 3. If this happens, the spreadsheet will try to unload at this point, and the control scheme will fall apart. The same failure can happen with a real compressor.

If you are running this software on a Lotus 1-2-3 version other that 2.3, you will get some irritating beeps at the end of the \Z macro. This is because the macro is displaying the three grahs it created with the :PrintPreview function. To view the graphs, use the /Graph Name Use function and F10 instead. Since this is the last operation the macro attempts, no calculations will be screwed up. (You can eliminate the beeps by erasing cell AL29).

To simulate On-Off control, set the Unload Point to 99% and use the following settings in the Control Hardware section: Fully Throttled Power=0.1%, 0% Fully Unloaded Power, and 1 second for both Unloading Time Constants.

Never set the minimum discharge pressure below 0.1 psig. This could cause the macro to enter an infinite loop. Besides, it's likely to be true, except when the compressor is turned off completely for a long time.

The Throttle-only Power final calculation is based on the assumption that the compressor throttles under conditions defined by measured capacity and pressure, NOT median pressure. Be careful when comparing the Throttle-only power with the cycle power because if those two pressures are different, you aren't really comparing apples with apples. If you want to change it, replace 'L7' with '(L9+L10)/2' in the calculation cell, for a better comparison. The Ideal Load-Unload Power calculation uses the median pressure value for its calculation.

Throttle-only Power and Ideal Load Unload Power for capacities above the unload can be calculated directly by entering all of the variables and then hitting F9, CALCULATE, rather than the Alt-Z macro. These two calculations do not depend on any of the iterative calculations.

The circular reference indicator is okay – REALLY!!! The formulas in cells A26 and B26 are anchored to cells A29 and B29. This would seemingly cause a self-reference when the formulae are copied down to cells A/B29, but A29 always gets overwritten with 0.01 in the macro, and the reference to B29 is buried as part of a logical condition that cannot be reached during the first iteration. As for cells A/B26 being circular even after the macro is complete, those specific cells aren't used for any calculations.

Figure 62: Spreadsheet Instructions

fill the iteration range, and the model is run using the desired pressure values as the actual compressor switch settings. The operating cycle starts and ends with the minimum pressure switch being triggered. Since there will be a mechanical response time and because proportional control mechanisms have inherent offsets, these switch setting values will not exactly match the actual maximum/minimum pressure range spanned by the compressor.

The macro first checks that the calculation time interval is large enough to define a complete cycle in 90 steps, increases it if not, and then makes sure that the last discharge pressure and last throttle position of the modeled cycle are the same as the first values. Once these criteria are met, the macro adjusts the operating pressure switch setting values based on the results of the first run, and repeats the calculations. This is typically done twice. After three runs, the difference between desired and modeled maximum/minimum pressures is normally less than 0.5 psig.

The pressure switches are initially set to the desired maximum and minimum values. By letting the compressor model simulate a cycle and then adjusting the switches to account for mechanical response deviations, the macro sets the pressure switches in a manner quite similar to that of a human mechanic.

After the macro has finished with the thermodynamic calculations, the power formula is applied to each iteration, and the average power per cycle is calculated. The throttling-only power and ideal load-unload power are also calculated for comparison. Finally, the system pressure, compressor discharge pressure, throttle position (inlet pressure), and power are plotted for the full cycle.

3. Iterative Calculations

The first four columns are used as indicators and timers for the other columns. They define whether the compressor is reloading, fully loaded, throttling, unloading, or idling. The next fifteen columns perform the thermodynamic calculation. Three columns used as markers for the macro follow. The power calculation column is last.

Reload (I_1): The reload/full load sequence timer/indicator. Column I_1 initially starts at 0.01 seconds and increases continuously by time intervals until the throttling sequence calculation begins. On rare occasions when the throttle starts to close before it has fully opened, the reload indicator continues to increase until the throttle sequence's approximation of throttle position (column TP_1) results in a lower value than the throttle position as calculated by the reload/full load curve (column TP_2).

If the simulation determines that the throttle won't have time to completely close after unloading at the end of the cycle before the compressor must reload again, the first value for column I_1 will be set to a value greater than 0.01 for subsequent calculations. This is necessary in order that reloading throttle position value looked up for the curve fit (and based on the time value in column I_1) is the proper value corresponding to previous run's last throttle position.

Throttle (I_2): The throttling sequence timer/indicator. This column records elapsed time from the beginning of the time the discharge pressure enters the proportional control range up to the point when the compressor unloads.

Unload/Idle Indicator (I_3): The primary unloading and idle sequence indicator. This timer starts as soon as the compressor unloads, i.e. when the throttle position drops below the set unload point. It increases until the system pressure drops to the minimum pressure setpoint (end of cycle).

Unload/Idle Indicator (I_4): The secondary unloading and idle sequence indicator. Timer I_4 starts as soon as the discharge pressure drops below the linear decay approximation region and enters the exponential decay region. It increases concurrently with column I_3, but normally lags by several time intervals. Column I_4 also stops at the end of the cycle.

Total time (t): Timer counter. This column indicates total elapsed time since the beginning of the cycle.

Pressure Ratio (PR): System pressure ratio is the ratio between absolute system pressure (P_{sys}) and absolute atmospheric pressure (P_{atm} = 14.7 psia). As with all columns, values taken from columns to the "right" are from the previous iteration, and values used from columns to the "left" (none in this particular calculation) are from the current iteration.

$$PR = P_{sys}/P_{atm}$$

Plant Demand (D): Plant demand is assumed to vary with pressure. This is because air flow through an orifice increases with pressure in the lines. Demand is calculated in cfm.

$$D = scfm_{load} / PR$$

The value of $scfm_{load}$ is taken from one of the intermediate calculations, Plant Demand (scfm), and is fixed for all iterations.

Plant Pressure Drop (dP_{pl}): dP_{pl} indicates the drop in pressure in psi due to plant air demand during a single time increment, DT. The calculation ignores positive pressure contributions from the compressor.

$$dP_{pl} = - D \times DT \times P_{sys} / (V_r \times 60 \text{ sec/min})$$

Discharge Flow Rate (cfm_{dis}): The air compressor discharge flow rate is expressed in cfm, not scfm, based on the discharge pressure from the previous iteration (P_{dis}).

$$cfm_{dis} = cfm_r \times P_{atm} / (P_{dis} + 14.7)$$

where P_{dis} is the compressor discharge pressure in psig. Rated capacity and pressure are normally found in manufacturer's literature.

Throttle Position (TP_1): Throttle position during reload and full load. As described before, reloading is approximated with an arctangent curve. This column indicates throttle position (in percent of fully open) while the compressor reloads and maintains full load. At first, the value is looked up in a table that scales the fixed characteristic reloading curve to the reloading time defined in column I_1. Once the throttle opens completely it remains open as long as the reload/full load indicator continues to increase. For systems with large receivers, this calculation and will likely be insignificant because reloading time is small compared to cycle time. However, there are occasions when the reloading time is relatively long, and it can significantly affect the compressor's performance.

Throttle Position (TP_2): Throttle position during throttling. This column indicates the throttle position as though it were solely dependent on the proportional control mechanism. Once I_2 starts to increase, the throttle will gradually close as the discharge pressure continues to rise and approach the maximum pressure switch setting ($P_{max,sw}$). The equation for throttle position, in percent open, is

$$L = (P_{max,sw} - P_{dis}) / \Delta P$$

The calculation is performed as long as indicator/timer I_2 is not blank.

Throttle Position (TP_3): Throttle position when unloading or idle. Column TP_3 is used to calculate the throttle position after the compressor unloads until the end of the cycle. As stated before, the curve is a simple exponential decay, with the throttle position gradually approaching 0% open. The basic formula is

$$TP_3 = (T_{max} - T_{min}) \times e^{-(I3)/\tau}$$

where T_{max} is the maximum throttle position, normally near 100% open, T_{min} is the minimum throttle position, normally near 0% open, and τ is the exponential time constant for suction, in seconds. The time constant can be estimated as 1/4 of the time it takes for the throttle to close across 98% of the possible range.

Throttle Position (TP): Combined throttle position. This column combines the estimates of the previous three column into one curve that defines the compressor throttle position. It does this by selecting the minimum non-blank value.

Compressor Pressure Increase (dP_{c1}): Tank pressure increase due to compressor discharge. Similar to dP_{pl}, dP_{c1} indicates the positive pressure change in the receiver due solely to compressor contributions. This calculation assumes that the air is being injected into the receiver volume. The formula is

$$dP_{c1} = cfm_{dis} \times TP \times P_{dis} \times DT / (V_{tank} \times 60 \text{ sec/min})$$

Whenever the compressor is unloaded, the pressure contribution will be 0.

Compressor Pressure Increase (dP_{c2}): Modified pressure increase due to compressor. When the compressor is reloading and the discharge pressure is still below that of the system pressure, the volume being pressurized does not include the receiver. It includes only the oil separator volume and discharge piping upstream of the check valve(V_{sep}). dP_{c2} adjusts dP_{c1} when appropriate. The calculation is

$$dP_{c2} = dP_{c1} \times V_{tank} / V_{sep}$$

Otherwise it leaves the value as is was previously calculated, dPc2 = dPc1.

Compressor Discharge Pressure (P_{dis1}): The reload/full load discharge pressure calculation adds the appropriate compressor pressure contribution and subtracts the appropriate plant demand from the previous iteration's discharge pressure.

$$P_{dis1} = Pdis_{prev} + dP_{c2} \ (+ \ dP_{pl})$$

There are several special conditions, however. For example, no plant demand is subtracted if the discharge pressure is below the system pressure. Also, the discharge pressure can never exceed the system pressure, which could happen in the iteration when the oil separator fills up if the formula did not prevent it.

Compressor Discharge Pressure (P_{dis2}): Discharge pressure when throttling is always calculated in the same manner. No decisions must be made because the previous system pressure ($Psys_{prev}$) will equal the previous discharge pressure ($Pdis_{prev}$) always, by the time throttling begins.

$$P_{dis2} = Psys_{prev} + dP_{pl} + dP_{c2}$$

Compressor Discharge Pressure (P_{dis3}): Discharge pressure when unloading is calculated as an exponential decay, much like column TP_3. But because the discharge line has its own plumbing, the decay rate will be different than that of the throttle. In fact, based on laboratory testing, discharge pressure decays linearly above a certain pressure. Thus, in the model, discharge pressure decays linearly above a specified pressure, and exponentially below that pressure. In the spreadsheet, the linear pressure drop rate, the limiting pressure, the minimum discharge pressure, and the exponential decay time constant all can be adjusted. The two formulas are

$$P_{dis3} = Pdis_{prev} - K_{unl} \times dt \qquad P > Pdis_{max}$$

and,

$$P_{dis3} = Pdis_{min} + (Pdis_{max} - Pdis_{min}) \times e^{-I4/\tau} \qquad P <= Pdis_{max}$$

Compressor Discharge Pressure (P_{dis}): The combined discharge pressure calculation combines the estimates of the previous three calculations into one column that defines the compressor discharge pressure. Like the combined throttle position calculation, it does this by selecting the minimum not-zero value. Combining is necessary because the previous calculations sometimes overlap.

System Pressure (P_{sys}): The system pressure is simply discharge pressure whenever they are rising together. During the early stages of reloading, and after the compressor unloads, the discharge pressure is less than the system pressure. Under these two conditions, system pressure is calculated as the previous system pressure minus the change in pressure due to plant demand.

$Pdis_{last}$: This is a marker for the macro. $Pdis_{last}$ shows the last discharge pressure of the cycle.

$Psys_{min}$: This is a marker that helps the macro determine the minimum system pressure during the cycle.

Throttle Pos. (TP_{last}): TP_{last} is a marker that shows the last throttle position of the cycle.

Power (kW): The compressor real power for each iteration is based on the throttle position (which is directly proportional to suction pressure), and discharge pressure. Variable compressor efficiency characteristics are included each of the two power equations that are multiplied together.

$$kW = \%FLP_{TP} \times \%FLP_{dis}$$

The formulas used to calculate the percent of full load power due to throttling ($\%FLP_{TP}$) and the percent of full load power due to the discharge pressure ($\%FLP_{dis}$) are shown in the intermediate calculation range of the spreadsheet.

Graph X-axis: This column is used to display the x-axis, time, for all of all graphs.

4. Other Calculations

Plant Demand ($scfm_{load}$): This is the fixed air load required by plant equipment and leaks. Air flow is expressed in cfm under standard atmospheric conditions.

$$scfm_{load} = Q_{cap} \times \%C \times (14.7 + P_r)/(14.7 + P_m)$$

where,

$\%C$ = plant demand in percent capacity

p_r = pressure at which compressor capacity is rated

p_m = pressure at which $\%C$ was measured

Rated Full Load Power (FLP_r): See Appendix F.

Throttle Only Power (TOP): Throttle Only Power is calculated as described in section III.C.1. The result, %P is multiplied by FLP to indicate power requirements in kW.

Ideal Load-Unload Power (LDUP): This uses the same basic formula to calculate power while the compressor is loaded. Since the compressor cycles, the maximum and minimum pressures are averaged to estimate power when loaded. Also, the measured capacity is converted to the equivalent capacity required at the average power.

During the time when the compressor is unloaded, the percent fully unloaded power is used. The equation is:

$$LDUP = FLP_r \times P_{out} \times [\%C_{adj} + \%FLP_u \times (1 - C\%_{adj})$$

where

$$\%C_{adj} = \text{capacity adjusted for average power}$$

$$= p_{ave} / (14.7 + p_{meas}) \times \%C$$

$$P_{out} = \text{power modification due to outlet conditions}$$

$$= \%FLP_u/FLP_t\% + (1-\%FLP_u/\%FLP_t) x p_{ave}/(14.7+p_r)$$

$$p_{ave} = 14.7 + (p_{min} + p_{max}) / 2$$

Note: Since LDUP uses the average pressure and TOP uses measured pressure, they should be compared carefully.

5. Default Values

This section is included so that variables that might get inadvertently changed during operation can be reset if necessary. The values shown are not rigid - that is why they are variables - but they are shown here for reference in case of possible accidents.

Variable	Value	
Volume of air space in oil separator and discharge lines	2	ft_3
Fully throttled power	70	%
Fully unloaded power (atmospheric separator blowdown)	16	%
(25 psig separator blowdown)	25	%
Max. dp for proportional response	6	psi
Total throttle reloading time (properly working)	20	sec
(clogged lines)	90	sec
Maximum throttle position for unloading curve	100	%
Minimum throttle position for unloading curve	0.1	%
Tau(suction/throttle)	5.5	sec
Maximum pressure for exponential discharge unloading	90	psig
Minimum pressure for discharge	2.5	psig
Unload rate above maximum pressure	8	psi/sec
Tau(discharge)	9	sec
Reloading arctangent values		
Phase coefficient	-0.7	
Frequency coefficient	0.4	
Amplitude coefficient	1.1	
Change time	38.2	sec
Minimum inlet pressure	-0.5	psig
Change in pressure	13.3	psi
pi	3.1416	
curve increment	0.423	sec

6. Running Time

Run time will vary depending on variable input. Typical times are

10MHz 286	300 - 350	sec.
33MHz 386	35 - 50	sec.
33MHz 486	15 - 20	sec.

APPENDIX H: SMART CONTROLLER HARDWARE AND SOFTWARE

This appendix describes the hardware and software used to install the smart controller.

1. HARDWARE.

a. The BCC-52 Single Board Computer. In most microprocessor-based control systems, the heart is a small single board computer (SBC) built around one of several available microcontrollers. Microcontrollers are typically similar in design, including a microprocessor, clock and communication circuitry, a small amount of RAM, as well as some on-chip ROM. The ROM may be programmable, or delivered with prewritten firmware intended to support the intended application. The rest of the SBC is composed of whatever integrated circuits are considered important to support the microcontroller for the application. Normally, these will include a small amount of RAM, a parallel I/O controller, a serial I/O controller, address decoding circuitry, and perhaps some ROM.

The SBC chosen for this project was a Micromint BCC-52. A block diagram of the board is shown in Figure 63. This is a compact low cost computer built around the Intel 8052AH-BASIC microcontroller chip. Probably the most striking feature of this chip is that it has a small version of the BASIC language stored directly onto the microcontroller, and thus "wakes up" in BASIC. This feature dispenses with both the need to write in assembly language and the cost associated with assemblers and compilers. It also makes the SBC extremely easy to debug; one can develop, run, and debug programs in BASIC code.

In addition to the microcontroller with embedded language, the BCC-52 has numerous features which make it an attractive choice for the SBC developer. Up to 48 kbytes combined RAM and ROM may be placed on the board. Expansion boards are available if more memory is needed. An EPROM programmer allows the developer to burn in ROM programs on the board itself. Typically, this requires a separate device. A programmable parallel interface provides three 8-bit ports which can be configured for input or output. Three serial ports are also included, one of which provides a console RS-232 port through which the user may communicate with the BCC-52. The computer's default processing mode is interpreted BASIC, but if speed is crucial the microcontroller can be switched to run 8052 machine language routines directly.

b. Input/Output Board. In order to interface the SBC to the physical system for measurement and control of compressor operation, a number of simple interfaces were required. Inputs were two eight bit analog-to-digital converters (ADC) for collecting data from two pressure sensors. Three output bits of Port B were used for control of the ADC trigger, a two position relay, and an LED. A line diagram of the Input/Output board is shown in Figure 64.

For converting the analog signals from the pressure sensors into binary values for the BCC-52, two ADC7821 analog-to-digital converters were used. Though these chips are high speed devices and overqualified for their job, they were selected for availability and previous experience. They were set for use as unipolar, 0 to +5V conversion. The voltage delivered by the suction pressure sensor ranged from +1 to +6V for 0 to -15 psig. A voltage divider reduced the range to within the 5V maximum range. The 0 to 250 psig, +1 to +6V discharge pressure sensor did not require similar modification because pressure never exceeded 200 psig. Binary ADC output was sent to ports A and C of the SBC.

Bit two of the output port from the BCC-52 (port B) was used with a pull-up resistor arrangement to trigger the ADCs. Two other bits of port B were used for output. Bit 0 controlled the transistor that supplied current to open and close the relay. An LED parallel to the transistor was configured to turn on whenever the switch was energized and the compressor was unloaded. The final switch was designed to light an LED whenever the compressor was unloaded.

On the board, but independent from the described circuit, a 7808 voltage regulator was used to supply the +8V reference voltage required by the pressure transducers from the +12V provided by the power supply.

c. Sensors, Actuator, and Power Supply. The same pressure sensors

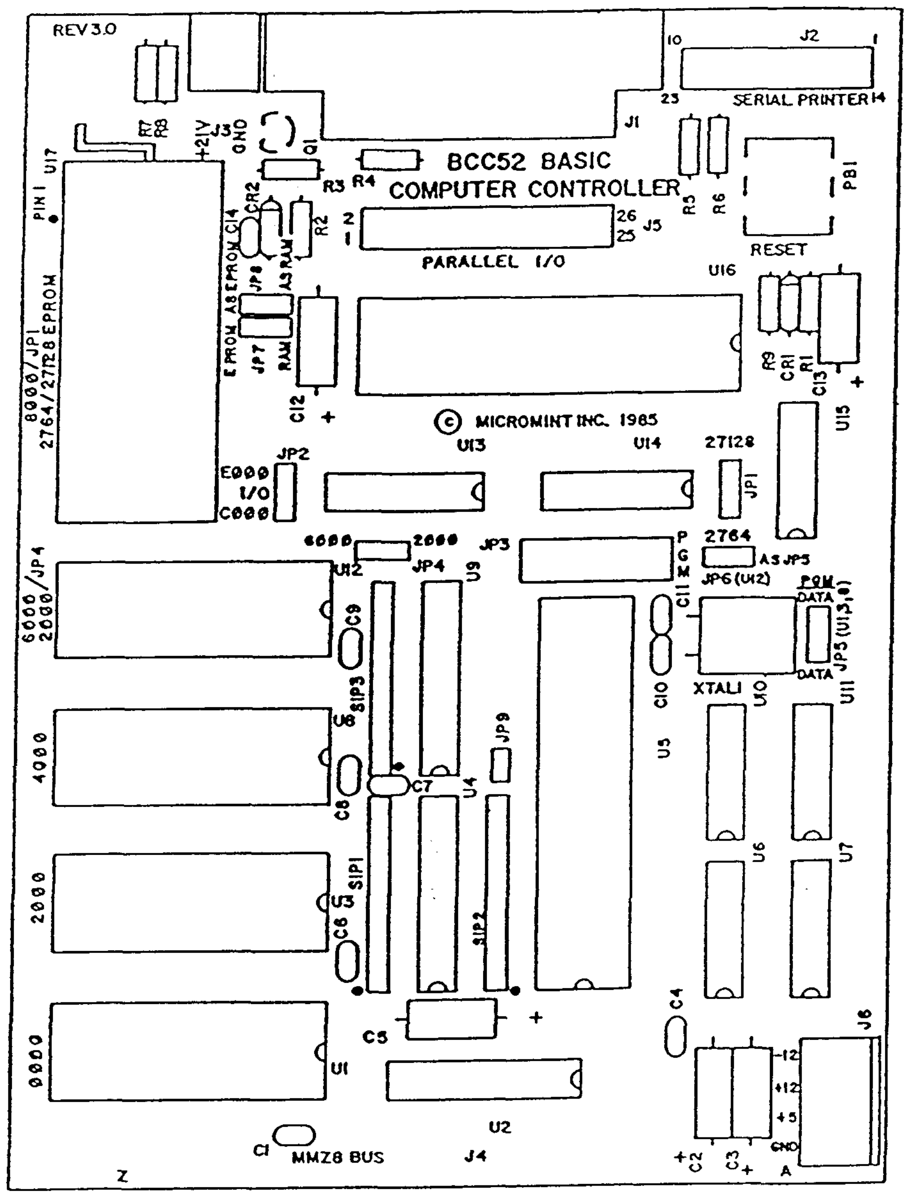

Figure 63: BCC-52 Silkscreen

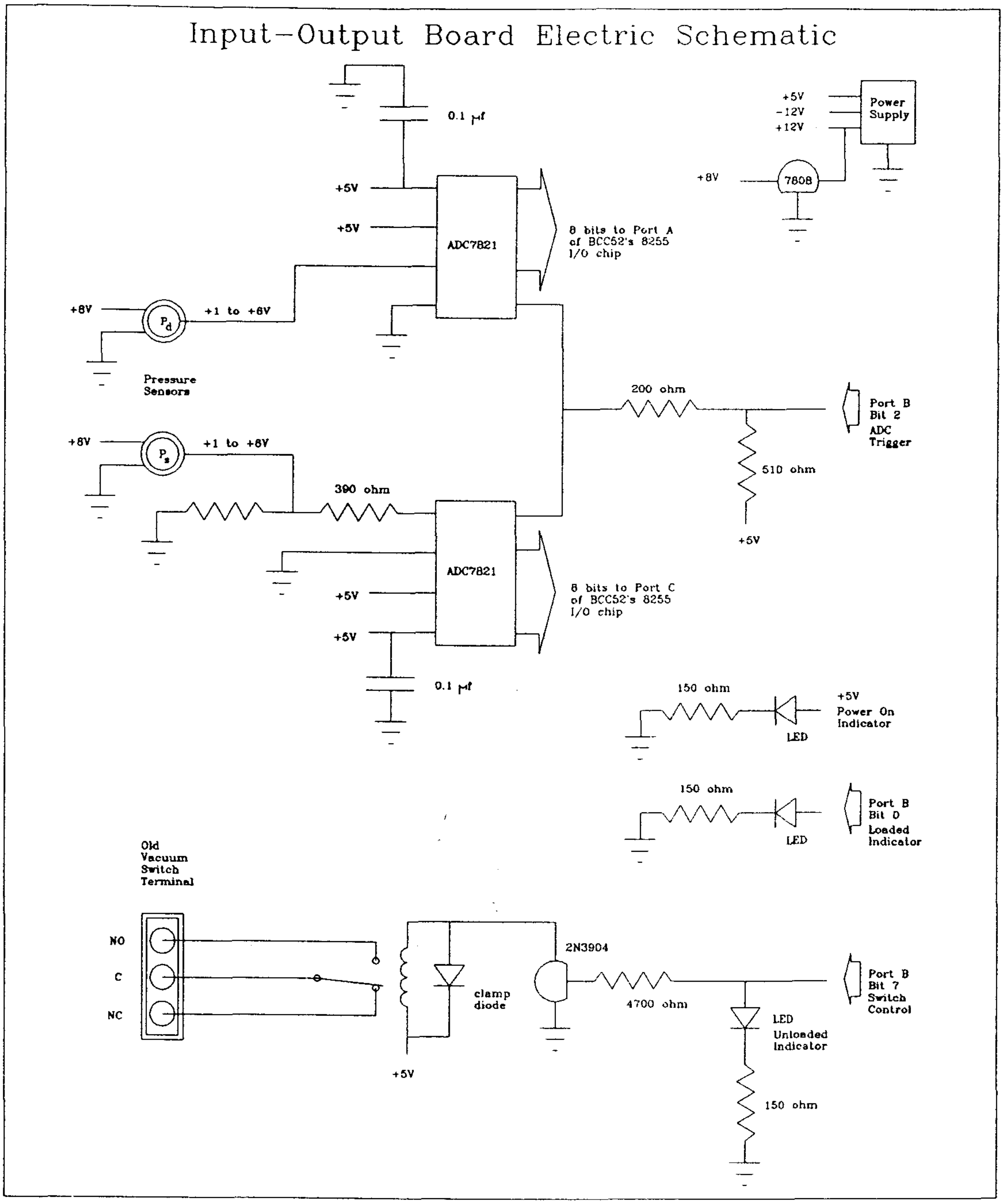

Figure 64: I/O Board Schematic

described in Appendix C, Data Acquisition System, were used for the smart controller.

The only actuating device was an Aromat 5V with a maximum current of 1 amp. The control coil voltage was provided by a 2N3904 transistor from bit 7 of port B. Three output terminals (C, NC, NO) replaced the vacuum switch connectors.

A Micromint modular power supply was used to supply +12V, -12V, +5V, and GND voltages required by the SBC and I/O board.

2. SOFTWARE

a. The BASIC-52 Programming Language. Though programming languages are not usually considered subjects requiring comment, BASIC-52 and its operating environment are unusual. The language is significantly more powerful than a stripped down version of BASIC.

BASIC-52 offers a higher level language than assembly language to work with. Specifically, floating point math can be used, as well as IF, FOR-NEXT, WHILE, and other traditional structured language commands. Since the language was developed specifically for control applications and ROM space was limited, the language was significantly compressed. Most graphical statements were discarded, as well as file linking options. In place of these features, BASIC-52 offers powerful bit control of memory, clock operation, pointers, special function registers, and useful commands that help the language operate in a stand-alone environment. There is also an assembly language interface.

b. Implementation. For major changes in code, the program was downloaded in ASCII format to a PC, edited using a text editor (QEdit), and then uploaded to the SBC. Procomm was used for communication through the RS-232 serial port. When only minor changes were required, the simplest method was to use Procomm and turn the PC into a "dumb" terminal. Then the BCC-52's line editor, which is similar to EDLIN in DOS and equally minimal, was used to write lines of code.

While the program was in development, code was stored in an ASCII file and uploaded to the SBC each day before resuming work. Once the program was completed, which happened several times, the code was burned onto the EPROM on the board. Subsequent revisions required that the EPROM be erased with UV light before new code could be burned in.

c. Program Flowchart. A flowchart of the program is shown in Figures 65 through 68.

d. Annotated Program Listing. An annotated program listing is also included in this appendix.

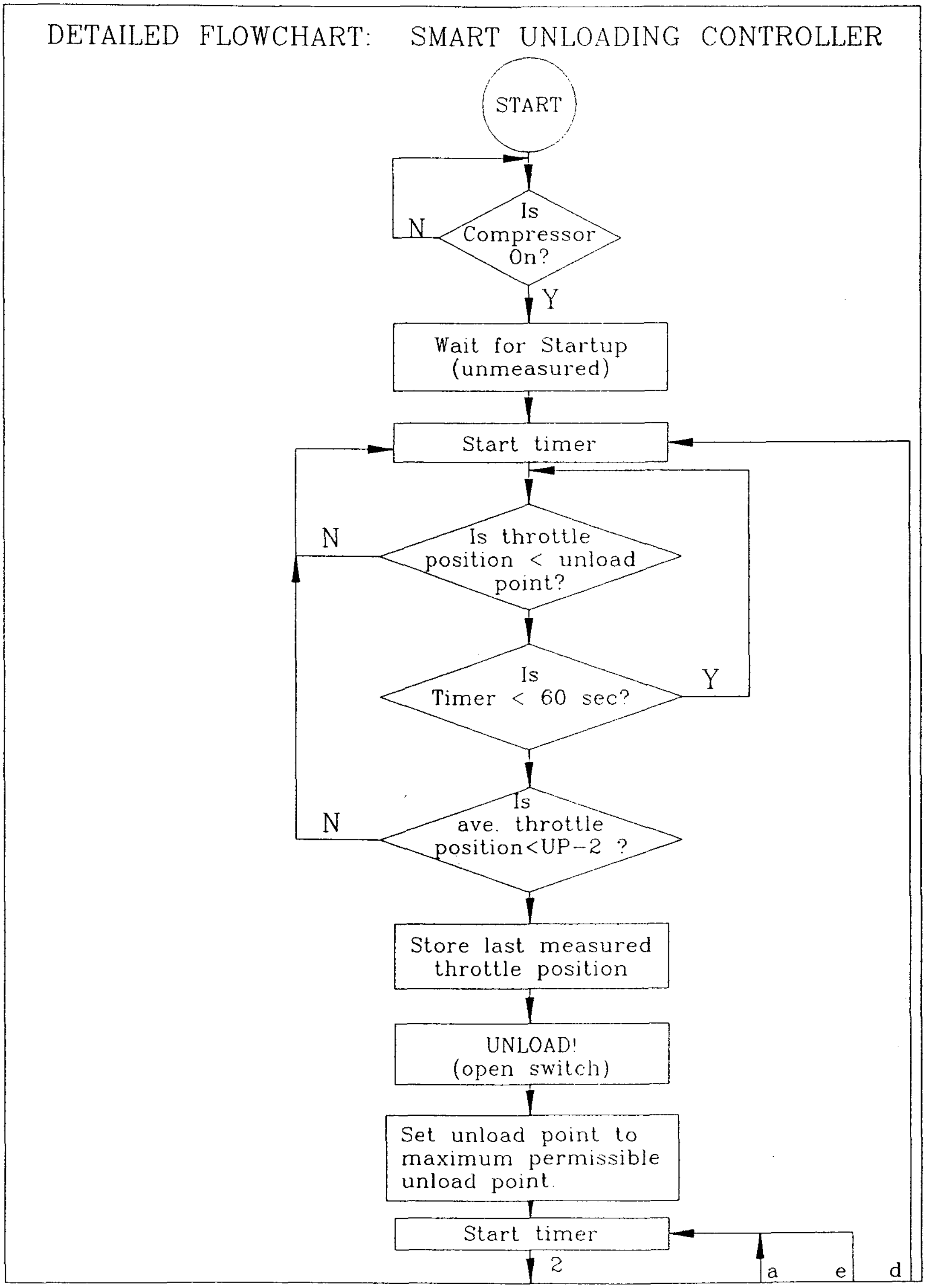

Figure 65: Smart Control Program Flowchart, 1 of 4

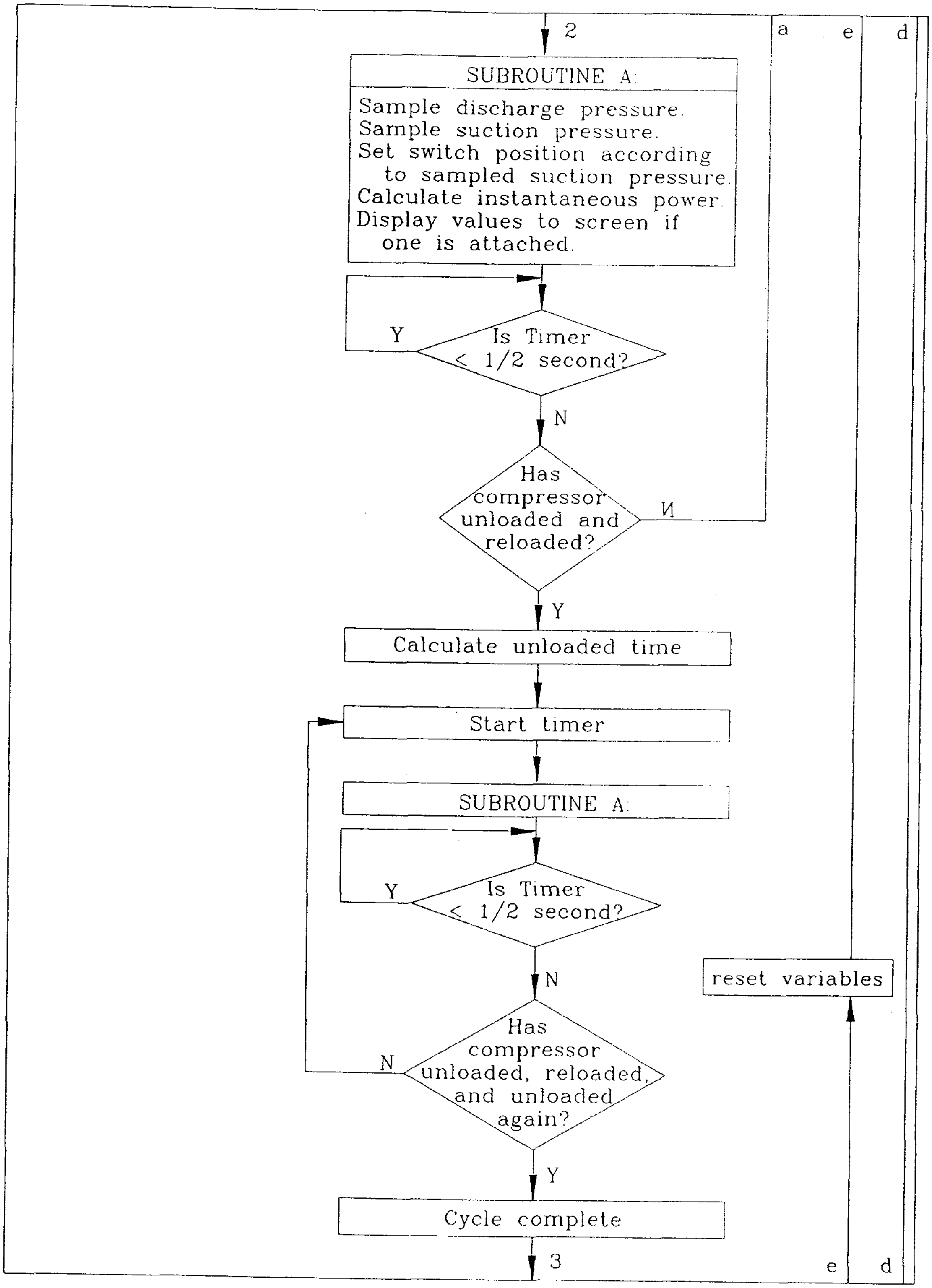

Figure 66: Smart Control Program Flowchart, 2 of 4

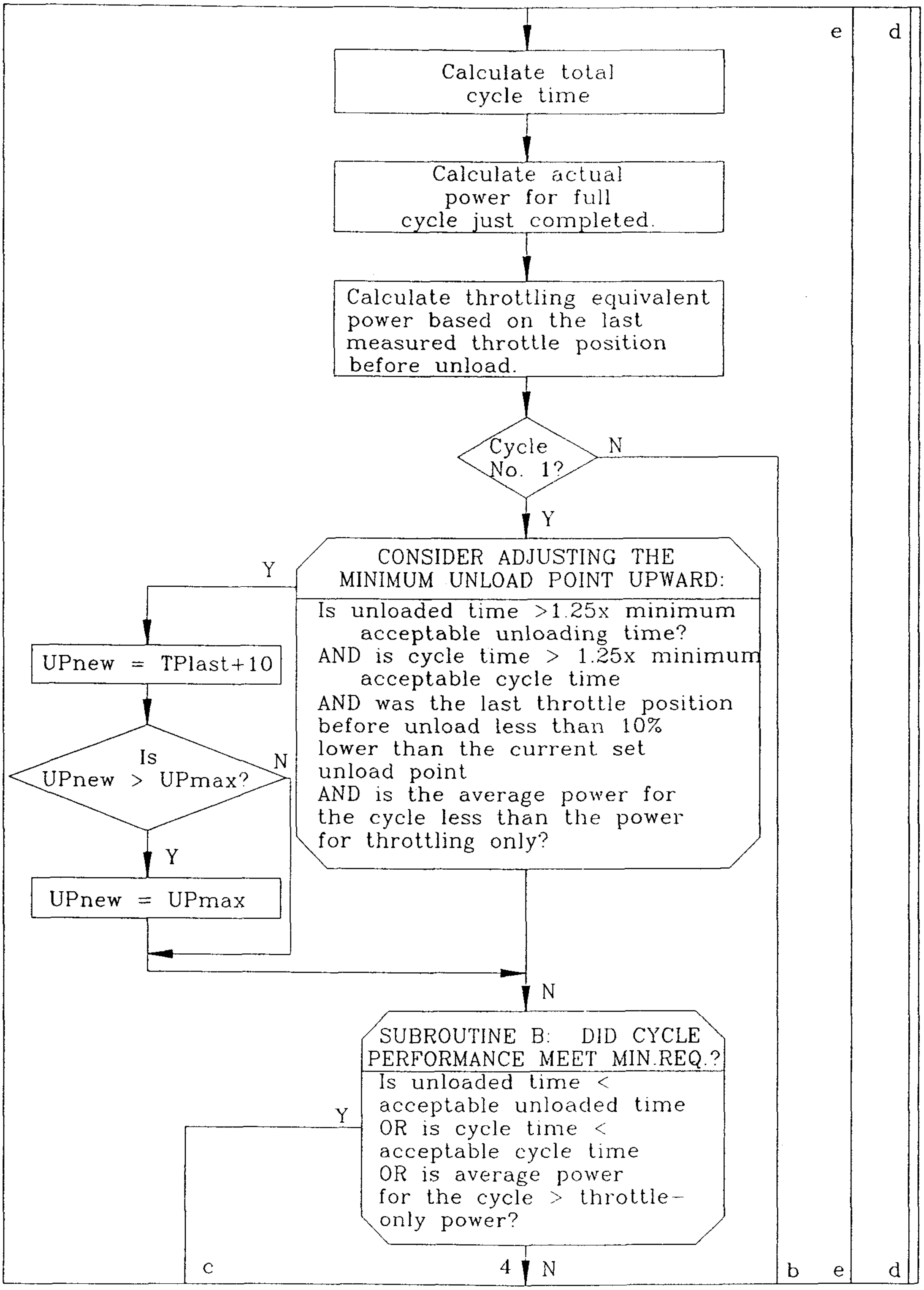

Figure 67: Smart Control Program Flowchart, 3 of 4

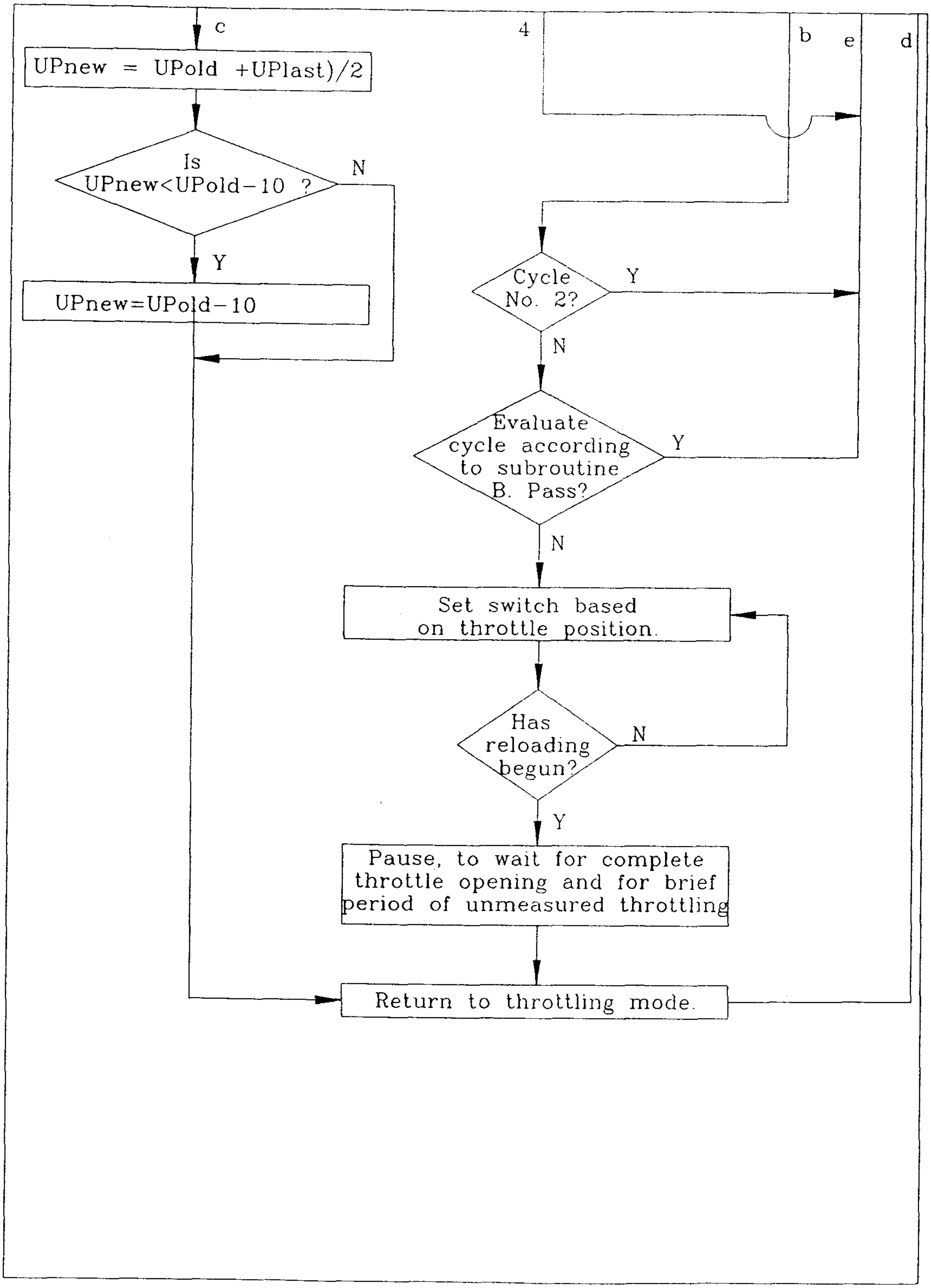

Figure 68: Smart Control Program Flowchart, 4 of 4

SMART CONTROLLER PROGRAM LISTING

```
1    ONEX1 995
10   REM                    Jon Maxwell, August, 1992
20   REM
30   REM    A SMART CONTROLLER TO REDUCE AIR COMPRESSOR ENERGY USE
40   REM
50   REM    This program seeks to reduce the average demand required by
60   REM by an air compressor by sensing demand and directing the compres-
70   REM sor to operate using full load-idle cycling instead of throttling,
80   REM when conditions are appropriate. Inlet and outlet pressure are
95   REM the measured parameters and define the compressor's status. The
96   REM vacuum switch, which defines the unload point, is the controlled
97   REM device.
98   REM
99   REM ----------------------------------------------------------------
100  GOSUB 3100 : REM          assign initial variable values
300  REM ----------------------------------------------------------------
310  TIME=0 : PRINT " **INITIALIZATION, POWER-UP & UNMEASURED THROTTLING***"
320  CLOCK 1 : REM             start clock
330  XBY(0C803H)=99H : REM        configure ports 0C800/1/2 In/Out/Input
340  XBY(0C801H)=80H : REM        red lt. on, confrm closed vac.sw.init
345  VSPOS=1 : REM             indicate vacuum switch position
350  GOSUB 1000 : REM          retrieve input /output pressures
360  IF TP>90.AND.PD<10 GOTO 350
370  REM                       wait for compressor to turn on
400  IF TIME<INITIM GOTO 400 : REM       wait for warm-up
405  REM                ::::::::::::::::::::::temporary:::::::::::
410  GOSUB 1000 : REM          retrieve operating conditions
520  REM
530  REM ----------------------------------------------------------------
560  PRINT "          ****************** INITIAL *****************"
562  REM  Compressor has been throttling for a while. 'Assume' an optimal
564  REM  max.capacity for throttling 95%. Unload if below 95% for several
566  REM  mins.&adjust the setting down later if necessary. This point
568  REM  shouldn't be confused with the fixed unload point for ld-unlding.
570  REM ----------------------------------------------------------------
575  PRINT "          ************ MEASURED THROTTLING ***********"
610  TIME=0 : THPOS=0 : THCNT=0
625  GOSUB 1000 : REM          retrieve conditions
630  IF TP>UP GOTO 610 : REM       check if load is <up% for THRTIM mins.
635  FOR I=1 TO O1000 : NEXT I : REM   pause
640  THPOS=TP+THPOS
650  THCNT=THCNT+1 : REM             chk if ave.ld<(up-avdff) aft 1min
660  THAVPS=THPOS/THCNT
670  IF THAVPS>(UP-AVEDFF).AND.TIME>60 GOTO 610
675  IF TIME<THRTIM*60 GOTO 625
678  REM
679  REM ----------------------------------------------------------------
680  PRINT "               ************UNLOAD!******************"
690  XBY(0C801H)=XBY(0C801H).XOR.81H
710  VSPOS=0 : REM             open vacuum switch
715  TIME=0
720  STRTIM=TIME : REM            store starting time
730  CYCLN=1 : REM            indicate first cycle after unloading
740  TPSTP=TP : REM             store current %capacity/throttle pos.
750  REM
800  REM ----------------------------------------------------------------
805  PRINT "          ******** FIRST LOAD-UNLOAD CYCLE **************"
810  GOSUB 1500
812  REM ----------------------------------------------------------------
```

```
815  PRINT "**** Determine if optimum unload point should be adjusted ****"
820  GOSUB 2500
840  PRINT "unltim:",UNLTIM,"cyctim:",CYCTIM,"avepwr:",AVEPWR,"up:",UP,
850  PRINT "thopwr:",THOPWER,THOPWR,"tpstp",":",TPSTP
860  REM
865  IF UTIMMN>UNLTIM.OR.CTIMMN>CYCTIM.OR.AVEPWR>=UP GOTO 980
870  REM ------------------------------------------------------------
880  PRINT "      **** Monitor during subsequent load-unload cycles ****"
890  TPSTP=LULUP : REM           set LULUP as unload point
900  STRTIM=TIME : REM           set timer for cycling
910  CYCLN=CYCLN+1 : REM            increment cycle number counter
920  GOSUB 1500 : REM            analyze 1 cycle
930  REM
940  IF CYCLN=2 GOTO 900 : REM     partial cycle;no decisions based on it
942  UPSPWR=FLTPWR+(100-FLTPWR)*UP/100
944  UPDPWR=(MINPWR/FLTPWR+(1-MINPWR/FLTPWR)*(PSIX-10)/PSIX)*100
946  UPPWR=UPSPWR*UPDPWR/100 : REM  calculate min.accept.throttle only power
948  REM                       conserv. assume thr.pwr @ psix-10
950  IF UTIMMN<UNLTIM.AND.CTIMMN<CYCTIM.AND.AVEPWR<UPPWR GOTO 900
960  REM                       decide if should switch to throttling
970  REM ------------------------------------------------------------
980  PRINT " ****************** RETURN TO THROTTLING ********************"
990  GOSUB 3500 : REM           wait to resume fully reloaded oper.
992  GOTO 575 : REM            start over
994  CLOCK 0
995  XBU(0C801H)=080H : PRINT "Stopped because of Interrupt. Switch reset"
996  END : REM              **************END*******************
997  REM ===========================================================
998  REM ===========================================================
999  REM
1000 REM --------Subroutine to retrieve suction & discharge pressure------
1004 REM
1005 GOSUB 4500 : REM                 print status
1010 REM
1020 A=0 : C=0 : REM                  reset counters for averaging
1040 FOR I=1 TO NS
1050 XBY(0C801H)=4H.OR.XBY(0C801H) : REM      trigger bit3 of outport high
1060 XBY(0C801H)=4H.XOR.XBY(0C801H) : REM      trigger low;initiate two A->D
1070 REM                           conversions;(XOR complements)
1080 A=A+XBY(0C800H) : REM                retrieve value from ADC chans
1090 C=C+XBY(0C802H) : REM                   A&C and add to previous
1100 NEXT I
1105 A=A/NS : C=C/NS : REM              average ADC values
1110 A=5/255*A : REM                  convert from bits to volts
1115 A=A-1 : REM                    convert 1-6V to 0-5V(no>5)
1117 A=99/100*(A-0.05) : REM *adl.corr.due to Vref not perfect for transducr
1120 A=250/5*A : REM                  convert volts to psig
1123 IF A>PSIX THEN PSIX=A : REM         check that assumed max.disc.press
1126 REM                          isn't exceeded & reset if it is.
1130 PD=100*A/PSIX : REM                convert psig to %max psig
1140 REM                          -------------------------
1150 C=5/255*C : REM                  convert from bits to volts
1155 C=C*6/5-1 : REM  convert form (1-6V)*5R/6R to 0-5 V (inp may be 5-6V)
1157 C=14/15*(C-0.03) : REM  *addl.corr.due to Vref not perfect for transdcr
1160 C=-15/5*C : REM                  convert volts to psig
1163 IF C>TPN.AND.C<0 THEN TPN=C : REM       check assumd max&min.suct.pres
1164 IF C<TPX.AND.C>14.5 THEN TPX=C : REM     aren't exceeded & reset if are
1170 TP=100*(TPX-C)/(TPX-TPN) : REM         convert psig to throttle pos.
1190 RETURN
1500 REM ------------------------------------------------------------
1510 REM ----Subroutine to analyze compressor power for 1 cycle-----------
1520 REM
```

```
1525 FLAG1=0 : FLAG2=0 : FLAG3=0 : PWR=0 : PWRCNT=0 : UNLTIM=0
1560  GOSUB 1000 :  REM             retrieve suction/discharge conditions
1565  GOSUB 2000 :  REM             set proper vacuum switch position
1570  IF TIME<0.5 GOTO 1570 : TIME=0 :  REM pause for 1/2 second
1580 SCTPWR=FLTPWR+(100-FLTPWR)*TP/100
1590 DISPWR=MINPWR/FLTPWR*100+(1-MINPWR/FLTPWR)*PD
1600 PWR=SCTPWR*DISPWR/100+PWR :  REM  calculate power based on conditions
1610 PWRCNT=PWRCNT+1
1620  IF TP>TPSTP THEN FLAG1=1 :  REM   set flag1 if throttle has
1625  REM                        finished closing and has opened back
1626  REM                        up past the initial unloading point.
1627  REM
1630  IF FLAG1=0 GOTO 1560 :  REM     if neither condition is met
1635  IF TP>TPSTP GOTO 1560 :  REM     throttle closed enough to compl cycle.
1640 AVEPWR=PWR/PWRCNT :  REM          calculate ave.power for 1 full cycle.
1645 CYCTIM=TIME-STRTIM
1650  RETURN
1660  REM -------------------------------------------------------------
2000  REM ---Subroutine to determine vacuum switch position when cycling---
2005  REM
2010  IF TP<MAKBRK THEN VSPOS=0 : XBY(0C801H)=XBY(0C801H).OR.1H
2015  IF TP<MAKBRK THEN FLAG3=1 : XBY(0C801H)=XBY(0C801H).AND.07FH
2017  IF TP<MAKBRK GOTO 2490
2018  REM                        send 1 to switch and green light
2019  REM                        send 0 to red light
2020  IF TP>=LULUP THEN VSPOS=1 : XBY(0C801H)=XBY(0C801H).AND.0FEH
2030  IF TP>=LULUP THEN FLAG2=1 : XBY(0C801H)=XBY(0C801H).OR.80H
2032  IF UNLTIM=0.AND.TP>LULUP THEN UNLTIM=TIME-STRTIM
2035  IF TP>=LULUP GOTO 2490
2036  REM                        send 0 to switch and green light
2037  REM                        send 1 to red light
2038  REM                        measure unloaded time (exception)
2040  IF FLAG2=0.AND.FLAG3=1.AND.VSPOS=0 THEN UNLTIM=TIME-STRTIM
2042  REM                        measure unloaded time (normal)
2045  IF FLAG3=0.AND.FLAG2=0 GOTO 2490
2047  REM                        if initial unloading (exception)
2050  IF FLAG2=0 THEN VSPOS=1 : XBY(0C801H)=XBY(0C801H).AND.0FEH
2055  IF FLAG2=0 THEN XBY(0C801H)=XBY(0C801H).OR.80H
2057  REM                        if reloading, close switch (normal)
2060  IF FLAG2=1 THEN VSPOS=0 : XBY(0C801H)=XBY(0C801H).OR.1H
2065  IF FLAG2=1 THEN XBY(0C801H)=XBY(0C801H).AND.07FH
2070  REM                        if unloading, open switch (normal)
2072  REM
2074  REM                         close vacuum switch if tp>lulup
2076  REM                         open vacuum switch if tp<makbrk
2078  REM                         open v.s. if tp<lulup & unloading
2080  REM                         open v.s. if tp<lulup & unloading
2082  REM                         (lulup is always bigger than makbrk)
2084  REM                         record time as tp exceeds makbrk (or
2086  REM                         lulup, if tp never drops below makbrk)
2490  RETURN
2499  REM -------------------------------------------------------------
2500  REM --------- Determine if optimum unload point is valid -----------
2510  REM
2520  REM                         assumes that capacity is the same
2530  REM                         immediately before unloading and for
2540  REM                         first cycle after unloading.
2550  REM
2560 THOPWR=FLTPWR+(100-FLTPWR)*TPSTP/100 :  REM calc ave.power when thrtlng
2570  REM
2575  IF AVEPWR>=THOPWR.AND.TPSTP>UP-20 THEN UP=(UP+TPSTP)/2 :  GOTO 2999
2576  IF AVEPWR>=THOPWR THEN UP=UP-10 :  GOTO 2999
```

```
2578  IF UNLTIM<UTIMMN.AND.TPSTP>UP-20 THEN UP=(UP+TPSTP)/2 :  GOTO 2999
2579  IF UNLTIM<UTIMMN THEN UP=UP-10 :  GOTO 2999
2580  IF CYCTIM<CTIMMN.AND.TPSTP>UP-20 THEN UP=(UP+TPSTP)/2 :  GOTO 2999
2585  IF CYCTIM<CTIMMN THEN UP=UP-10 :  GOTO 2999
2590  REM      don't reduce min.cap.for throttling(UP)if ave.power, complete
2600  REM      cycle time, and unloaded time all meet limiting requirements.
2620  REMNew minimum power -s halfway between measured tpstp and previous
2630  REMup.  This is effictively proportional control that has been
2640  REMtempered because of the ssumption stated in lines just above.
2650  IF TPSTP<UP-10 GOTO 2999
2660  REM      if unloading point was between 1% & 10% lower than UP, conside
2670  REM      increasing UP.  Don't try to increase UP above LULUP
2700  IF UNLTIM>1.25*UTIMMN.AND.CYCTIM>1.25*CTIMMN.AND. EQVCAP<UP-2 THEN UP=TPSTP+10
2710  REM      increase UP by a fixed 1% if capacity was close to max. for
2720  REM      ld-unlding decision, UP is less than ld-unld pt for cycling,
2730  REM      cycle time is twice as long as the minimum, unloaded time is
2740  REM      twice as long as the minimum, and increasing it will not cause
2750  REM      UP to exceed ld-unld UP.
2999  RETURN
3000  REM
3099  REM -----------------------------------------------------------------
3100  REM                        *** INITIAL VARIABLE ASSIGNMENT ***
3110  NS=25
3120  PSIX=75 :  REM psig    Guess smaller number if not sure
3130  TPX=-13.5 :  REM psig   Guess bigger number if not sure
3140  TPN=-0.5 :  REM psig    Guess smaller number if not sure
3150  THRTIM=1 :  REM minutes
3160  MINPWR=16 :  REM %full capacity power
3170  FLTPWR=68 :  REM %full capacity power
3180  UP=90 :  REM %open       Guess bigger number if not sure
3190  LULUP=95 :  REM  %open  Guess smaller number if not sure
3200  MAKBRK=40 :  REM %open
3210  UTIMMN=20 :  REM secs
3220  CTIMMN=60 :  REM secs
3230  AVEDFF=2 :  REM %load
3240  INITIM=30 :  REM secs
3250  PRNTSG=0
3490  RETURN
3500  REM -----------------------------------------------------------------
3510  REM ----------- Subroutine used to return to throttling -------------
3520  REM
3525  FLAG2=0 : FLAG3=0
3530  GOSUB 1000 :  REM              check conditions
3540  GOSUB 2000 :  REM              set vacuum switch as necessary
3550  IF VSPOS=0 GOTO 3530 :  REM      wait for reloading to begin
3560  STRTIM=TIME
3670  IF TIME-STRTIM<60 GOTO 3670
3680  REM                          allow time for throttle to fully open
3685  STRTIM=0
3990  RETURN
4500  REM -----------------------------------------------------------------
4506  IF PRNTSG=1 GOTO 4590
4510  PRINT " throttle position:  ",TP,"% open"
4520  PRINT "discharge pressure:  ",PD/100*PSIX,"psig"
4530  PRINT "optimal unld point:  ",UP
4540  PRINT "   switch position:  ",VSPOS
4550  PRINT "      elapsed time:  ",TIME-STRTIM
4555  PRINT "          max psig:  ",PSIX," psig"
4560  IF DISPWR>0 THEN  PRINT "instantaneous pwr: ",SCTPWR/100*DISPWR," %"
4570  PRINT
4580  REM FOR J=1 TO 1000 :  NEXT J
4585  PRNTSG=1 :  GOTO 4700
```

```
4590 PRNTSG=0
4700  RETURN
     REM ----------------------------------------------------------------
     REM                    ****** VARIABLE DESCRIPTIONS ******
     REM
     REM           - transducer performance both 0-5V, 0->-15psi,0->250psi
     REM     A      - variable used when signal is retrieved from Port A
     REM     avedff - %load differential between min.throt.abs.&min.thr.ave
     REM              should reflect consideration for cycling losses
     REM     avepwr - ave.power for first load-unld cycle after thring (%)
     REM     C      - variable used when signal is retrieved from Port C
     REM     clock1 - turns time incrementer on (clock0 turns it off)
     REM     ctimmn - minimum acceptable time for 1 complete ld-unld cycle
     REM     cyctim - measured time for 1 complete load-unload cycle
     REM     cycln  - cycle number, when load-unloading
     REM     dispwr - power adjustment due to discharge pressure (%)
     REM     eqvcap - equivalent capacity as if throttling, based on averag
     REM              power measure over 1 load-unload cycle(%, may be -)
     REM     flag1  - indicates if reloading has exceeded UP for 1st cycle
     REM     flag2  - indicates if reloading has exceeded LULUP
     REM     flag3  - indicates if thr.pos.is still above MAKBRKn when unldng
     REM     fltpwr - power fully throttled, high disch press. (%)
     REM     initim - initial time waiting for compressor to start up
     REM     lulup  - Unload point when load-unload cycling (%)
     REM     makbrk - point during reloading when vacuum switch closes(thr%)
     REM     minpwr - minimum possible unloaded power (%)
     REM     mnctim - min. acceptable time for 1 complet ld-unld cycle(sec)
     REM     ns     - number of signals to average w/ ADC
     REM     oldtim - timer counter
     REM     pd     - discharge pressure (% of psix)
     REM     psix   - default compr.max oper press(psig;changed dynamicly)
     REM     pwr    - single calculated power measurement (%)
     REM     pwrcnt - counter used in avepwr loop to follow # of measrmnts
     REM     sctpwr - power adjustment due to suction pressure (%)
     REM     strtim - stores 'initial time' for various operations
     REM     thavps - average throttle position after several samples
     REM     thcnt  - counter to get thavps from thpos
     REM     thopwr - power for throttling-only oper. @ a given%Cap. (%FLP)
     REM     thpos  - throttle position for a single gosub
     REM     thrtim - time load must throttle below UPopt b/f switch to
     REM              cycle mode (minutes)
     REM     time   - special BASIC-52 variable(secs.,automaticly increms.)
     REM     tp     - throttle position (%open)
     REM     tpn    - suction pressure w/ fully open thr.(psig)
     REM     tpstp  - throttle stop indicates first thrpos measrd. after UL
     REM     tpx    - suction pressure w/ fully closed thr.
     REM     unltim - measured time from begin unlding to ~1/2 reloaded
     REM     up     - UP is %Capacity below which load-unloading is better.
     REM          - To start out, set assumed optimum unldpt. high
     REM     upspwr - pwr adjustment due to suct.press.when calcing uppwr
     REM     updpwr - pwr.adjustment due to disc.press.when calcing uppwr
     REM     uppwr  - power required when throttling at up %capacity
     REM     utimmn - minim. acceptable time from unlding to reloading(sec)
     REM     vspos  - indicator of vacuum switch position (0=open,1=closed)
     REM     xby    - used to address a specific byte address in data mem.
```

APPENDIX I: SMART CONTROLLER HARDWARE COST

This is definitely a prototype design. The intent was not to develop a production design with simple integrated circuits and sensors, but to validate the smart control concept. With this in mind, the following costs are estimated:

Qty.	Item	Each	Total
(1)	Micromint BCC-52 single board computer		$189.00
(1)	Micromint PS11 modular regulated power supply		19.00
(1)	EPROM		8.00
(4)	8k RAM	5.00	20.00
(2)	Omega 24x series pressure transducers	138.00	276.00
(1)	Aromat 5V 1A dc coil relay		2.50
(200')	Small gauge wire		16.00
(1)	3 Screw terminal board		0.50
(2)	1/4" tees	0.50	1.00
(1)	8V Voltage regulator		1.50
(2)	Analog-Digital Converters	10.00	20.00
(3)	LEDs	0.17	0.50
(1)	2N3904 transistor		1.00
(8)	1/4 watt resistors	0.06	0.50
(2)	.1 ^{n}f capacitors	0.20	0.40
(1)	26 pin Berg connector and cable		4.00
(1)	24' extension cord		8.00
(1)	I/O prototype board		5.00
(misc)	Pipe wrap, teflon tape, makeshift case, connecting wire		10.00
TOTAL COST			$583.00