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**ENGINEERING REPORT**

OXYGEN BOOST COMPRESSOR STUDY

74-410521A April 15, 1974

*NAS 9-10465*

(NASA-CR-141938) ENGINEERING REPORT:  
OXYGEN BOOST COMPRESSOR STUDY (AirResearch  
Mfg. Co., Phoenix, Ariz.) 12 p HC \$3.25

N75-28427

CSCL 13I

Unclas  
31065

G3/37



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA  
A DIVISION OF THE GARRETT CORPORATION

# ENGINEERING REPORT

OXYGEN BOOST COMPRESSOR STUDY

74-410521A

April 15, 1974

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REV	BY	APPROVED	DATE	PAGES AND/OR PARAGRAPHS AFFECTED
NC	LST	RWG	3-11-74	Original Issue
A	LST	RWG	4-15-74	Report revised to present the latest test results and recommendations.



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## ENGINEERING REPORT OXYGEN BOOST COMPRESSOR STUDY

### 1. INTRODUCTION

The information in this report presents the initial results of a study program conducted by AiResearch on an oxygen boost compressor to support an AiResearch-designed, self-contained life support system.

The material presented herein includes a description of the compressor, the results of the preliminary analysis, the test results, and recommendations for follow-on efforts.

### 2. DESCRIPTION OF COMPRESSOR

The compressor is used to compress breathing oxygen to a self-contained life support system for extra-vehicular activities (EVA) in connection with the space shuttle program. The concept employs high pressure bottles in a manner similar to the SCUBA equipment used by underwater divers, thereby eliminating the need for an umbilical cord.

The compressor system is intended to utilize gaseous oxygen obtained from the cryogenic storage bottles used for the cabin oxygen supply. The cabin requires a flow of approximately 1.25 lb per hour at a pressure of 900 psia.

The program conducted by AiResearch is intended to provide a compressor that can be used to fill the EVA storage bottles from the 900-psia cabin supply and compress the oxygen to a final bottle pressure of 4000 psia. Because of the relatively low inflow rate, proper management of the inflow gas is critical for two functions: 1) the flow rate must be controlled within limits, and 2) the pressure energy within the inflowing gas is to be used to provide the best refill time.

The compressor selected for this application consists of a free-piston type. The inflow or cabin supply is used to operate a piston with a large area. The large area piston is connected to a smaller piston that is constructed as a compressor. A mechanism referred to as an automatic cycle valve was devised to store the inflowing gas in a capacitance chamber until the energy buildup is sufficient to initiate a cycle. At this point, the cycle valve mechanism suddenly releases the stored energy into the large (motor piston) area. The gas stored in the capacitance chamber is then expanded behind the motor piston to drive the compressor piston through a compression cycle. The expanded gas is then released into the cabin for use as the cabin supply. The required cabin flow of 1.25 lb per hour is maintained by feeding the capacitance chamber through an orifice.



### 3. PRELIMINARY OXYGEN BOOST COMPRESSOR ANALYSIS

#### 3.1 Introduction

An analysis of the boost compressor was performed to demonstrate its operability and to provide preliminary sizing of various parameters so that detailed design could proceed.

The results of the analysis consisted of computer printouts which recorded the values of system variables as a function of time. The data obtained from the computer printouts was then reduced and the plot shown in Figure 1 was prepared to present the performance of the compressor. Figure 1 is a plot of storage tank pressure versus a cabin flow-time parameter denoted

$$\frac{W_c t}{V_t}$$

where

$W_c$  = cabin flow rate, lb per hour,

$t$  = time, hours, and

$V_t$  = storage tank volume, in.<sup>3</sup>.

This section of the report presents details of the analysis and a comparison of the analysis with initial test data.

#### 3.2 Analysis

A schematic diagram of the system is shown in Figure 2. Tank pressure,  $P_5$ , was computed by integrating the mass flow rate of gas through the check valve and applying the ideal gas law for an assumed tank volume. Tank temperature was assumed to be equal to compressor temperature,  $T_4$ .

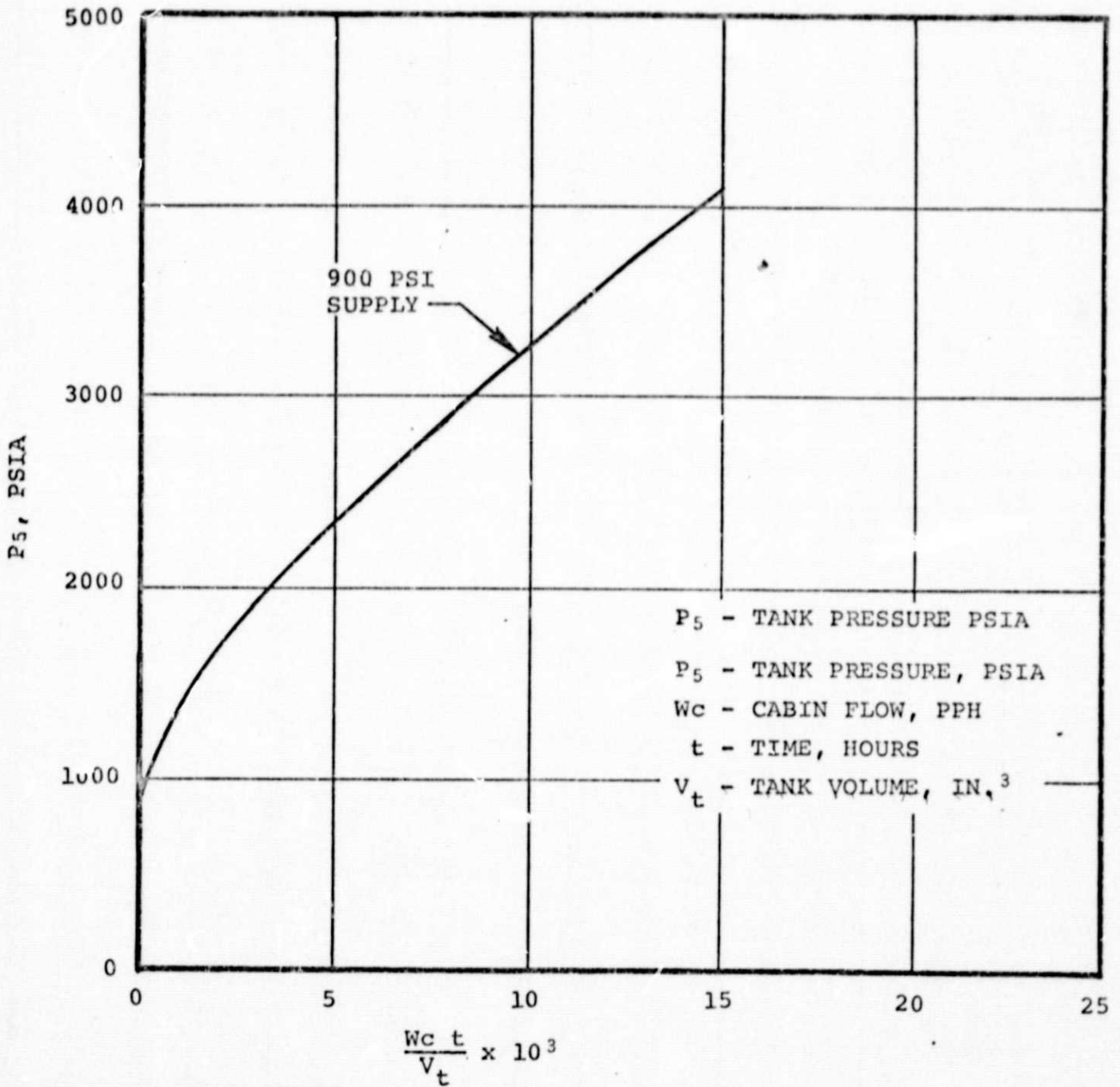


FIGURE 1

PERFORMANCE OF GASEOUS OXYGEN COMPRESSOR



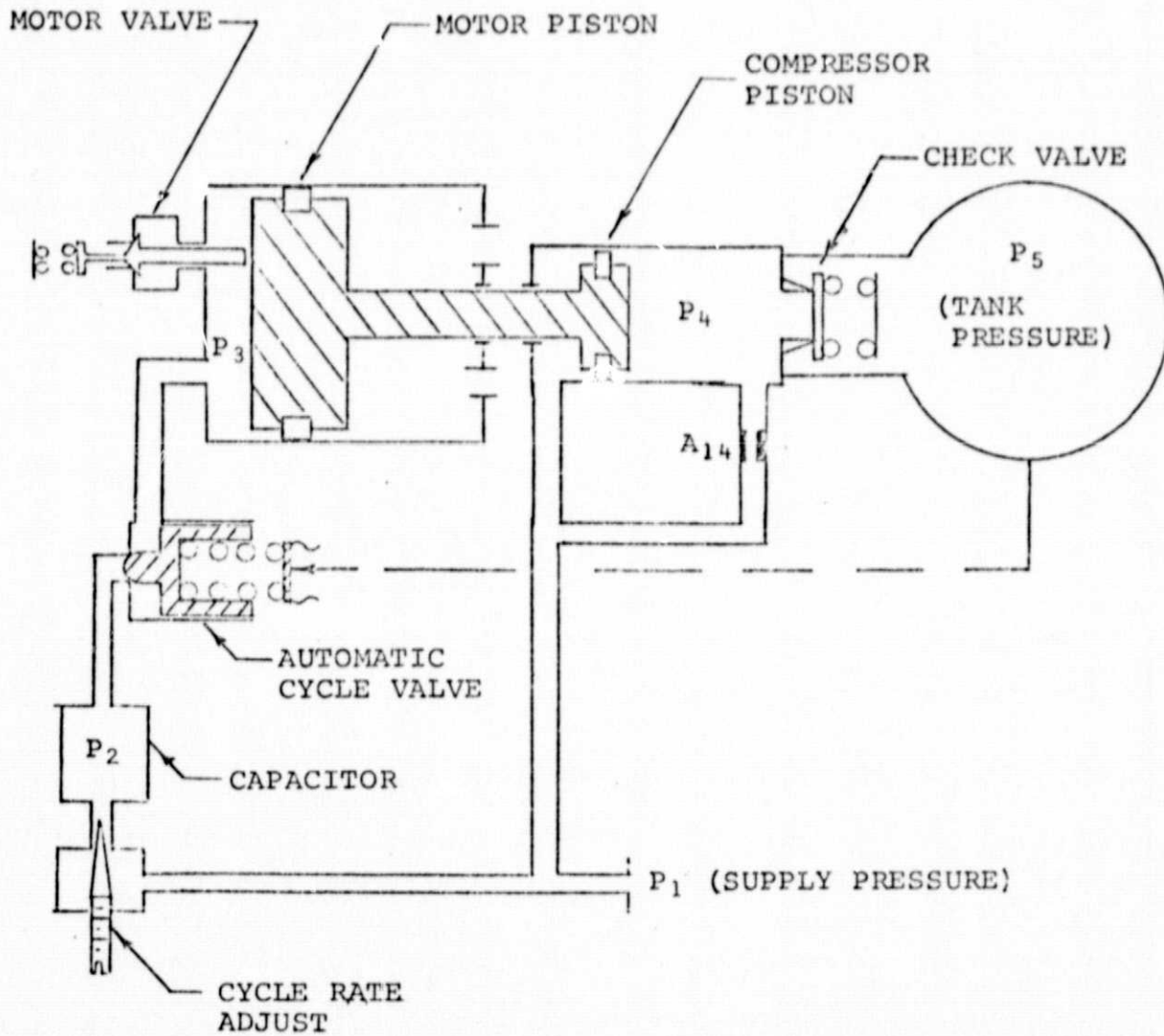


FIGURE 2

SCHEMATIC OF BOOST COMPRESSOR



The mass flow rate of gas through the check valve was computed using the relationship,

$$W_{45} = \frac{K P_4 A_{45} N_{45}}{\sqrt{T_4}}$$

where

$W_{45}$  = mass flow rate, lb per second,

$K$  = oxygen gas constant, 0.5586  $\sqrt{\text{deg R}}$  per second,

$P_4$  = compressor pressure, psia,

$A_{45}$  = check valve effective area, in.<sup>2</sup>,

$N_{45}$  = compressor flow factor, dimensionless, and

$T_4$  = compressor temperature, deg R.

The check valve effective area was computed as a function of a force unbalance resulting from pressures  $P_4$  and  $P_5$  and from the check valve spring force.

The compressor gas temperature,  $T_4$ , was computed from the isentropic relationship,

$$T_4 = T_1 \left( \frac{P_4}{P_1} \right)^{\frac{\gamma - 1}{\gamma}}$$

where

$T_1$  = supply gas temperature, deg R,

$P_1$  = supply gas pressure, psia, and

$\gamma$  = ratio of specific heats, dimensionless.

Compressor pressure,  $P_4$ , was computed by integrating the net flow ( $W_{14} - W_{45}$ ) of gas into the compression chamber and then applying the ideal gas equation of state. The volume of the compression chamber was computed as a function of compressor piston area and stroke. The position of the compressor piston was obtained by integrating the piston acceleration twice. Piston acceleration was obtained by dividing the net force unbalance on the motor-compressor piston by the mass of the moving part.



The net force unbalance on the piston was obtained by considering the effect of compressor pressure ( $P_4$ ), supply pressure ( $P_1$ ), seal friction (assumed to be 50 pounds), ambient pressure, and motor pressure ( $P_3$ ).

Motor pressure,  $P_3$ , was computed by integrating the net flow rate of gas into the motor expansion chamber and then applying the ideal gas equation of state. Adiabatic expansion of the gas was assumed. The net flow rate of gas into the expansion chamber consisted of flow in from the automatic cycle valve and out through the motor valve. The motor valve area was obtained from a summation of forces considering the motor valve spring force and the motor pressure. The valve was held open by the spring force when motor pressure dropped below 25 psig. The motion of the piston at the end of the return stroke served to close the motor valve.

The design of the motor valve described above resulted from a study of several alternative methods, including the use of side vents and operating the motor valve from the automatic cycle valve. The resulting design of the motor valve was considered to be the best in terms of performance and simplicity.

Other ideas were considered for the basic operation of the device. One approach was to let supply pressure "cock" a negative rate spring and then use the energy stored in the spring to provide the compressive energy. Another approach was to create a negative rate effect by a properly designed kinematic linkage between the motor and compressor piston. Each of these methods was simulated with the computer; however, each was considered to be unnecessarily complex and showed no significant performance gain over the schematic arrangement presented herein.

The flow of gas into the motor expansion chamber was computed from the capacitor pressure,  $P_2$ , and the flow area of the automatic cycle valve. The flow area of the automatic cycle valve was considered to be a two-position function depending upon the direction of the summation of forces. Opening forces consisted of capacitor pressure times seat area plus motor pressure times shoulder area. The closing force consisted of a spring preload proportional to tank pressure. The stepped areas on the automatic cycle valve, in conjunction with the capacitor volume, provided an oscillating effect; the cycle valve remained closed until the capacitor pressure,  $P_2$ , increased sufficient to overcome the spring force. The cycle valve remained open until the motor pressure,  $P_3$ , dropped sufficient for the spring force to close the valve.



The computer analysis showed that the tank pressure feedback on the automatic cycle valve mechanism was necessary. Without tank pressure feedback, the spring preload on the cycle valve was constant; therefore, the energy released during the expansion in the motor chamber was constant for each cycle. However, the quantity of energy required to complete a cycle was proportional to the pressure level in the tank. Therefore, if the preload was set high enough for the unit to cycle at high tank pressure, energy in the form of excessive impact speeds at the end of the compression stroke was wasted during the low pressure cycles. With tank pressure feedback, the energy supplied by the capacitor is more nearly equal to that required at each tank pressure condition.

The capacitor pressure was obtained by integrating the net mass flow into the capacitor and then applying the ideal gas equation of state. The flow rate of gas into the capacitor was set by the supply pressure and the cycle rate needle valve adjustment.

### 3.3 Comparison of Analysis with Initial Testing

Performance of the compressor end of the device was considered to be acceptable and was very similar to that obtained analytically. Some problems were encountered in causing the device to cycle at all pressure levels, however. The problem was believed to result from insufficient opening of the automatic cycle valve mechanism, thereby preventing a complete stroke of the piston. The cycle valve did not open completely because of the rate provided by the feedback spring. The problem was avoided and the unit was made to cycle by altering the schematic as shown in Figure 3. Until the motor piston moves, motor pressure acts upon a decreased area, thus insuring that motor pressure builds up to a high value before the piston can move. In addition, an orifice was placed between the cycle valve and the expansion chamber to maintain an elevated pressure on the cycle valve shoulder so that it would remain open long enough for the piston to complete its stroke.

Also shown in Figure 3 is a simplified motor valve which is simply an orifice. Preliminary computer performance runs indicate that the modifications shown will result in proper operation of the compressor.

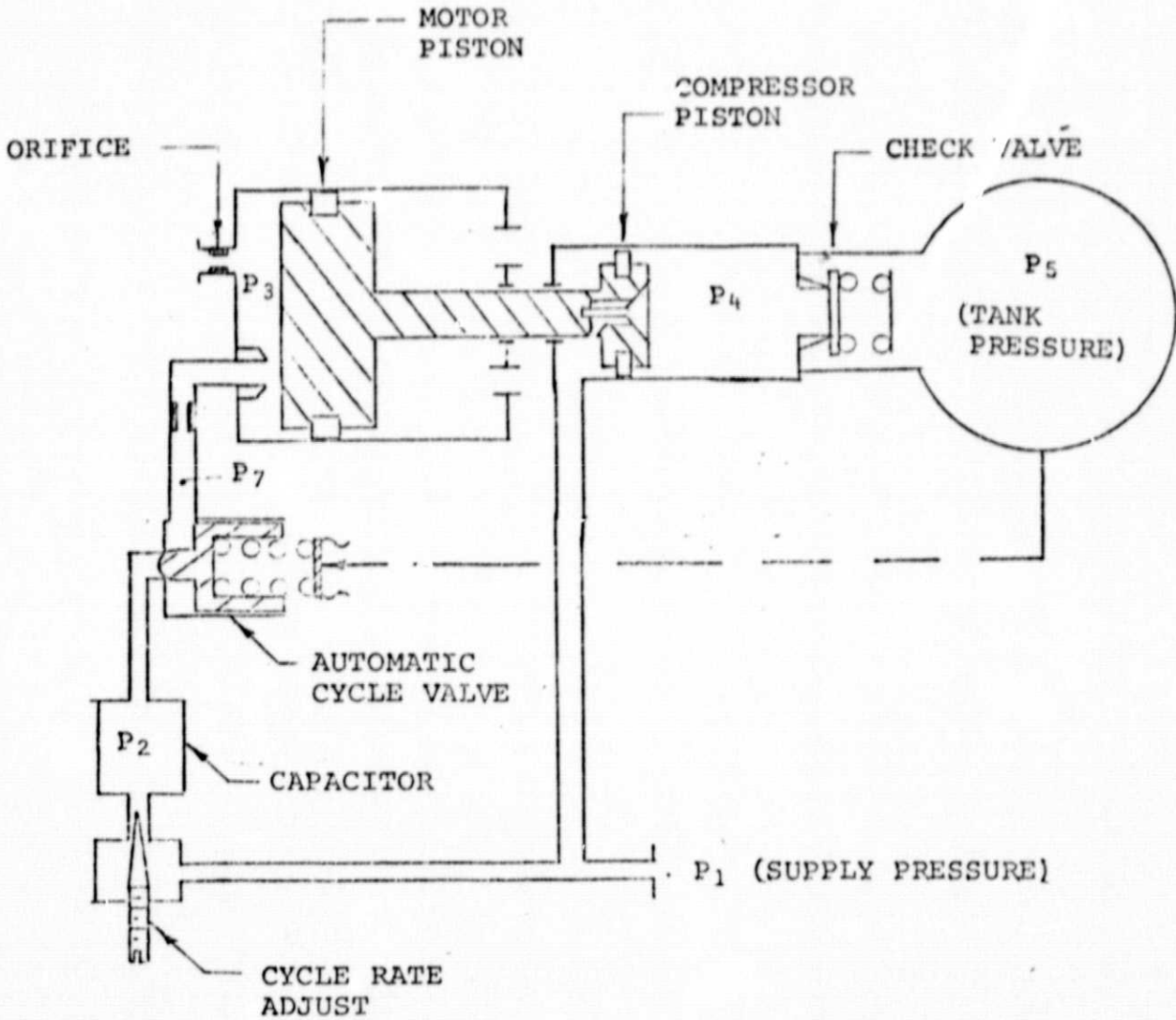


FIGURE 3

SCHEMATIC OF MODIFIED BOOST COMPRESSOR





4. TEST RESULTS

4.1 Weight and Leakage

The weight and leakage characteristics of the unit are tabulated below.

Weight

Weight of the preliminary rough cut block version:  
8.4 pounds

Leakage

Compressor Piston Shaft Seal (Item 48): 0.000167 lb per hr at a supply pressure of 900 psig

Compressor Piston (Item 14): 0.025 lb per hr at 900 psig

Compressor Check Valve (Item 37): 4.5 psi drop over a 12-hour period with a pressure of 300 psig in the capacitance chamber

4.2 Component Performance

4.2.1 Compressor Section - This test was conducted by driving the compressor piston, Item 13, by cycling a solenoid switcher valve. The compressor section was supplied with dry air from a separate source at 900 psig and was connected to a 16.86 cubic inch volume. The pressure to the solenoid was supplied through an orifice into a 1.3 cubic inch volume. This pressure was manually varied to provide a complete stroke of the motor piston as the bottle pressure increased. The following tabulation presents the results of the tests.

<u>Bottle Pressure</u> psig	<u>Number of Cycles</u> <u>for Sample</u>	<u>Compressor Flow</u> lb per cycle
1000	10	0.000542
2600	17	0.001035
1000 - 3000	117	0.000834

Note: Theoretical flow of the compressor is 0.001118 lb per cycle.



The pressure for this test was limited to 3000 psig due to bottle capacity and handling limits imposed on test personnel. Test data showing compressor section efficiency following successful development of the automatic cycling mechanism is presented on Figure 4, which follows.

Several test runs are shown on Figure 4 where the compressor supply was dry air at 900 psig, and the compressor was pumping into a 26 cubic inch bottle. As shown, two curves are plotted on Figure 4 which presents the number of cycles versus receiver bottle pressure. The theoretical plot on Figure 4 presents a curve of 100 percent compressor efficiency for comparison with the test results. Data Plot No. 1 depicts a complete bottle pumping cycle from 900 psig to 4000 psig. As shown, the compressor efficiency is reduced at the higher pressure and with an increased number of cycles. An investigation into the reason for this characteristic was performed and the results are presented in paragraph 4.4, SUMMARY OF RESULTS, herein.

Data Plot No. 2 in Figure 4 shows a test run where the feedback mechanism was adjusted to provide maximum compressor efficiency. As noted, high efficiency in the compressor section is possible; however, the motor gas consumption rate is increased. The reason for the increase in gas consumption (Plot No. 2) and the loss of efficiency (Plot No. 1) is the same. This matter is also discussed in paragraph 4.4.

4.2.2 Automatic Cycle Mechanism and Feedback - Subsequent tests were conducted to establish automatic cycling controlled by compressor pressure feedback. This series of tests consisted mainly of obtaining data to correct the mechanism such that automatic cycling was accomplished. During the first tests, the automatic cycling mechanism would not function. This was found to be due to the low pressure at which the motor piston, Item 13, would actuate. Because of the low pressure level, the secondary area of the automatic cycle valve was not being backpressured; thus, a toggling function was not created. The correction to obtain a toggle action was provided by incorporation of a seat near the outer perimeter of the motor piston such that the piston area was greatly reduced in the retract position. This is shown in Revision "A" of Drawing PA878169. (The parts list for the drawing is in process.) The seat on the motor piston effectively provided the desired results by providing backpressure on the secondary area of the automatic cycle valve. This action allows the valve to open fully prior to control motion of the motor piston, thereby providing the desired inrush of high pressure into the motor piston chamber. Subsequent tests were conducted on the automatic cycle system to establish the proper feedback spring rates and to control the motor gas usage.



During the tests, a common supply pressure of 900 psi was applied to the motor and compressor. The supply to the motor was restricted so that the flow was 1.25 lb per hr. The capacitance volume and feedback forces were then altered one at a time to establish data such that a proper feedback schedule was obtained.

The curves shown on Figure 5 present the functional characteristics of the automatic cycle mechanism. Data Plot No. 1 shows the actual test results. Plot No. 2 presents the theoretical capacitance pressure as a function of the receiver bottle pressure at which the automatic cycle valve opens. The theoretical curve is based on energy balance equations for ideal operation of the current configuration. Plot No. 2 is the actual capacitance pressure results which were used for the feedback function of Data Plot No. 1 in Figure 4.

The above-noted data shows an excess of capacitance pressure near the higher bottle pressures. Although the excess capacitance pressure is the result of the spring rate in the feedback mechanism, the efficiency is still low at high bottle pressures. This effect is also caused by the "piston terminal velocity" characteristic discussed in paragraph 4.4, SUMMARY OF RESULTS.

Figure 6, which follows, shows the time required for the pressure in a 26 cubic inch receiver bottle to increase from 900 psig to 4000 psig. This data was obtained by conducting tests with the motor inflow orifice preset to flow 1.25 pounds per hour with a supply pressure of 900 psig and with the compressor operating on automatic cycling. This data also shows the loss of efficiency which is characteristic at the high receiver bottle pressures.

#### 4.3 Mechanical Functions

During the preceding tests, several mechanical changes were made to correct deficiencies. Some items were deleted when a function was found not to be necessary. These changes occurred in the motor piston area and are depicted on Drawing PA878169, Rev. A. The piston, Item 13, was constructed as an assembly with three major pieces to carry Belleville washers to absorb excess energy at the end of the stroke. During cycle tests, the screw threads yielded, resulting in failure of the assembly. The changes resulted in modifying this piston to a one-piece component with the Belleville washers retained in the forward housing. Subsequent tests have proved this arrangement to be satisfactory. As a result of functional tests, the check and relief valve, Item 30, was found to be unnecessary and troublesome. This valve was replaced by a simple orifice, 0.031 inch in diameter. This change also permitted deletion of side port vents which are no longer used. The major benefit of this change is a very significant reduction of noise.

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Follow-on development efforts on the automatic cycle valve revealed that the secondary area of the valve had to be increased. This increase in area was required to accommodate the interaction encountered between the motor piston and the automatic cycle valve at certain conditions. In this regard, the piston would not stroke fully at the lower pressure conditions. Incorporation of the larger secondary area in the valve permitted automatic cycling of all bottle pressures between 900 psig and 4000 psig. Small dimensional changes also occurred in the valve seat areas; however, none was significant.

During the automatic cycling tests, the compressor check valve developed a leak. Subsequent disassembly of the unit revealed that the silastic seat seal had developed an extruded type buildup around the poppet lip. This buildup was near the high pressure side, thus indicating that gas pressure had been trapped under the seal. In addition to the seal condition, the seat also showed fretting. This indicated that some high pressure absorption was taking place in the material. This situation is being evaluated with the silastic seal supplier.

In line with the foregoing, it was considered that a thinner layered seal, bonded to a metal backup plate, would provide a decided improvement in the valve leakage characteristics. However, due to time limitations in the program, this concept was not fabricated nor tested. It is noted that the storage bottle could be charged several times without a leakage problem, provided that a slow bleed-off of bottle pressure followed the filling. It was determined that a new seat seal was required after two to three hours of operation.

4.4 Summary of Results

The results obtained in the overall analytical and test program indicate that a functional oxygen boost compressor using a flow rate type cycle system is feasible. The program did, however, reveal that problems with efficiency are still to be resolved. Evaluation of the test and analytical data indicated that the transfer of expanding gas energy was not being properly converted into mechanical energy since the motor piston was not reaching the velocity required to provide completion of the compression cycle.

Plot No. 2 of Figure 4 shows that good compression efficiency is possible. Figure 5 shows that the automatic cycle valve pressure is more than sufficient, although the cycle was still short due to slow piston velocity. A phenomenon called "piston terminal velocity" was subsequently analyzed. In this regard, the terminal velocity of a piston supplied with a gas through an orifice is determined by the following equation:

$$v = \frac{KA_{12} C_{12} \sqrt{T}}{A_p}$$





Then for deceleration ( $F = ma$  and  $V^2 = 2ax$ ), the following derivation is obtained:

$$A_{12} = \left( \frac{A_p}{KC_{12}} \right) \sqrt{\frac{2FX}{T M_p}}$$

where

$A_{12}$  = Effective area of the supplying orifice, inches<sup>2</sup>

$A_p$  = Area of the motor piston, inches<sup>2</sup>

$F$  = Accelerating force, pounds

$X$  = Distance of deceleration, inches

$K$  = Constant

$C_{12}$  = Orifice factor, a function of  $\left( \frac{P_1}{P_2} \right)$

$T$  = Absolute temperature of inlet gas, deg R

$M_p$  = Mass of the motor piston, lb-ft/sec<sup>2</sup>

The variables  $X$  and  $F$  may be extracted from the compressor efficiency data and the pressure to be compressed. Figure 7, which follows, depicts a plot of force versus stroke that was interpolated from previous data. This curve represents the condition with a bottle pressure of 400 psig and an automatic cycle valve pressure of 800 psig. It is a plot of the force and stroke as seen by the motor and compressor pistons.

An evaluation of the foregoing indicates that the automatic cycle valve in the supply is too small. In this regard, the valve area should be increased by changing the seat diameter from 0.25 inch to 0.50 inch to provide for sufficient energy management.

A last-minute test has been conducted to verify this analytical theory. The area of the automatic cycle mechanism was increased by increasing the diameter of the valve by 0.05 inch. The efficiency was improved as a result of this change. The change in efficiency was approximately the same as the difference between Data Plot 1 and Data Plot 2 or Figure 4.

It is noted that the demonstrator compressor to be delivered will not be equipped with this modification.



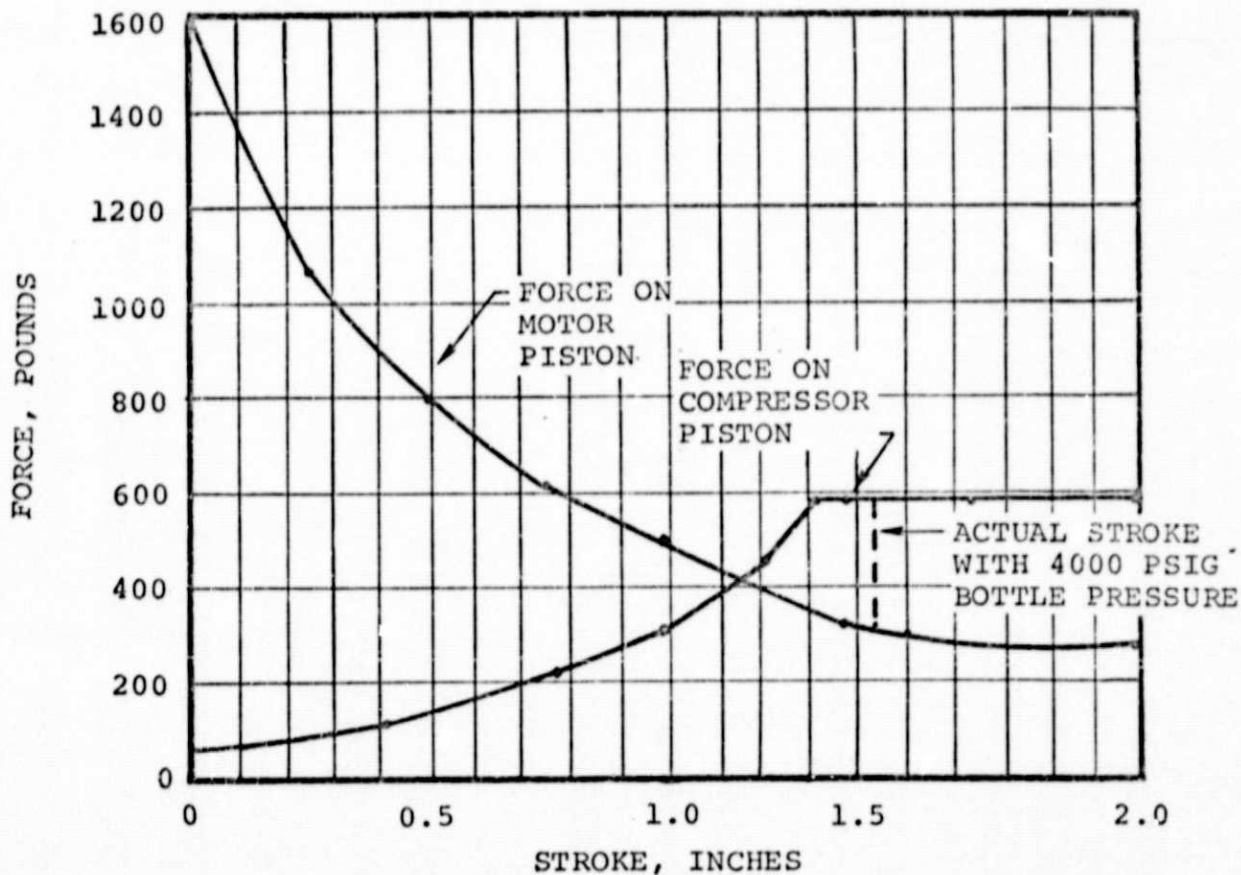


FIGURE 7

FORCE VERSUS STROKE CHARACTERISTICS OF COMPRESSOR



## 5. RECOMMENDED FOLLOW-ON EFFORTS

### 5.1 Recommended Follow-On Analysis

It is suggested that the follow-on effort include an analytical study to evaluate the experimental data which has been produced. Also, the effect of the schematic changes already discussed should be thoroughly evaluated and the design optimized, so that performance limits will be known.

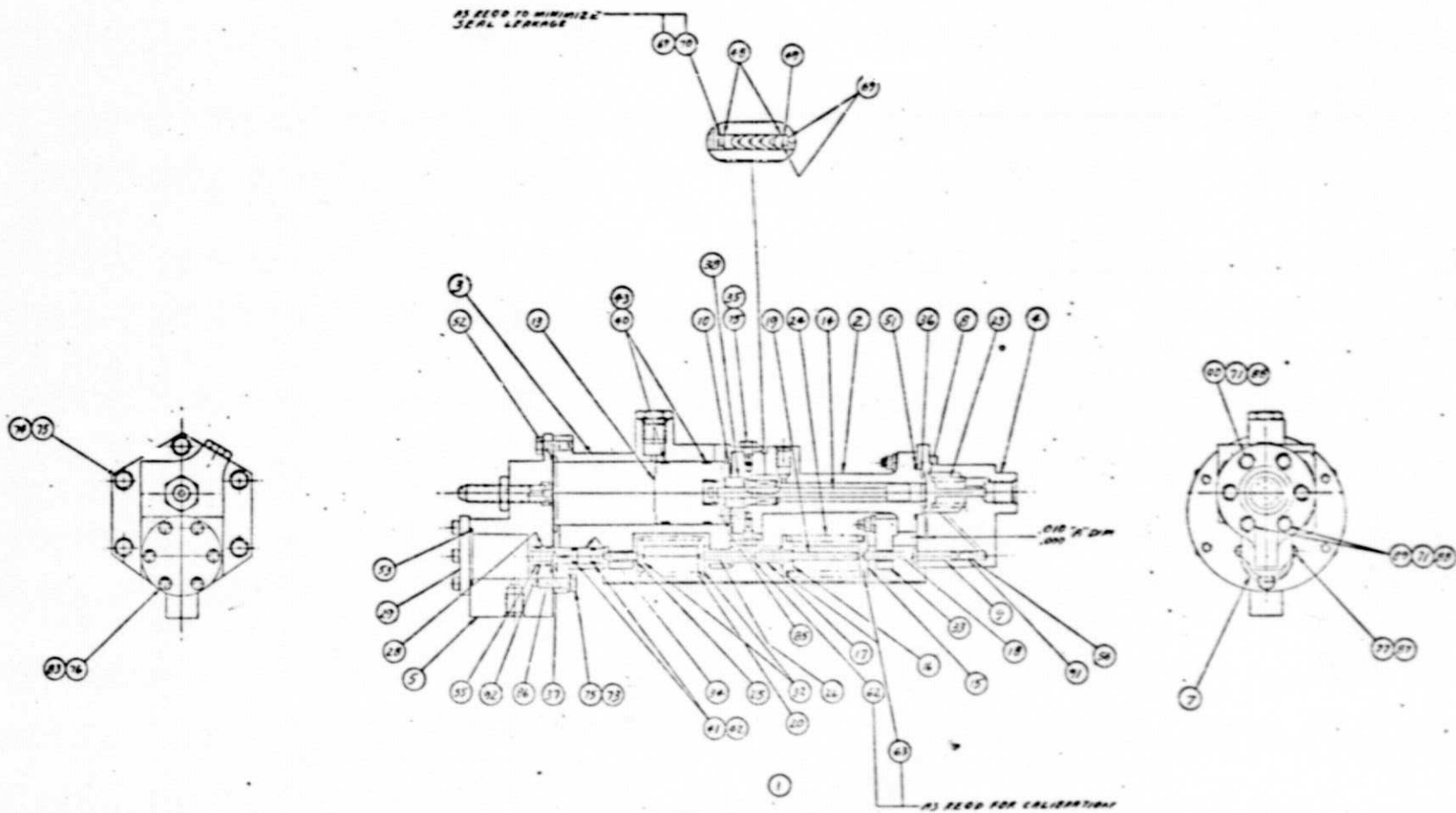
### 5.2 Recommended Tests

AiResearch considers that the analytical and test efforts presented in this report have shown the feasibility of a flow rate cycle system operating with a free piston-type compressor. It is therefore recommended that further development efforts be made. One such effort would consist of further development to alter the area of the automatic cycling valve so that further studies to refine the use of expansion energy could be accomplished.

It is also recommended that several high pressure valve seat configurations be fabricated and tested to increase service life and efficiency.

Following the foregoing, it is also recommended that the empirical data of the test program be correlated with the computer program data. This would provide basic data for designing a flight-weight unit.





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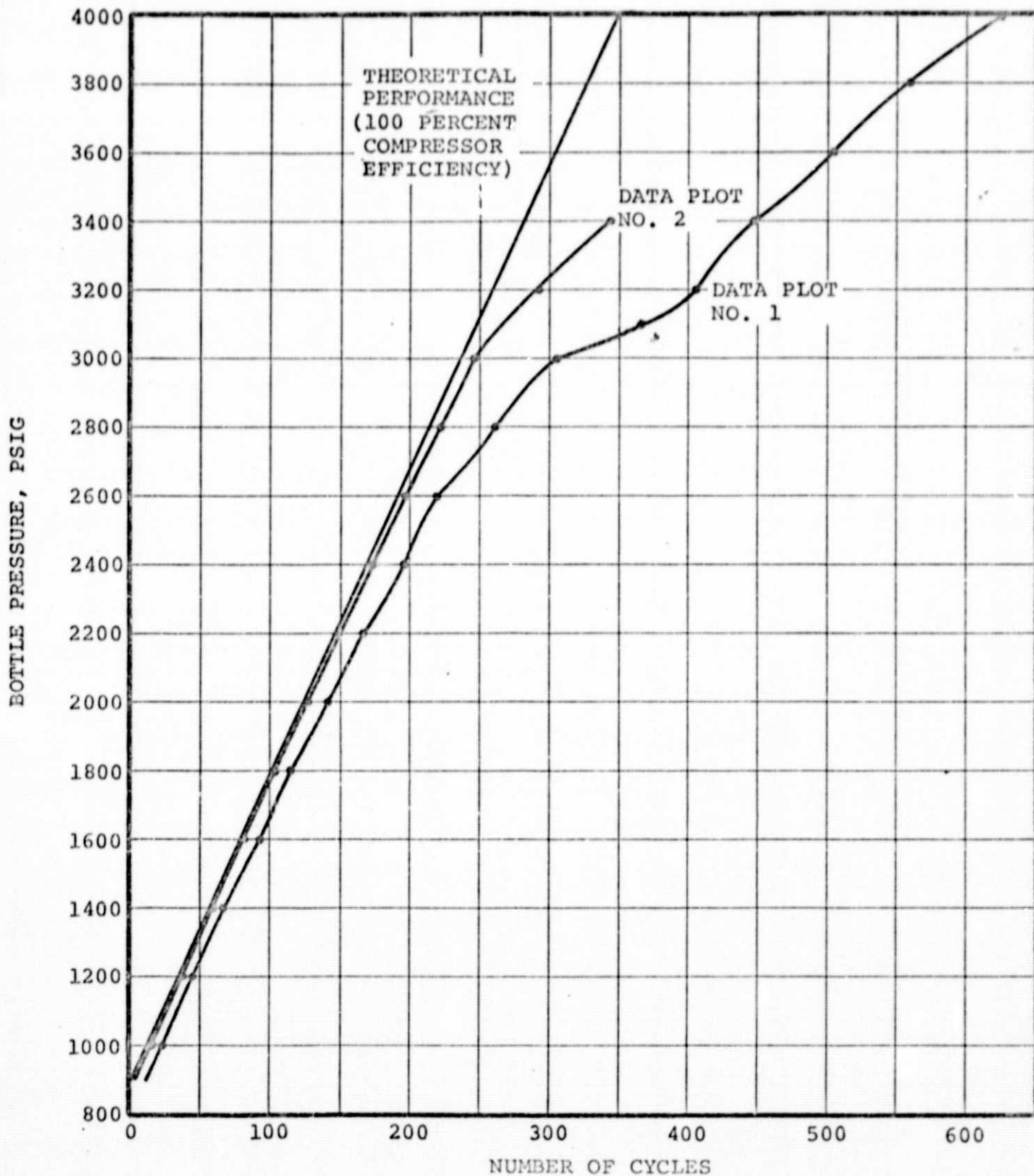


FIGURE 4

COMPRESSOR PRESSURE VERSUS CYCLES



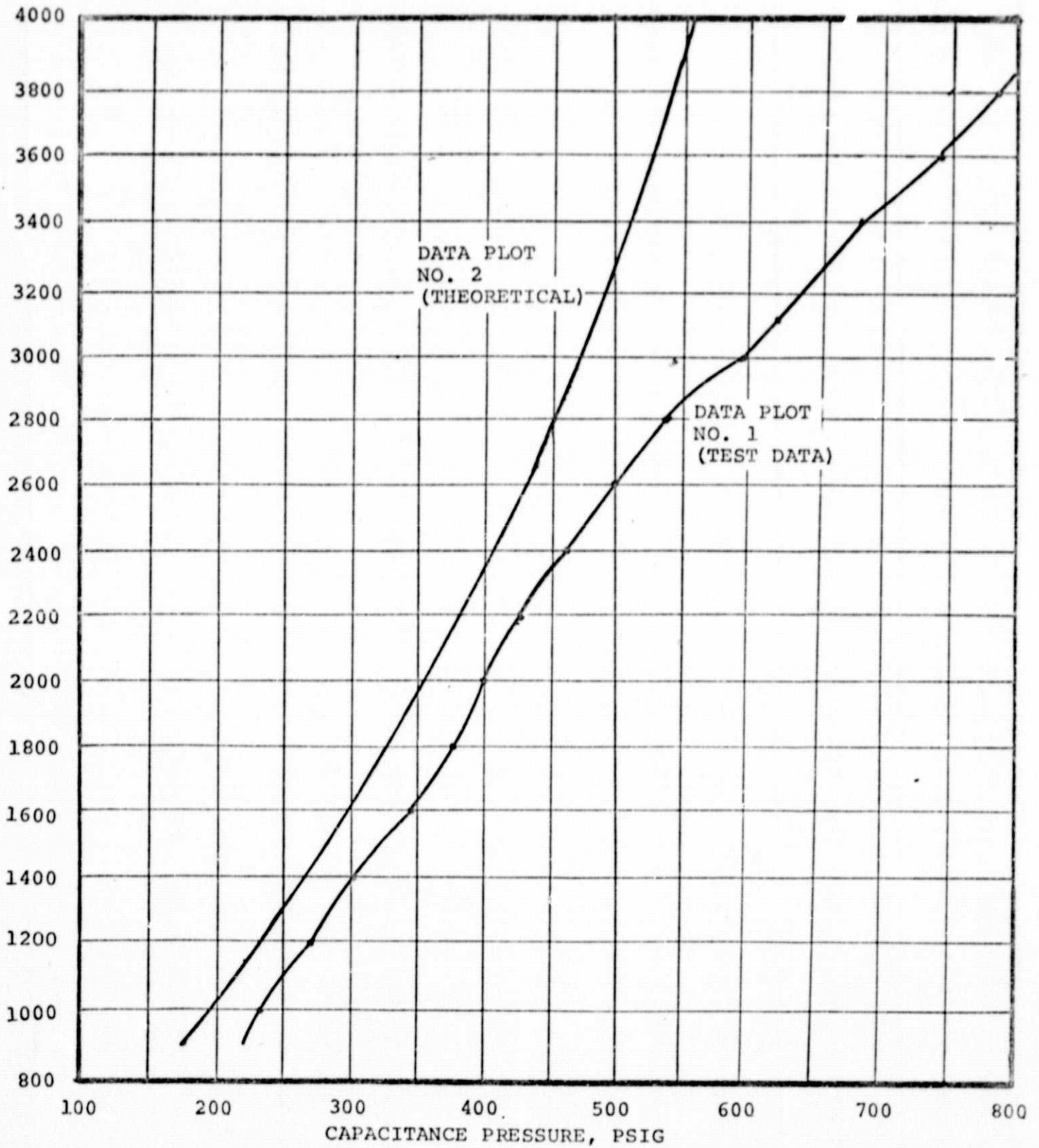


FIGURE 5  
CHARACTERISTICS OF AUTOMATIC CYCLE MECHANISM

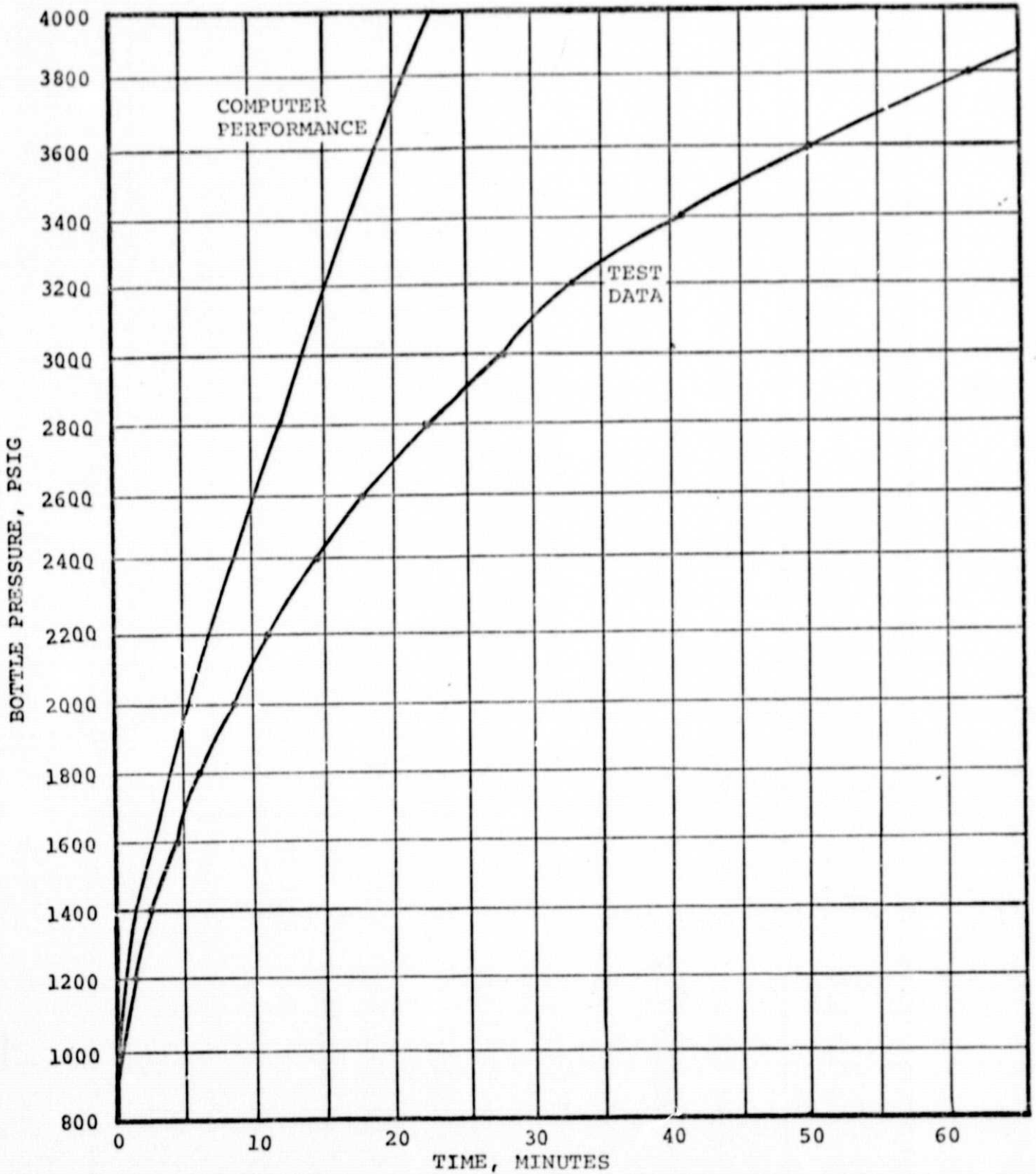


FIGURE 6

COMPRESSOR PRESSURE VERSUS TIME