1976

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INTRODUCTION

Preliminary tests of prototype compressors for an underwater breathing system revealed excessive power usage that appeared to be connected to high pressure build-ups in the compressor cylinders.

The compressor design was therefore reviewed, the probable cause of the difficulty was identified using analytical approaches and a design modification was suggested. A 35% reduction in power requirement was the result.

This paper presents the way this was accomplished in terms of a historical case study. By 'historical' is meant that the analysis of the valves, which at first were suspected (wrongly, as it turned out) of causing the difficulty, is also presented.

It is felt that there is merit in this way of presentation since in some other situations the valves might prove to be the source of misfunction, and not the manifold design.

DESCRIPTION OF COMPRESSORS

A first prototype compressor was designed and fabricated as a nominal one horsepower compressor. Two of these compressors will be used in an underwater breathing system. They are both attached to an underwater diving base, from here on referred to as a diving platform. The supply compressor "pushes" breathing gas out to a diver from the ambient atmosphere of the diving platform against a nominal pressure differential of about 80 psi. The return compressor "pulls" gas from the diver and returns it back to the diving platform against a pressure differential of approximately 35 psi. The diving platform is under an ambient pressure equal to the water pressure at the depth the platform is located at. For this investigation, the "design" depth of 1,000 feet of sea water was chosen.

The breathing gas is Helium, with Oxygen and Nitrogen added such that their partial pressures are the same as in the earth atmosphere at sea level.

The compressors are of the reciprocating type and are hermetically sealed by stainless steel bellows. The valves that were selected after extensive preliminary testing by the designer were reed valves. They consist of multiple finger type reeds riveted to a seat plate. Each finger seals off a circular port. Inlet and outlet valving is achieved by installing reeds on opposite sides of a common seat plate. Teflon gaskets seal the inlet and outlet reed sections from each other and from the ambient atmosphere. Valves open into, respectively from, discharge and suction plenum chambers. From these chambers, pipes convey the breathing gas to other parts of the system.

PROBLEMS ENCOUNTERED WITH FIRST PROTOTYPE.

The prototype compressor was tested in a specially designed pressure chamber. It was found that the power consumption was almost twice the value expected. As a matter of fact, a larger drive motor had to be used to complete the tests.

The flow rate was satisfactory.

A pressure transducer was installed for cylinder pressure measurements and showed a peak-to-peak pressure difference of about 185 psi instead of the discharge minus suction pressure difference of about 80 psi. A typical measurement is sketched in Figure 1 and compared to an ideal indicator diagram. The area enclosed by the pressure volume trace is equal to the indicated energy per cycle and is almost twice that of the ideal trace.

It was therefore concluded that if the reason of the overpressure could be found then this would also serve as the reason for the excessive power requirement.

BREATHING GAS PROPERTIES

To start analytical detective work, the properties of the breathing gas were needed.

In the following, the subscripts O, N and H denote Oxygen, Nitrogen and Helium. By the law of mixtures [1],

$$p_{total} = p_O + p_N + p_H$$

Since p_O and p_N are known, as well as p_{total}, which is given by the depth,

$$p_H = p_{total} - p_O - p_N$$

The molecular weight is

$$M_{total} = r_O M_O + r_N M_N + r_H M_H$$

where

$$r_O = \frac{p_O}{p_{total}}, \quad r_N = \frac{p_N}{p_{total}}, \quad r_H = \frac{p_H}{p_{total}}$$

The specific weight is

$$\gamma_{total} = r_O\gamma_O + r_N\gamma_N + r_H\gamma_H$$

Also of interest is that

$$\gamma(\text{at } p_{total}) \cong \gamma(\text{at } 14.7 \text{ psi}) \frac{p_{total}}{14.7}$$

if we assume that the temperature is the same. The gas constant of the gas mixture is given by

$$R_{total} = g_O R_O + g_N R_N + g_H R_H$$

Where

$$g_O = r_O \frac{M_O}{M_{total}}, \quad g_N = r_N \frac{M_N}{M_{total}},$$

$$g_H = r_H \frac{M_H}{M_{total}}$$

The specific heats are

$$c_{p_{total}} = g_O c_{pO} + g_N c_{pN} + g_H c_{pH}$$

$$c_{v_{total}} = g_O c_{vO} + g_N c_{vN} + g_H c_{vH}$$

The adiabatic number is therefore

$$k_{total} = \frac{c_{p_{total}}}{c_{v_{total}}}$$

The properties of the individual gases at atmospheric pressure are summarized [2]

Type of Gas	M lb/P Mole	γ lb/Ft^3 at 14.7 psi	R Ft-lb/ lb- °R	C_p BTU/lb-°R	C_v BTU/lb-°R	K
Oxygen, O_2	32.0	0.083	48.3	0.219	0.157	1.40
Nitrogen, N_2	28.0	0.073	55.2	0.248	0.178	1.40
Helium He	4.0	0.010	386.0	1.241	0.745	1.67

Thus, given the following values at a suction pressure of 459 psi (this corresponds to a diving platform location at 1000 feet):

$$\gamma_{total} \quad (\text{at } p_{total}) = 0.403 \ \frac{lb_m}{ft^3}$$

$$R_{total} = 311 \quad \frac{ft\ lb_f}{lb_m\ {}^\circ R}$$

$$k_{total} = 1.65$$

At discharge pressure of 541 psi, we get:

$$\gamma_{total} \quad (\text{at } p_{total}) = 0.461 \ \frac{lb_m}{ft^3}$$

$$R_{total} = 320 \quad \frac{Ft\ lb_f}{lb_m\ {}^\circ R}$$

$$k_{total} = 1.65$$

That this is different enough from the values we would get for pure Helium can bee seen from Figures 2 to 5. Shown in Figure 6 is the speed of sound. At the suction and discharge condition it is:

$$c_s = 35410 \quad \frac{in}{sec}$$

$$c_d = 35730 \quad \frac{in}{sec}$$

The speed of sound was calculated from:

$$c = \sqrt{\frac{k_{total}\ p_{total}\ 386}{\gamma_{total}\ (\text{at } p_{total})}}$$

VALVE ANALYSIS

First, an analysis is presented whose aim is to establish average Mach numbers of flow through the valves. These Mach numbers are a measure of pressure drop, valves with M $>$ 0.2 being possibly unacceptable and valves with M$<$0.2 being possibly acceptable.

Tne volumetric efficiency is [1]

$$\eta = 1 - \frac{V_c}{V} \left[\left(\frac{p_d}{p_s}\right)^{\frac{1}{n}} - 1 \right]$$

For the case under investigation

V_c = clearance volume = 0.84 in^3

V = swept volume = 5.3 in^3

p_d = nominal discharge pressure = 541 $\frac{lb_f}{in^2}$

p_s = nominal suction pressure = 459 $\frac{lb_f}{in^2}$

n= polytropic coefficient = k = 1.65

This gave η = 0.98 which had to be considered good. Even while the clearance volume is large, the pressure ratio is nearly unity, which negates the large clearance volume effect.

The volume taken in at suction condition is:

$$V_S = \eta V = 5.21 \ in^3$$

Thus, the mass delivered per cycle is:

$$m = \gamma_S V_S = \gamma_d V_d$$

where

γ_s = specific weight at suction= 0.403 $\frac{lb_m}{ft^3}$

γ_d=specific weight at discharge= 0.461 $\frac{lb_m}{ft^3}$

Thus, m = 0.00122 lb_m/cycle. The volume that is discharged is:

$$V_d = \frac{\gamma_s}{\gamma_d} \ V_s = 4.55 \ in^3$$

The time the valves are open is estimated to be for the suction valve

$$t_s \cong \frac{30}{n}$$

where

n = rotational speed = 1725 RPM

Thus, t_s = 0.017 sec. For the discharge valve,

$$t_d \doteq \frac{27}{n} = 0.016 \ sec$$

Flow areas are a function of valve lift. We may, however, argue that the valve reeds will very quickly reach their limit displacement. Thus, we take the area that shows the most flow restriction under full lift condition. This gave for both suction and discharge valves:

$$A_s = A_d = 0.39 \text{ in}^2$$

The mean velocities are therefore:

$$v_s = \frac{V_c}{A_s t_s} = 766 \ \frac{\text{in}}{\text{sec}}$$

$$v_d = \frac{V_d}{A_d t_d} = 741 \ \frac{\text{in}}{\text{sec}}$$

The average Mach numbers are, therefore:

$$M_s = \frac{v_s}{c_s} = 0.02$$

$$M_d = \frac{v_d}{c_d} = 0.02$$

These Mach numbers are quite small and it had to be concluded, therefore, that the designed valve flow areas are sufficiently large and cannot be held responsible for the large measured pressure build-ups.

This conclusion was later verified by actually calculating approximate pressure build-ups due to valve action alone by use of an existing computer program. This program was constructed according to the principles laid down in references [3,4,5]. It incorporated the dynamics of the valve reeds and utilized the actual instantaneous flow areas.

Inputs to the program were the geometries of the compressor, the gas properties, instantaneous flow and force area [3,4] and the fundamental natural frequencies and modes of the valve reeds. The natural frequency of a reed for both suction and discharge was determined to be 307 Hz. This agreed well with the frequency of the wave ripples that showed up on the cylinder pressure data taken. Flow and force areas were almost identical for suction and discharge.

Results verified the classical analysis. The valves were not responsible for the high pressure build-ups. A sketch of a typical pressure-volume diagram obtained by the program is given in Figure 7. The pressure peaks at the beginning of discharge and suction are due to the effects of the inertia of the valve reeds. Once the reeds have moved out of the way, the overpressure appears to be of very reasonable magnitude.

The mass delivered per cycle calculated by the program was $m = 0.00121 \ lb_m/\text{cycle}$, only slightly lower than the ideal value of $m = 0.00122 \ lb_m/\text{cycle}$ which was calculated before with the classical approach.

PLENUM PRESSURE CHANGES

Suspicion centered therefore on the plenum design. For instance, it was felt that the measured cylinder pressure increase during discharge might be due to an immediate increase in discharge plenum pressure due to the inertial properties of the gas. This effect was encountered in a similar fashion in refrigeration compressors [6,7,8,9,10].

Unfortunately, computer programs that were developed in this area were all tailored to certain compressor types and a modification to the needs of this study would have exceeded the time and budget limitations . Rather than going to a measurement program, it was decided to use a simplified formula for pressure increases in the plenums by making the assumption that mass flow into the plenum is a known quantity not influenced by the pressure increase itself.

Development of the formula is given elsewhere in these proceedings. According to it, the plenum pressures are given by

$$p = \begin{cases} = \bar{A}\, e^{-\bar{B}t}(e^{\bar{B}t}-1) & 0 \le t \le T \\ = \bar{A}\, e^{-\bar{B}t}(e^{\bar{B}T}-1) & T \le t \end{cases}$$

where

$$\bar{A} = \frac{cm}{gTA} \quad , \quad \bar{B} = \frac{cA}{V}$$

and where

c = speed of sound [in/sec]
m = mass per cycle [lb_m/cycle]
T = time valve is open [sec]
A = pipe crossectional area [in^2]
V = plenum volume

The following values were taken for the discharge system

c =	35700	in/sec
A =	0.31	in^2
V =	0.81	in^3
T =	0.0157	sec
m =	0.00122	lb_m/cycle
g =	386	in/sec^2

The result is an almost immediate over pressure of 23.5 psi, lasting during the entire opening of the valve, which has to be matched by the cylinder pressure so that the valve will open at all. For the suction valve the values were

$$
\begin{aligned}
c &= 35410 && \text{in/sec} \\
A &= 0.31 && \text{in}^2 \\
V &= 0.81 && \text{in}^3 \\
T &= 0.0174 && \text{sec} \\
m &= 0.00122 && \text{lb}_m\text{/cycle} \\
g &= 386 && \text{in/sec}^2
\end{aligned}
$$

The result is an almost immediate suction plenum pressure change of -21 psi. The cylinder pressure during suction will have to be lower than that in order for the valves to open. This is quite conclusive, especially if we remember that resonance conditions in the pipe may change the plenum pressure even further.

Therefore, the conclusion was clearly that the plenum volumes had to be enlarged. Plenum pressure curves for plenum sizes of 0.81, 10, 100 in^3 are shown in Figures 8 and 9, to give the reader an appreciation of how much larger the plenums actually had to be in this case.

The formula also explained the ratio of over pressure between 600 feet and 1000 feet diving depth, which was measured to be, for 80 psi compression pressure,

$$r = 0.64$$

It is, for small plenum sizes, theoretically given by

$$r = \frac{(cm)_{600\ \text{feet}}}{(cm)_{1000\ \text{feet}}}$$

The mass per cycle delivered at 600 feet is roughly

$$m_{600} = m_{1000} \left(\frac{\gamma_{600}}{\gamma_{1000}}\right)$$

Thus, the theoretical value was found to be

$$r = \frac{(c\gamma)_{600}}{(c\gamma)_{1000}} = 0.65$$

This agreed well with the experimental value.

SIZE OF REQUIRED PLENUM VOLUMES

If the formula used in the previous chapter is evaluated at valve closing time t = T, we get as maximum over pressure (or under pressure)

$$p_{max} = \bar{A}\ (1 - e^{-\bar{B}T})$$

The required volume for a certain allowable value of p_{max} is therefore

$$V = \frac{\bar{c}}{\ln\left[\frac{1}{1 - \frac{p_{max}}{\bar{A}}}\right]}$$

where

$$\bar{c} = cAT$$

$$\bar{A} = \frac{cm}{gTA}$$

For instance, for an allowable over pressure of P_{max} = 5 psi in the discharge plenum, we need for the case under investigation a volume V = 734 in^3.

Note that only as $V \to \infty$ will $p_{max} \to 0$.

Plots of required volumes as function of p_{max} are given in Figure 10.

PRELIMINARY RESULTS FOR COMPRESSOR WITH LARGER PLENUMS

The plenum chambers of the compressor were redesigned to have larger volumes. For chambers of 40 in^3 size, the power consumption was reduced by about 35%. Measurements [11] on a multi-cylinder unit showing total overpressure (defined as the difference between total pressure variation and ideally expected pressure variation) and the total power consumption (this includes the drive motor, friction and other losses) as function of plenum volume are shown in Figure 11. The figure shows that the original plenum was clearly too small. These measurements were made after the theoretical reasoning led to a redesign of the plenums. Thus, the value of sound theoretical approaches to compressor design in the drawing board stage was demonstrated.

SUMMARY

A case study was presented that traced the excessive power consumption of a diving compressor to gas inertia effects in the discharge and suction plenums. Since the compressor was by all reasonable standards well designed, it was felt that presentation of such a case study was worthwhile. Similar effects may be (and have been) encountered in the design of other compressors. This case study also presents the first practical application of an approximate equation for gas inertia induced pressure built-up in plenums. Development of this formula is presented elsewhere in these proceedings.

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ACKNOWLEDGEMENTS

The authors would like to acknowledge the help and guidance received from Mr. R.C. Bentz, Project Engineer, Naval Coastal Systems Laboratory, and Mr. P.L. Fontecchio, Senior Project Engineer, Metal Bellows Company.

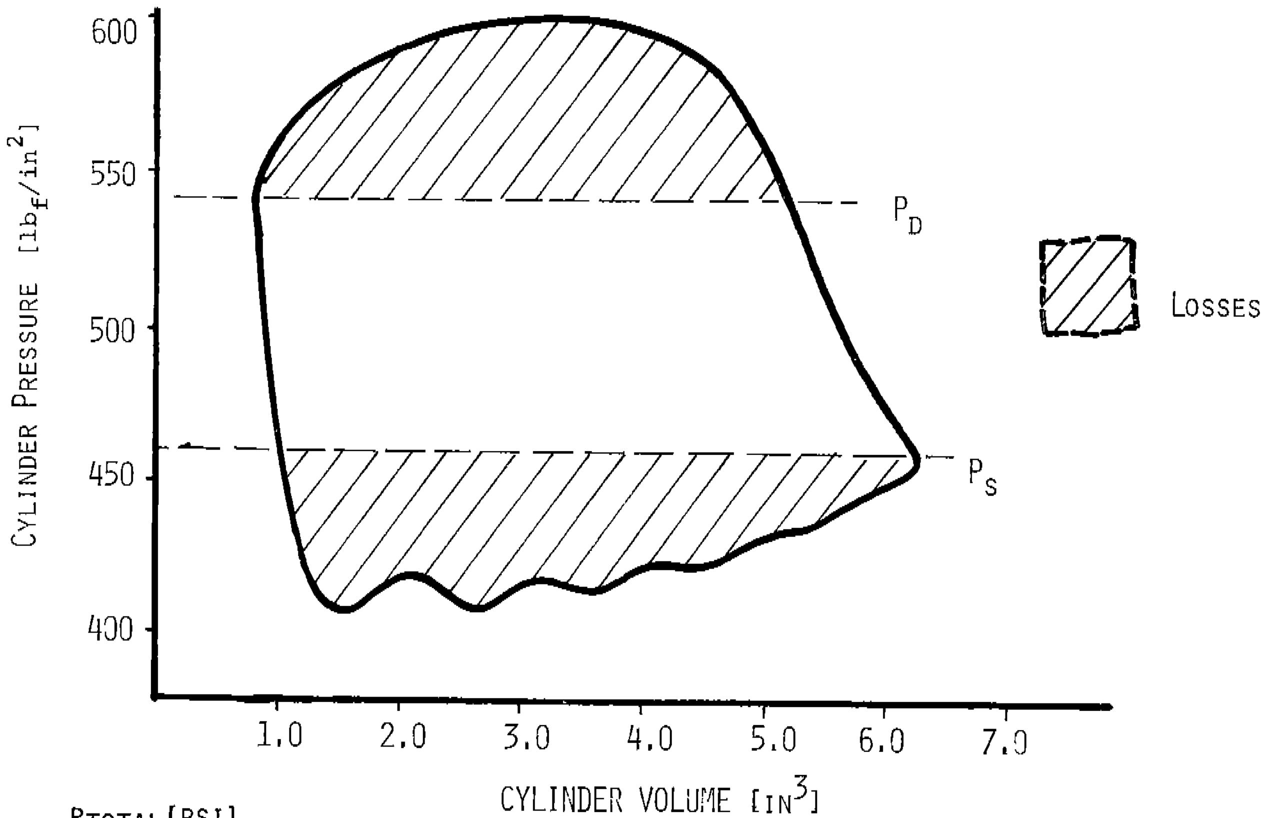

Figure 1. Typical Measured Indicator Diagram

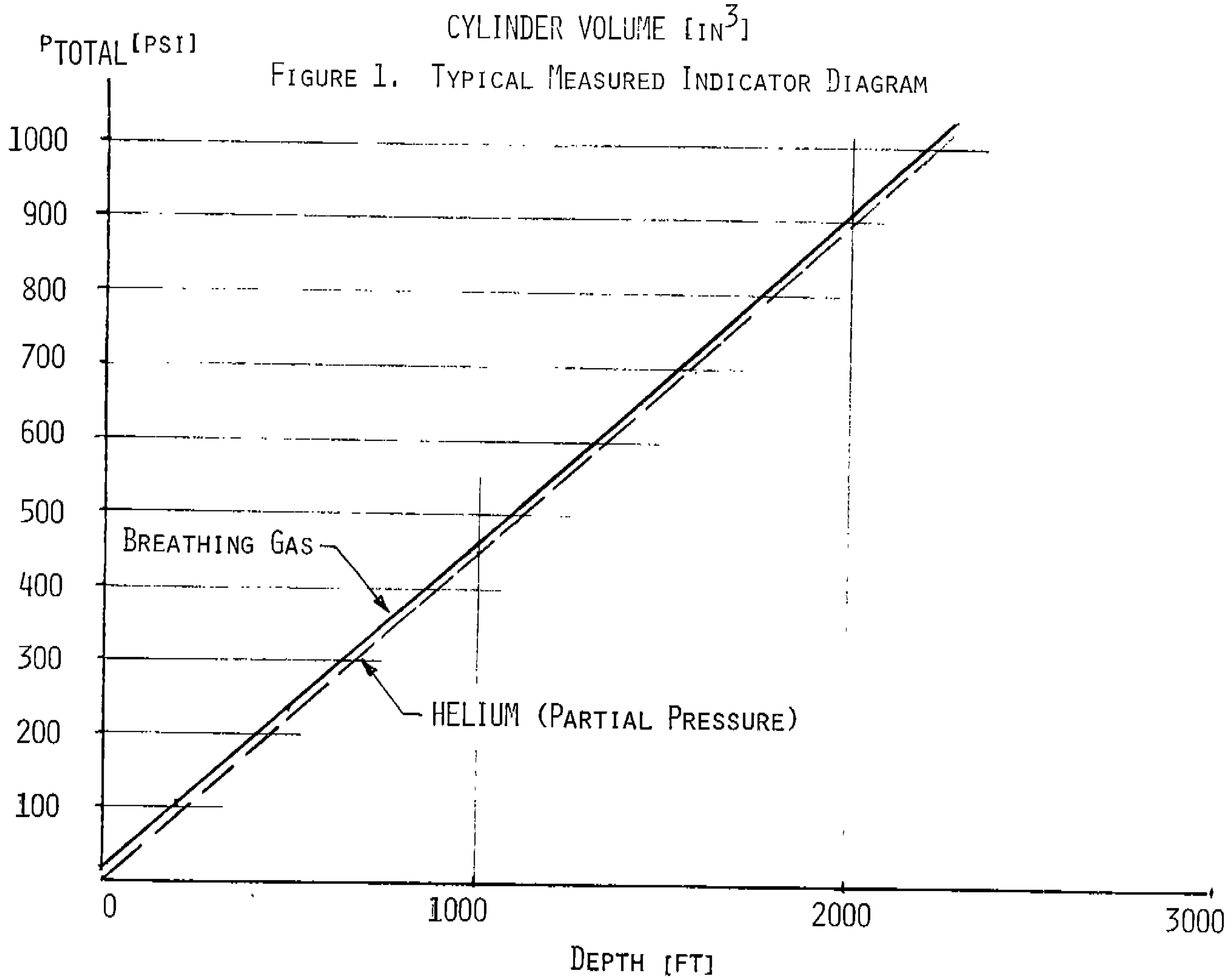

Figure 2. Pressure of Breathing Gas

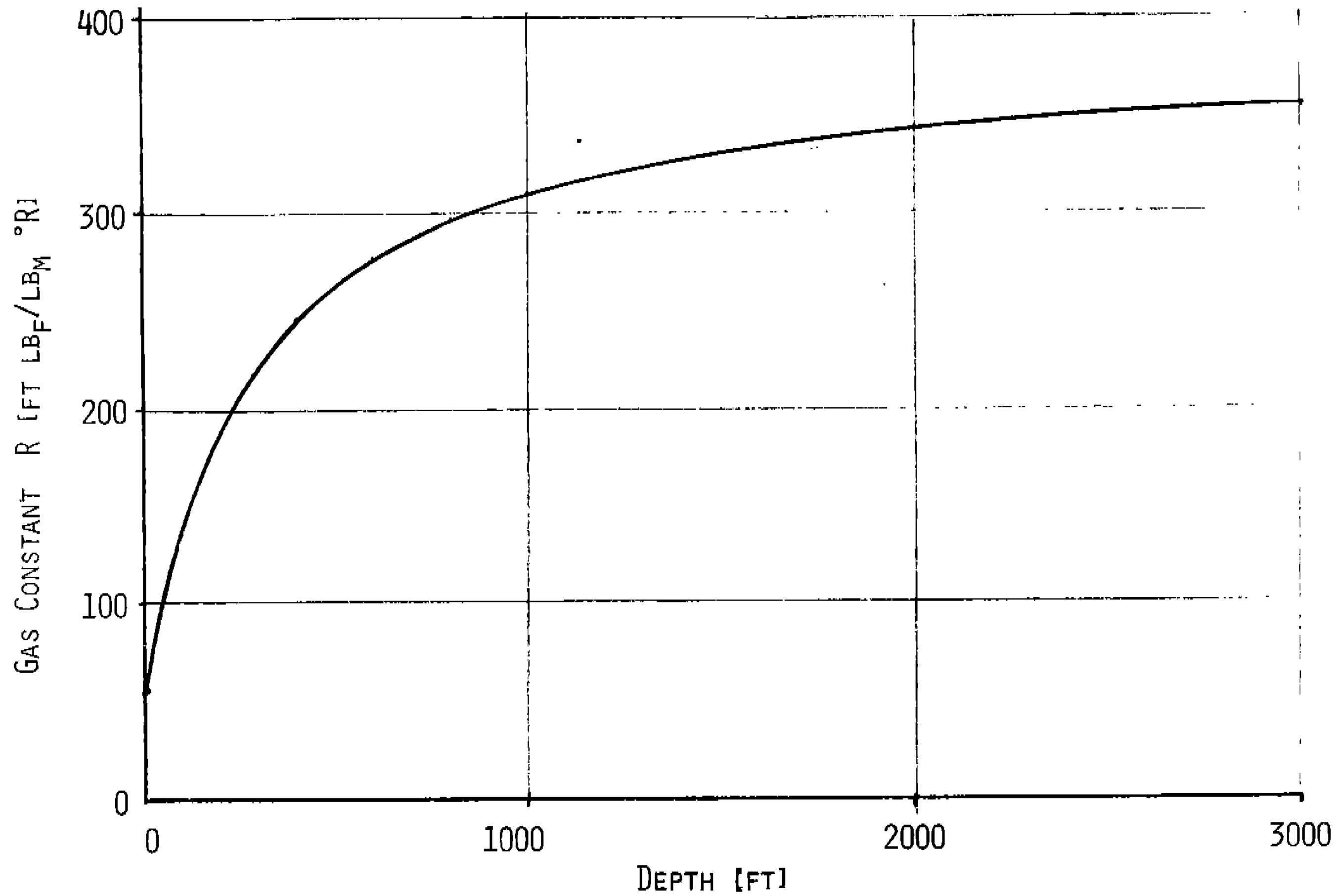

Figure 3. Gas Constant

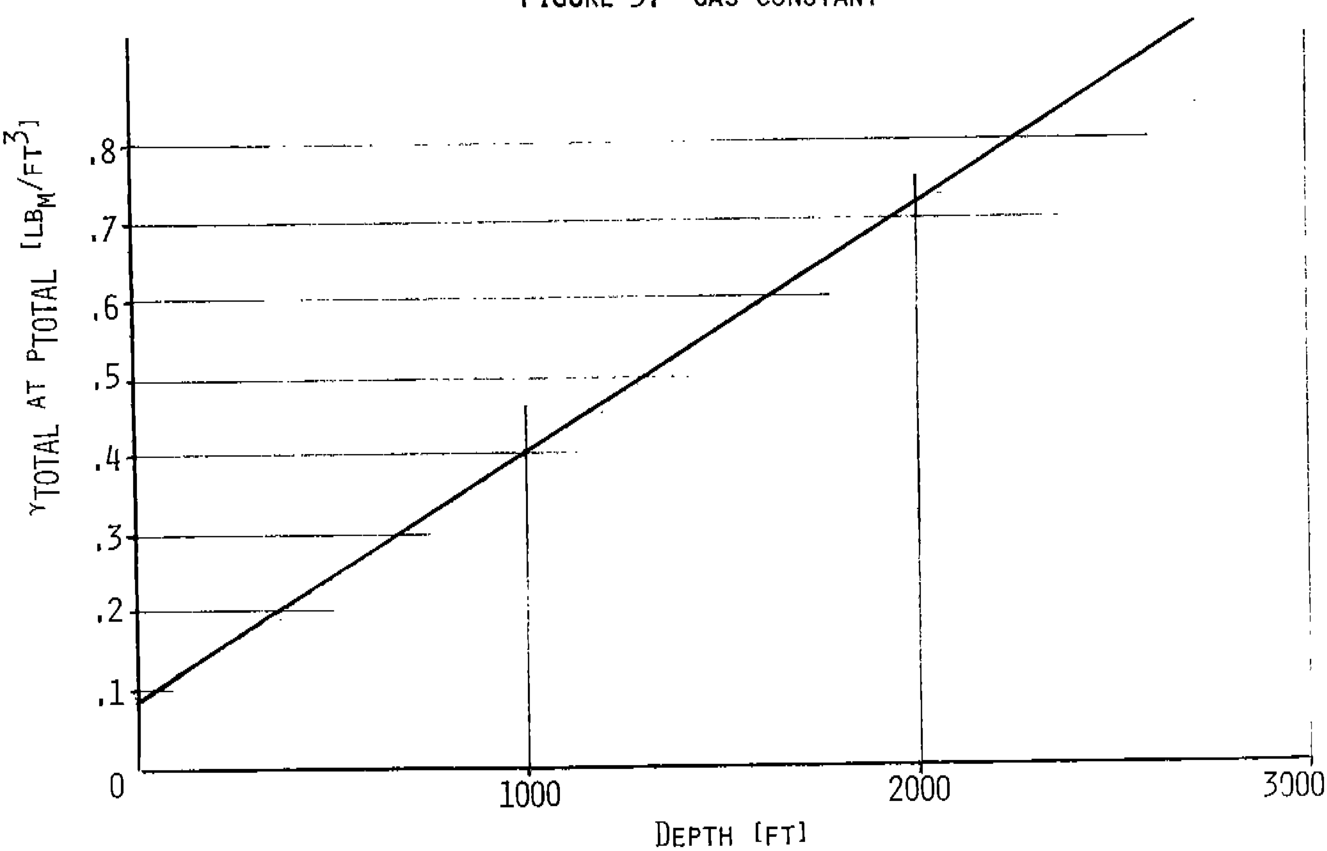

Figure 4. Specific Mass

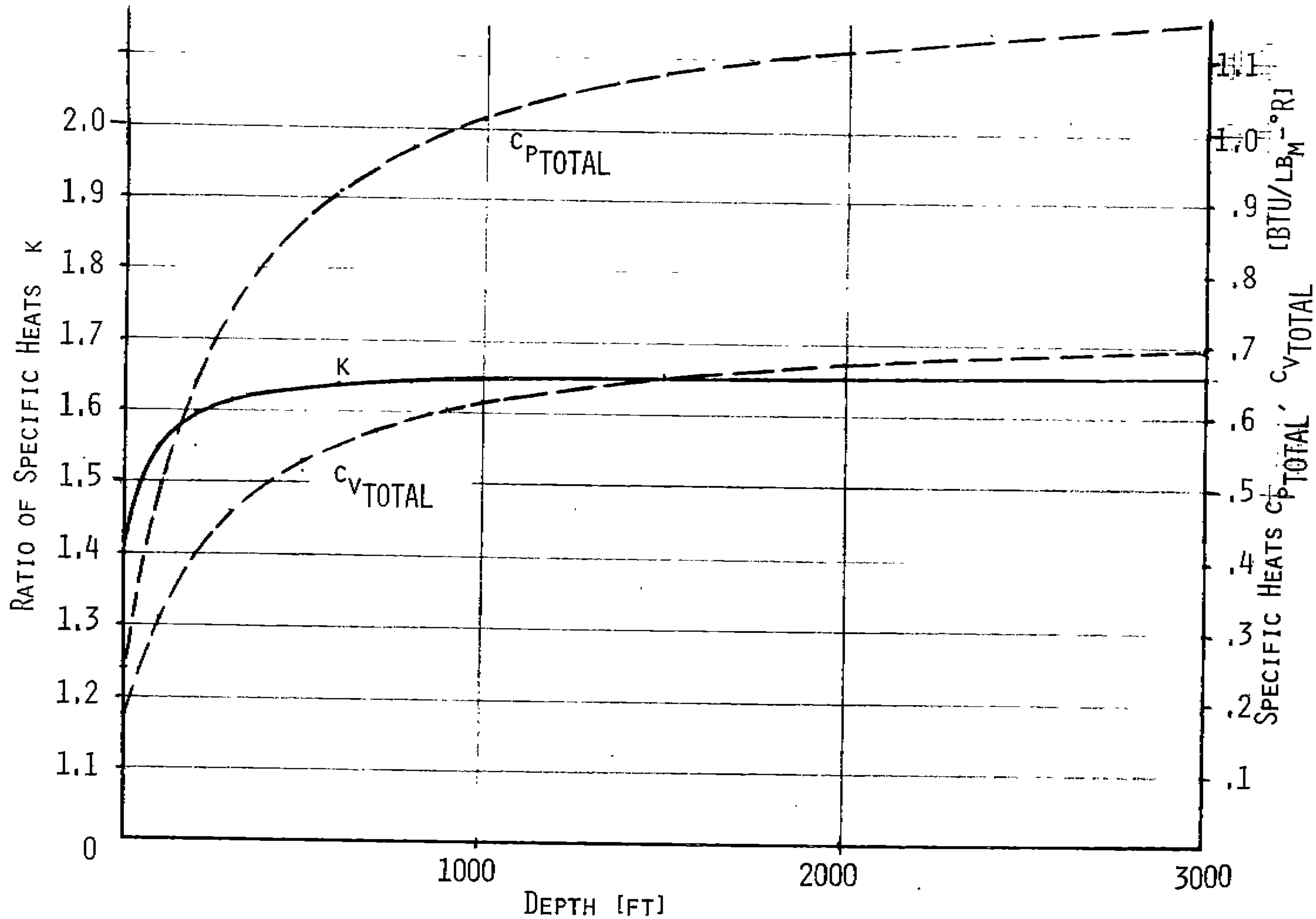

Figure 5. Specific Heats

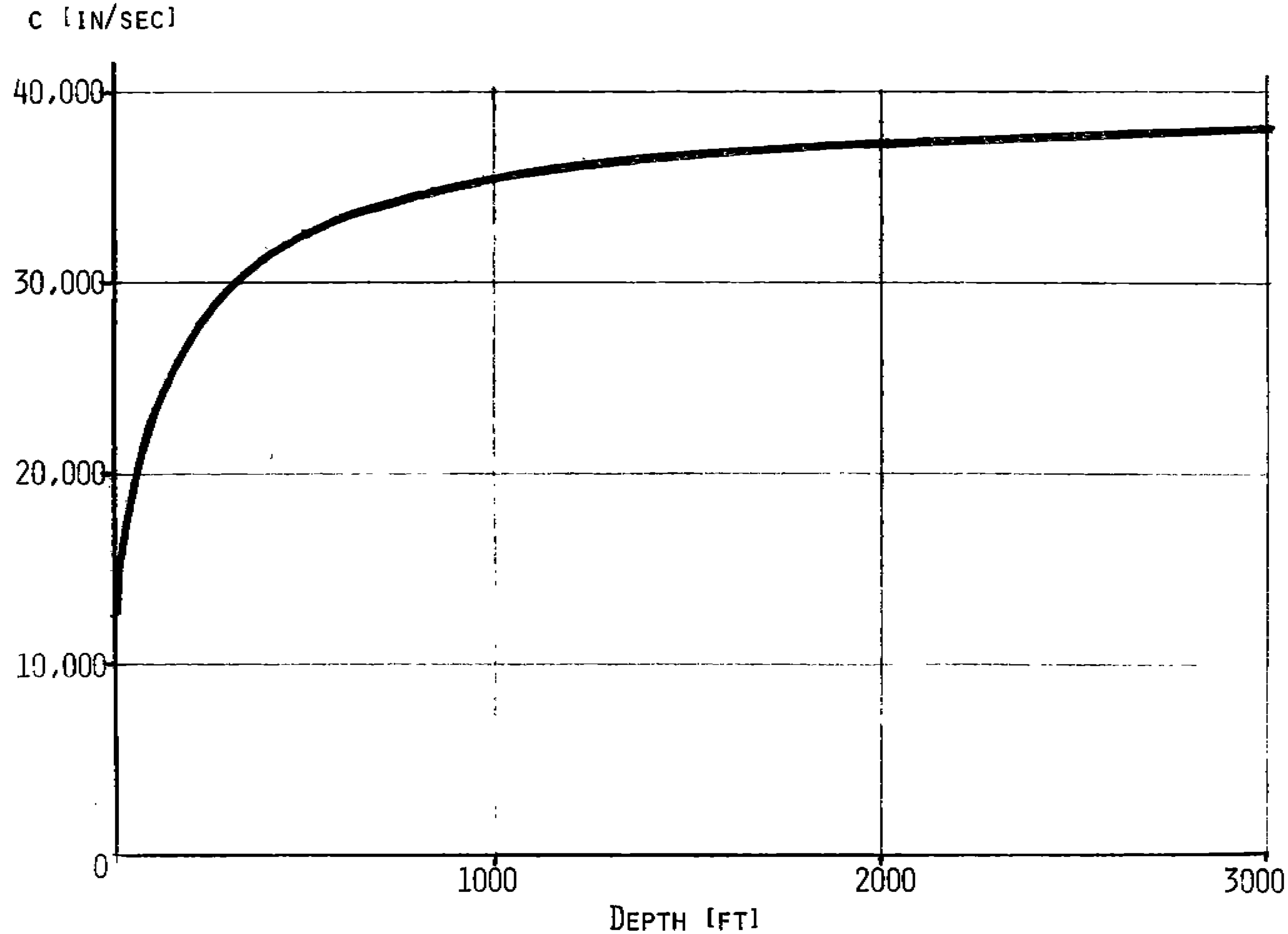

FIGURE 6. SPEED OF SOUND

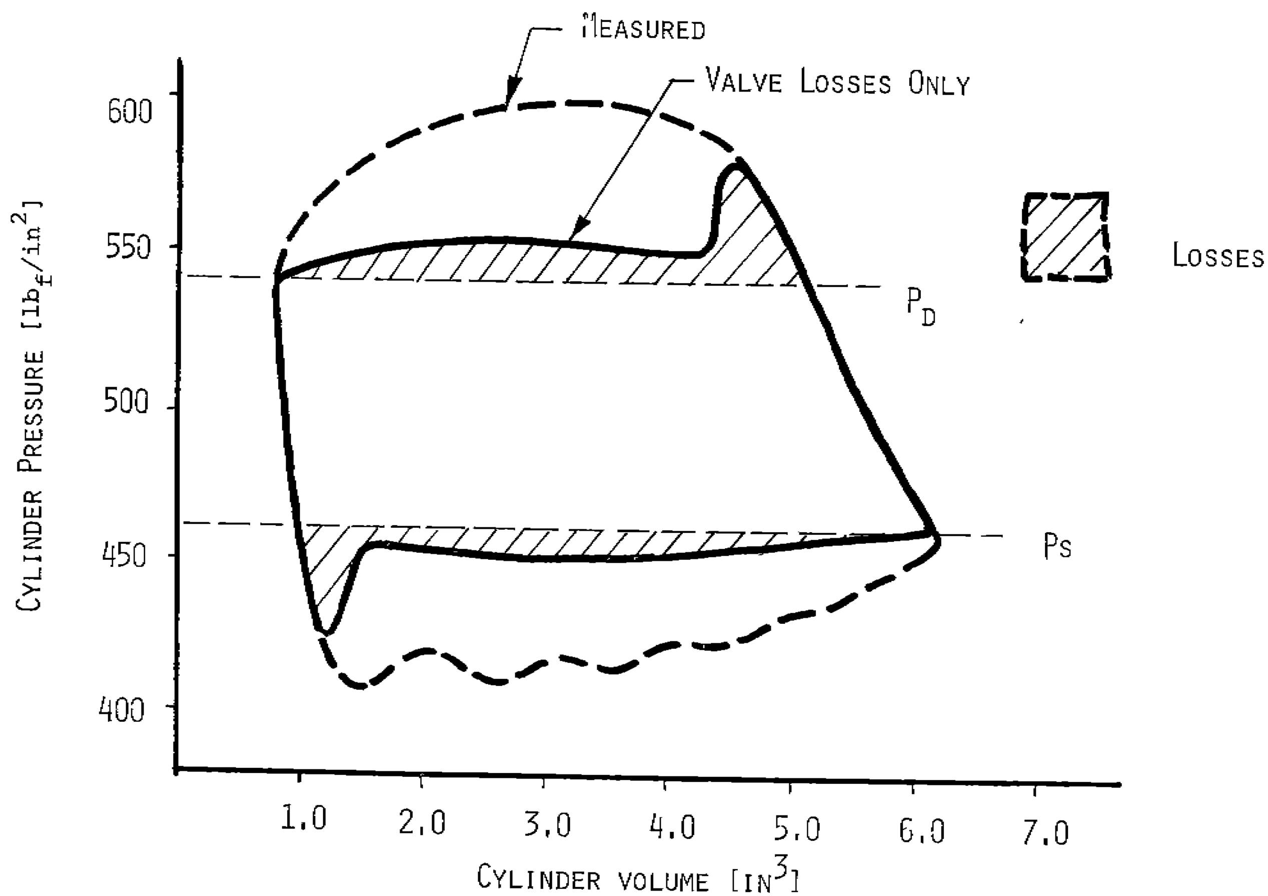

FIGURE 7. EFFECT OF VALVES ON INDICATOR DIAGRAM

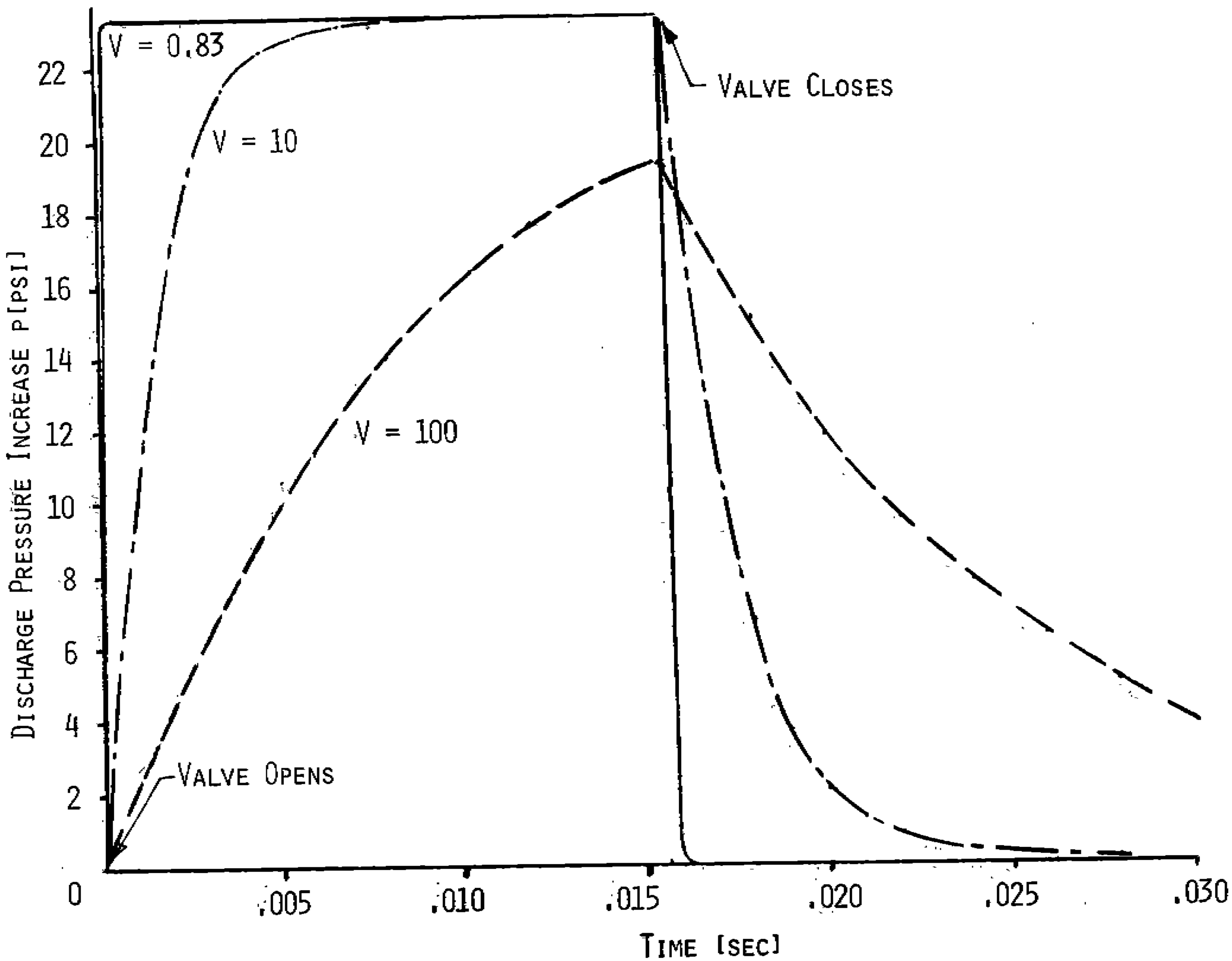

FIGURE 8. DISCHARGE PRESSURE INCREASE

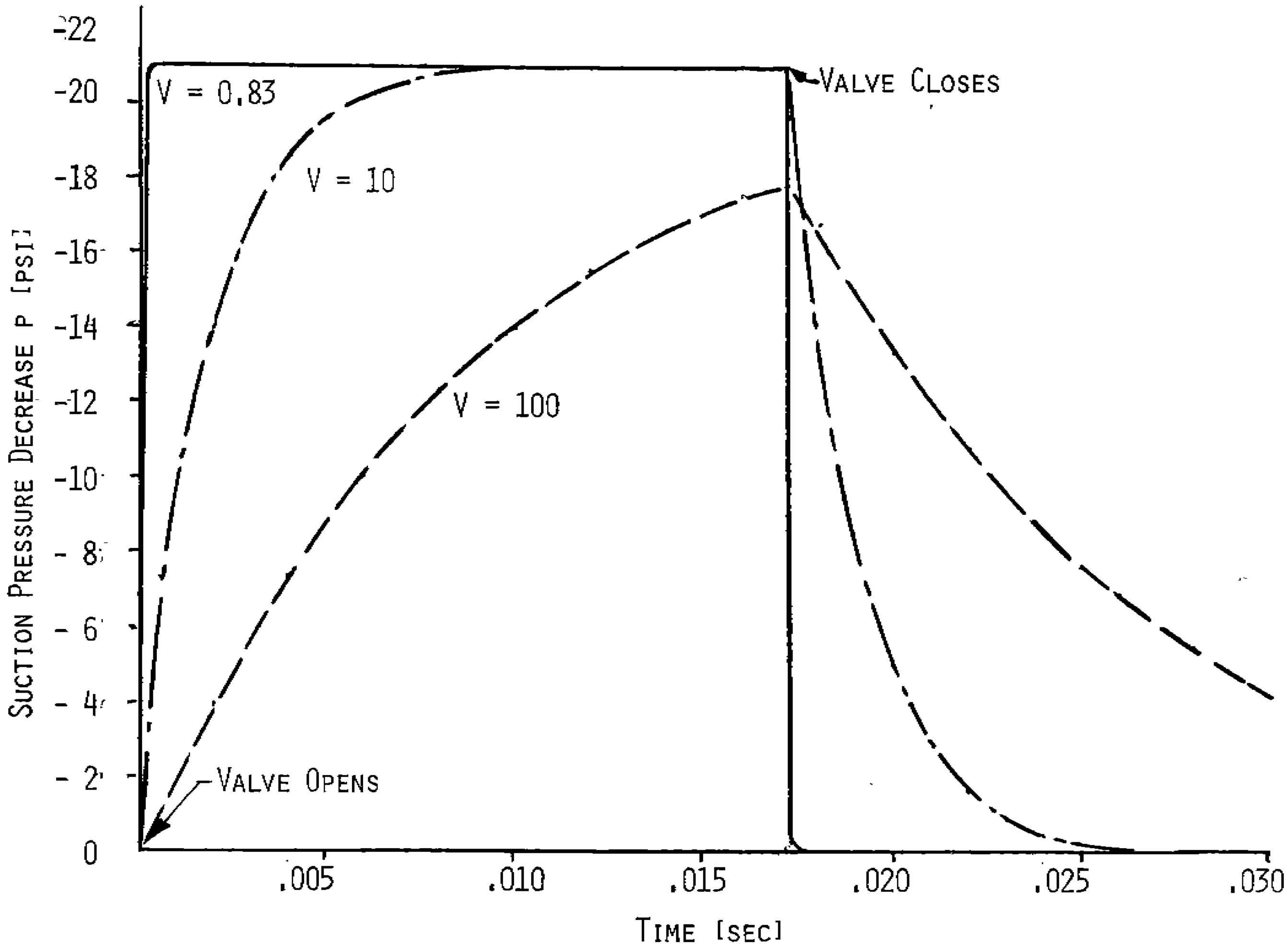

Figure 9. Suction Pressure Decrease

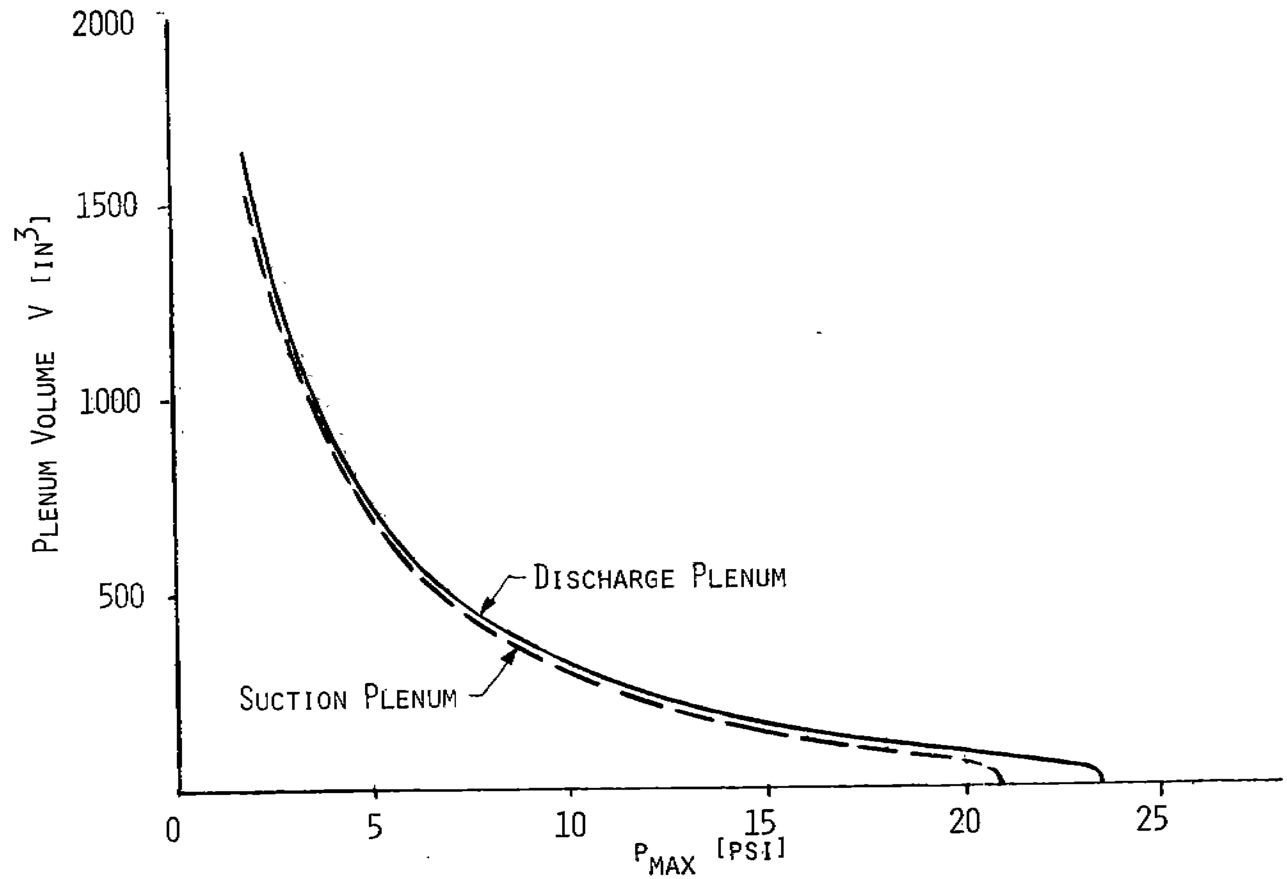

FIGURE 10. NECESSARY PLENUM VOLUME

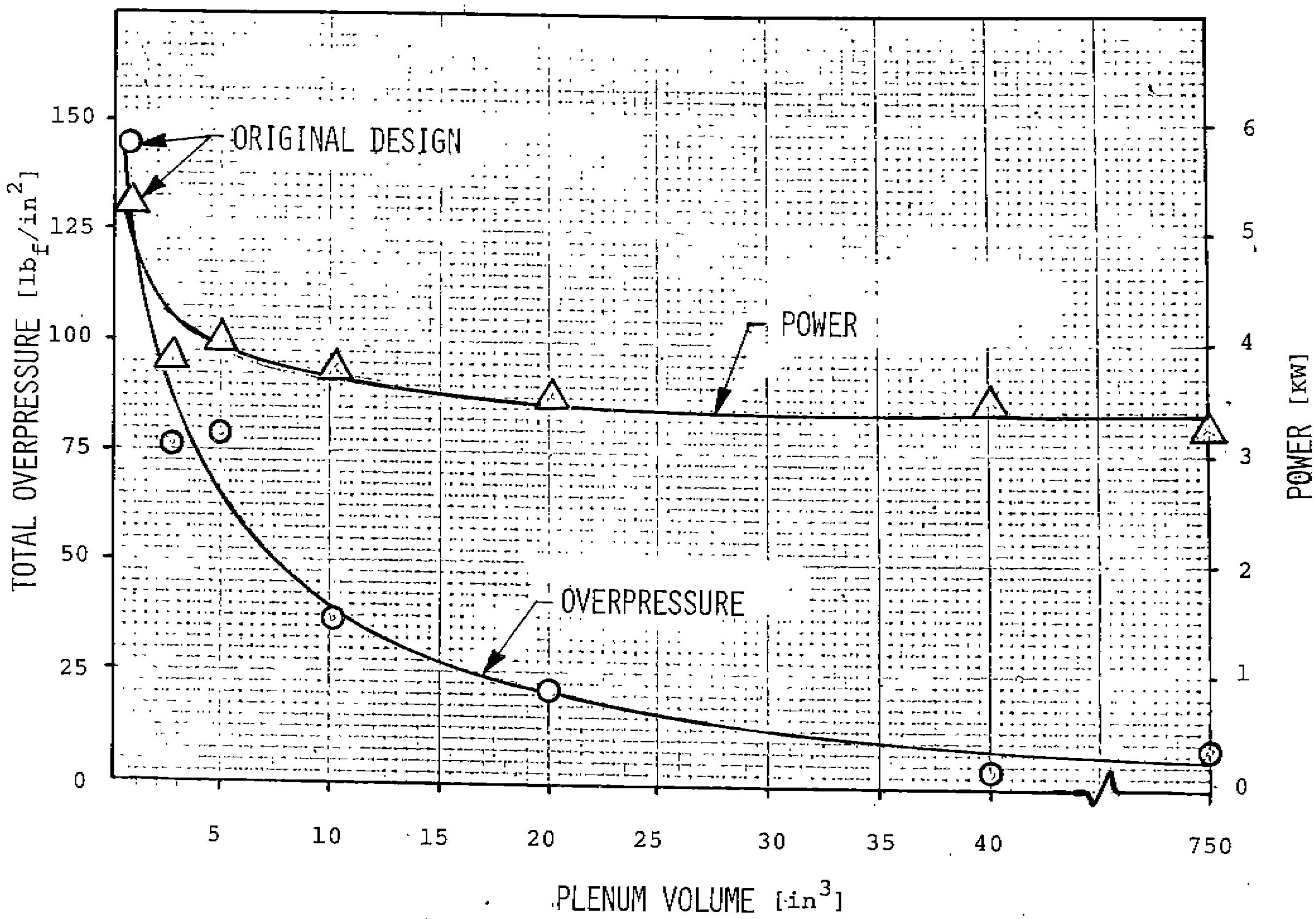

FIGURE 11 MEASURED OVERPRESSURE AND POWER REDUCTION AS FUNCTION OF PLENUM SIZE.