A STUDY OF THE CAUSE, MEASUREMENT, AND REDUCTION OF PRESSURE SURGES

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A STUDY OF THE CAUSE, MEASUREMENT,

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SYMBOLS AND ABBREVIATIONS

A Area of flow, sq. ft.

a Velocity of propagation of pressure wave, fps

app Appendage

B Bore of pump, square inches

C_m Volume of air in desurger at maximum pressure, cubic feet

C_o Volume of air in desurger at operating pressure, cubic feet

Cu.in. Cubic inch

C_x Volume of air in desurger at minimum pressure, cubic feet

cfs Cubic feet per second

cps Cycles per second

d Inside diameter of pipe, inches

E Modulus of elasticity of pipe, psi

f Pressure wave traveling in direction of + x

f_1 Pressure wave traveling in direction of - x

fps Feet per second

F Average pump volumetric factor

g Acceleration due to gravity, 32.2 ft per second^2

G Number grids deflection on oscilloscope

H Pressure head in feet of fluid

H_o Normal flow pressure, feet of fluid

H_s Maximum surge head, feet of fluid

K Bulk modulus of fluid, psi

$\bar{K}$ Friction loss constant

L Length of pipe, feet

lbs/ft^3 Pounds per cubic foot

n Polytropic exponent of gas expansion

P	Pressure, psi
P_m	Maximum pressure, psi
P_o	Normal line pressure, psi
P_x	Minimum pressure, psi
ΔP	Pressure variation or surge, psi
P_m^*	Maximum pressure, psia
P_o^*	Operating pressure, psia
P_x^*	Minimum pressure, psia
psi	Pounds per square inch
psia	Pounds per square inch absolute
psig	Pounds per square inch gage
Q_m	Instantaneous rate of flow at Pm, cfs
Q_o	Mean flow rate, cfs
Q_x	Instantaneous flow rate at P_x, cfs.
S	pump stroke, inches
SPS	Surges per second
t	Wall thickness of pipe, inches
T	Time, second
T_1	Time at maximum surge
Thru	Through-flow
V	Extinguished velocity, feet per second
V_c	Velocity of flow after partial valve closure, fps
V_o	Mean velocity of flow, fps
V_d	Volume entering desurger, FT^3
w	Specific weight, pounds per cubic foot
x	Distance measured from volume end of conduit to point in conduit under consideration
*	Denotes absolute values

GREEK LETTERS

ρ^*	Pipe line characteristic
σ^*	Desurger characteristic

CHAPTER I

STATEMENT OF THE PROBLEM

Pressure variations, commonly called pressure surges, occur in all systems transporting liquids. These pressure surges, caused by the acceleration and deceleration of the fluid column, arise from the valve action of pumps, from water hammer due to valve closure or similar sudden restrictions, or from a combination of both. The magnitude of these pressure surges is dictated by the severity of the velocity change. Figures 1, 2, and 3 show time-pressure recordings of some typical pressure variations. The annoying effects of pressure surges appear in many forms, such as:

1. Failure of system due to over stressing.
2. Failure of system due to fatigue caused by high magnitude high frequency surges.
3. Loss of pump efficiency.[1]
4. Dangerous vibration in series--paralleling of fluid pumps. (Test by International Derrick and Equipment Co., Beaumont, Texas.)
5. Errors in correct metering caused by inertial effect of pressure sruges.[2]
6. Interference of pump surges with bottom hole precussion drilling devices.

[1] E. C. Fitch, "The Effect of Pressure Surges on the Efficiency and Operation of a Piston Pump," Masters Thesis, 1951.

[2] E. C. Fitch and Harry M. Wyatt, "Effect of Transient Pressure on Flow Metering," World Oil, January, 1952.

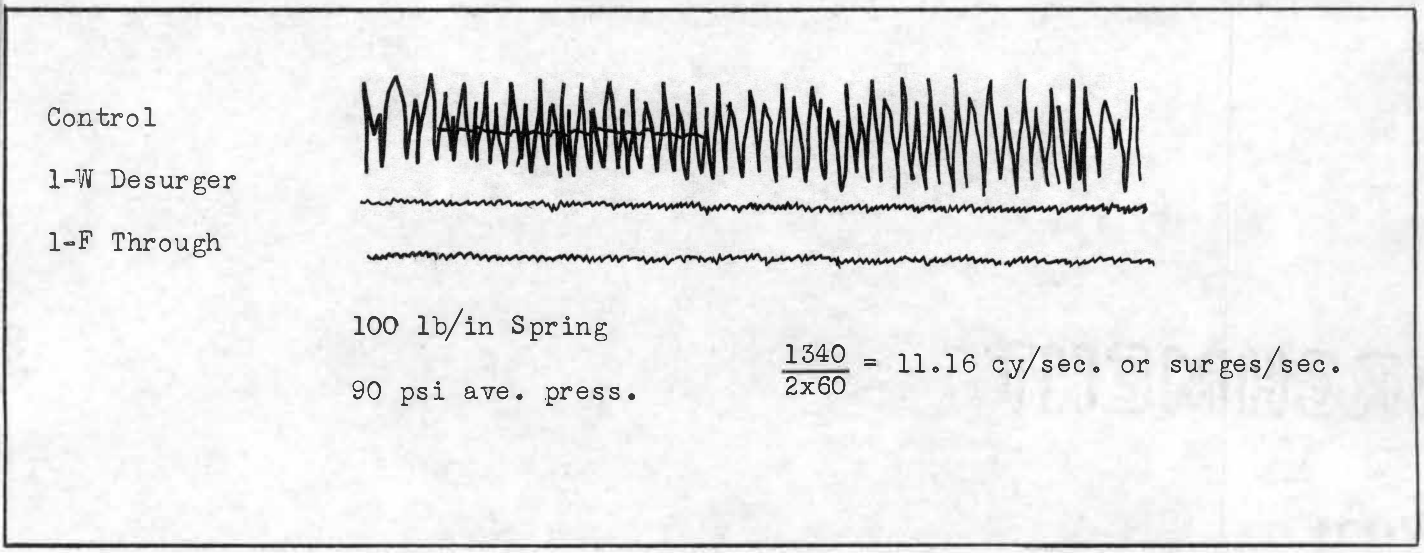


Figure 1 Engine Indicator Card Showing Pressure Variation of a Piston Pump.

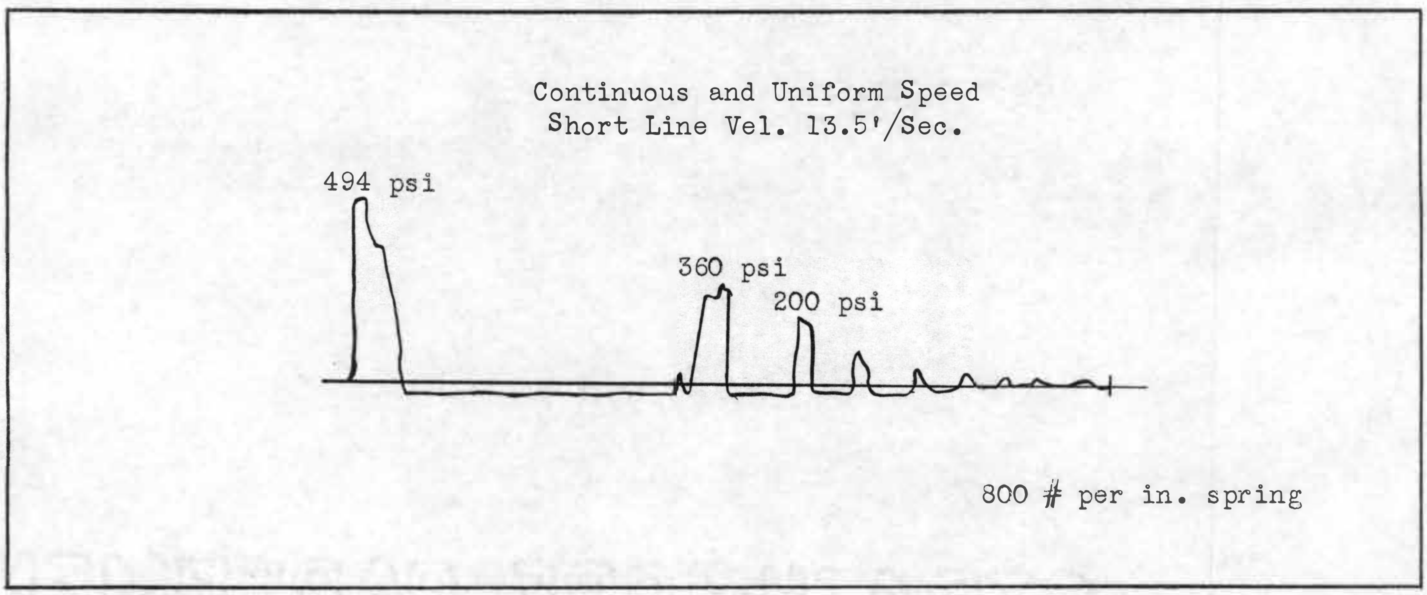
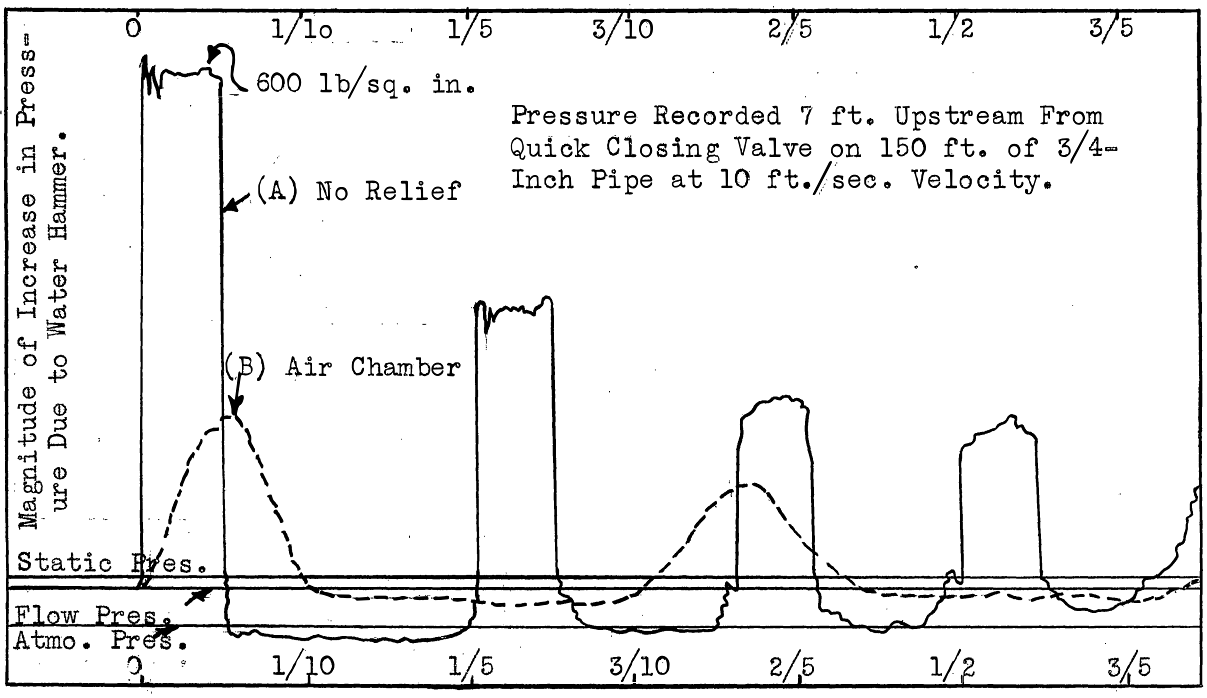


Figure 2 Engine Indicator Card Showing Pressure Surge Created by Instant Valve Closure.

Time in Seconds After Closure of Quick Acting Valve on Test Pipe



Typical Water Hammer Pressure-Time Diagrams Showing

Curves:

(A) Maximum Water Hammer Pressure Wave in Test Pipe with No Relief Device.

(B) Maximum Water Hammer Pressure Wave in Test Pipe with Air Chamber.

Figure 4

This investigation as recorded in the following thesis was initiated to study the cause of pressure variations in piping and to observe the actual efficiency of commercial surge removing devices now on the market.

CHAPTER II

HISTORY AND SUGGESTED SOLUTION TO PROBLEM

From the time the first pipe line was used, pressure surges have been an ever present but undesirable phenomena. With the design trend moving toward higher operating pressures and higher speed pumps, the problem of pressure surges has been correspondingly magnified.

In order to discover a method of reducing or completely relieving a fluid system of pressure surges, many investigations have been made. The first significant contribution to water-hammer theory appears to be that of Michaud[3] published in 1878, where the author noted the oscillation characteristics of water-hammer and considered the influence of the elasticity of the walls of the conduit and the compressibility of water as a form of air reservoir of variable capacity.

In 1904, the Journal of American Water Works Association presented a translation of experiments by Professor Joukowsky[4] in which he developed the theory relative to water-hammer for a closed conduit. Joukowsky first established the rate of propagation of pressure waves and proved that the maximum water-hammer pressure was

$$H_s = \frac{a\,V}{g} \tag{1}$$

[3]S. Michaud, "Water-Hammer in Conduits; Study of the Means Used for Diminishing the Effects," Bulletin de la Societe Vaudoise abs Engeneurs et Architects, Lausanne, 1878.

[4]Joukowsky, "Water-Hammer," Proceedings, American Water Works Association, 1904, p. 344.

where

a = velocity of propagation of pressure wave, fps

V = extinquished velocity, fps

g = acceleration due to gravity, 32.2 feet per second^2

The value of a in feet per second has been shown to be

$$a = \frac{12}{\sqrt{\frac{w}{g}\left(\frac{1}{K} + \frac{d}{Et}\right)}} \qquad (2)$$

where

w = specific weight of fluid (62.4 for water), lbs per cu.ft.

K = bulk modulus of fluid flowing in pipe, psi

E = modulus of elasticity of pipe, psi

d = inside diameter of pipe inches

t = wall thickness of pipe inches.

Figure 4 gives a graphic solution to this equation.

Probably one of the most important works on water-hammer and one on which virtually all of our present theory of water-hammer is based was published in 1903 and extended to 1913 by Allievi[5]. Allievi's works gave the mathematical analysis of water-hammer and presented simple charts for the determination of the maximum pressure rise for uniform closures of valves in simple conduits.

In the American Society of Civil Engineering Transactions of 1920, Mr. N. R. Gibson[6] developed the basic theory of water-hammer as the arithmetic summation of a series of instantaneous water-hammer waves.

[5]Lorenzo Allievi, "General Theory of Perturbed Flow of Water in Pressure Conduits," Annali della Societa degli Ingegneri ed Architetti Italiani, Milan, 1903.

[6]N. R. Gibson, "Pressure in Penstock Caused by the Gradual Closing of Turbine Gates," Transactions A.S.C.E., 1920, Vol. 83.

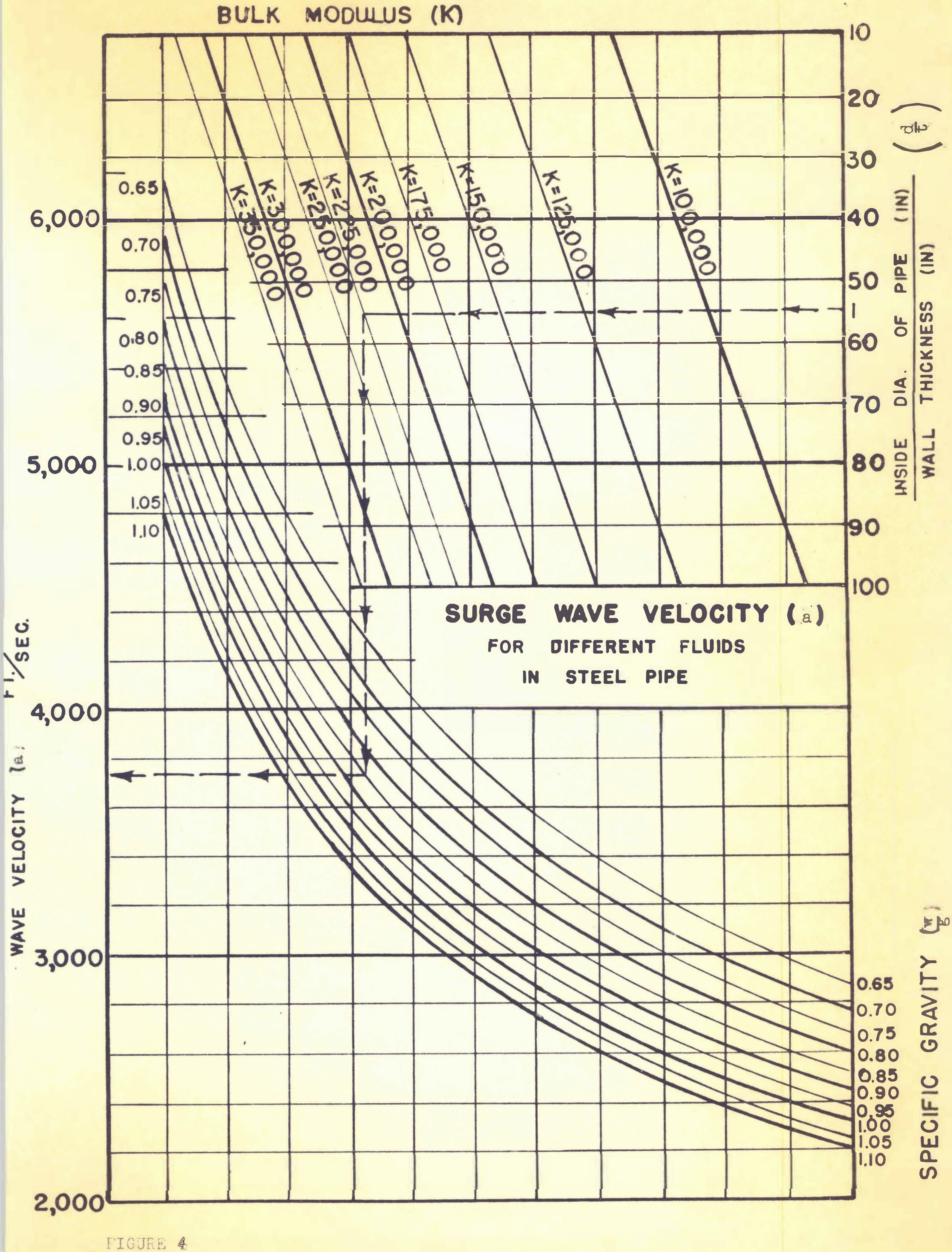

BULK MODULUS (K)
K=350,000
K=300,000
K=250,000
K=225,000
K=200,000
K=175,000
K=150,000
K=125,000
K=100,000
SURGE WAVE VELOCITY (a)
FOR DIFFERENT FLUIDS
IN STEEL PIPE
WAVE VELOCITY (a) FT./SEC.
6,000
5,000
4,000
3,000
2,000
0.65
0.70
0.75
0.80
0.85
0.90
0.95
1.00
1.05
1.10
10
20
30
40
50
60
70
80
90
100
INSIDE DIA. OF PIPE (IN) / WALL THICKNESS (IN) (d/t)
SPECIFIC GRAVITY
0.65
0.70
0.75
0.80
0.85
0.90
0.95
1.00
1.05
1.10

FIGURE 4

The American Society of Mechanical Engineers formed the Committee on Water Hammer and held the first symposium in 1933. As a result of the success of the first symposium on water hammer, the second symposium was held in 1937 and included contributions from engineers in Canada, Great Britain, Switzerland, Italy and Brazil in addition to those from the United States.

Although the studies as mentioned above were necessary for the development of water-hammer theory, solutions based on these theories were extremely involved and not always reliable or practical.

Industry's answer to the problem of pressure variations was the development of surge dampeners or pulsation dampeners, all of which were based on the concept that if fluid could be accumulated as the pressure was increasing and discharged as the pressure was decreasing, a smooth, non-varying flow pressure could be approached. These commercial devices were helpful, but since there sizing was based on a "hit" or "miss" method, they were in many cases inadequate to do the required job.

No doubt the greatest contribution to this science was presented in May, 1954 by Mr. Edwin J. Waller[7] in his "Fundamental Analysis of Unsteady Pressure Variations in Pipeline Systems." In this paper, Mr. Waller presents a method for the solution of surge problems which has been verified by tests on both laboratory and field installations.

Data presented in this thesis was taken before December 8, 1949; since this date several worthy reports have been written, but as they were not available to the writer they were not included in this thesis.

[7]E. J. Waller, "Fundamental Analysis of Unsteady Pressure Variations in Pipeline Systems," Publication No. 92, Oklahoma Engineering Experimental Station of Oklahoma A. and M. College, 1954.

CHAPTER III

BASIC CONCEPTS OF WATER HAMMER THEORY

The three basic concepts of water hammer theory are:

1. Rigid Water Column Theory presented by Mr. R. W. Angus in a report entitled "Water Hammer in Pipes, Including Those Supplied by Centrifugal Pumps."[8]

2. Elastic Water Column Theory presented by Mr. F. M. Wood in a report entitled "The Application of Heavisides Operational Calculus to the Solutions of Problems in Water Hammer."[9]

3. Solution of water hammer problems by impedance matching in fluid system presented by Mr. E. J. Waller in his "Fundamental Analysis of Unsteady Pressure Variations in Pipeline Systems."[10]

Rigid Water Column Theory

When a closed pipe is filled with moving water, the laws governing the changes of pressure and discharge depend upon the conditions under which the flow occurs. If the water is considered to be incompressible and the velocity of water which passes through any section of the pipe remains constant, Bernoulli's energy equation applies at any two sections

[8] Angus, R. W., "Water Hammer in Pipes, Including Those Supplied by Centrifugal Pumps: Graphical Treatment," Bulletin 152, University of Toronto Press, 1938.

[9] Wood, F. M., "The Application of Heavisides Operational Calculus to the Solutions of Problems in Water Hammer," Transactions A.S.M.E., Vol. 59, Paper Hyd-59-15, November, 1937, pp. 707-713

[10] E. J. Waller, "Fundamental Analysis of Unsteady Pressure Variations in Pipeline Systems."

of the pipe. However, when the motion is unsteady, that is, when the discharge at each section is varying rapidly from one instant to the next, rapid pressure changes occur inside the pipe and the Bernoulli equation is no longer applicable. These pressure changes are referred to as "water hammer" due to the hammering sound which often accompanies the phenomena.

In order to obtain the basic physical laws of water hammer, the effect of rapid changes in flow are considered for a pipe line of uniform area A and length L. The pipe line is connected to a reservoir at its upper end and has a control gate at the lower end for regulating the discharge of water into the atmoshpere. In the presentation of this theory the following assumptions are made:

1. The water in the pipe is incompressible.

2. The pipe walls do not stretch regardless of the pressure inside the pipe.

3. The pipe line remains full of water at all times and the minimum pressure inside of the pipe is in excess of the vapor pressure of water.

4. The hydraulic losses and velocity head are negligible when compared with the pressure changes.

5. The velocity of water in the direction of the axis of the pipe is uniform over any cross section of the pipe.

6. The pressure is uniform over a transverse cross section of the pipe and is equal to the pressure at the center line of the pipe.

7. The reservoir level remains constant during the gate movement.

If the flow at the control gate is altered, an unbalanced external force will act at the gate on the mass of the water column. The magnitude of this unbalanced force is determined through the application of

Newton's second law of motion and found equal to

$$\frac{H_s}{H_o} = \frac{K_1}{2} + \sqrt{K_1 + \frac{K_1^2}{4}} \qquad (3)$$

where

$$K_1 = \left(\frac{L\ V}{g\ H_o\ T_1}\right)^2 .$$

H_o = Normal flow pressure, feet of fluid

L = Length of pipe, feet

T_1 = Time at maximum surge, seconds.

Elastic Water Column Theory

The same assumptions used in Rigid Water Column Theory are applicable in this approach with the exception that the elasticity of the pipe walls and the compressibility of the water under the action of a pressure change are also taken into account. An element of water which is bounded by two parallel forces normal to the axis of the pipe is considered. The condition of dynamic equilibrium requires that the unbalanced force acting on the element of water be made equal to the product of the element's mass and acceleration; that is, Newton's second law of motion is satisfied. The condition of continuity for the element requires that all available space inside the boundaries of the element be occupied by water at all times. The equations resulting from the conditions of dynamic equilibrium and continuity are then solved simultaneously to obtain the fundamental water hammer equations, which are expressed as follows:

$$H_s - H_o = f\left(T_1 - \frac{x}{a}\right) + f_1\left(T_1 + \frac{x}{a}\right) \qquad (4)$$

$$V_c \quad V_o = \frac{g}{a}\left[f\left(T_1 - \frac{x}{a}\right) \quad f_1\left(T_1 + \frac{x}{a}\right)\right] \tag{5}$$

where

x = distance measured positive from valve end of conduit to point in conduit under consideration

f = pressure wave traveling in direction of + x

f_1 = pressure wave traveling in direction of - x

V_c = velocity of flow after partial valve closure, fps

V_o - mean velocity of flow, fps.

<u>Solution by Impedance Matching</u>

In the development of this concept the following assumptions were made:

1. One directional flow.

2. The principle of superposition was valid for this case.

3. The elasticity of the fluid was expressable in explicit terms.

4. The stress on the faces of the fluid element was expressed in terms of the deformation of element.

5. Turbulence when it occurs may be expressed in terms of the deformation, which enables one to make a considerable simplification of the differential equations of motion.

In this approach Mr. E. J. Waller through an analysis based on fundamental hydrodynamic theory was able to define boundary conditions in terms of the physical parameters of the system. With the boundary conditions thus defined a solution to the system differential equations was possible and by the use of fundamental wave mechanics this solution was interpreted. This enables one to analyze an existing pipe line system by considering the known physical properties of the fluid, pipe, pump, etc., along with the flow conditions (including measurements of instantaneous pressure) for existence of adverse pressure variations in the system and to design components that when placed in the system would change it so that adverse pressure variations were no longer present. This concept was pertinent to the over all problem, but not essential to the work performed in this investigation.

CHAPTER IV

SOLUTION

There are numerous methods for reducing pressure surges or water-hammer. Any control valve or device which slowly changes the velocity of flow in the pipe or which stores and dissipates energy from the fluid is effective in reducing pressure surges or water-hammer pressure. Some of these are listed below:

1. Slow closing valves.
2. Spring operating relief valves.
3. Surge tanks.
4. Automatic surge suppressors actuated hydraulically or electrically.
5. Mechanical shock absorbers or cushions.
6. Air chambers.
7. Mechanical pneumatic arrestors or fluid impact absorbers.

Unfortunately none of the above relief measures are practical or adequate under all circumstances. For example reducing water-hammer pressures by closing valves slowly is desirable but since this requires personnel it is not economically practical. Moreover, the modern trend is toward quick closing faucets and automatic flush valves for both domestic and industrial use. In many industrial processes quick acting valves are a necessity.

Spring operated relief valves may be adequate under some conditions where it is possible to drain off the fluid discharged from the relief port opening during the water-hammer pressure rise. The necessity of providing a drain is often annoying, moreover, frequent operation tends to wear all parts quickly so that such valves require considerable attention and maintenance.

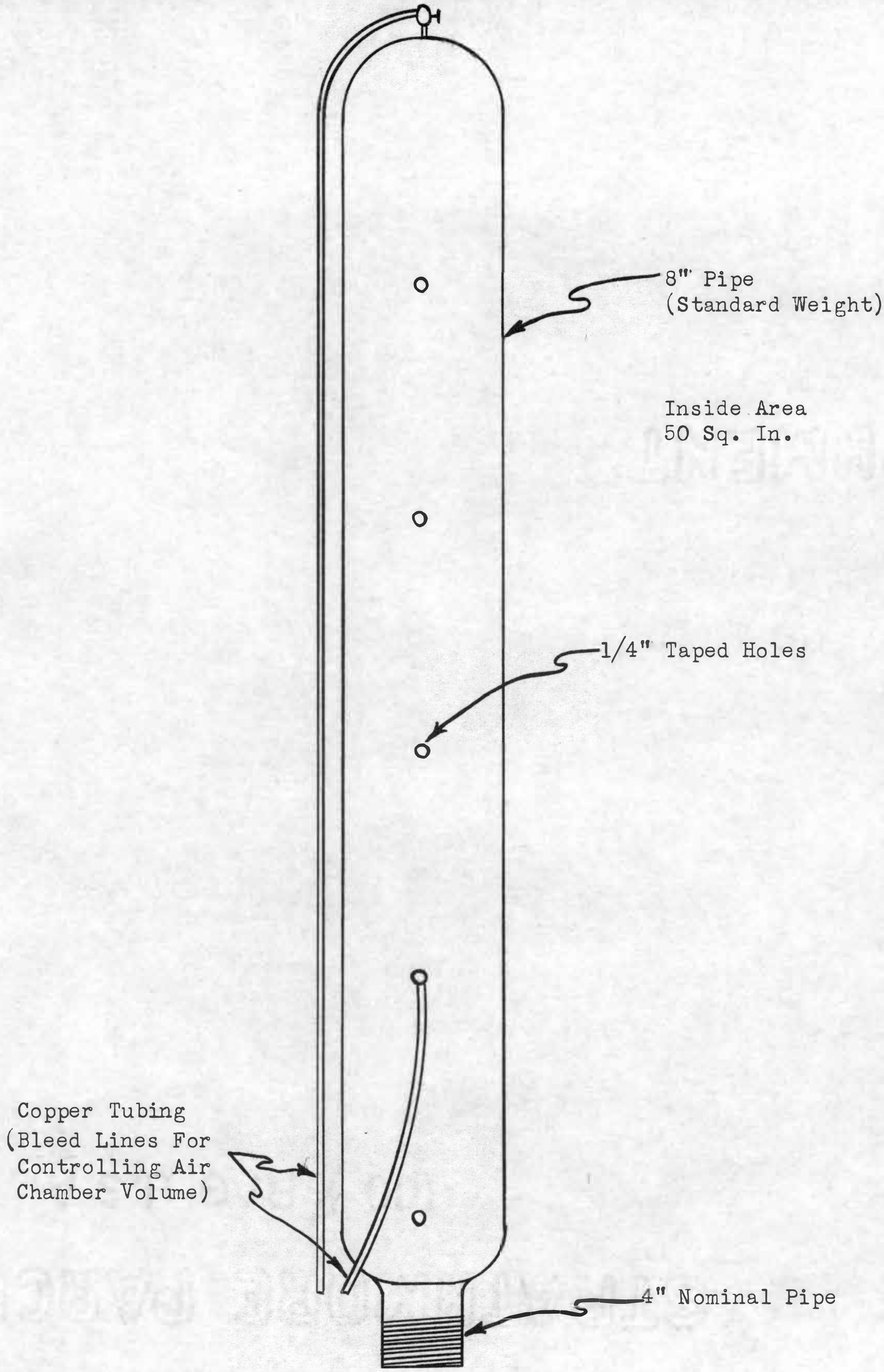
Surge tanks and automatic surge suppressors are not economic protective devices for pipe line systems. The former often overflows and discharges large amounts of flowing fluid and is not adaptable to high pressure systems. The latter is usually designed for very large pipelines and is quite expensive.

Mechanical shock absorbers have not proved entirely satisfactory due to sluggishness of moving parts, small shock absorbing capacity, and high maintenance cost.

Air chambers when functioning are admitted to be a most economical and fairly efficient protective device and have been widely used even though it has been difficult under repeated shocks to keep adequate amounts of air in the chamber. If the air is replenished at a pressure equal to static pressure in the pipe, it expands when a valve is opened and the pressure drops to flow pressure. A portion of the air is thus carried out with the fluid and the volume of air is reduced quickly to that which would exist if the air in the chamber were originally at atmospheric pressure. The remaining air is either absorbed by the water under repeated shocks or leaks from the chamber due to faulty construction.

Of the seven devices listed above the two most important are (6) air chambers and (7) mechanical pneumatic arrestors or fluid impact absorbers.

Air chambers (check Figure 5) are probably the most commonly used of all the different types of surge removing devices and therefore will be discussed first.



AIR CHAMBER

Figure 5

AIR CHAMBER SIZING.

The size of the air chamber and air space are selected so that the air chamber will be able to store a portion of the excess volume of the compression wave not absorbed by the fluid itself or the expanding pipe without raising the air pressure beyond a predetermined limit.

To be effective the air vessel should be placed as close as possible to the source of pressure surge.

<u>Theory for the Sizing of Air Chambers for Reciprocating Pumps</u>.

If it is assumed that the velocity and hence the instantaneous quantity of the fluid discharged by a simplex pump varies almost sinusoidally with respect to time or angle of crank rotation, it is apparent that this instantaneous flow rate is alternately less and greater than the mean flow rate. The mean flow rate may be defined as the product of the total quantity of fluid discharged during a 360° rotation of the cranks and the number of revolutions of the cranks per unit of time. Making the pump double-acting or increasing the number of cylinders results in a less widely fluctuating quantity-time curve since the velocity of piston will be increasing during the interval that the velocity of another is decreasing. Obviously the flow rate variations would approach zero as a limiting value as the number of equally spaced cranks was infinitely increased. Since there are practical limits to the number of cylinders employed in reciprocating pumps, the variations from the mean flow rate must be removed by other means. A chamber of adequate capacity located at the discharge of the pump has been proven effective in eliminating volumetric and hence pressure fluctuations in countless installations.

In order to determine the correct capacity of the air chamber to be used to accommodate the volume of liquid pumped in excess of that flowing through the associated piping system at the mean flow rate, it will first be necessary to examine the flow characteristics of the system. If a constant back pressure P_o is maintained, the pump, equipped with a surge suppressor and operating between the pressure limits P_x and P_m where $\frac{(P_x + P_m)}{2} = P_o$, will furnish fluid at its rated mean flow Q_o If the back pressure effective at the pump is maintained by pipe friction with or without flow controlling devices, this pressure P_o will equal the pressure loss in the line at the mean flow rate Q_o. Let P_m be the maximum allowable surge pressure. When a portion of the liquid is at pressure P_m, its instantaneous flow rate becomes Q_m, which is greater than Q_o. Because over a period of one cycle the pump is delivering fluid at the mean rate Q_o, during a portion of this cycle the instantaneous rate will be reduced to Q_x, where $Q_o - Q_x = Q_m - Q_o$. In order to simplify calculation $Q_o - Q_x = Q_m - Q_o$ is actually an assumption. But by so assuming a number of variables are eliminated from our calculation. These assumptions do not materially change the true values and it does give a simple workable solution. If the reduced system pressure which impels a flow of Q_x is P_x, the approximate pressure relationship becomes $P_m - P_o = P_o - P_x$, this assumption considerably simplifies the mathematics of unsteady flow. Assuming then that P_o is known or can be calculated and the allowable pressure rise $P_m - P_o$ is specified, the capacity of the required suppressor can be determined. The pressure relationship is: $P_x = 2P_o - P_m$ Using the starred symbols P_x, P_o^*, and P_m^* for absolute values of the corresponding gauge pressure P_x, P_o, and P_m the equation for the calculation of the required desurger capacity C_x is derived in the following manner.

Let:

B = bore of pump, square inches

C_m = volume in chamber at pressure P_m , cubic feet

C_o = volume in chamber at pressure P_o , cubic feet

C_x = volume in chamber at minimum P_x , cubic feet

F = average pump volumetric factor

n = polytropic exponent of gas expansion

P_m = maximum pressure, psi

P_o = normal line pressure, psi

P_x = minimum pressure, psi

P_x = chamber charging pressure, psi

P_m^* = P_o^* + allowable pressure increase, psia

P_o^* = operating pressure, psia

P_x^* = P_o^* - allowable pressure increase, psia

Q_m = instantaneous flow rate at P_m , cfs

Q_o = mean flow rate, cfs

Q_x = instantaneous flow rate at P_x , cfs

S = pump stroke, inches

* = denotes absolute values

The pump volumetric factor is defined as the ratio of the average fundamental volumetric variation per stroke divided by the volumetric displacement per piston. For pump volumetric factor F, see Table I. Since pressure fluctuations, created by every piston stroke, may occur rapidly at frequencies of one or more cycles per second, the cushioning gas within the chamber will be compressed and expanded nearly adiabatically. A value of 1.4 for n should thus be employed for air-charged suppressors.

From basic gas law

$$P_o^* C_o^n = P_m^* C_m^n \tag{6}$$

and

$$C_m = C_o \left(\frac{P_o^*}{P_m^*}\right)^{\frac{1}{n}}$$

Volume of liquid, V_d , entering the desurger is

$$V_d = C_o - C_m = C_o \left[1 - \left(\frac{P_o^*}{P_m^*}\right)^{\frac{1}{n}} \right] \text{cu. ft.} \tag{7}$$

The volume, C_x , occupied by the gas at pressure P_x is

$$C_x = C_o \left(\frac{P_o^*}{P_x^*}\right)^{\frac{1}{n}} \tag{8}$$

The required volume to be accumulated by the chamber as dictated by the pump is:

$$V_d = (0.7854\ B^2 SF) = C_o \left[1 - \left(\frac{P_o^*}{P_m^*}\right)^{\frac{1}{n}} \right] =$$

$$(0.7854\ B^2 SF) = \frac{C_x}{\left(\frac{P_o^*}{P_x^*}\right)^{\frac{1}{n}}} \left[1 - \left(\frac{P_o^*}{P_m^*}\right)^{\frac{1}{n}} \right] \tag{9}$$

$$C_x = \frac{(0.7854\ B^2 SF)\left(\frac{P_o^*}{P_x^*}\right)^{\frac{1}{n}}}{1 - \left(\frac{P_o^*}{P_m^*}\right)^{\frac{1}{n}}} \text{cu. in.} \tag{10}$$

When applying equation (10) if calculated value of C_x is smaller than actual volume of desurger (C_o) that you have selected the unit has sufficient desurging capacity; if C_x is greater than C_o, a larger desurger is required.

Pump	Type	Factor F
Simplex	Single acting	0.60
	Double acting	0.25
Duplex	Single acting	0.25
	Double acting	0.15
Triplex	Single acting	0.13
	Double acting	0.06
Quadruplex	Single acting	0.10
	Double acting	0.06
Quintuplex	Single acting	0.06
	Double acting	0.02
Sextuplex	Single acting	0.06
Septuplex	Single acting	0.02

Table I. Reciprocating Pump Volumetric Factor

Theory for Sizing of Air Chambers for Valve Closures

1. Let H_m^* = maximum or allowable pressure head, ft

H_s^* = maximum pressure surge head, ft

H_o^* = static pressure head, ft

then $H_m^* = H_s^* + H_o^*$.

2. Lorenzo Allievi[9] shows that the pressure surge in a pipe line equipped with an air chamber depends on the two parameters, ρ^* and σ^*, where friction is not considered. He then shows that, without frictional effects, chambers of normal size are ineffectual in controlling upsurges. Louis Bergran in a dis-

[9] Lorenzo Allievi, 1903.

cussion of Allievi's paper,[10] describes a simple, effective, differential orifice for use in conjunction with an air chamber. In his early work with water-hammer Mr. Allievi describes the pipe line characteristic ρ^* as

$$\rho^* = \frac{aV_o}{2g\, H_o^*} \tag{11}$$

where

ρ^* is defined as pipe line characteristic as far as water-hammer surges is concerned. a, V_o, g, and H_o^* have been defined earlier.

The air chamber characteristic developed by Allievi is σ^*,

where

$$\sigma^* = \frac{2g\, C_o\, H_o^*}{A\, L\, V_o^2} \tag{12}$$

where

σ^* = air chamber characteristic

C_o = the volume of air in the air chamber at absolute pressure head H_o^* cu ft. g, H_o^*, V_o have been defined earlier

A = area of flow

L = length of pipeline in question.

3. Algebraic manipulation of Equations (11) and (12) gives

$$\sigma^* = \frac{C_o\, a}{A\, L\, V_o\, \rho^*} \tag{13}$$

or

$$C_o = \sigma^*\, \rho^*\, Q_o\, \frac{L}{a} \tag{14}$$

[10]Lorenzo Allievi, "Air Chambers for Discharge Pipes," Transactions A. S. M. E., Vol. 59, 1937.

4. For ease in calculating certain simplified assumptions have been made concerning the transient wave that follows power interruption. It is assumed that, (a) there is a check valve on the discharge side of the pump which closes immediately on power failure, and (b) the air chamber is situated near the pump. In addition, it is assumed that, (c) the pressure-volume relationship for the air in the chamber may be expressed by

$$H^{*}\ C^{1.4} = \text{a constant.}$$

It is further assumed that, (d) the ratio of the total head loss for the same flow into and from the air chamber is 2.5 to 1, (e) the air in the chamber is subjected to a head of H^{*}, (f) the head loss (surface friction and loss at the orifice) varies with the square of the velocity; and (g) during the transient condition following power failure, the condition of continuation of flow in the discharge line is maintained ---- that is, the water column remains intact throughout the length of the line. These assumptions and simplifications are necessary because the variables involved are so numerous and they permit a solution which yields useable results. Under the conditions imposed by the assumptions this entire transient is completely described by fixing the variables $\overline{K}$, $2\,\rho^{*}$, and $2\,\rho^{*}\,\sigma^{*}$. This variable $\overline{K}$ will be defined so that $\overline{K}H^{*}$ is the total head loss for a flow of Q_o down the pipeline and into the air chamber where Q_o is the initial rate of flow in the pipeline in cubic feet per second.

When $2\,\rho^{*}$, σ^{*}, and $\overline{K}$ are fixed it makes possible computation by the graphical method of the complete transient which follows power failure. In Figure 6 the maximum upsurges have

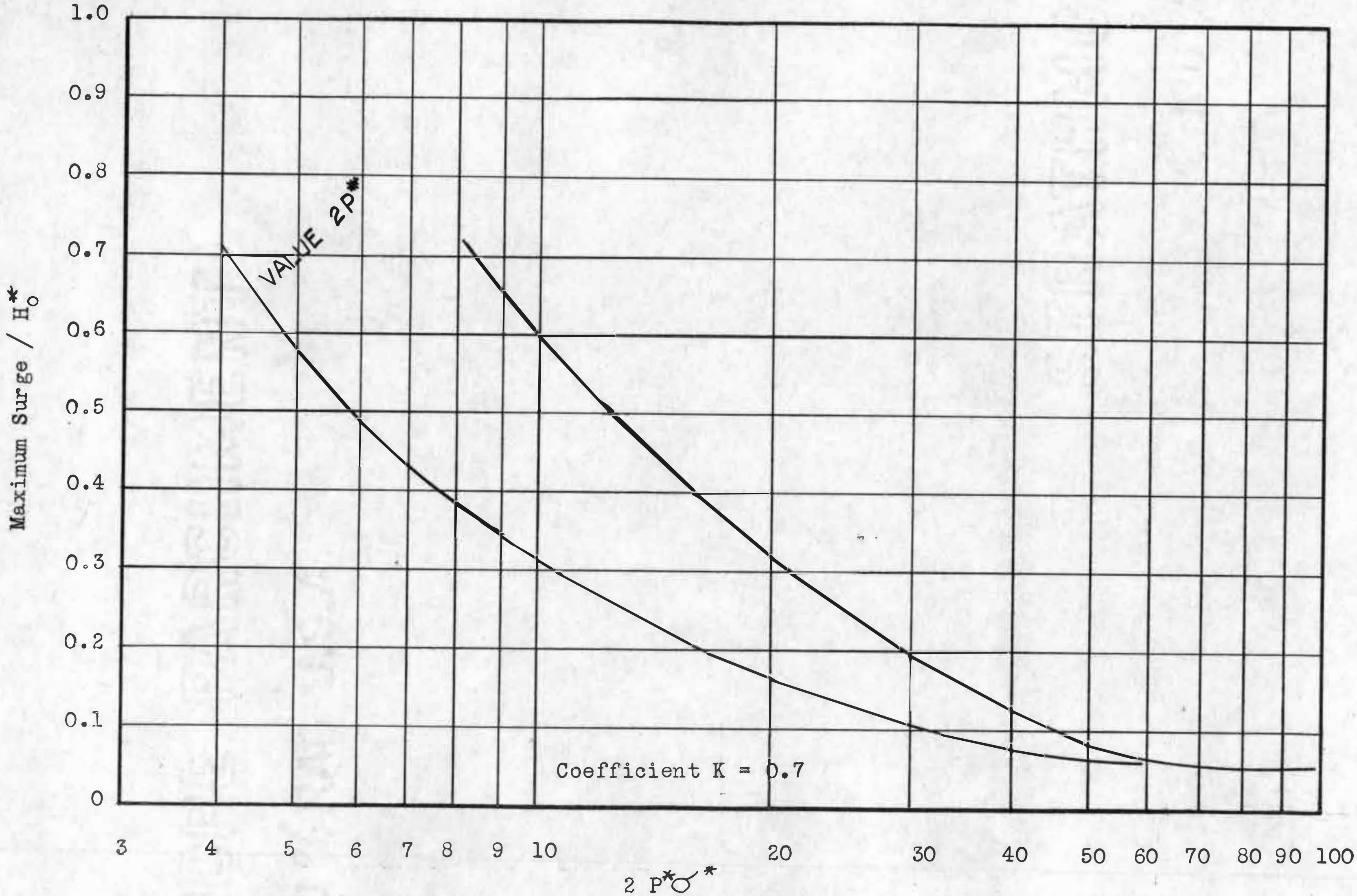


FIGURE 6 Maximum Surge for Pump-Discharge Line Equipped With Throttled Air Chamber.

been plotted in terms of these variables. Maximum upsurges at the pump are plotted as percentages of H_o^* for various values of the descriptive variables.

Ordinarily, when an air chamber is being designed for a pump-discharge line, the values of L, a, V_o, Q_o, A, H_o^*, and g will be known. From these values 2ρ can be computed. The allowable, maximum surge values may be dictated by specifications, operating conditions, or the profile of the discharge line. With the values of $2\rho^*$ and the maximum allowable surges known, values of $\bar{K}$ and $2\rho^* \sigma^*$ may be chosen from Figure 6 such that the surge limitations are met. When $2\rho^* \sigma^*$ has been determined, C_o can be computed from Equation 14. The volume of the air chamber is then determined by considering that the chamber must contain adequate air above the upper emergency level to control the surges to desirable limits, and enough water below the lower emergency level to prevent unwatering. With allowances for the volume between the upper and lower emergency levels, the total required volume of the air chamber can be computed.

5. C_o is the minimum volume of air that must be maintained within the air chamber for it's efficient operation. Therefore, the total amount would depend on the type of installation and frequency of service. For a system in operation 24 hours daily only absorption must be considered, but for a system under intermittent use the air chamber must be checked at each starting.

CHAPTER V

INSTRUMENTS FOR SURGE STUDY

As mentioned earlier, if a suitable solution to the dampening of pressure surges is to be developed, it is a certainty that acceptable instruments must be used for the recording and studying of these surges.

The most common type of instrumentation for measuring transient pressures usually consists of two primary units, the first, is the pick-up or transmitter which converts a pressure change into some measurable electrical impulse such as resistance, potential or capacitance and sends out a signal as a change in potential, and the second is the receiver which accepts the transmitted signal and produces a written or photographic record.

Two of the most rigid requirements for a pick-up device are the need for sensitivity and stability. High sensitivity may be termed the ability to produce a strong output signal with small pressure changes even at high static pressures. A pick-up with good stability is one in which the transmitted signal remains constant for a unit pressure change regardless of frequency, vibration, temperature or static pressure. Other desirable features of a pick-up recommend that it be easily attached to the pipeline, is small, rugged, and requires a minimum of auxiliary equipment operating at a safe voltage.

Pick-up devices can conveniently be classified by the method in which they change or produce an electrical signal.

1. Strain gauge and wheatstone bridge combinations in which unbalance of the bridge produces a change in potential.
2. Condenser type pick-up instruments in which a high frequency is

modulated. This requires an oscillator which may be considered a part of the electrical circuit of the transmitter.

3. Crystal type pick-up instruments which may be used to produce or change the potential of a circuit.

4. Electromagnetic instruments which use a moving coil or magnet and are self generating.

5. Electrokinetic instruments which utilize the phenomena of the streaming potential of a liquid through a porous solid. The device is self generating. Other instruments often used are bourdon gauges, bourdon recorders and engine indicators.

CHAPTER VI

EQUIPMENT

Due to the tremendous scope of water-hammer theory and the limited finances available for this particular study the author restricted his laboratory investigation to the determination and comparison of the efficiency of the surge removing ability of two commercial available desurgers and the common air chamber. The two desurgers used in this study were the Wade Shokstop and the Fluidynamic Desurger. The Wade Shokstop, Figure 7, manufactured by Wade Manufacturing Company, a division of Woodruff and Edwards, Incorporated of Elgin, Illinois is an appendage-type device containing metallic bellows and an air chamber which is precharged for each installation. The Fluidynamic Desurger, Figure 8, manufactured by Westinghouse Air Brake Company of Wilmerding, Pennsylvania, is a through-flow device which incorporates a combination of two surge-removing techniques, throttling orifices and a variable volume chamber.

After the causes of pressure variations in piping were investigated, the next step in this study was the building of equipment to test and measure the magnitude of surges with and without the use of desurgers. Photograph 1 shows the test set-up as it appears in the Hydraulic Laboratory at Oklahoma A & M College. The pump, as it appears in Photograph 2, was manufactured by National Cooperatives, Inc., Chicago, Illinois, and is a single piston double acting pump with a 1.75 inch stroke and a 1.5 inch bore. (3.0925 $inch^3$ per stroke). The power was supplied by a shaft connected by means of a Reeves Pulley to a 5 horsepower electric motor with variable speed drive. The RPM of the pump was determined by a strobotac, Photograph 2, which had

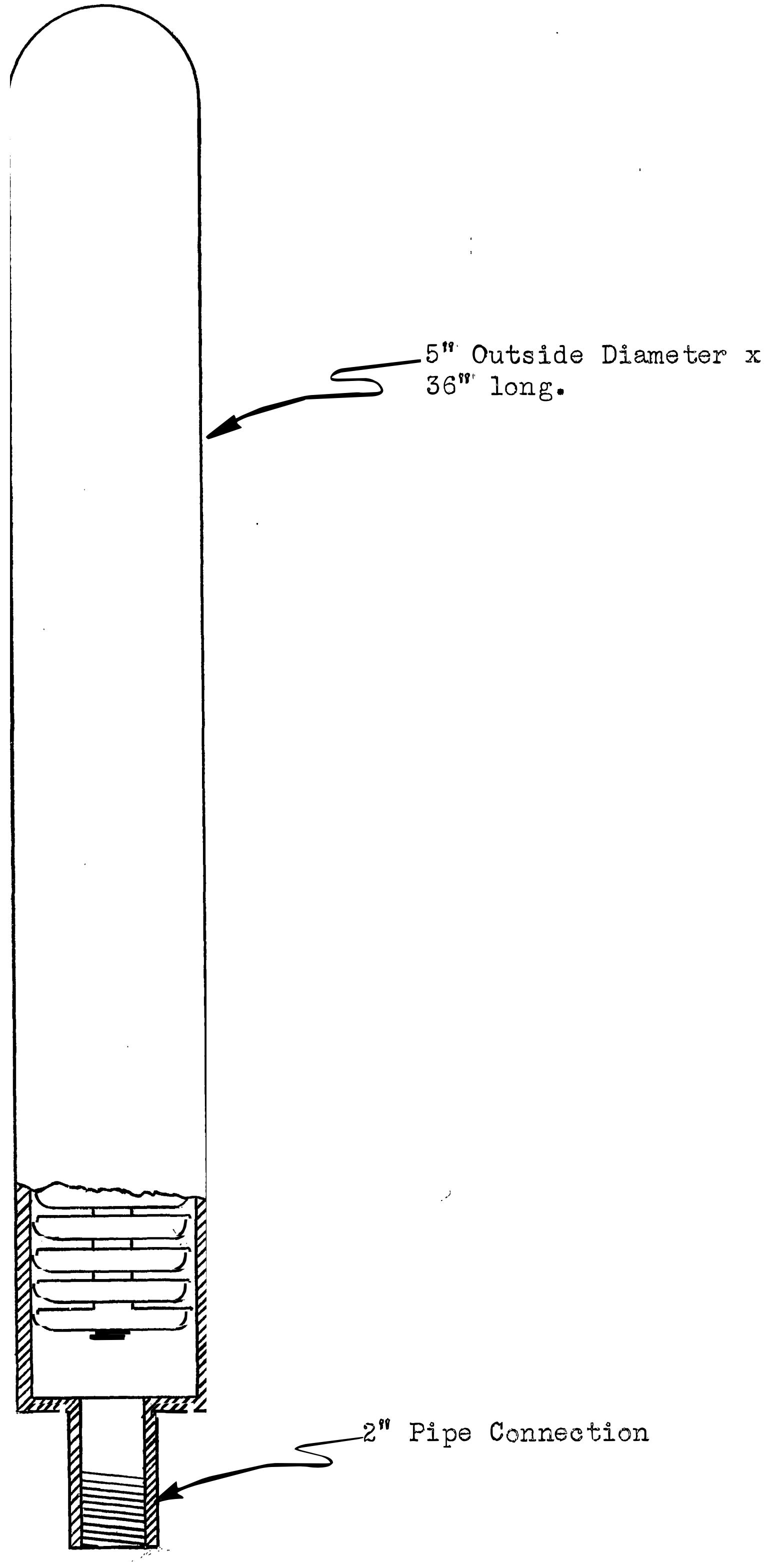


Figure 7 Wade Shokstop Desurger

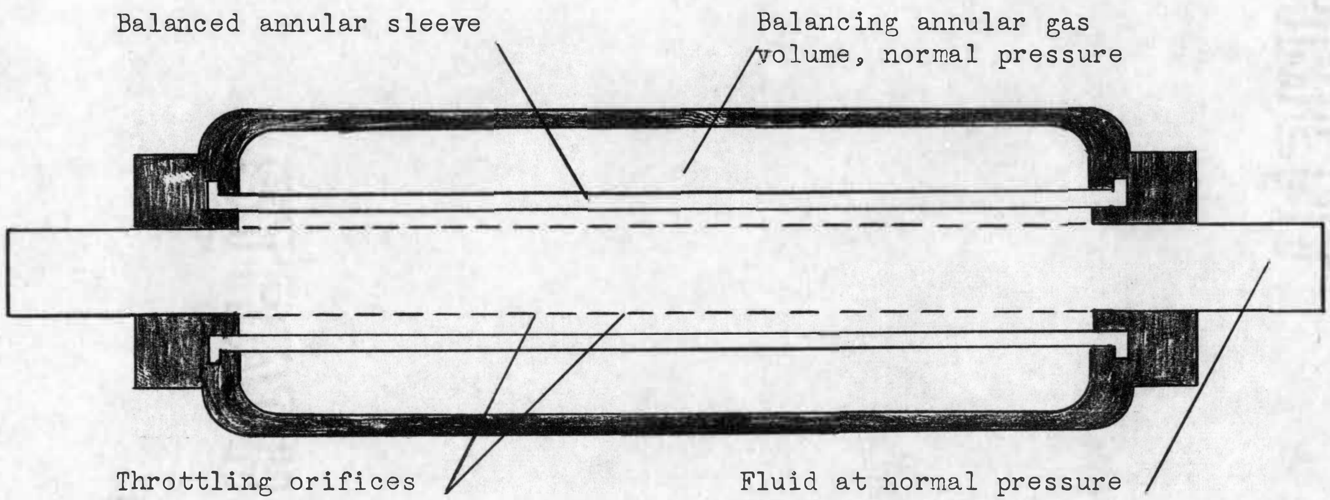


Figure 8 Fluidynamic Desurger.

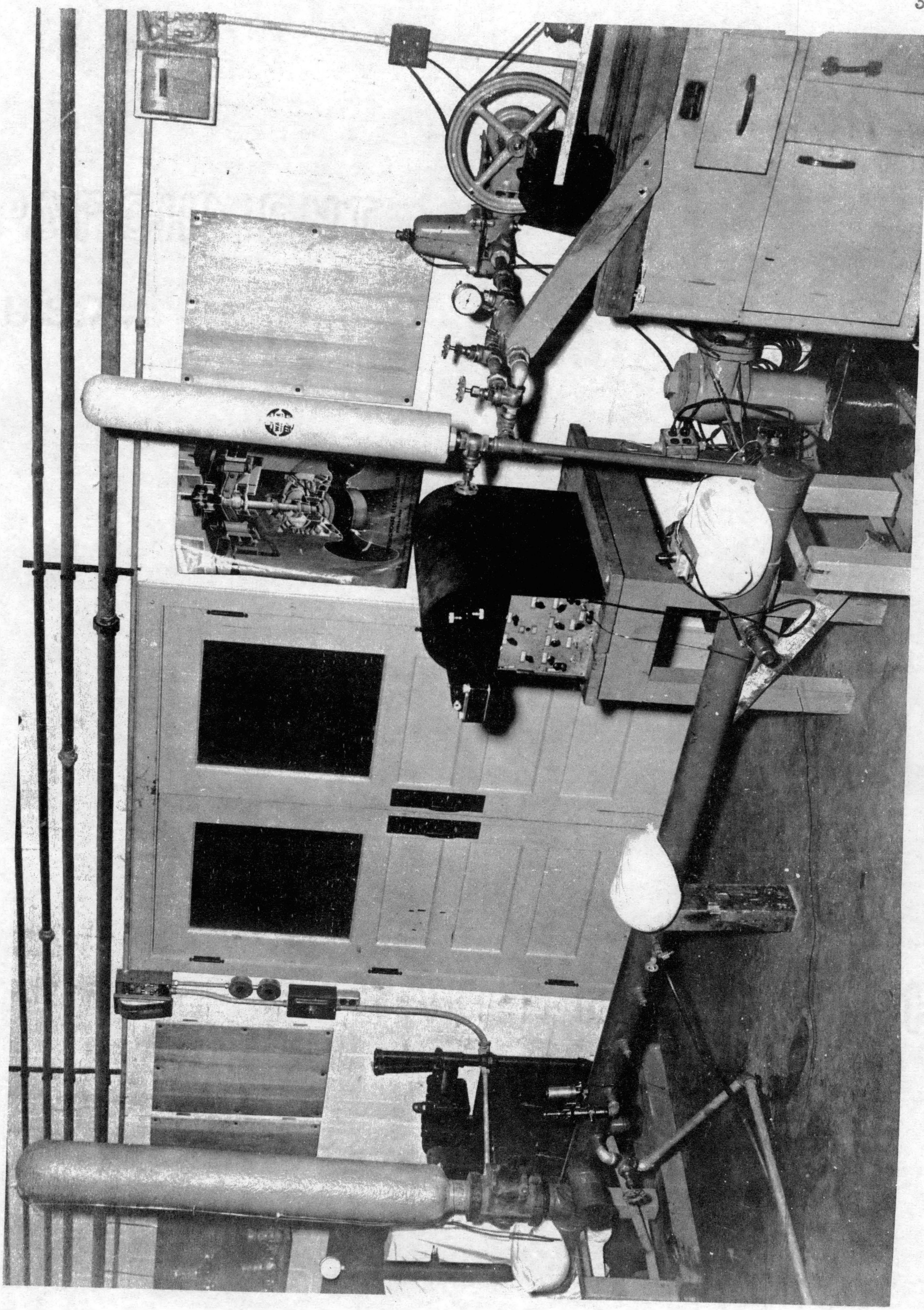


Photo No. 1

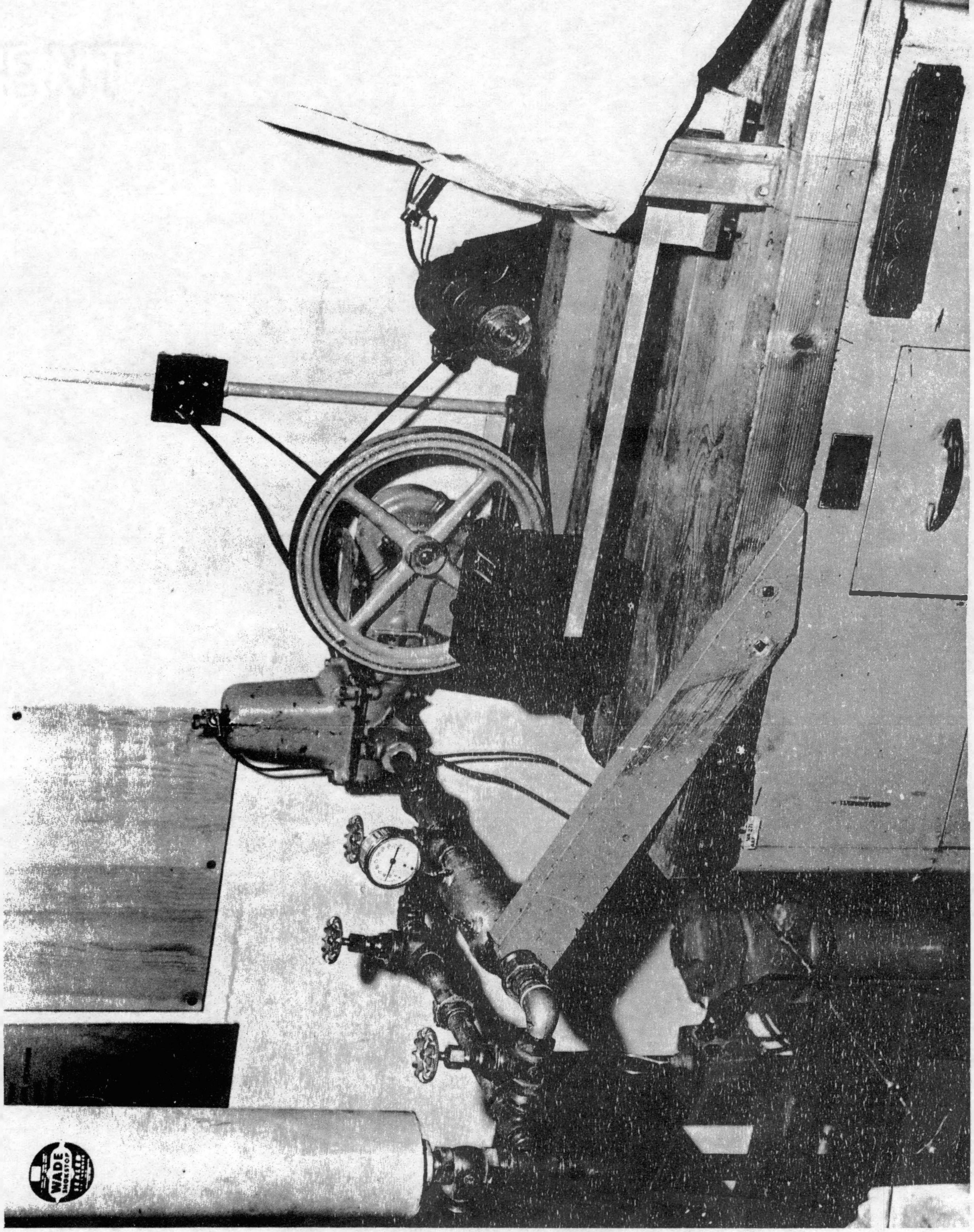


Photo No. 2

been checked with a revolution counter at various points within the working range. Water was supplied to the suction side of the pump by means of a 1-1/2 inch line from a six inch supply riser. The discharge from the pump was carried by a one inch line into the first test section which consisted of the Fluidynamic desurger with a bypass built around it so that tests could be run using the desurger as a through flow device or as an appendage with flow going around the bypass. The Fluidynamic could also be completely shut off from the system when running efficiency tests on the Wade Shokstop and on the air chamber. (In the remainder of this thesis the Fluidynamic Desurger will be designated as the 1-F desurger or surge suppressor, with the Wade Shokstop designated as the 1-W desurger or surge suppressor.)

Downstream from the 1-F desurger and bypass, the 1-W desurger was installed in a vertical position as shown in Photographs 1 and 2. The one inch line led directly from the 1-W desurger to the six inch instrument run as shown in Photograph 1. The air chamber was located at the very end of the instrument run (Photograph 1) which actually placed the air chamber in the most advantageous location since part of the energy of the pressure surges would be absorbed by the system before reaching the air chamber. The air chamber was constructed so that a controlled amount of air could be maintained in the chamber at all times (see Figure 5). During the entire period of testing the amount of air or desurging volume in the three devices were as follows: air chamber 1180 cu. in., 1-W desurger 573 cu. in., 1-F desurger 79 cu. in. Fluid was discharged from the instrument run through a 1-1/2 inch line into the laboratory sump. At the end of the 1-1/2 inch pipe run was a control valve for varying the discharge pressure.

Since it was necessary to measure high frequency, high magnitude surge impulses in order for this study to be successful it was deemed advisable to build or develop an electronic surge measuring device for it was felt by advisers concerned that a mechanical device would have too much inertia of working parts. An electrokinetic transducer which utilizes the phenomena of the streaming potential of a liquid through a porous solid was chosen for this task. Since sufficient funds were not available for the purchase of such equipment it was necessary to have one built. Mr. Gordon Smith of the R.A.D. Laboratory, under the direction of Professor Norton and Professor Fristoe, built such an instrument according to the instruction as outlined by Mr. Milton Williams, Humble Oil and Refining Company, Houston, Texas, in his technical article, "An Electrokinetic Transducer," which appeared in the October, 1948 issue of "The Review of Scientific Instrument." The electrokinetic transducer which consists of a pickup, amplifier, voltage regulator, and oscilloscope is shown in Photograph 3. Photograph 4 shows the transducer installed in a working position. Other instruments used were as follows: Strobotac to measure the RPM of the pump shaft from which the frequency of the surges generated by the pump could be determined, cantilever type Bacharach engine indicator with 100 and 150 pounds springs to record the pressure variations in the test section, bourdon type pressure gauge to record the pressure surges but with little success, bourdon type pressure gauge with air seal between gauge and test section to give the average pressure during test runs. (See Photograph 1).

In order to obtain conclusive results it was a must that all instruments be calibrated and functional before any test runs could be made. The bourdon gauges were periodically checked against a standard. Since there were no equation or calibration charts for the electrokinetic transducer it was necessary to start from the very beginning.

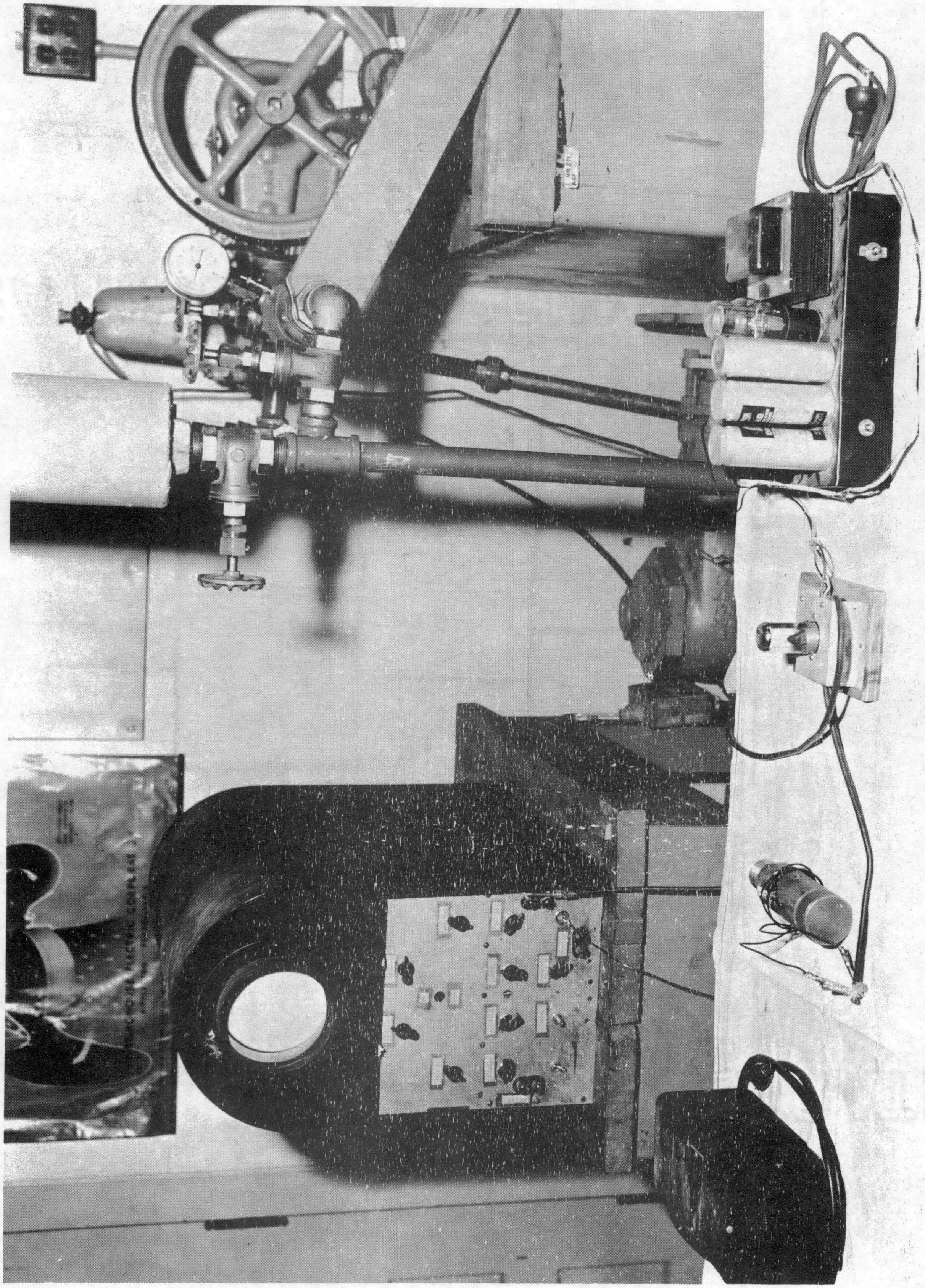


Photo No. 3

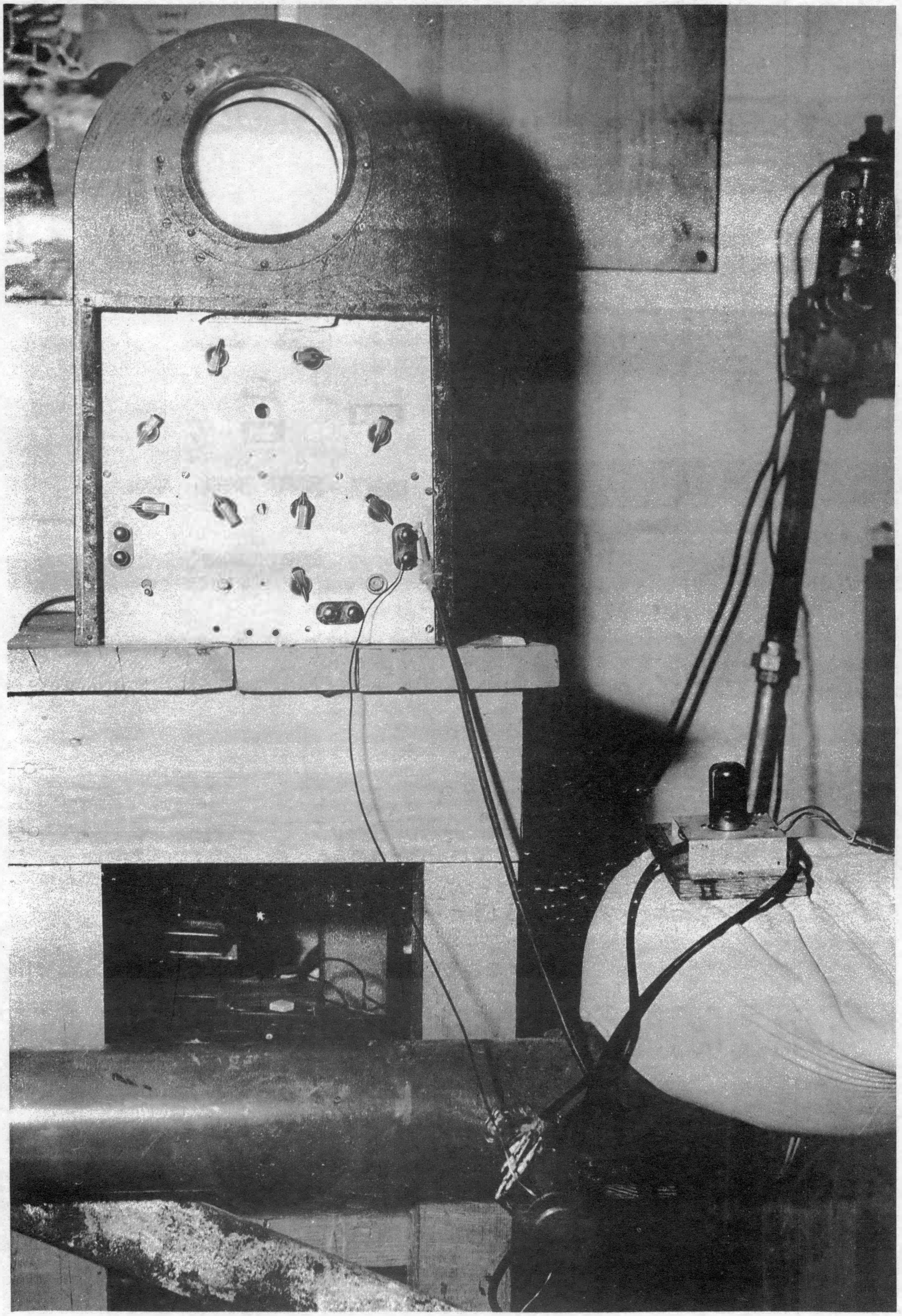


Photo No. 4

CHAPTER VII

CALIBRATION OF ELECTROKINETIC TRANSDUCER

Two methods were used in the calibration of the electrokinetic transducer, (1) the pressure relief method and, (2) repetitive transient wave method. Calibration was first performed by the pressure relief method. The pressure relief calibrator as shown in Figure 9 consists of a volume reserve chamber, bleeder valve, bourdon pressure gauge and an instant valve. Calibration was accomplished in the following manner: the transducer was screwed directly into the volume chamber, after which the chamber was partially filled with water. The system was charged to the desired pressure with compressed air and this pressure was recorded. The pressure was then released by opening the instant valve; the deflection thus created on the oscilloscope was noted. In this manner information on the magnitude of the deflection of the oscilloscope caused by a certain pressure reduction could be studied. A series of such runs were made with values of deflection plotted against the change in pressure as recorded by the bourdon gauge. Calibration by this method did not prove too successful since comparative data could not be obtained. For an example of calibration by this method please consult Figure 10.

The electrokinetic transducer was then calibrated by the repetitive transient wave method. A small war surplus three stage compressor with electric drive arranged as shown on Photograph 5 was used to calibrate by the second method. This setup was so arranged that variation in pressure surges could be produced by manipulation of a control valve. This variation in pressure was of a standing wave form, that is, it could

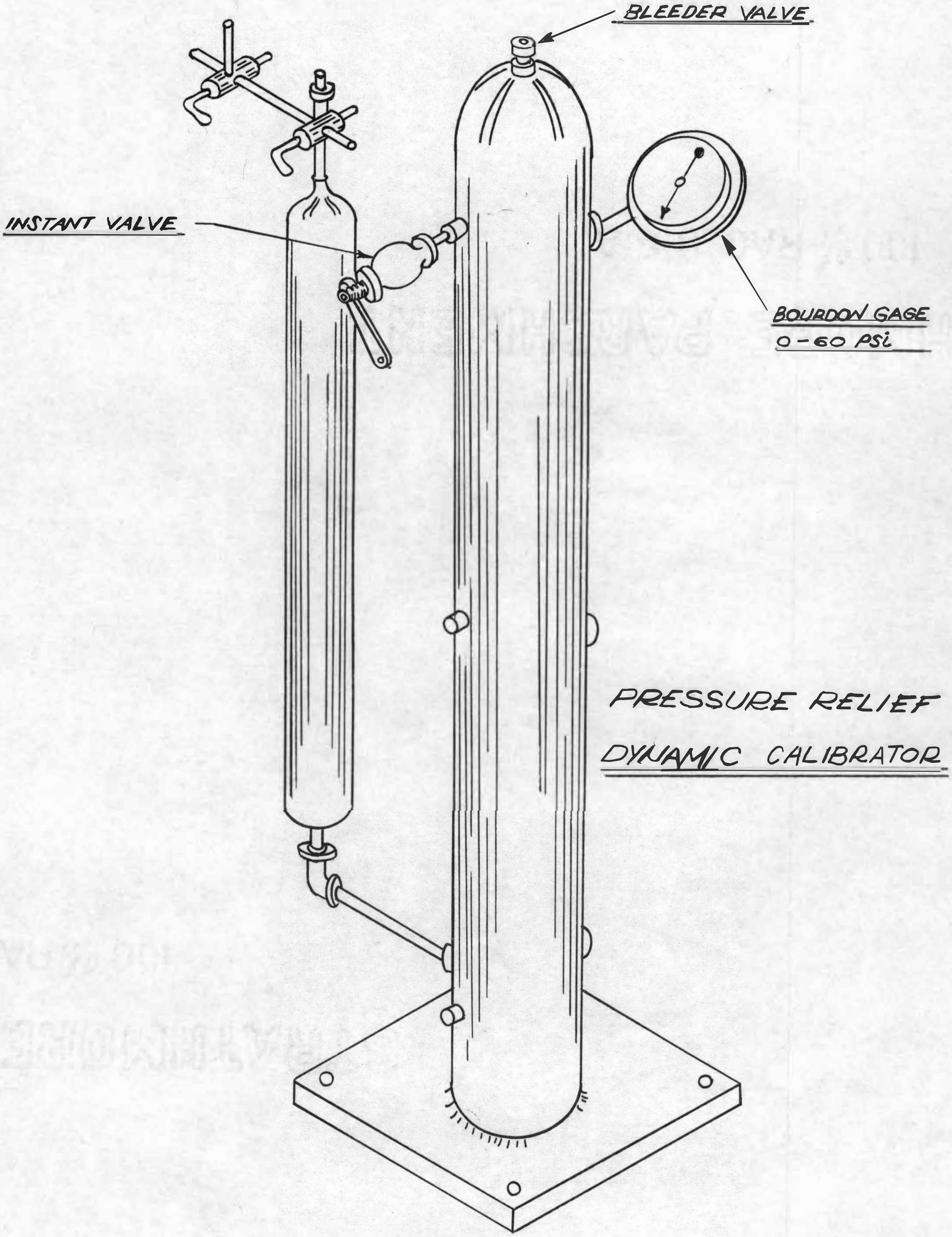

BLEEDER VALVE
INSTANT VALVE
BOURDON GAGE
0 - 60 PSI
PRESSURE RELIEF
DYNAMIC CALIBRATOR

Figure 9

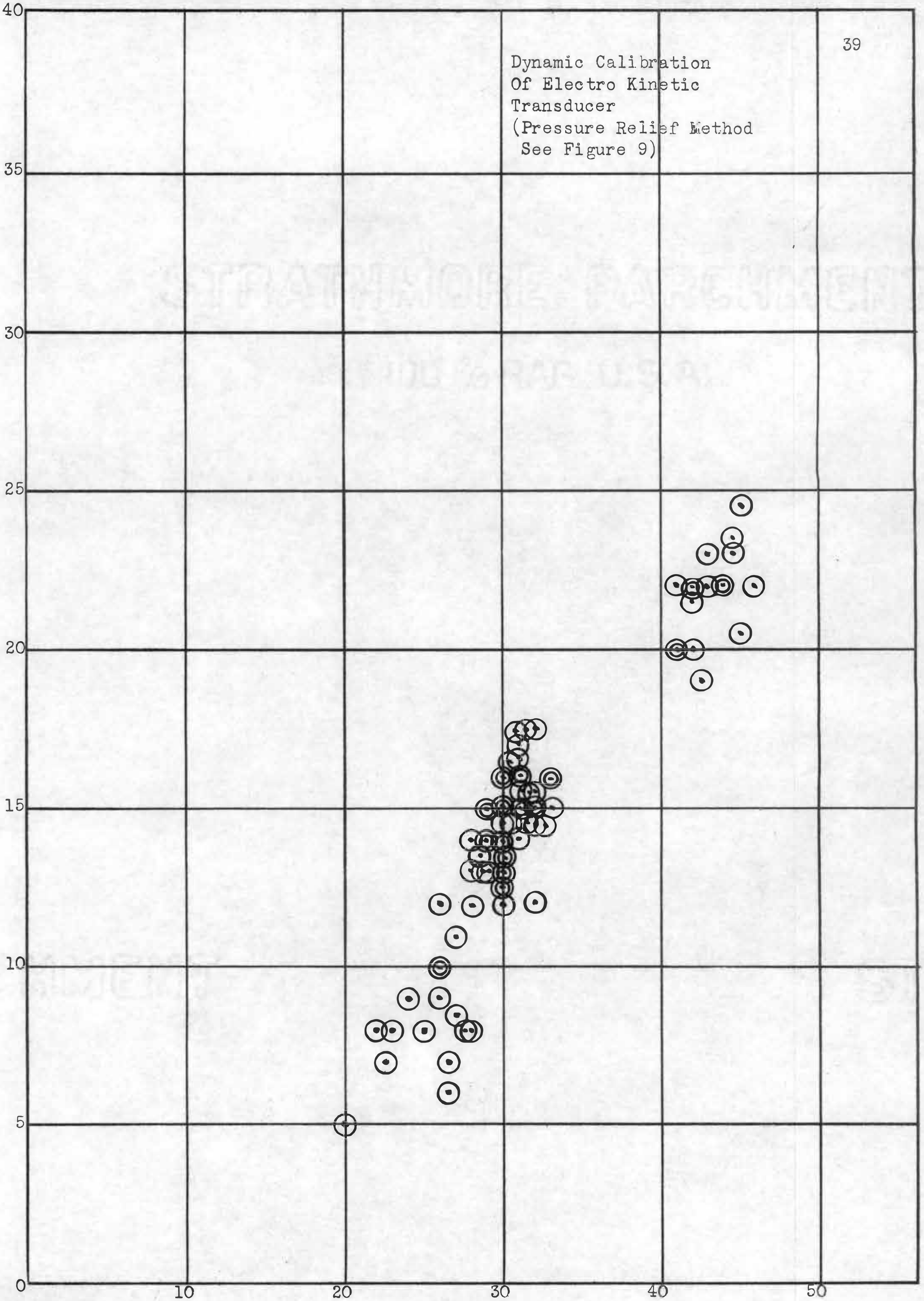


Figure 10 ΔP=Pressure Variation (P.S.I.)

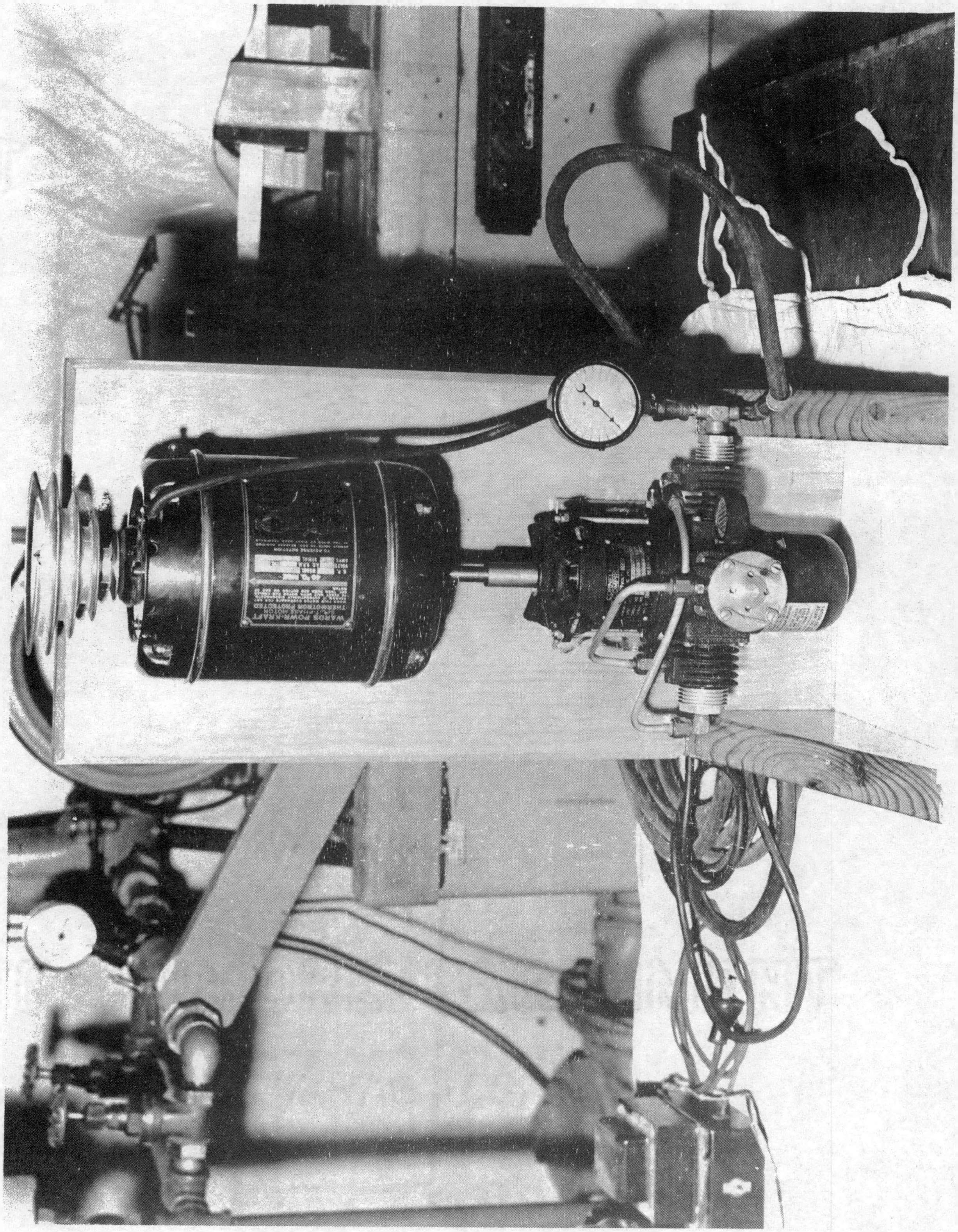


Photo No. 5

be created, picked up on the transducer and if the transducer was synchronized to the speed of the pressure impulses the pressure wave would appear to remain stationary on the oscilloscope. The magnitude of the pressure wave thus formed was read in number of grid deflections on the oscilloscope and recorded versus the variation in pressure in pounds per square inch as read from the bourdon gauge. A series of such runs were made and values of pressure surge vs. grid deflections were plotted on cross-section paper. (See Figure 11). The equation of the line thus formed was found from basic analytic geometry to be

$$\triangle P = .7425\, G - 1.188 \tag{15}$$

where

$\triangle P$ = pressure surge psi

G = no. grids deflection on oscilloscope

Information as obtained from the second method of calibration was used in this experiment not only because it yielded far better results but also because it was felt that the calibration apparatus more nearly duplicated the situation that was present during actual testing of the desurgers. The results from use of the transducer compared favorably with results as obtained from use of the engine indicator.

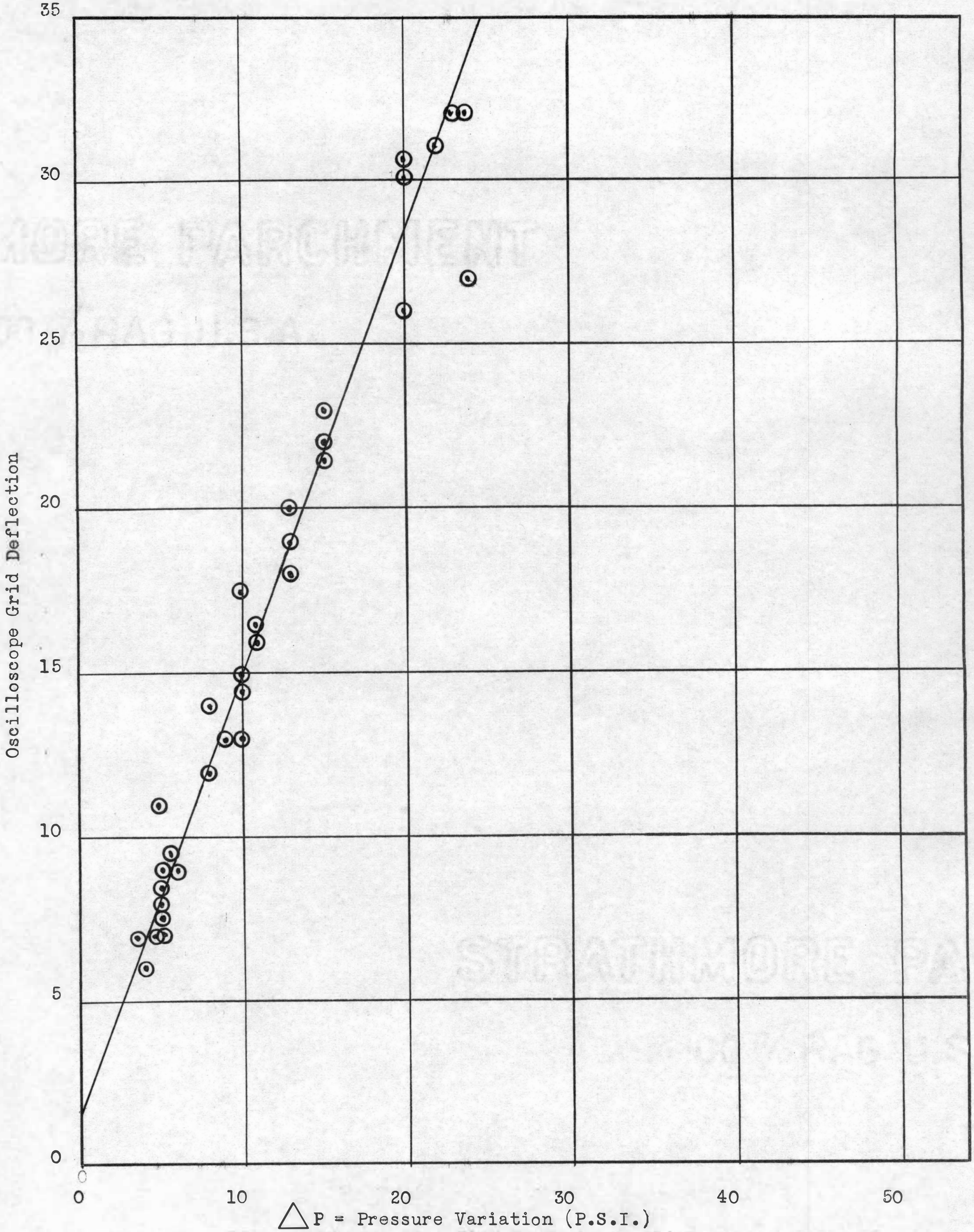


FIGURE 11 Dynamic Calibration of Electrokinetic Transducer (Transient Wave Method).

CHAPTER VIII

DETERMINATION OF THE EFFICIENCY OF COMMERCIAL DESURGING DEVICES NOW ON THE MARKET

Test Procedure

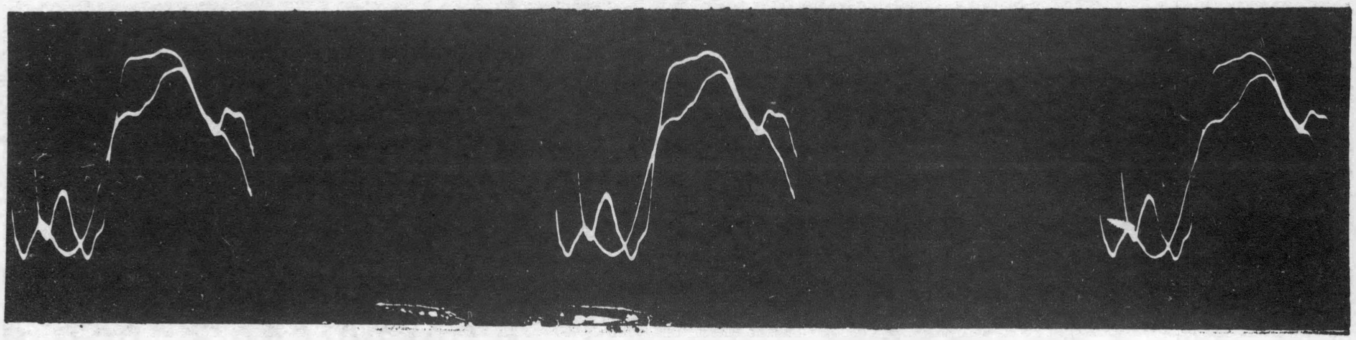
The first test runs were made with all desurging devices eliminated from the system. All discharge valves were opened wide, the pump was then started and brought up to the desired speed. With the pump running at the desired speed the discharge valve was slowly closed until the operating pressure was increased to the test pressure. The RPM of the pump was then checked with the strobotac and counter. The next step was to obtain an engine indicator card of the pressure wave created by the pump without any desurging effect; this was called a "control run". After taking the indicator card the number of grid deflections as read on the oscilloscope were recorded for the same operating conditions. The average operating pressure as indicated by bourdon gauge, with snubber, was recorded. After a complete "control run" was made the next step was to direct flow through or by one of the desurgers or air chambers, and keeping all other conditions exactly the same, the above outline procedure was repeated. Thus by comparison of the two sets of readings the efficiency of the desurging device could be calculated.

Figure 1 gives an example of the data obtained using the engine indicators. For this particular run each inch height of the trace represents a pressure surge of 100 psi. Comparison of the magnitude of the pressure surge of the no control run with the magnitude of the pressure surge as indicated when using one of the desurgers will give the efficiency of removal of that particular surge device.

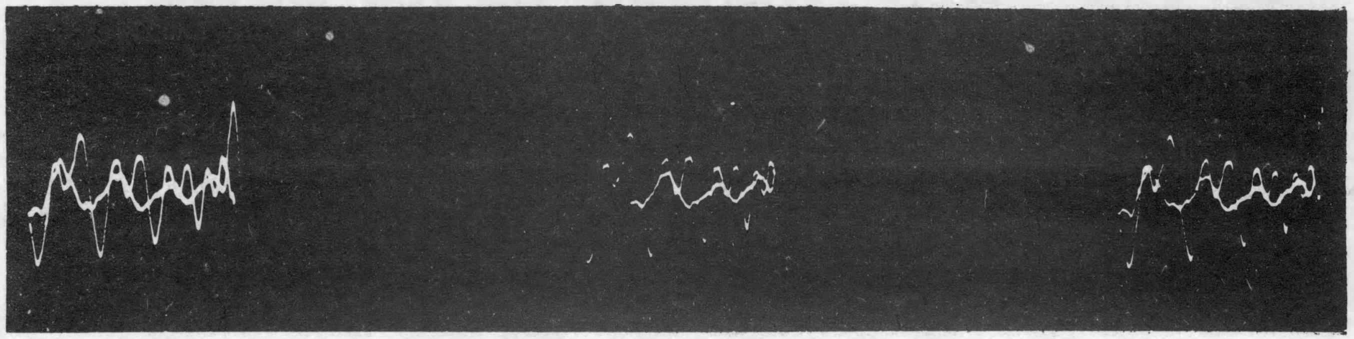
The pressure variation and efficiency of the desurgers was also checked using the oscilloscope. Figure 12 shows photographs taken of pressure impulses as they appeared on the oscilloscope when operating with an average discharge pressure of 90 psi. Figure 12a shows the variation in pressure impulses created by the test pump without a desurger working or a control run as it is called in this report. Note that there are two different traces in each of the three Figures 12a, 12b, and 12c. The trace with higher magnitude of pressure variation is caused by the forward stroke of the piston while the other trace which is slightly smaller is caused by the reverse stroke of the piston. This variation is due to the difference in piston areas which is equal to the cross section of the piston rod. That is the area of the piston during forward stroke minus the area of the piston during reverse stroke is equal to cross section of the piston rod. These photographs were taken in the following manner. The system was started and the desired operating conditions were obtainéd, the oscilloscope was synchronized to the speed of the pressure wave impulses which caused the impulses to appear to be stationary. It was then a fairly simple operation to photograph the resulting image. By using calibration chart Figure 11 efficiency of unit can easily be determined.

By increasing the speed of the pump the magnitude of the pressure surges can be increased; by adjustment of the discharge valve a constant discharge pressure can be maintained; by recording and comparing surge pressures and RPM the efficiency of the desurgers relative to frequency and magnitude of pressure surges can be determined.

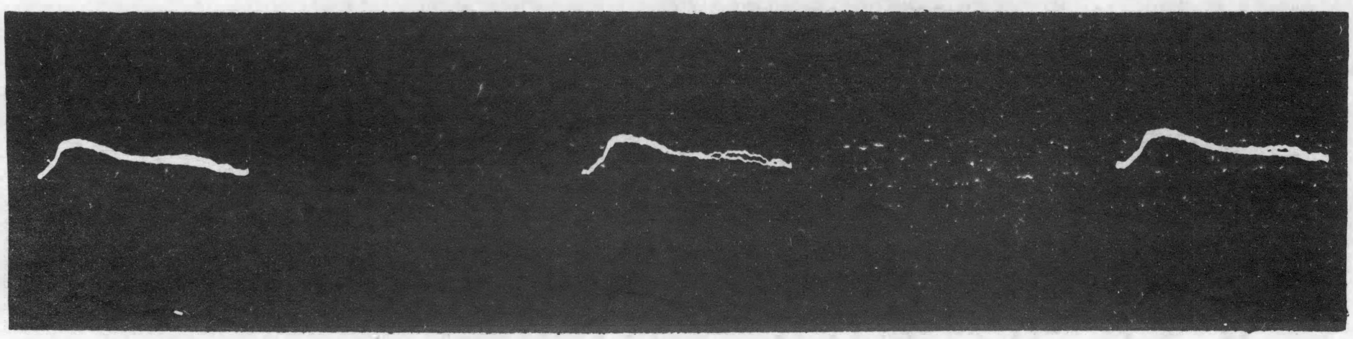
This procedure was used first with the 1-F desurger as an appendage



12a. Pipe Line No Control.



12b. Pipe Line With Air Dome.



12c. Pipe Line With Surge Suppressor (1-F).

FIGURE 12. Ocilloscope Record of Surges at 90 psi Operating Pressure.

at an operating pressure of 70 psi, then repeated at an operating pressure of 90 psi discharge.

Following this the 1-F desurger was switched to a through flow device under the same two conditions with the same information recorded. In this manner the efficiency of the unit could be determined as well as the comparison of the efficiency of the 1-F unit under two different types of installation.

Efficiency tests were then run on the 1-W desurger and the air chamber making sure that the data was taken under exactly the same conditions as to speed, temperature, pressure surge and discharge pressure as were present when working with the 1-F desurger. Only in this manner could reliable comparison between the three desurgers be made.

The same procedure was used for each desurging device, first with the discharge pressure held at 70 psi and with the frequency varying from five cycles to nineteen cycles per second with the discharge pressure held at 90 psi with the frequency varying from five cycles to nineteen cycles per second. The data which was recorded in this manner is presented in graph form, see Figures 13 through Figures 32.

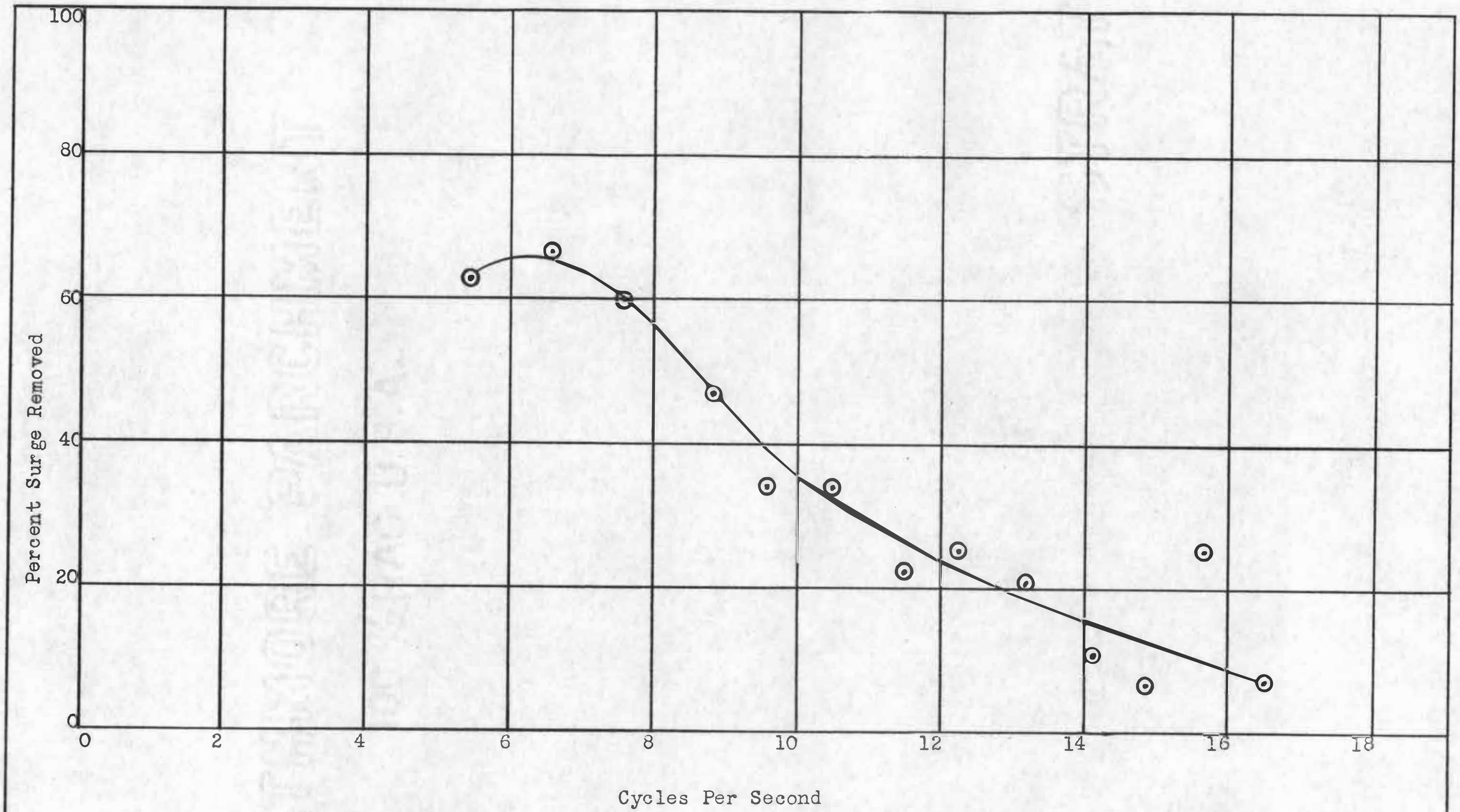


Figure 13 Percent Surge Removed vs Cycles Per Second
With 70 psi. Average Flow Pressure
Using Air Dome

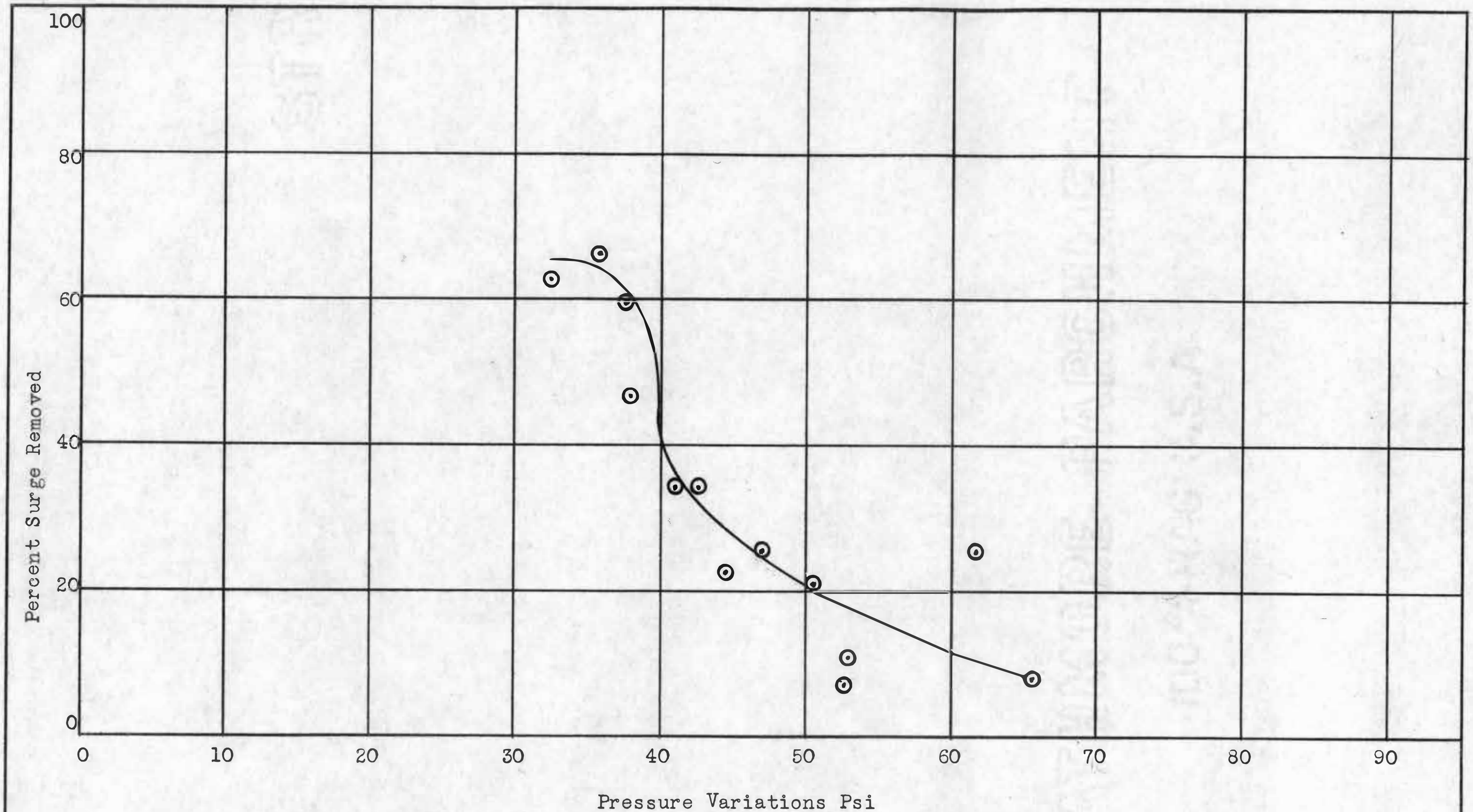


Figure 14 Percent Surge Removed vs Pressure Variations
With 70 psi. Average Flow Pressure
Using Air Dome

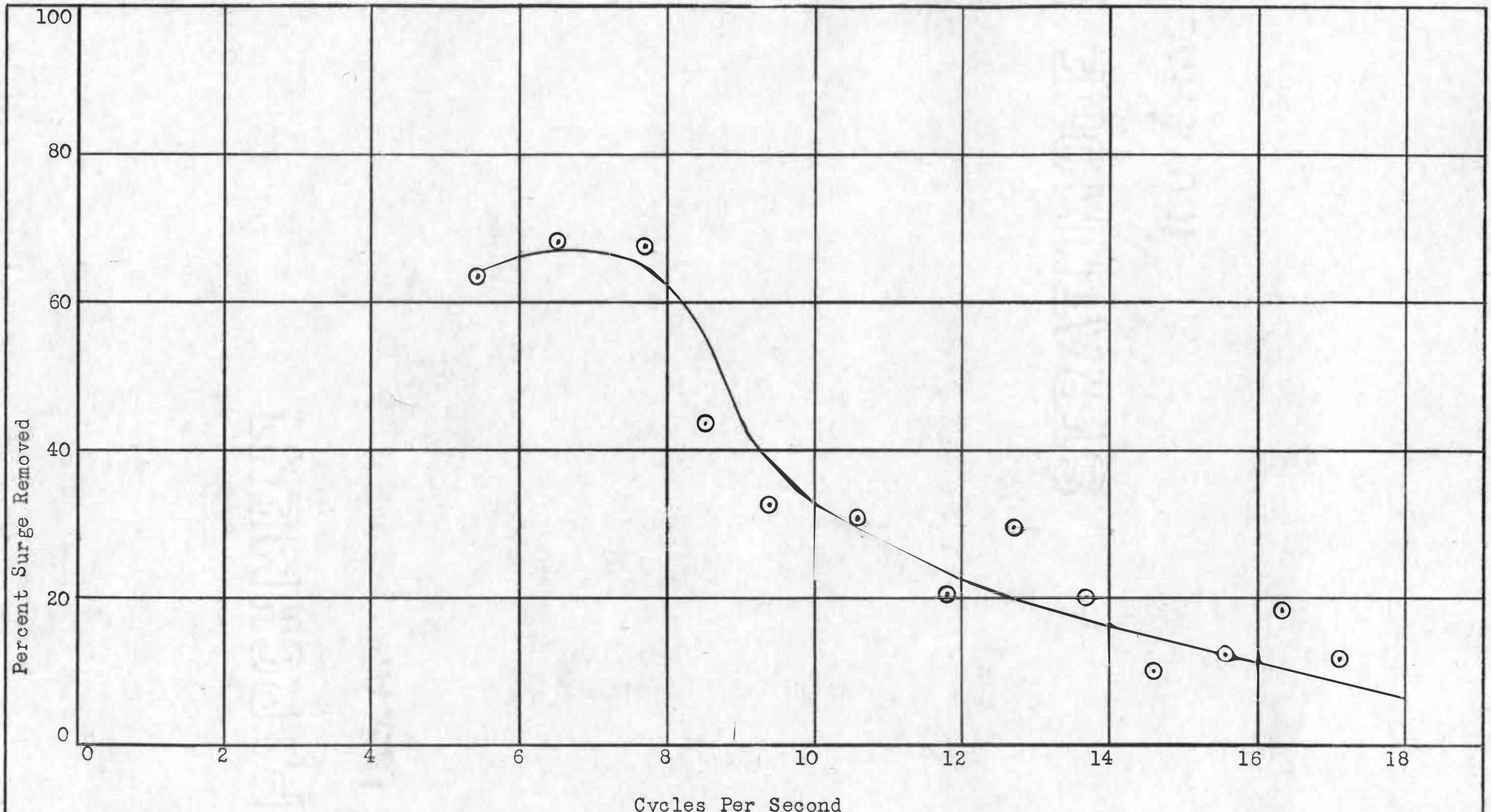


Figure 15 Percent Surge Removed vs Cycles Per Second
With 90 p.s.i. Average Flow Pressure
Using Air Dome

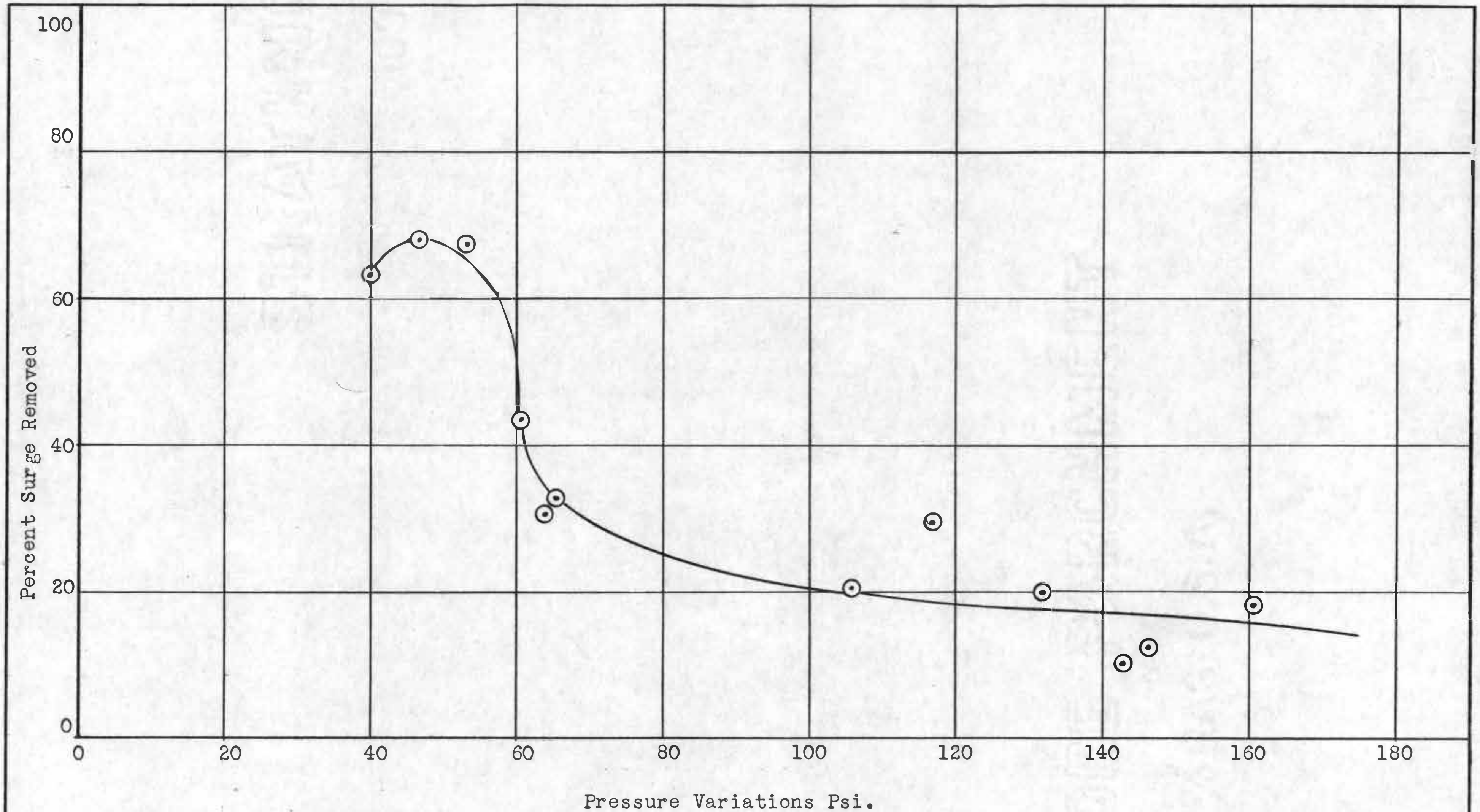


Figure 16 Percent Surge Removed vs Pressure Variations
With 90 p.s.i. Average Flow Pressure
Using Air Dome

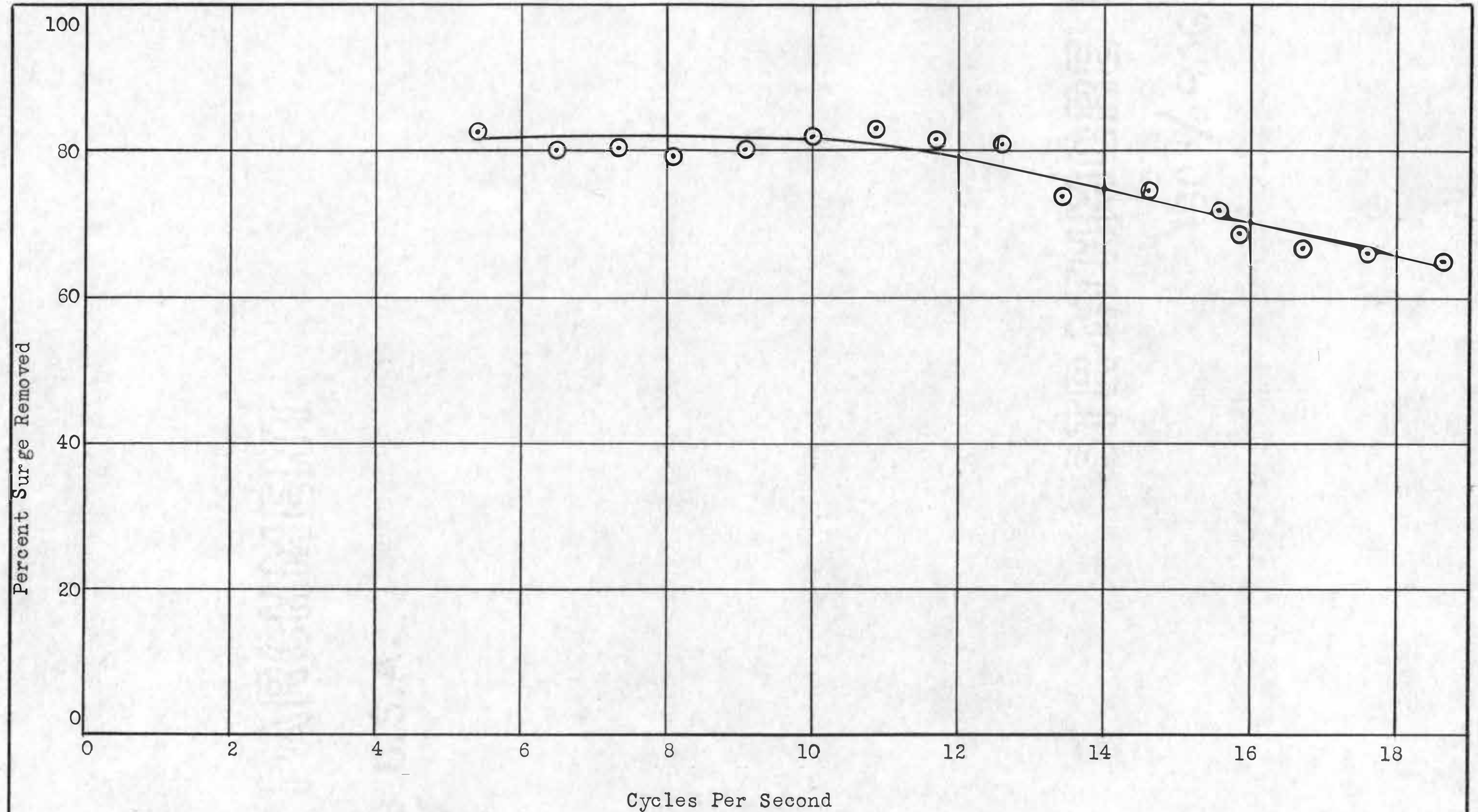


Figure 17 Percent Surge Removed vs Cycles Per Second
With 70 p.s.i. Average Flow Pressure
Using 1-W Surge Suppressor as Appendage

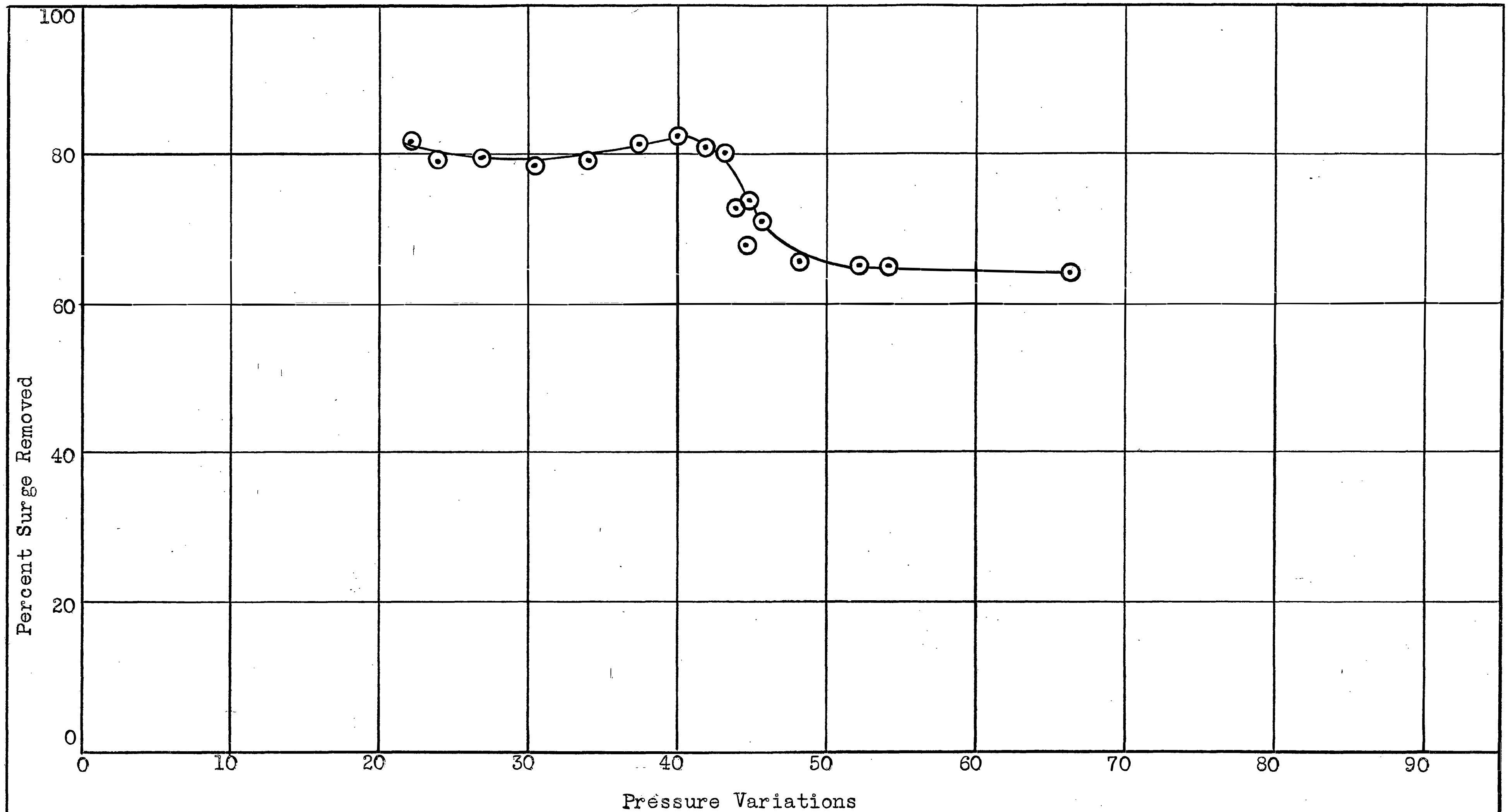


Figure 18 Percent Surge Removed vs Pressure Variations
With 70 psi. Average Flow Pressure
Using 1-W Surge Suppressor as Appendage

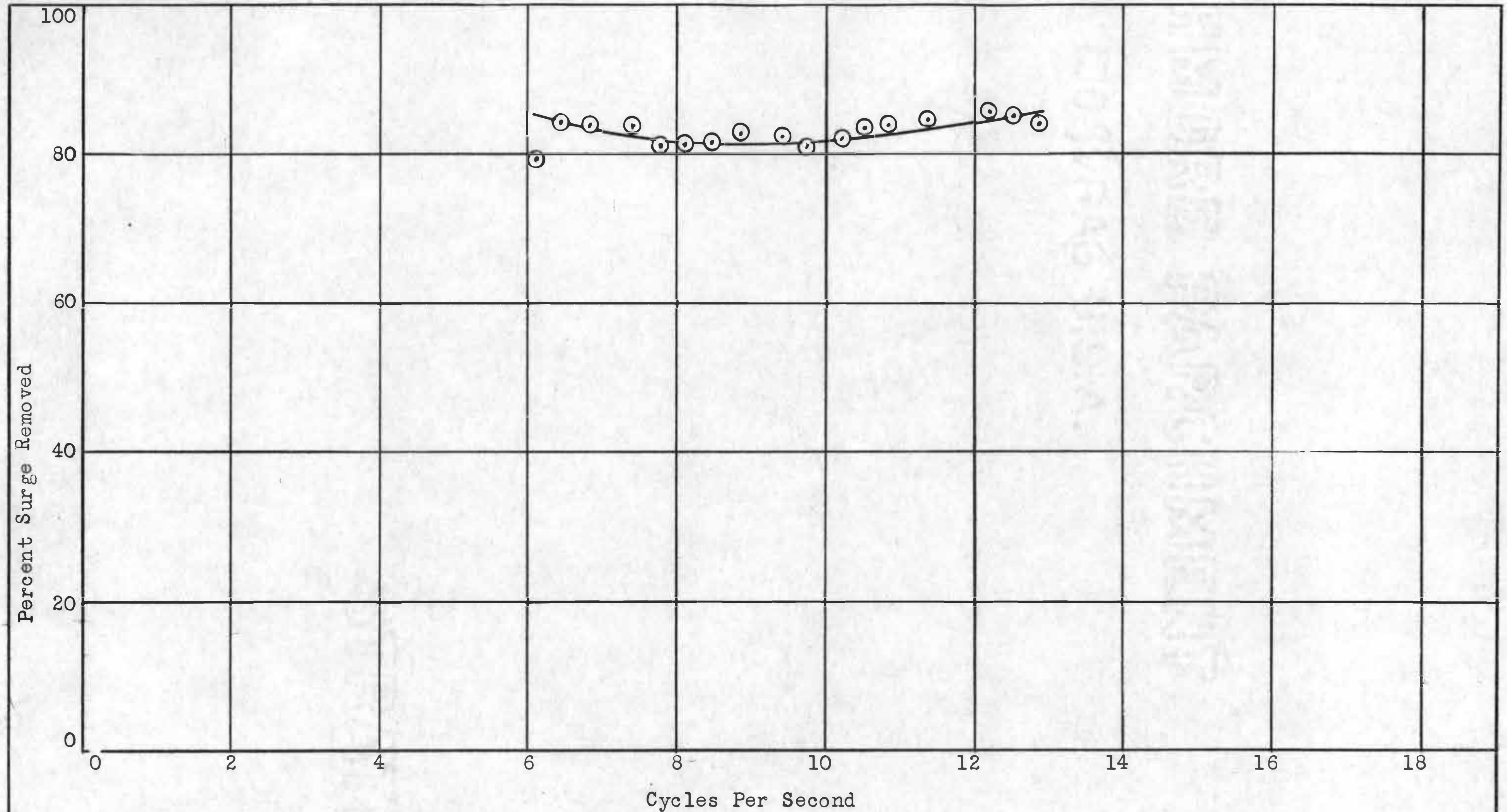


Figure 19 Percent Surge Removed vs Cycles Per Second
With 90 p.s.i. Average Flow Pressure
Using 1-W Surge Suppressor as Appendage

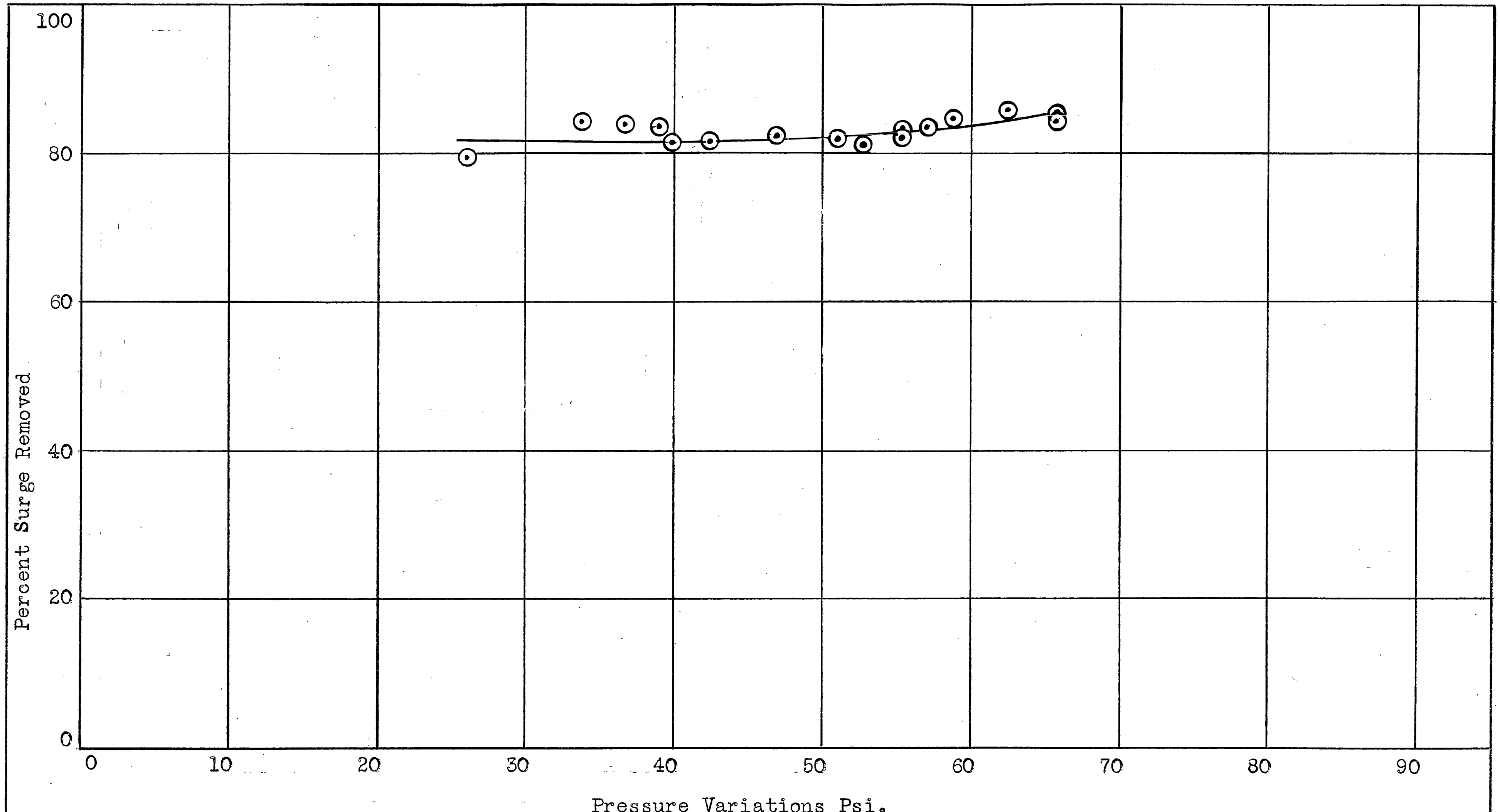


Figure 20 Percent Surge Removed Vs Pressure Variations
With 90 p.s.i. Average Flow Pressure
Using 1-W Surge Suppressor as Appendage

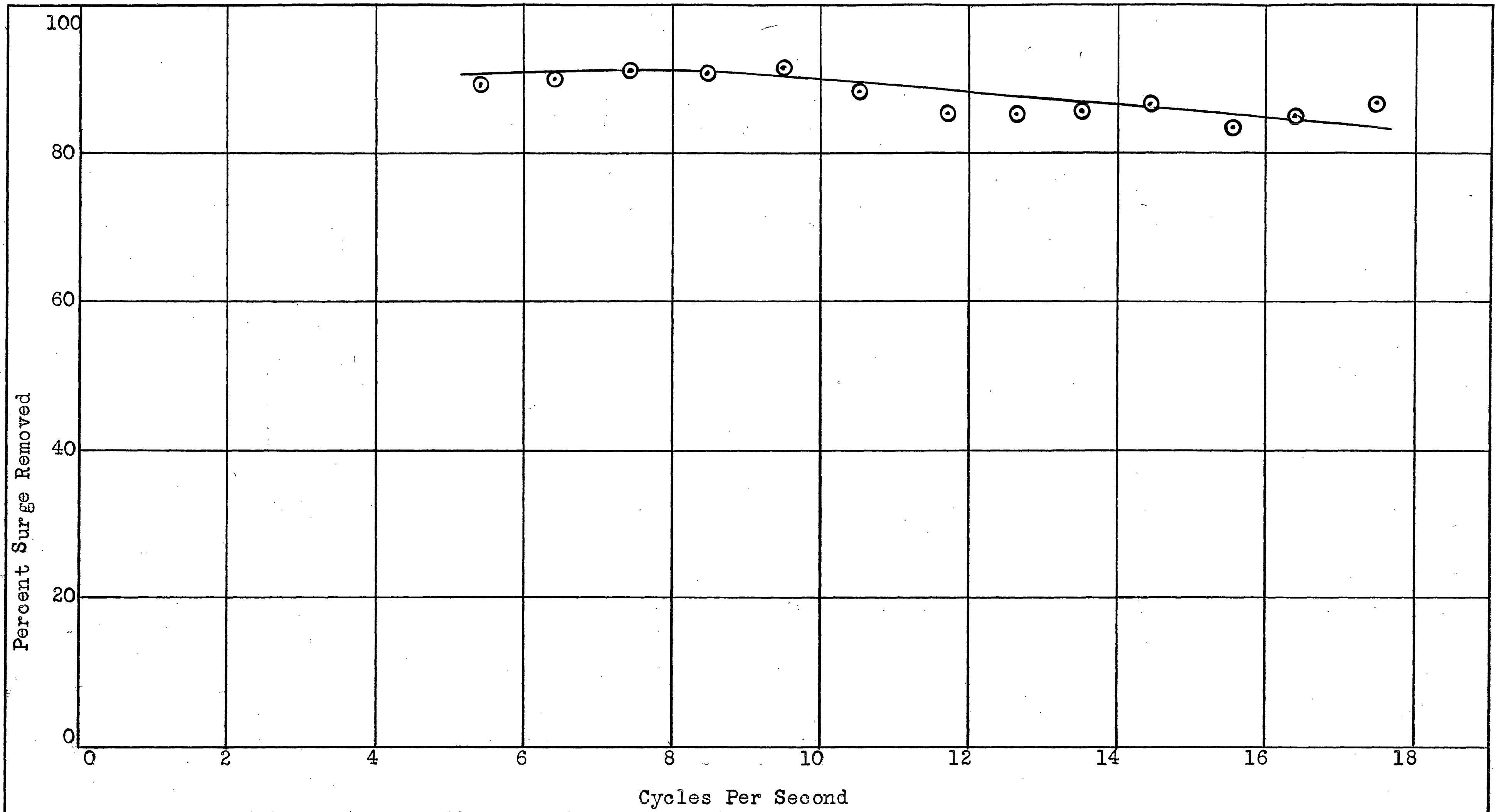


Figure 21 Percent Surge Removed Vs Cycles Per Second
With 70 p.s.i. Average Flow Pressure
Using 1-F Surge Suppressor as Appendage.

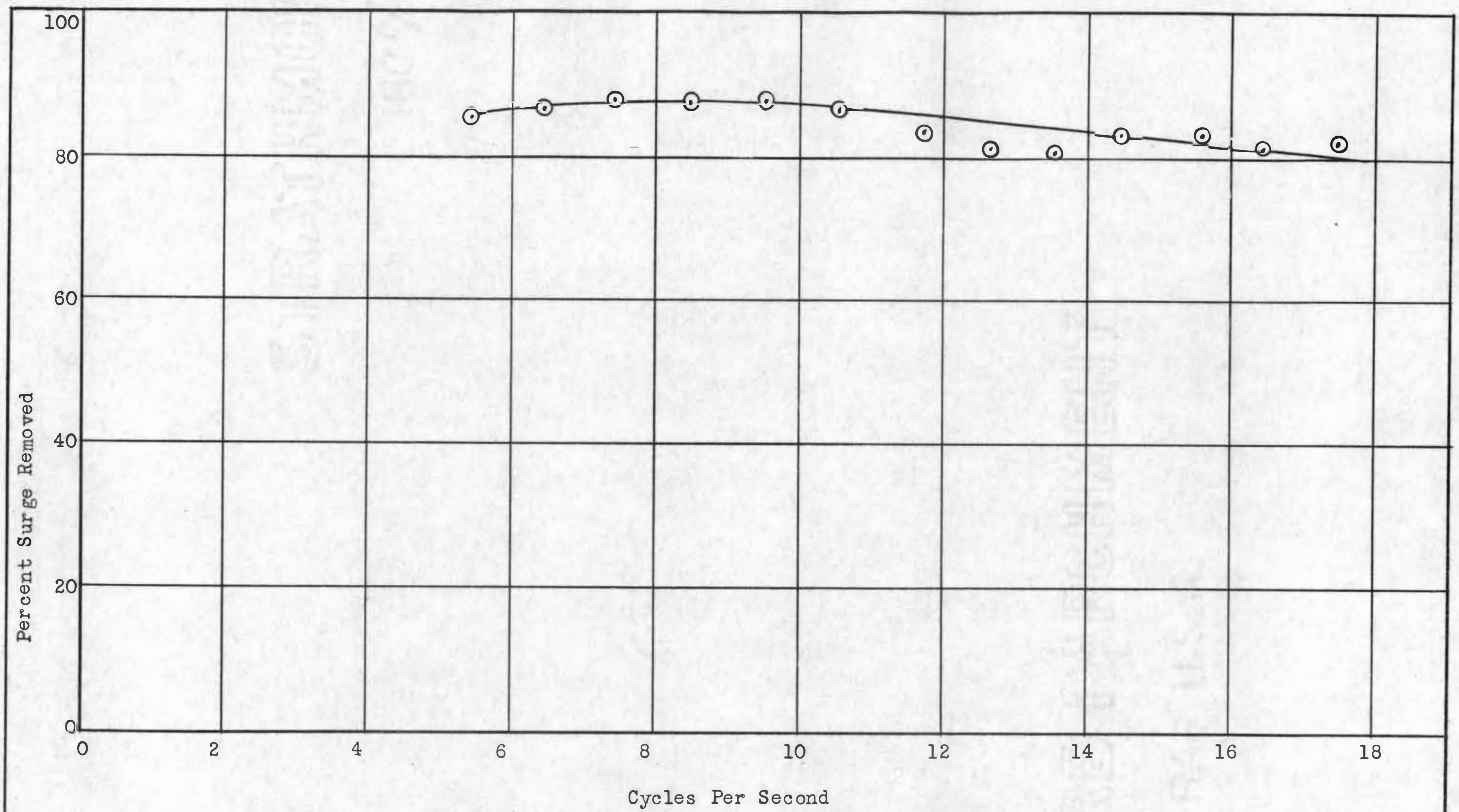


Figure 22 Percent Surge Removed vs Cycles Per Second
With 70 p.s.i. Average Flow Pressure
Using 1-F Surge Suppressor with Thru Flow

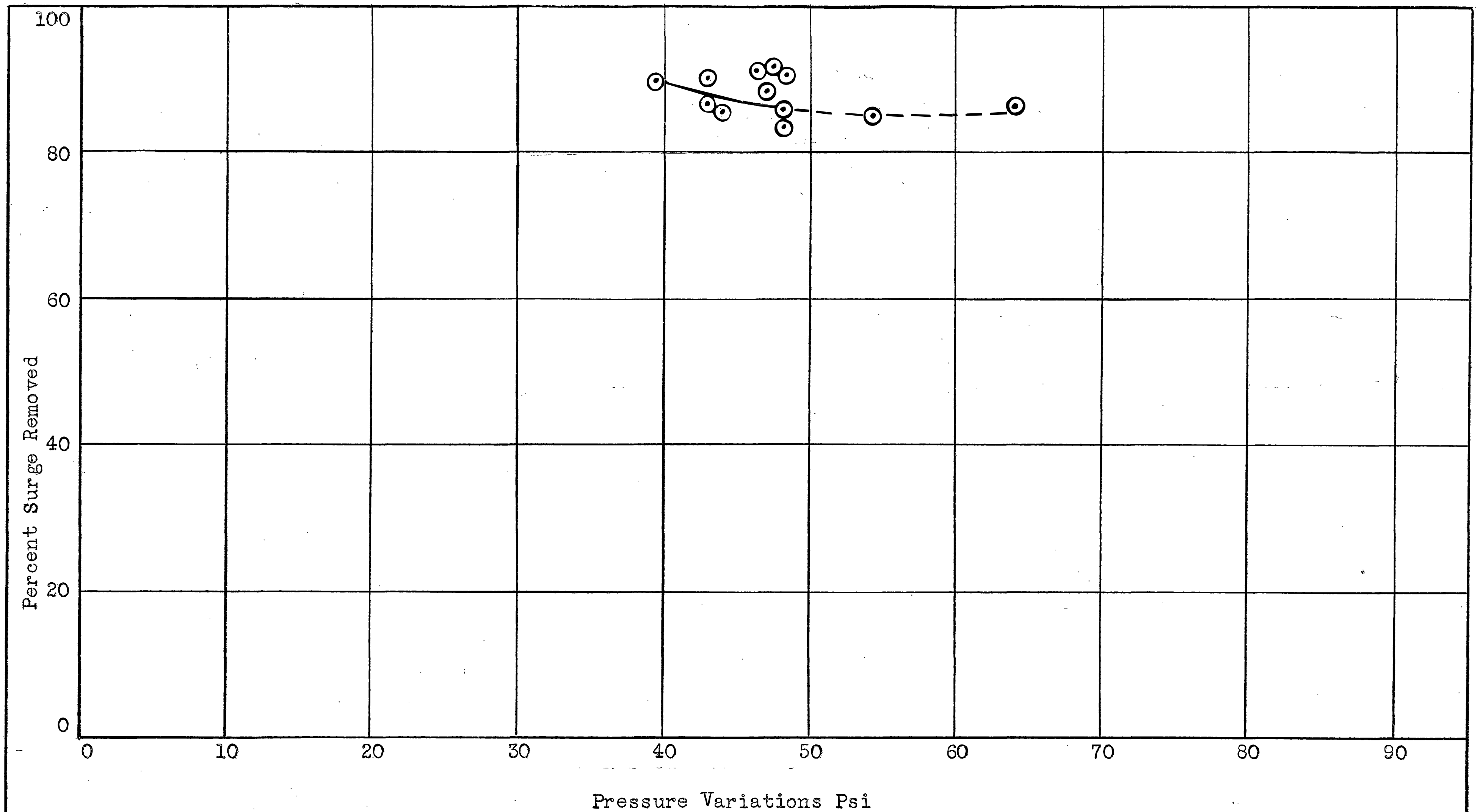


Figure 23 Percent Surge Removed vs Pressure Variations
With 70 p.s.i.Average Flow Pressure
Using 1-F Surge Suppressor as Appendage

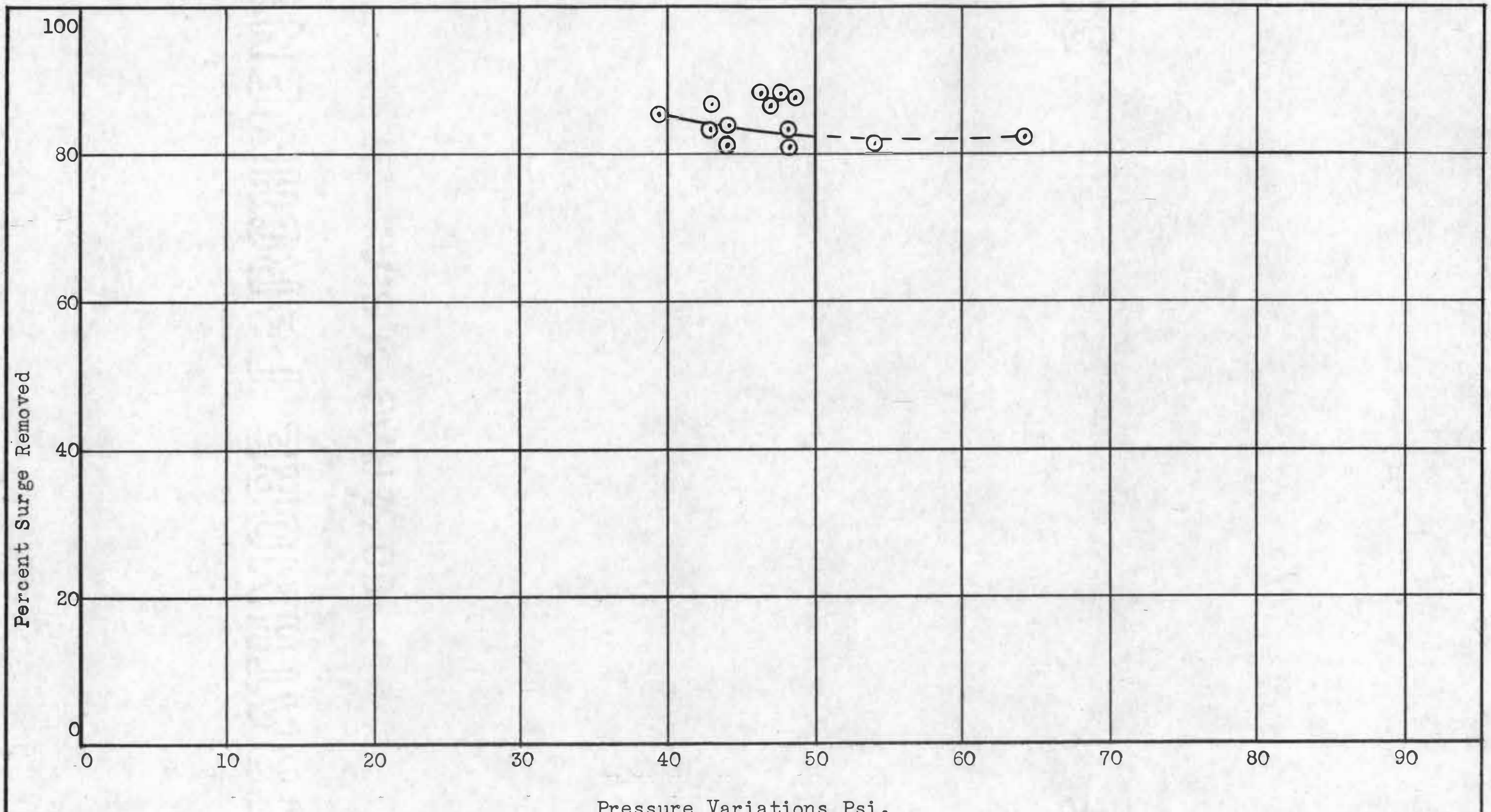


Figure 24 Percent Surge Removed vs Pressure Variations
With 70 psi. Average Flow Pressure
Using 1-F Surge Suppressor With Thru Flow

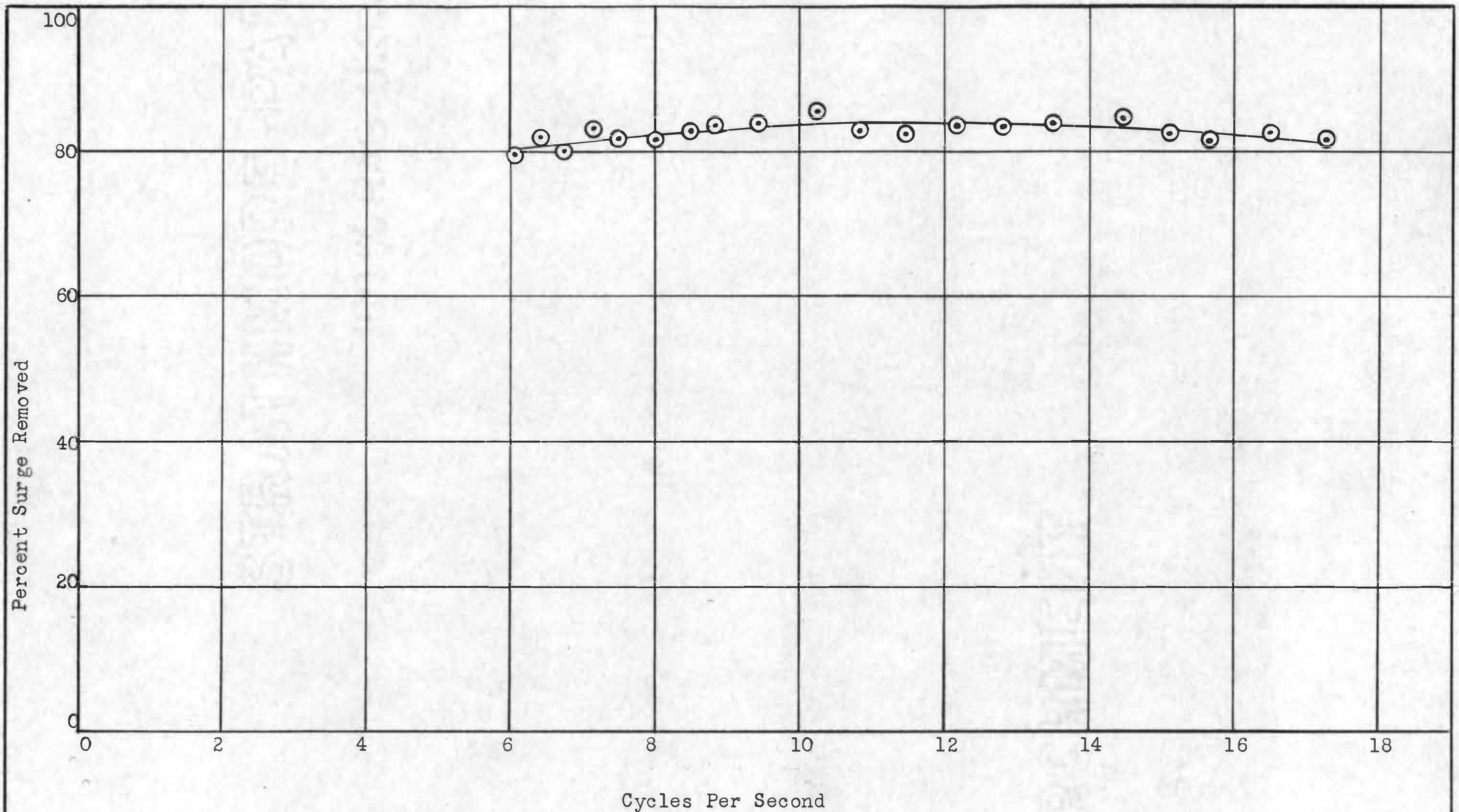


Figure 25 Percent Surge Removed vs Cycles Per Second
With 90 psi. Average Flow Pressure
Using 1-F Surge Suppressor as Appendage

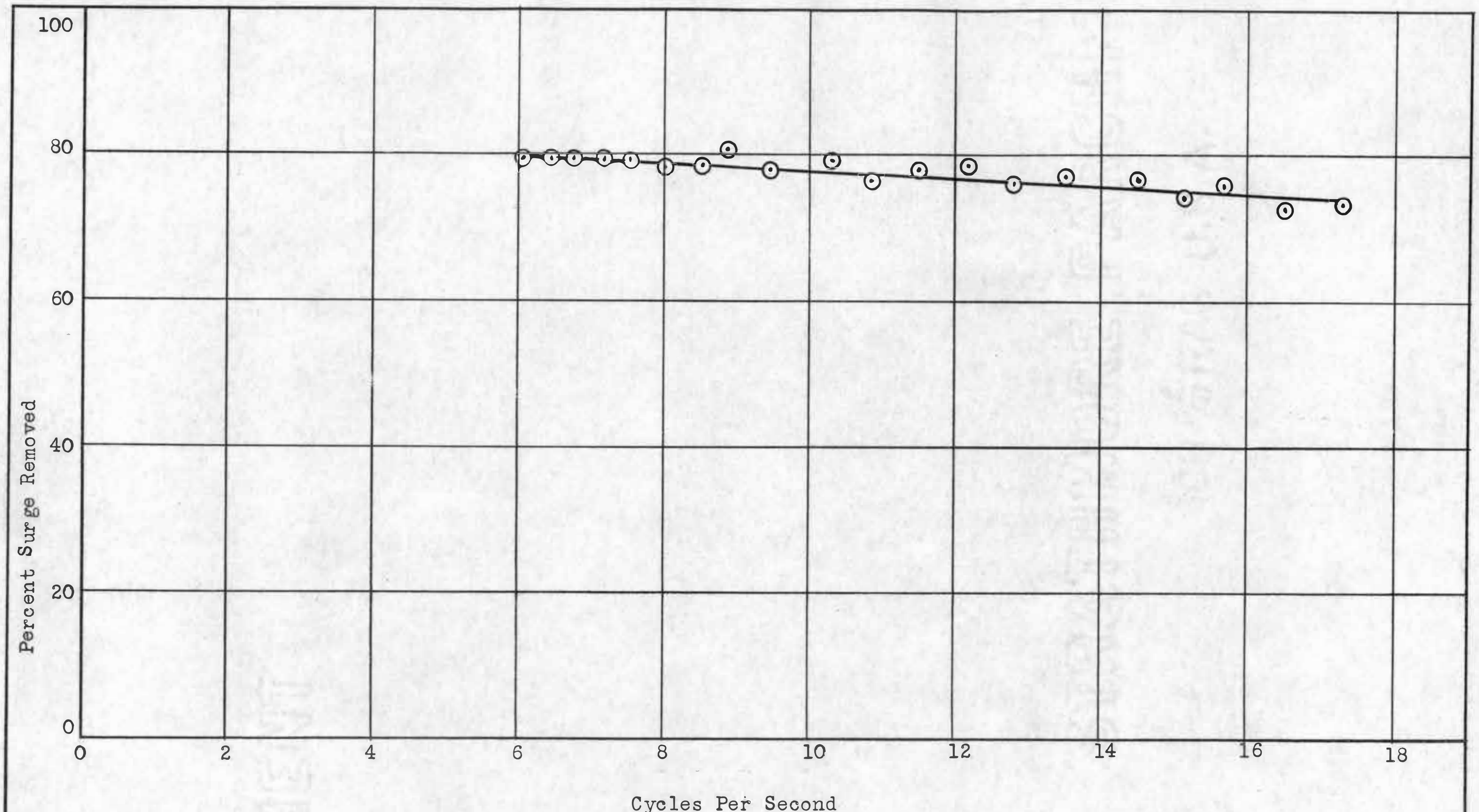


Figure 26 Percent Surge Removed vs Cycles Per Second
With 90 p.s.i. Average Flow Pressure
Using 1-F Surge Suppressor with Thru Flow

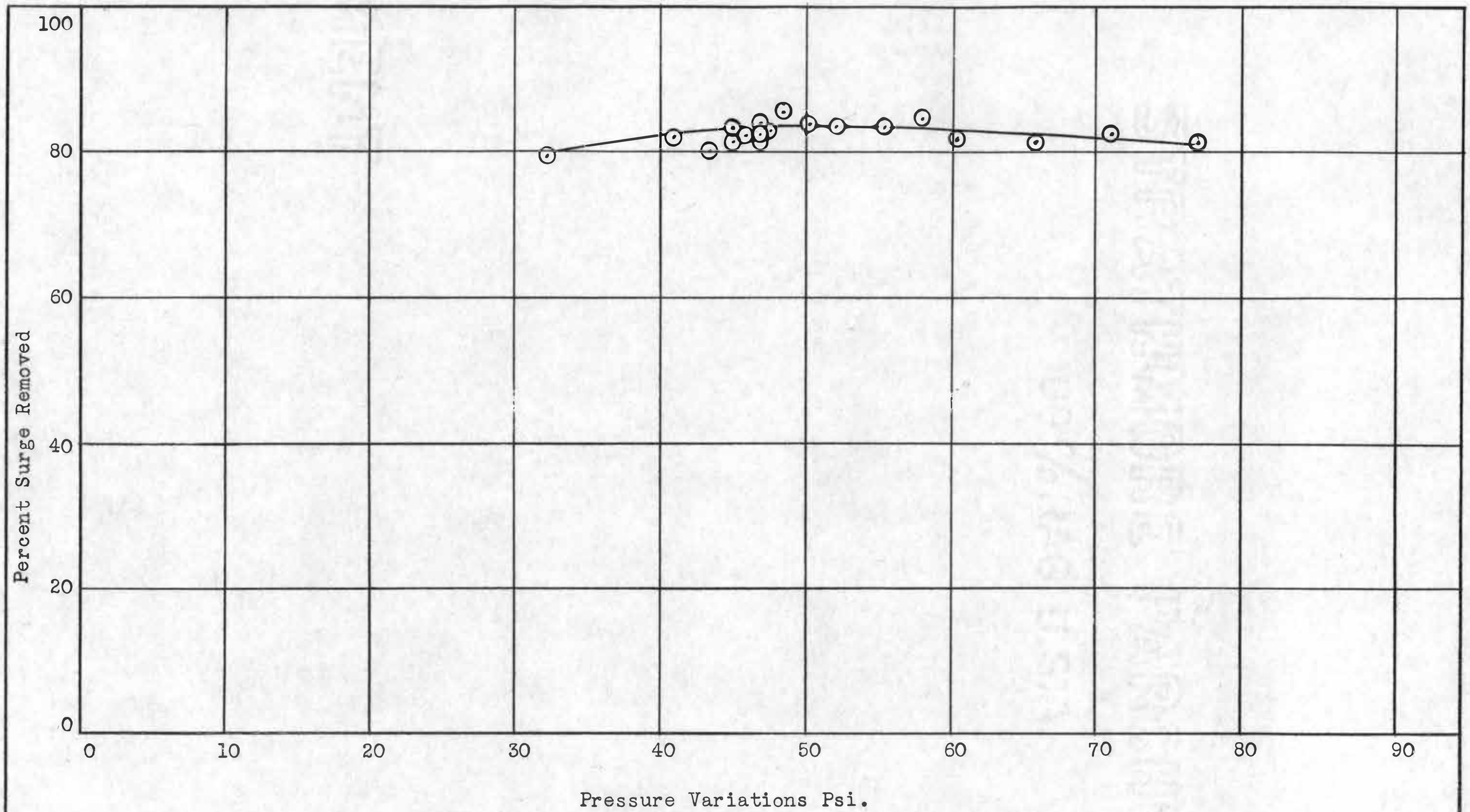


Figure 27 Percent Surge Removed vs Pressure Variations
With 90 p.s.i. Average Flow Pressure
Using 1-F Surge Suppressor as Appendage

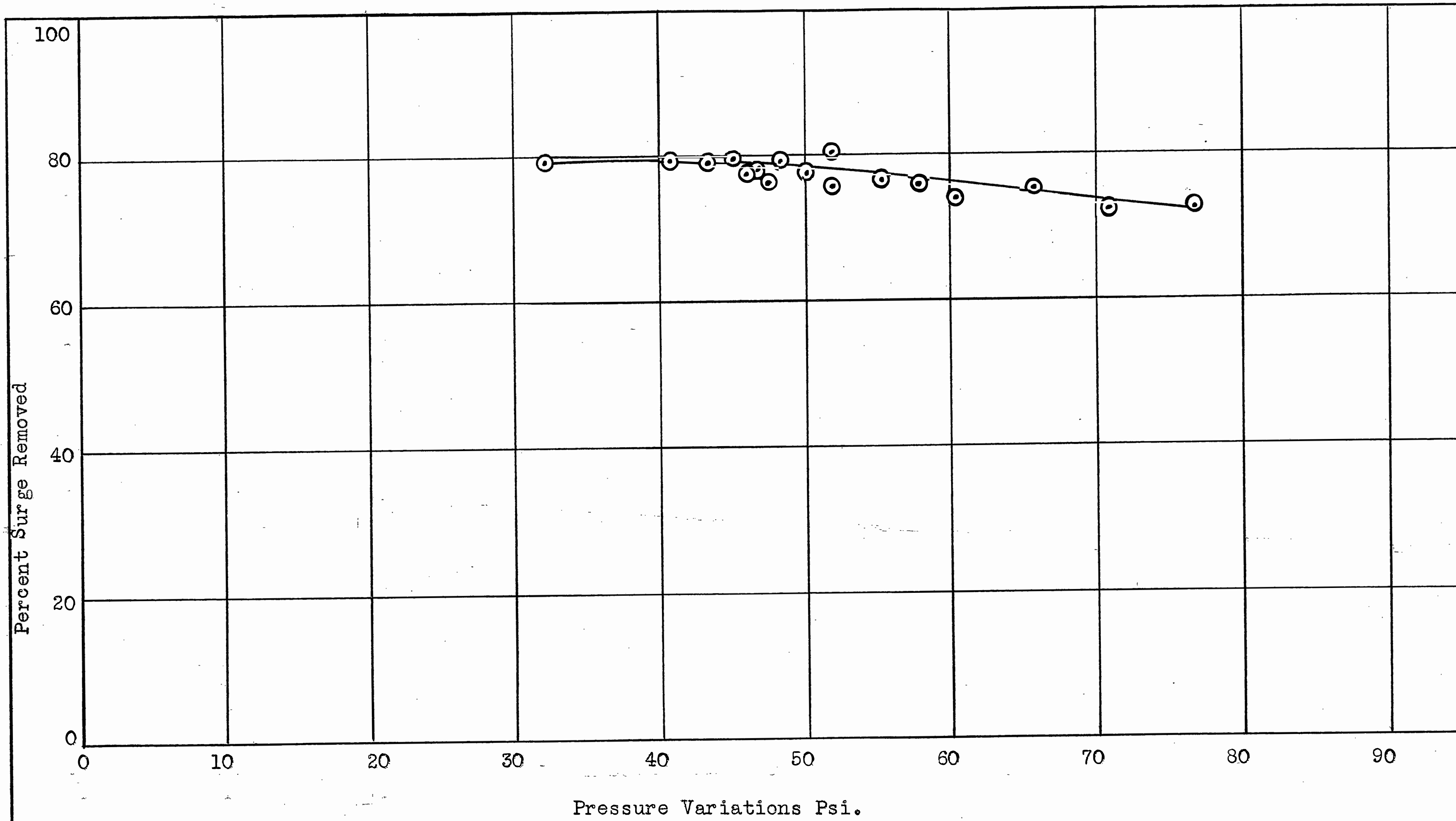


Figure 28 Percent Surge Removed vs Pressure Variations
With 90 p.s.i. Average Flow Pressure
Using 1-F Surge Suppressor with Thru Flow

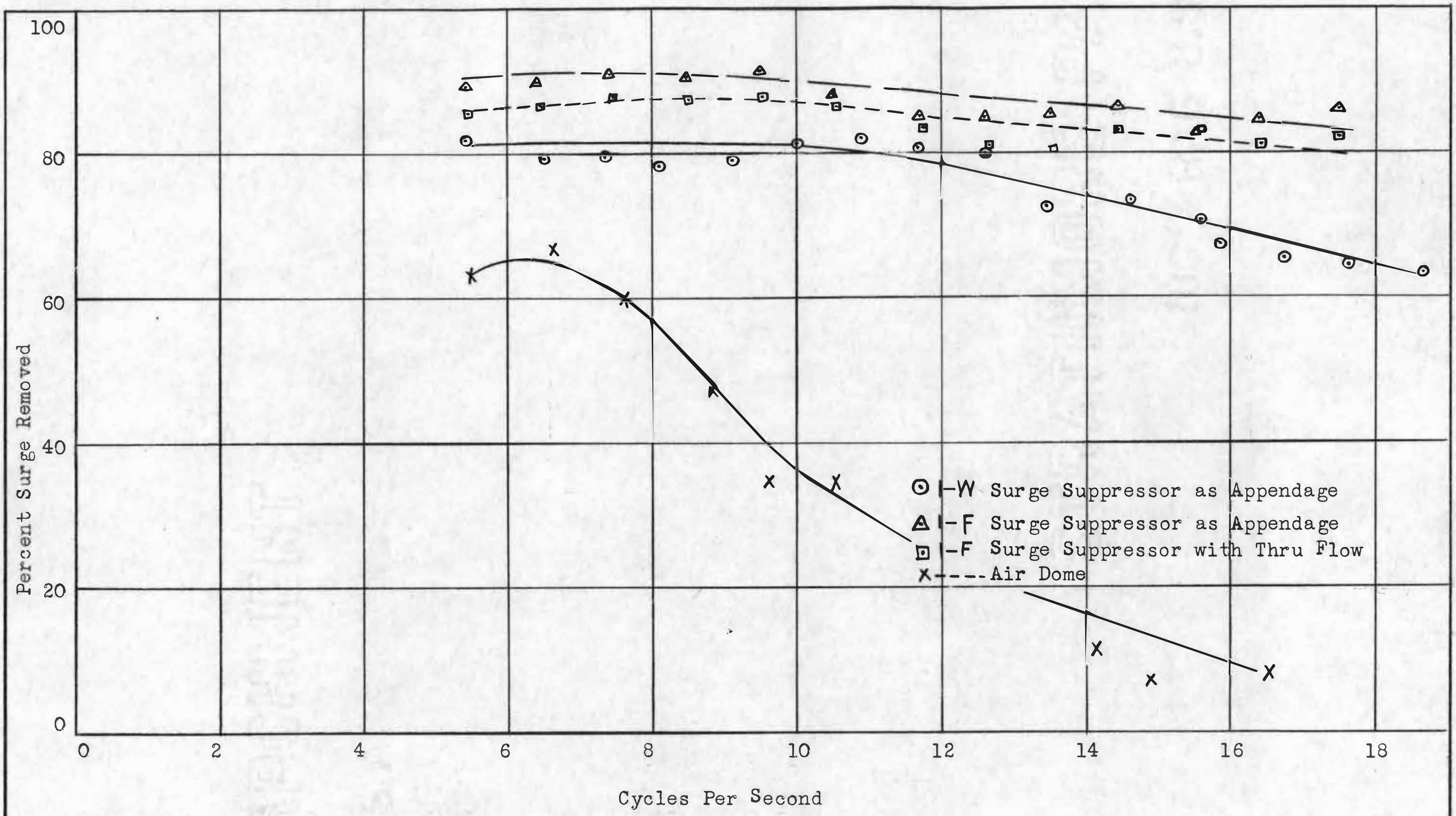


Figure 29 Comparison of Percent Surge Removed vs Cycles Per Second With 70 p.s.i. Average Flow Pressure

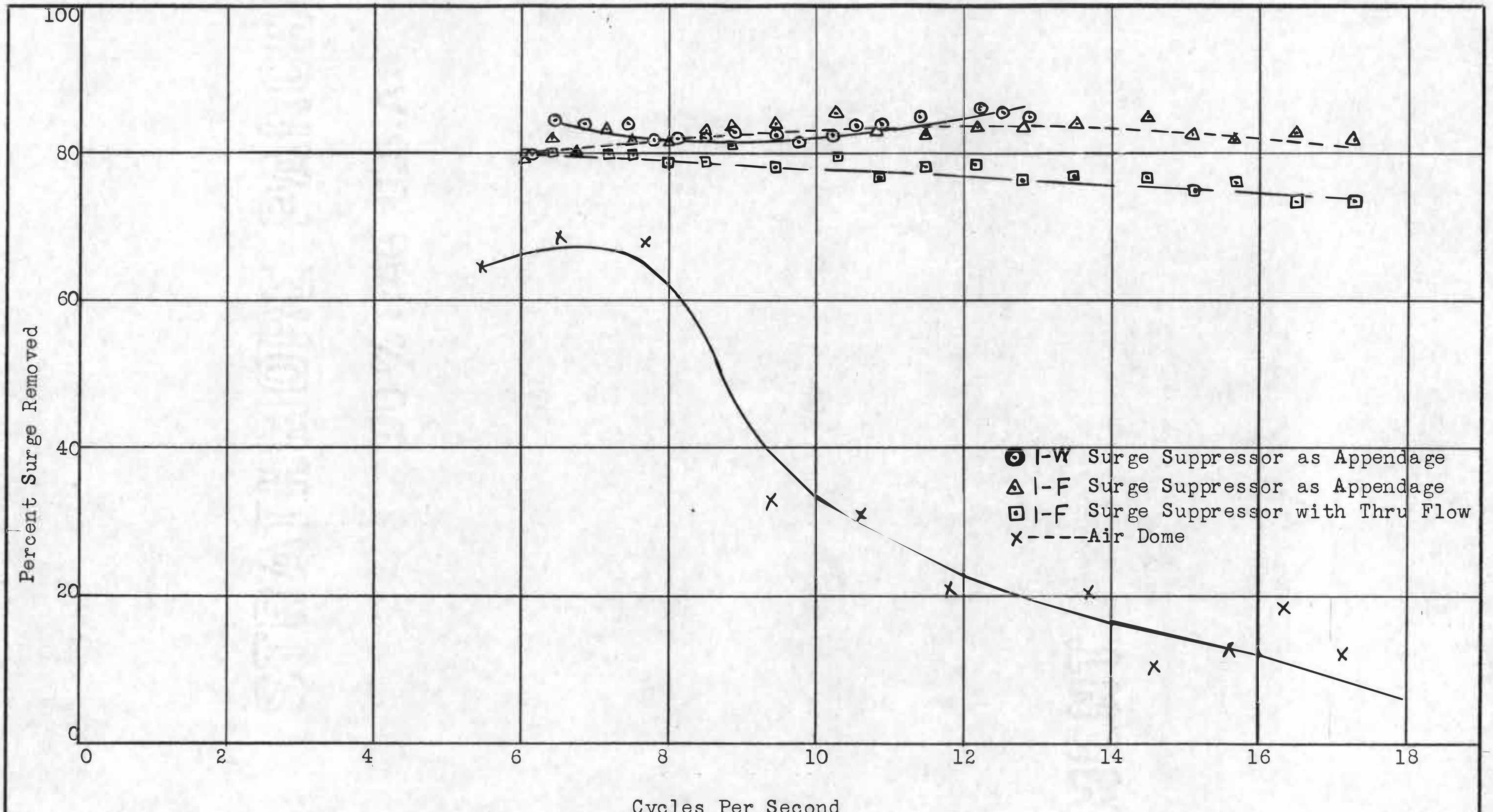


Figure 30 Comparison of Percent Surge Removed vs Cycles Per Second With 90 p.s.i. Average Flow Pressure

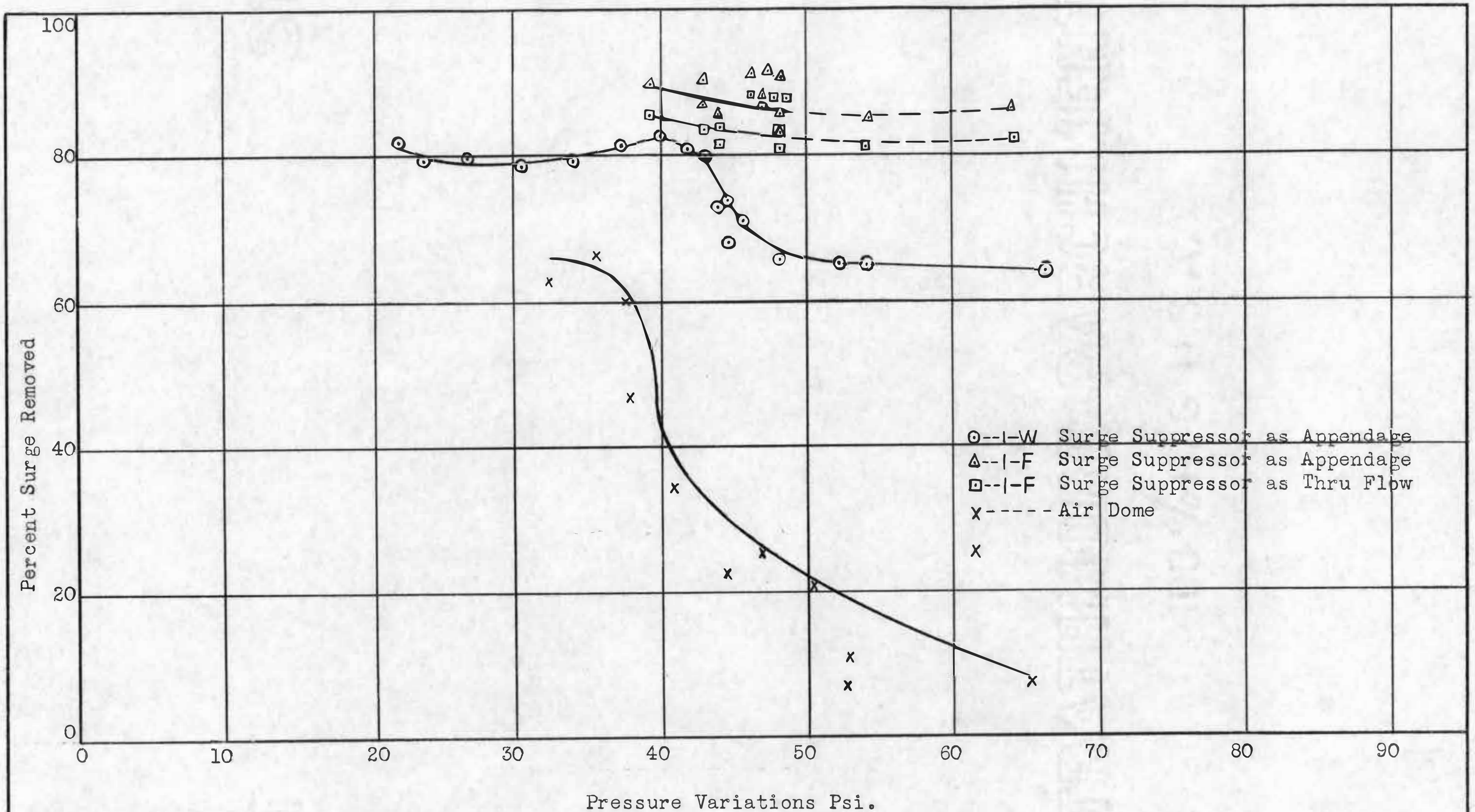


Figure 31 Comparison of Percent Surge Removed vs Pressure Variations With 70 p.s.i. Average Flow Pressure

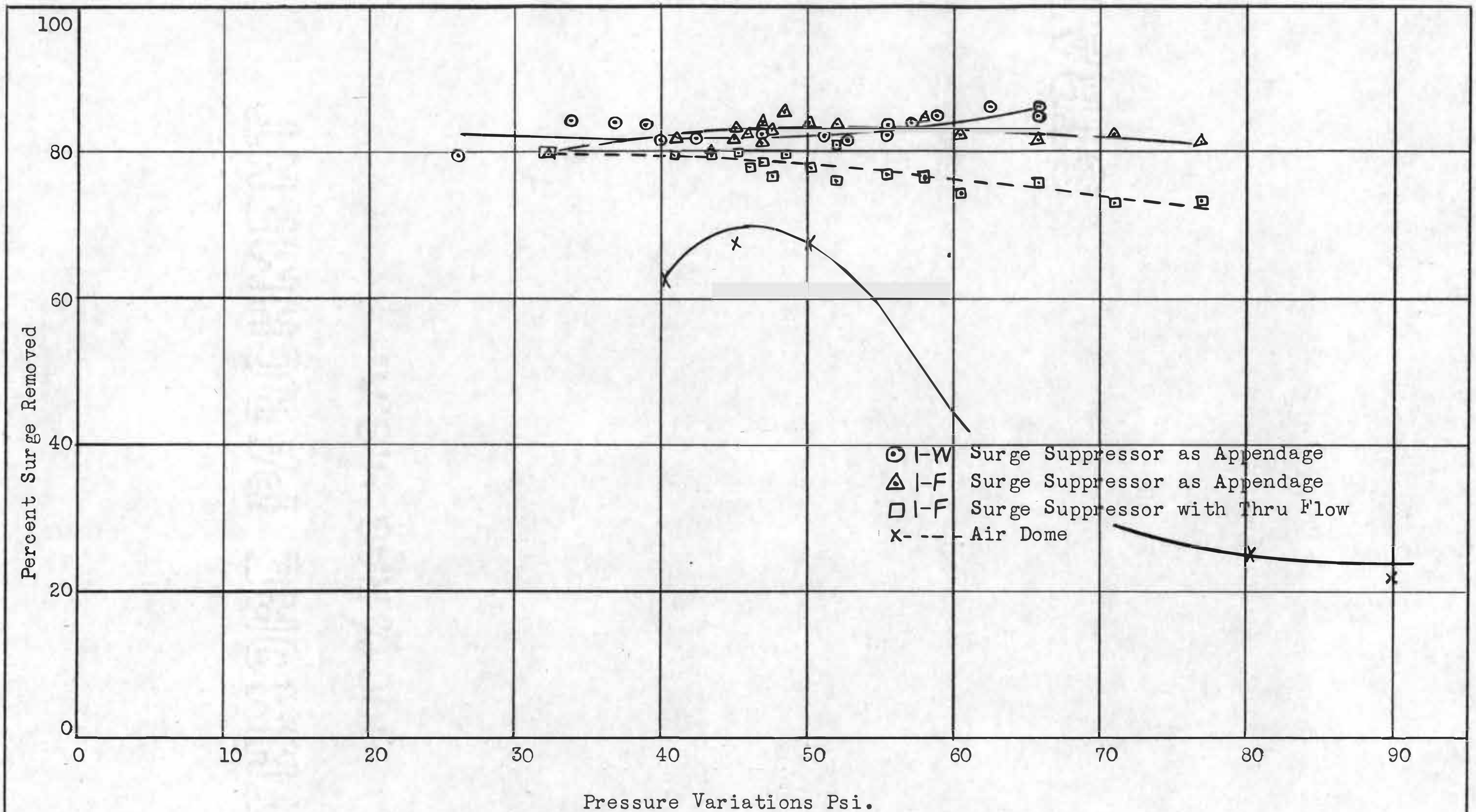


Figure 32 Comparison of Percent Surge Removed vs Pressure Variations
With 90 p.s.i. Average Flow Pressure

ORIGINAL EXPERIMENTAL DATA

Dates Tests Were Run:

July 18, 1949
July 19, 1949
October 24, 1949
October 25, 1949
November 1, 1949
November 2, 1949
November 3, 1949
November 12, 1949
November 28, 1949
November 29, 1949
December 5, 1949
December 7, 1949

Data for Pressure Relief Method of Calibration of Transducer	Run No.	Oscilloscope Deflection No. of Grids	Relief Pressure Psi	Vertical Gain Position
Initial Reading 1.5 Grids Deflection	1	38	53	1
	2	40	53	1
	3	39	50	1
	4	33.5	46	1
	5	33	45	1
	6	34	44	1
	7	33	44	1
	8	20	35	1
	9	19	34.5	1
	10	18.5	34.5	1
	11	19	35	1
	12	18	36	1
	13	21	36	1
	14	19	36	1
	15	15	28	1
	16	16.5	30.5	1
	17	15	30	1
	18	16	30	1
	19	16.5	30	1
	20	30	17	1
	21	30	16.5	1
	22	5.5	29.5	2
	23	6.0	30	2
	24	6.0	30	2
	25	6.0	30	2
	26	6.25	31	2
	27	8.25	41.5	2
	28	8.0	41	2
	29	8.5	41	2

Run No.	Deflection	Pressure	Gain
30	8.5	41	2
31	8.0	41	2
32	11.5	52	2
33	11.5	52	2
34	12.5	55	2
35	12.0	55	2
36	12.0	54.5	2
37	13.5	29.5	Max.
38	15	31	Max.
39	15.5	31	Max.
40	14	30	Max.
41	15.5	31	Max.
42	14	31	Max.
43	16	30	Max.
44	15	30	Max.
45	16	31	Max.
46	14	28	Max.
47	13	28	Max.
48	15	29	Max.
49	12	26	Max.
50	9	24	Max.
51	7	22.5	Max.
52	8	23	Max.
53	8	22	Max.
54	8.5	27	Max.
55	10	26	Max.
56	9	26	Max.
57	11	27	Max.
58	8	25	Max.
59	10	26	Max.
60	14	29	Max.
61	13	28.5	Max.
62	15.5	31	Max.
63	14	29	Max.
64	15	30	Max.
65	16	31	Max.
66	15.5	31	Max.
67	14.5	30	Max.
68	15	31	Max.
69	15	31	Max.
70	16	31	Max.
71	14	29	Max.
72	16.5	31	Max.
73	16	31	Max.
74	16	31	Max.
75	16	30	Max.
76	17.5	32	Max.
77	16.5	31	Max.
78	17	31	Max.
79	17.5	31.5	Max.
80	17.5	31.0	Max.
81	16.5	30.5	Max.
82	20.5	45	Max.
83	22	46	Max.
84	20	41	Max.
85	19	42.5	Max.

Run No.	Deflection	Pressure	Gain
86	22	42	Max.
87	20	42	Max.
88	22	44	Max.
89	22	44	Max.
90	22	44	Max.
91	22	44	Max.
92	23	44.5	Max.
93	23.5	44.5	Max.
94	24.5	45	Max.
95	23	43	Max.
96	20	41	Max.
97	21.5	42	Max.
98	22	42	Max.
99	22	43	Max.
100	22	45	Max.
101	22	45	1
102	22	45	1
103	21	44	1
104	20	44	1
105	20.5	44	1
106	19	40	1
107	17	39	1
108	18	39	1
109	20	40	1
110	18	40	1
111	16	39	1
112	19	39+	1
113	20	41	1
114	18	39	1
115	20	39	1
116	18	39	1
117	18.5	38.5	1
118	17	37	1
119	19	40	1
120	21	42	1
121	19.5	42	1
122	13	29.5	1
123	13	29	1
124	13.5	28.5	1
125	15	29	1
126	13	30	1
127	12	30	1
128	12.5	30	1
129	14	30	1
130	13.5	30	1
131	14	30	1
132	14	30	1
133	14.5	30.5	1
134	13	30	1
135	11.5	27.5	1
136	12	28	1
137	13.5	30	1
138	14	30	1
139	12	32	1
140	15	32	1
141	15	33	1

Run No.	Deflection	Pressure	Gain
142	15	32	1
143	15	32	1
144	16	33	1
145	14.5	32	1
146	16	33	1
147	14.5	32.5	1
148	13.5	30	1
149	13.5	30	1
150	13	30	1
151	15.5	31.5	1
152	14.5	31	1
153	14.5	32	1
154	5	20	2
155	6	26.5	2
156	7	26.5	2
157	8	28	2
158	7.5	27.5	2
159	8	27.5	2
160	6	26.5	2

DATA FOR EFFICIENCY DETERMINATION OF DESURGING UNITS

Place: Hydraulic Laboratory, Oklahoma A and M College

Oscilloscope Readings
Average Pressure: 90 psi
Wade Shokstop Desurger

Run No.	R P M X 4	'scope Deflec	Condition	Deflec. Removed	R P M	C P S	% Surge Removed	Δp psi
1	910	11	Control	----	183	6.1	----	26.1
2	910	2.25	Wade	8.75	183	6.1	79.5	----
3	960	15.75	Control	----	193	6.43	----	33.9
4	960	2.5	Wade	13.25	193	6.43	84.2	----
5	1030	17.25	Control	----	208	6.83	----	36.8
6	1030	2.75	Wade	14.5	208	6.83	84.0	----
7	1090	18.5	Control	----	222	7.40	----	39.0
8	1090	3.0	Wade	15.5	222	7.40	83.75	----
9	1150	19.0	Control	----	233	7.77	----	39.9
10	1150	3.5	Wade	15.5	233	7.77	81.50	----
11	1210	19.0	Control	----	243	8.10	----	39.9
12	1210	3.5	Wade	15.5	243	8.10	81.50	----
13	1265	20.5	Control	----	254	8.46	----	42.4
14	1265	3.75	Wade	16.75	254	8.46	81.70	----
15	1320	23	Control	----	265	8.83	----	46.7
16	1320	4	Wade	19	265	8.83	82.60	----
17	1380	25	Control	----	282	9.40	----	50.1
18	1380	4.5	Wade	20.5	282	9.40	82.10	----
19	1442	26.5	Control	----	292	9.73	----	52.7
20	1442	5	Wade	21.5	292	9.73	81.2	----
21	1508	28	Control	----	305	10.2	----	55.3
22	1508	5	Wade	23	305	10.20	82.2	----
23	1558	28	Control	----	315	10.50	----	55.3
24	1558	4.5	Wade	23.5	315	10.50	83.5	----
25	1614	29	Control	----	325	10.83	----	57.0
26	1614	4.75	Wade	24.25	325	10.83	83.6	----
27	1675	30	Control	----	340	11.33	----	58.7
28	1675	4.5	Wade	25.5	340	11.33	84.8	----
29	1801	32.2	Control	----	365	12.17	----	62.5
30	1801	4.5	Wade	27.7	365	12.17	86.0	----
31	1858	34.0	Control	----	375	12.50	----	65.7
32	1858	5.0	Wade	29.0	375	12.50	85.3	----
33	1919	34.0	Control	----	385	12.84	----	65.7
34	1919	5.2	Wade	28.8	385	12.84	84.7	----

Engine Indicators Readings with 100#/inch Spring

Run No.	R P M X 4	Δp psi	Condition	R P M	C P S	% Surge Removed	Δp Removed
1	910	6	Wade	227	7.6	87.8	43
2	910	49	Control	227	7.6	----	--
3	960	5	Wade	240	8.0	88.8	40

Run No.	RPM	Δp	Condition	RPM	CPS	%	Δp
4	960	45	Control	240	8.00	----	--
5	1030	6	Wade	258	8.63	87.8	43
6	1030	49	Control	258	8.63	----	--
7	1265	6	Wade	316	10.53	88.5	46
8	1265	52	Control	316	10.53	----	--
9	1675	5	Wade	418	13.98	93.2	68
10	1675	73	Control	418	13.98	----	--

Oscilloscope Readings
Average Pressure: 90 psi
Fluidynamic Desurger
Pressure Loading 54 psi

Run No.	R P M X 4	'scope deflec	Condition	Deflec. Removed	R P M	C P S	% Surge Removed
1	1697	22.5	Control	----	425	14.15	----
2	1697	4.0	Fd App	18.5	425	14.15	82.3
3	1697	5.0	Fd thru	18	425	14.15	80.1
4	1810	23.0	Control	----	452	15.1	----
5	1810	3.75	Fd App	19.25	452	15.1	83.7
6	1811	5.0	Fd thru	18	452	15.1	78.2
7	1902	26.0	Control	----	480	16.0	----
8	1902	4.25	Fd App	21.75	480	16.0	83.7
9	1902	6.25	Fd thru	19.75	480	16.0	76.0
10	2010	28.00	Control	----	503	16.75	----
11	2010	4.5	Fd App	23.5	503	16.75	84
12	2010	6.5	Fd thru	21.5	503	16.75	77
13	2130	29.50	Control	----	533	17.8	----
14	2130	4.5	Fd App	25.0	533	17.8	85
15	2130	7.0	Fd Thru	22.5	533	17.8	77
16	2230	31.0	Control	----	558	18.6	----
17	2232	5.5	Fd App	25.5	558	18.6	82.3
18	2232	8.0	Fd Thru	23.0	558	18.6	74.5
19	2321	34.0	Control	----	580	19.33	----
20	2321	6.25	Fd App	27.75	580	19.33	82
21	2321	8.25	Fd Thru	25.75	580	19.33	76
22	2440	37.0	Control	----	610	20.3	----
23	2440	6.50	Fd App	31.5	610	20.3	86
24	2440	10.0	Fd Thru	27	610	20.3	73
25	2560	41	Control	----	640	21.3	----
26	2560	7.50	Fd App	33.5	640	21.3	82
27	2560	11	Fd Thru	30	640	21.3	73.5

Engine Indicators Readings 100#/inch Spring
Average Pressure: 90 psi
Fluidynamic Desurger
Pressure Loading 54 psi

Run No.	R P M X 4	Δp psi	Condition	Δp Removed	R P M	C P S	% Surge Removed
1	1697	62	Control	----	424	14.15	----
2	1697	3	App	59	424	14.15	95
3	1697	5	Thru	57	424	14.15	92

Run No.	RPM	Δp	Condition	Δp	RPM	CPS	%
4	1811	86	Control	----	452	15.05	----
5	1811	4	App	82	452	15.05	95
6	1811	7	Thru	79	452	15.05	92
7	1902	148	Control	----	475	15.82	----
8	1902	4	App	144	475	15.82	97.3
9	1902	7	Thru	141	475	15.82	95.3
10	2010	99	Control	----	502	16.7	----
11	2010	3	App	96	502	16.7	97
12	2010	4	Thru	95	502	16.7	96
13	2130	90	Control	----	532	17.7	----
14	2130	3	App	87	532	17.7	96.7
15	2130	5	Thru	85	532	17.7	94.5
16	2232	120	Control	----	558	18.6	----
17	2232	3	App	117	558	18.6	97.5
18	2232	5	Thru	115	558	18.6	96
19	2321	120	Control	----	580	19.33	----
20	2321	3	App	117	580	19.33	97.5
21	2321	8	Thru	112	580	19.33	93.3
22	2560	108	Control	----	640	21.3	----
23	2560	3	App	105	640	21.3	97.2
24	2560	6	Thru	102	640	21.3	94.5

Engine Indicators Readings 100#/inch Spring
Average Pressure 90 psi
Fluidynamic Desurger
Pressure Loading 54 psi

Run No.	R P M X 4	Δp psi	Condition	Δp Removed	R P M	C P S	% Surge Removed
1	900	45	Control	----	225	8.33	----
2	900	1	App	44	225	8.33	97.8
3	900	3	Thru	42	225	8.33	93.3
4	960	50	Control	----	240	8.0	----
5	960	3	App	47	240	8.0	94
6	960	6	Thru	44	240	8.0	88
7	1020	52	Control	----	255	8.5	----
8	1020	4	App	48	255	8.5	92.3
9	1020	6	Thru	46	255	8.5	88.5
10	1060	50	Control	----	265	8.83	----
11	1060	3	App	47	265	8.83	94
12	1060	5	Thru	45	265	8.83	90
13	1118	52	Control	----	280	9.24	----
14	1118	3	App	49	280	9.24	94.3
15	1118	4	Thru	48	280	9.24	92.3
16	1180	56	Control	----	295	9.84	----
17	1180	5	App	51	295	9.84	91.1
18	1180	6	Thru	50	295	9.84	89.3
19	1259	65	Control	----	315	10.48	----
20	1259	5	App	60	315	10.48	92.3
21	1259	6	Thru	59	315	10.48	90.8
22	1320	60	Control	----	330	11.0	----
23	1320	4	App	56	330	11.0	93.3
24	1320	6	Thru	54	330	11.0	90.0
25	1400	70	Control	----	350	11.65	----
26	1400	3	App	67	350	11.65	95.7
27	1400	6	Thru	63	350	11.65	90

Run No.	RPM	Δp	Condition	Δp	RPM	CPS	%
28	1510	62	Control	----	378	12.6	----
29	1510	2	App	60	378	12.6	96.7
30	1510	3	Thru	59	378	12.6	95.2
31	1605	69	Control	----	402	13.38	----
32	1605	3	App	66	402	13.38	95.6
33	1605	6	Thru	63	402	13.38	94.3

Oscilloscope Readings
Average Pressure: 90 psi
Fluidynamic Desurger
Pressure Loading 54 psi

Run No.	R P M X 4	'scope deflec	Condition	Deflec. Removed	R P M	C P S	% Surge Removed
1	900	14.5	Control	----	225	8.33	----
2	900	3	Fd App	11.5	225	8.33	79.5
3	900	3	Fd Thru	11.5	225	8.33	79.5
4	960	19.5	Control	----	240	8.0	----
5	960	3.5	Fd App	16.0	240	8.0	82.0
6	960	4	Fd Thru	15.5	240	8.0	79.5
7	1020	21.0	Control	----	255	8.5	----
8	1020	4.2	Fd App	16.8	255	8.5	80.0
9	1020	4.3	Fd Thru	16.7	255	8.5	79.5
10	1060	22.0	Control	----	265	8.83	----
11	1060	3.75	Fd App	18.25	265	8.83	83.0
12	1060	4.50	Fd Thru	17.50	265	8.83	79.6
13	1118	22.0	Control	----	280	9.24	----
14	1118	4.0	Fd App	18.0	280	9.24	81.8
15	1118	4.5	Fd Thru	17.15	280	9.24	79.6
16	1180	23.0	Control	----	295	9.84	----
17	1180	4.25	Fd App	18.75	295	9.84	81.5
18	1180	5.00	Fd Thru	18.00	295	9.84	77.3
19	1259	23.00	Control	----	315	10.48	----
20	1259	4.00	Fd App	19.00	315	10.48	82.6
21	1259	5.00	Fd Thru	18.00	315	10.48	77.3
22	1320	26.0	Control	----	330	11.00	----
23	1320	4.25	Fd App	21.75	330	11.00	83.6
24	1320	5.00	Fd Thru	21.00	330	11.00	81.8
25	1400	25.00	Control	----	350	11.65	----
26	1400	4.0	Fd App	21.00	350	11.65	84
27	1400	5.5	Fd Thru	19.50	350	11.65	78
28	1510	24.00	Control	----	378	12.6	----
29	1510	3.50	Fd App	20.50	378	12.6	85.5
30	1510	5.00	Fd Thru	19.00	378	12.6	79.2
31	1605	23.5	Control	----	492	13.38	----
32	1605	4.0	Fd App	19.5	402	13.38	83.0
33	1605	5.5	Fd Thru	18.0	402	13.38	76.6

Engine Indicators Readings 150#/inch Spring
Average Pressure 70 psi
Wade Shokstop Desurger

Run No.	R P M X 4	Δ p psi	Condition	Δp Removed	R P M	C P S	% Surge Removed
1	660	45	Control	---	165	5.5	----
2	660	6	Wade	39	165	5.5	86.7
3	1200	69	Control	----	300	10.0	----
4	1200	7	Wade	62	300	10.0	89.8
5	1401	86	Control	----	350	11.66	----
6	1401	14	Wade	72	350	11.66	83.7
7	1870	103	Control	----	490	15.58	----
8	1870	12	Wade	91	490	15.58	88.2
9	2234	135	Control	----	558	18.70	----
10	2234	15	Wade	120	558	18.70	88.8

Oscilloscope Readings
Average Pressure: 70 psi
Fluidynamic Desurger
Pressure Loading: 54 psi

Run No.	R P M X 4	'scope deflec	Deflec. Corrected	Condi-tion	Deflec. Removed	R P M	C P S	% Surge Removed
1	650	25.0	24.0	Cont	----	162	5.42	----
2	650	3.5	2.5	App	21.5	162	5.42	89.5
3	650	4.5	3.5	Thru	20.5	162	5.42	85.4
4	770	29.0	28.0	Cont	----	192	6.42	----
5	770	3.75	2.75	App	25.25	192	6.42	90.3
6	770	4.75	3.75	Thru	24.25	192	6.42	86.7
7	890	34.5	33.5	Cont	----	222	7.42	----
8	890	4	3	App	30.5	222	7.42	91.2
9	890	5	4	Thru	29.5	222	7.42	88.0
10	1020	36	35	Cont	----	255	8.48	----
11	1020	4.25	3.25	App	31.75	255	8.48	90.6
12	1020	5.25	4.25	Thru	30.75	255	8.48	87.8
13	1140	35.5	34.5	Cont	----	285	9.50	----
14	1140	4.0	3	App	31.5	285	9.50	91.2
15	1140	5.2	4.2	Thru	30.3	285	9.50	87.8
16	1263	35.0	34.0	Cont	----	318	10.52	----
17	1263	5.0	4.0	App	30.0	318	10.52	88.3
18	1263	5.5	4.5	Thru	29.5	318	10.52	86.7
19	1406	33.0	32.0	Cont	----	351	11.71	----
20	1406	5.75	4.75	App	27.25	351	11.71	85.2
21	1406	6.25	5.25	Thru	26.75	351	11.71	83.6
22	1520	33.0	32.0	Cont	----	380	12.66	----
23	1520	5.75	4.75	App	27.25	380	12.66	85.2
24	1520	7.00	6.00	Thru	26.00	380	12.66	81.4
25	1622	36.00	35.00	Cont	----	405	13.51	----
26	1622	6.00	5.00	App	30.00	405	13.51	85.7
27	1622	7.75	6.75	Thru	28.25	405	13.51	80.7
28	1735	31.0	30.0	Cont	----	434	14.45	Excessive Valve Clatter

Run No.	RPM	Deflec.	Corrected	Cond.	Removed	RPM	CPS	%
29	1735	5.0	4.0	App	26.00	434	14.45	86.7
30	1735	6.0	5.0	Thru	25.00	434	14.45	83.3

Oscilloscope Readings
Average Pressure: 70 psi
Wade Shokstop Desurger

Run No.	R P M X 4	'scope Deflec	Condition	Deflec. Removed	R P M	C P S	% Surge Removed
1	660	11.5	Control	----	165	5.40	----
2	660	2.5	Wade	9.0	165	5.40	81.8
3	780	12.5	Control	----	195	6.50	----
4	780	3.0	Wade	9.5	195	6.50	79.2
5	882	14.25	Control	----	220	7.35	----
6	882	3.25	Wade	11.00	220	7.35	79.8
7	970	16.50	Control	----	242	8.08	----
8	970	4.0	Wade	12.5	242	8.08	78.2
9	1092	18.5	Control	----	273	9.09	----
10	1092	4.25	Wade	14.25	273	9.09	79.2
11	1200	22.00	Control	----	300	10.0	----
12	1200	4.50	Wade	17.50	300	10.0	81.4
13	1303	26.0	Control	----	326	10.86	----
14	1303	5.0	Wade	21.0	326	10.86	82.3
15	1401	29.0	Control	----	350	11.67	----
16	1401	6.0	Wade		350	11.67	80.8
17	1510	30.5	Control	----	377	12.58	----
18	1510	6.5	Wade	24	377	12.58	80.0
19	1620	32.0	Control	----	405	13.42	----
20	1620	9.0	Wade		405	13.42	73.0
21	1752	33.0	Control	----	438	14.60	----
22	1752	9.75	Wade	23	438	14.60	73.8
23	1870	34.00	Control	----	467	15.58	----
24	1870	10.25	Wade		467	15.58	71.0
25	1900	33	Control	----	475	15.83	----
26	1900	11	Wade		475	15.83	67.7
27	2009	36	Control	----	502	16.72	----
28	2009	13	Wade	23	502	16.72	65.7
29	2114	41	Control	----	524	17.60	----
30	2114	15	Wade		524	17.60	65.0
31	2234	47	Control	----	558	18.61	----
32	2234	17.5	Wade		558	18.61	64.2

Oscilloscope Readings
Average Pressure: 90 psi
Air Dome

Run No.	R P M X 4	'scope Deflec	Condition	Deflec. Removed	R P M	C P S	% Surge Removed	Δp psi
1	655	26	Control	----	164	5.45	----	40
2	655	9.5	A. D.	16.5	164	5.45	63.5	----
3	780	34.5	Control	----	195	6.5	----	46.9
4	780	11	A. D.	23.5	195	6.5	68.0	----

Run No.	RPM	Deflec.	Condition	Removed	RPM	CPS	%	Δp
5	923	40	Control	----	231	7.68	----	53
6	923	13	A. D.	27	231	7.68	67.5	----
7	1020	46	Control	----	255	8.50	----	60.4
8	1020	26	A. D.	30	255	8.50	43.5	----
9	1124	52	Control	----	281	9.37	----	65.5
10	1124	35	A. D.	17	281	9.37	32.7	----
11	1270	50	Control	----	318	10.58	----	64.0
12	1270	35	A. D.	15	318	10.58	30	----
13	1400	29	Control	----	350	11.67	----	106
14	1400	23	A. D.	6	350	11.67	20	----
15	1522	32	Control	----	380	12.69	----	117
16	1522	22.5	A. D.	19.5	380	12.69	29.7	----
17	1643	36	Control	----	411	13.69	----	132
18	1643	30	A. D.	6	411	13.69	20	----
19	1750	39	Control	----	432	14.57	----	143
20	1750	35	A. D.	4	432	14.57	10.25	----
21	1870	40	Control	----	468	15.57	----	146.5
22	1870	35	A. D.	5	468	15.57	12.5	----
23	1960	44	Control	----	490	16.32	----	161
24	1960	36	A. D.	8	490	16.32	18.2	----
25	2052	54	Control	----	513	17.10	----	198
26	2052	50	A. D.	4	513	17.10	12.0	----

Oscilloscope Readings
Average Pressure: 70 psi
Air Dome

Run No.	R P M X 4	'scope deflec	Condition	Deflec. Removed	R P M	C P S	% Surge Removed	Δp psi
1	650	17.5	Control	----	162		----	32.4
2	650	6.5	A. D.	11	162	5	63	----
3	790	19.5	Control	----	197		----	35.7
4	790	6.5	A. D.	13	197	6.	66.6	----
5	910	22.5	Control	----	227		----	37.7
6	910	9	A. D.	13.5	227		60	----
7	1052	25	Control	----	263		----	38
8	1052	13	A. D.	12	263		47	----
9	1150	27.5	Control	----	287		----	41
10	1150	18	A. D.	9.5	287	9.	34.5	----
11	1260	30.5	Control	----	315		----	42.5
12	1260	20.0	A. D.	10.5	315		34.4	----
13	1380	33.0	Control	----	345		----	44.5
14	1380	25.5	A. D.	7.5	345		22.7	----
15	1470	35	Control	----	368		----	46.9
16	1470	26	A. D.	9	368		25.7	----
17	1583	38	Control	----	396		----	50.5
18	1583	30	A. D.	8	396	13.	21.1	----
19	1690	40	Control	----	422		----	53
20	1690	35.5	A. D.	4.5	422	14	11.25	----
21	1780	42.5	Control	----	445		----	52.5
22	1780	39.5	A. D.	3	445	14.	7.06	----
23	1880	47.0	Control	----	470		----	61.6
24	1880	35.0	A. D.	12	470		25.5	----
25	1970	50	Control	----	492	16.5	----	65.5
26	1970	46	A. D.	4	492		8.00	----

DATA FOR EFFICIENCY VERSUS ΔP AND EFFICIENCY VERSUS CPS

Wade Desurger 90 psi

Run No.	Cps	% Surge Removed	Δp psi
1	6.1	79.5	26.1
2	6.43	84.2	33.9
3	6.83	84.0	36.8
4	7.40	83.75	39.0
5	7.77	81.50	39.9
6	8.10	81.50	39.9
7	8.46	81.70	42.4
8	8.83	82.60	46.7
9	9.40	82.10	50.1
10	9.73	81.20	52.7
11	10.20	82.20	55.3
12	10.50	83.5	55.3
13	10.83	83.6	57.0
14	11.33	84.8	58.7
15	12.17	86	62.5
16	12.50	85.3	65.7
17	12.84	84.7	65.7

Fluidynamid Desurger appendage 90 psi

Run No.	Cps	% Surge Removed	Δp psi
1	6.08	79.3	32.2
2	6.43	82.0	40.7
3	6.73	80.0	43.3
4	7.16	83.0	45.0
5	7.50	81.7	45.0
6	8.00	81.5	46.7
7	8.50	82.5	46.7
8	8.83	83.6	51.9
9	9.43	84.0	50.1
10	10.26	85.3	48.4
11	10.83	83.0	47.5
12	11.46	82.2	45.9
13	12.16	83.6	46.7
14	12.80	83.5	51.9
15	13.50	83.80	55.3
16	14.46	84.7	57.9
17	15.10	82.4	60.4
18	15.66	81.6	65.7
19	16.50	82.4	70.9
20	17.27	81.7	76.7

Fluidynamic Desurger
Thru flow 90 psi

Run No.	Cps	% Surge Removed	Δp psi
1	6.08	79.3	32.2
2	6.43	79.5	40.7
3	6.73	79.5	43.3
4	7.16	79.5	45.0
5	7.50	79.5	45.0
6	8.00	78.3	46.7
7	8.50	78.3	46.7
8	8.83	80.7	51.9
9	9.43	78.0	50.1
10	10.26	79.2	48.4
11	10.83	76.5	47.5
12	11.46	77.8	45.9
13	12.16	78.3	46.7
14	12.80	76.0	51.9
15	13.50	76.8	55.3
16	14.46	76.3	57.9
17	15.10	74.3	60.4
18	15.66	75.8	65.7
19	16.50	73.0	70.9
20	17.27	73.2	76.7

Wade Desurger 70 psi

Run No.	Cps	% Surge Removed	Δp psi
1	5.40	81.8	22.2
2	6.50	79.2	23.8
3	7.35	79.8	26.8
4	8.08	78.2	30.5
5	9.09	79.2	34.0
6	10.0	81.4	37.4
7	10.86	82.3	40.0
8	11.67	80.8	41.9
9	12.58	80.0	43.2
10	13.42	73.0	44.0
11	14.60	73.8	44.6
12	15.58	71.0	45.7
13	15.83	67.7	44.6
14	16.72	65.7	48.2
15	17.60	65.0	54.3
16	18.61	64.0	66.3

Fluidynamic Desurger
Appendage 70 psi

Run No.	Cps	% Surge Removed	Δp psi
1	5.42	89.6	39.4
2	6.42	90.0	42.9
3	7.42	91.0	46.3
4	8.48	90.6	48.2
5	9.50	91.3	47.5
6	10.52	88.3	47.0
7	11.71	85.2	44.0

	Run No.	CPS	%	Δp
	8	12.66	85.2	44.0
	9	13.51	85.7	48.2
	10	14.45	86.7	43.0
	11	15.58	83.3	48.2
	12	16.40	85.0	54.3
	13	17.48	86.5	64.2
Fluidynamic Desurger Thru Flow 70 psi	1	5.42	85.4	39.4
	2	6.42	86.7	42.9
	3	7.42	88.0	46.3
	4	8.48	87.8	48.3
	5	9.50	87.8	47.5
	6	10.52	86.7	47.0
	7	11.71	83.6	44.0
	8	12.66	81.4	44.0
	9	13.51	80.7	48.2
	10	14.45	83.3	43.0
	11	15.58	83.3	48.2
	12	16.40	81.3	54.3
	13	17.48	82.3	64.2

CHAPTER IX

SAMPLE CALCULATIONS

Pressure Surge Determination

condition: 10" pipe (nominal)

1/2" thick pipe

Q_o = 5.8 cfs of water

P_o = 150 psi

A = .518 sq. ft.

$$H = \frac{aV_o}{g} \quad \text{equation (1), page 5}$$

$$a = \frac{12}{\sqrt{\frac{w}{g}\left(\frac{1}{K} + \frac{d}{Et}\right)}} \quad \text{equation (2), page 6}$$

$w = 62.4 \text{ lbs/ft}^3$

$g = 32.2 \text{ ft/sec}^2$

K = 300,000 psi

$d = 10.75 - 2(.50) = 9.75$

E = 30,000,000 psi

$$\frac{d}{t} = \frac{9.75}{.50} = 19.5$$

$$a = \frac{12}{\sqrt{\frac{62.4}{32.2}\left(\frac{1}{3 \times 10^5} + \frac{19.5}{3 \times 10^7}\right)}} = 4330 \text{ fps}$$

$$V = V_o - V_{final}$$

in case of complete stoppage of flow

$$V = V_o$$

$$V_o = \frac{Q_o}{A} = \frac{5.8}{.518} = 11.2 \text{ fps}$$

$$H = \frac{4330 \times 11.2}{32.2} = 1505 \text{ ft of water or surge pressure}$$

$$P = 0.433 \frac{aV_o}{g}$$

$$P = \frac{0.433 \times 4330 \times 11.2}{32.2} = 653 \text{ psi}$$

Pressure maximum

$$P_m = P_o + P_{surge} = 150 + 653 = 803 \text{ psi}$$

Compressibility

$$c = \frac{\dfrac{V_1 - V_2}{V_1}}{P_2 - P_1}$$

bulk modulus

$$K = \frac{1}{c}$$

Calculation for throttled air chamber size installed on reciprocating pump.

Pump:

$$S = 1.75 \text{ in}$$

$$B = 1.50 \text{ in}$$

$$F = .25 \text{ (simplex double acting pump)}$$

$$P_o = 90 \text{ psi}$$

$$P_x = 60 \text{ psi}$$

$$P_m = 120 \text{ psi}$$

$$n = 1.4$$

$$C_x = \frac{0.7854\, FB^2S\left(\dfrac{P_o^*}{P_x^*}\right)^{\frac{1}{n}}}{1 - \left(\dfrac{P_o^*}{P_m^*}\right)^{\frac{1}{n}}} \text{ cu.in. equation 10, page 20}$$

$$P_o^* = 105 \text{ psi}$$

$$P_x^* = 75 \text{ psi}$$

$$P_m^* = 135 \text{ psi}$$

$$\left(\frac{P_o^*}{P_x^*}\right)^{\frac{1}{n}} = \left(\frac{105}{75}\right)^{\frac{1}{1.4}} = (1.4)^{.714} = 1.272$$

$$\left(\frac{P_o^*}{P_m^*}\right)^{\frac{1}{n}} = \left(\frac{105}{135}\right)^{\frac{1}{1.4}} = (.778)^{.714} = .836$$

$$C_x = \frac{3.0925 \times .25 \times 1.272}{(1 - .836)} = 5.99 \text{ cu. in.}$$

There must be at least 6 cu. in. of volume in air chamber at all times for it to be effective.

Throttled Air Chamber for Pipe Line

condition: $L = 1000$ ft

$a = 4330$ fps

$V_o = 11.2$ fps

$Q_o = 5.8$ cfs

$A = .518$ sq. ft.

$P = 150$ psi

$H_o^* = 165 \times 2.31 = 381$ ft

$g = 32.2$

pipe line characteristic

$$\rho^* = \frac{aV_o}{2gH_o^*}$$ equation (11), page 22

$$\overset{*}{\varrho} = \frac{4330 \times 11.2}{2 \times 32.2 \times 381} = 1.98$$

$$2\overset{*}{\varrho} = 3.96$$

surge restricted to .5 H_o^* by specifications

from Figure 6 read value of $2\overset{*}{\varrho}\overset{*}{\sigma} = 13$ at intersection of

$$2\overset{*}{\varrho} = 4 \text{ and } \frac{\text{maximum surge}}{H_o^*} = .5$$

substituting these values in equation (14), page 24

$$C_o = \overset{*}{\varrho}\overset{*}{\sigma} Q_o \frac{L}{a} = \frac{13}{2} \times 5.8 \times \frac{1000}{4330} = 8.7 \text{ cu. ft.}$$

volume in desurger of throttled air chamber at P_o.

RPM of Pump

$$RPM = \frac{\text{strobotac reading}}{4}$$

$$RPM = \frac{2560}{4} = 640$$

$$cps = \frac{RPM \times 2}{60}$$

$$cps = \frac{640 \times 2}{60} = 21.3$$

$$\%\ \text{surge removed} = \frac{P(\text{control}) - P(\text{desurger})}{P\ (\text{control})} \times 100.$$

$$= \frac{41 - 7.50}{41} \times 100 = 82\ \%.$$

CHAPTER X

SUMMARY AND RESULTS

This study was performed first by investigating the causes of pressure surges and then trying to duplicate, in the laboratory, situations that were found present in the field.

In the transportation of fluids in a pipe line system, acceleration and deceleration of the fluid column is necessary. This acceleration and deceleration causes a change in velocity and a corresponding change in pressure, or a pressure variation. The severity of the velocity change dictates the magnitude of the pressure variation.

Before conclusive tests could be run it was necessary to build and calibrate a surge measuring device, the electrokinetic transducer.

Equipment was arranged so that a given set of conditions such as speed of pump and average pressure could be maintained. A small simplex double acting pump with variable speed drive was used to generate surges. The discharge pressure wave of the pump without desurger was recorded, than a desurger as "cut in" to the system and the discharge pressure wave again recorded. A comparison of the pressure wave with and without a desurger was made giving the apparent surge removal efficiency of the desurger tested.

The efficiency of three desurgers, Wade Shokstop, Fluidynamic, and Airdome were thus determined, relative both to magnitude and frequency of pressure surge.

All three desurging units tested proved to be effective surge removing devices.

AIRDOME

The Airdome when tested at an average pressure of 70 psi gave a maximum efficiency of 66 percent against a pressure variation of 35 psi. The efficiency of the Airdome dropped sharply with increased pressure variation to an efficiency of only 30 percent at a ΔP of 45 psi and as low as 14 percent at ΔP equal to 65 psi.

At 70 psi the efficiency of the Airdome relative to frequency of surges was a maximum of 65 percent at 6.5 surges per second (SPS) dropping off at 8 SPS. At 16 SPS the efficiency of the Airdome was only 10 percent.

When tested at an average pressure of 90 psi the maximum efficiency of the Airdome relative to pressure variation was 70 percent at ΔP equal 46 psi. The drop in efficiency was not as severe when operating at 90 psi as it was when operating at 70 psi, giving an efficiency of 25 percent at ΔP of 80 psi.

Relative to surges per second the efficiency curve at 90 psi was very similar to the 70 psi curve, giving maximum surge removal of 67 percent around 7 SPS.

WADE SHOKSTOP DESURGER

The Wade Shokstop exhibited good surge removing characteristics. When tested at 70 psi, relative to pressure variation, the Wade removed 82 percent surge at ΔP equal 22 psi, dropping gradually to 64 percent at a ΔP of 66 psi. Relative to surges per second, at 70 psi, the efficiency varied from 81 percent at 5.5 to 65 percent at 18 SPS. At 90 psi the Wade removed approximately 85 percent surges for the entire range of ΔP (26 psi - 66 psi) tested. The Wade averaged around 83 percent efficiency as the SPS varied from 6 to 13.

FLUIDYNAMIC DESURGER

Tests showed the Fluidynamic to be little effected by increase in both frequency and magnitude of pressure surges.

Tests at 70 psi with the Fluidynamic as an appendage showed an efficiency of approximately 87 percent versus ΔP and 90 percent versus SPS for the entire range tested. At 90 psi the Fluidynamic as an appendage gave an average efficiency of 83 percent for both pressure surge magnitude and frequency changes.

With the Fluidynamic as a through flow device results were comparable to unit installed as an appendage being some two to three percent lower under all conditions of tests.

Data as recorded during this study is presented in graph form, see Figures 13 through 32.

CHAPTER XI

CONCLUSIONS

With some refinement the electrokinetic transducer as used in this test would prove a valuable asset in the study and solution of surge problems. Calibration of the transducer produced a straight line relationship between deflection and magnitude of pressure surge. This was very convenient since it was then only necessary to compare the amount of deflection on the oscilloscope to secure the percent surge removal for a particular test run. Use of this transducer is limited to location where 60 cycle AC current is available. Redesign of this pickup to function as a portable unit independent of outside power would certainly seem justifiable.

Due to the increasing traffic in pipe line transportation and to the operators desire to increase the carrying capacity of existing facilities pressure surges caused by the stopping and starting of fluid are continuing to be of major concern to the pipeline operator. At the present time we are not far enough advanced in our knowledge to design a surge free pipe line system and stay within accepted economic practices. Therefore, a solution to pressure surge problems is not usually considered until danger of the surges show themselves by way of failure of some part of the system. The most common and the most economic solution for removal of pressure surges is the installation of an airdome or commercial desuring device. During this study three desurging units were tested with the operating limitations of each determined.

<u>AIRDOME</u>

The Airdome should not be used where the frequency of the surges created exceed eight (8) per second.

Use of the Airdome should be limited to installations where the pressure variation does not exceed 50 percent of the average operating pressure. Figure 14, operating pressure 70 psi, shows that once the pressure variation reaches 35 psi that the efficiency of the Airdome drops very sharply. This fact is confirmed by Figure 16 showing a drop of efficiency at around 45 psi when operating at 90 psi.

WADE SHOKSTOP DESURGER

The Wade desurger gave acceptable surge removing ability over the entire range of test pressure and frequency encountered. This device is recommended as long as the manufacturer is consulted before any applications are considered. This is a must since the desurging volume of this unit is small and no installation should be made without the recommendation of the seller.

FLUIDYNAMIC DESURGER

The Fluidynamic desurger gave the best all around results throughout the test. The unit apparently is not conscious of either pressure surge frequency or magnitude. This is undoubtedly due to the design of the unit which incorporates two acceptable surge removing principles, (1) throttling orifice and (2) compression chamber, into one simple workable unit. The efficiency of the Fluidynamic as an appendage was higher than as through flow. This was due to the placement of the unit which caused the pump to discharge straight, without turns, into the Fluidynamic with the main flow line coming off at right angles to this line. In this manner the surges were acted upon twice by the desurger before moving on down stream. Another desirable characteristic of the Fluidynamic desurger is its flexibility, it can be tuned to any application by varying the charge pressure.

The results of this study indicate that either the Wade or Fluidynamic desurger would be acceptable surge removing devices. The question then arises as to the desired efficiency, flexibility and economy of the proposed installation. These things can only be determined after a detailed study of the application with both the operator and the supplier of the desurging device.

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