THE EFFECT OF PRESSURE SURGES ON THE EFFICIENCY AND OPERATION OF A PISTON PUMP

THE EFFECT OF PRESSURE SURGES ON THE EFFICIENCY AND OPERATION OF A PISTON PUMP

By

ERNEST CHESTER FITCH, JR.

Bachelor of Science

Oklahoma Agricultural and Mechanical College

Stillwater, Oklahoma

1950

Submitted to the Faculty of the Graduate School of

the Oklahoma Agricultural and Mechanical College

in Partial Fulfillment of the Requirements

for the Degree of

MASTER OF SCIENCE

1951

THE EFFECT OF PRESSURE SURGES ON THE
EFFICIENCY AND OPERATION OF A PISTON PUMP

ERNEST CHESTER FITCH, JR.

MASTER OF SCIENCE

1951

THESIS AND ABSTRACT APPROVED:

J. P. Everett
Thesis Adviser

C. M. Leonard
Faculty Representative

D. C. McIntosh
Dean of the Graduate School

TABLE OF CONTENTS

ACKNOWLEDGEMENTS

It is a pleasure for the author to acknowledge his indebtedness to Professor Rollo E. Venn and the School of Mechanical Engineering for generously offering him a graduate fellowship and the opportunity to continue his education.

The author also wishes to express his sincere thanks to Professor Carroll M. Leonard for carefully reading the entire thesis and offering many valuable suggestions and criticisms, and to Mr. J. P. Everett for giving valuable advice and comments.

Especial appreciation is due Professor John Howard Dawson for his help in presenting the material of this thesis and under whose immediate direction this work was carried on in the Hydraulics Laboratory of the School of Civil Engineering.

Most of all the author is indebted to his wife, who has been a capable critic and efficient co-worker in every phase of this investigation.

LIST OF ILLUSTRATIONS

SYMBOLS AND ABBREVIATIONS

A	cross-sectional area of piston, square feet
C	calibration constant for watt-hour meter
c	celerity of pressure wave, feet per second
cu	cubic
D	diameter of cylinder, feet
E	linear modulus of elasticity of pipe, pounds per square inch
f	frequency of pump, cycles per second
fig.	figure
fps	feet per second
ft	feet
^oF	degree fahrenheit
g	acceleration of gravity, feet per second squared
hp	horsepower
HP_i	horsepower input to pump, horsepower
HP_o	water horsepower output of pump, horsepower
in.	inch
K	bulk modulus of elasticity of pipe, pounds per square inch
lb	pound
L	length of pipe, feet
L_s	length of stroke, feet
ME	mechanical efficiency of pump, percent
n_s	strokes per second
psf	pounds per square feet
psi	pounds per square inch
P_d	throttled discharge pressure, pounds per square inch

P_s	throttled suction pressure, pounds per square inch
P_t	total pressure or $(P_d - P_s)$, pounds per square inch
P'_d	throttled discharge pressure, pounds per square feet
P'_s	throttled suction pressure, pounds per square feet
Q	rate of flow, pounds per second
rev	revolution
rpm	revolutions per minute
RPM_p	pump speed, revolutions per minute
sec	second
sq	square
T	period of water hammer pressure, seconds
t	wall thickness of pipe, inch
T_w	time required for ten revolutions of watt-hour meter disk, seconds
V_a	actual volume pumped per second, cubic feet per second
V_d	velocity of fluid leaving pump, feet per second
V_s	velocity of fluid entering pump, feet per second
V_t	theoretical volume pumped per second, cubic feet per second
VE	volumetric efficiency of pump, per cent
ΔV	difference in fluid velocity, feet per second
W	watts pulled by constant load as indicated by watt meter, watts
w	specific weight of water, pounds per cubic feet
W_m	watt input to motor, watts
Z_d	elevation of discharge gauge, feet
Z_s	elevation of suction gauge, feet
Z_t	difference in elevation or $(Z_d - Z_s)$, feet
"	inch
#	pound
%	per cent

CHAPTER I

INTRODUCTION

The reciprocating pump possesses the inherent characteristic of delivering a pulsating flow of fluid. These pulsations serve no useful purpose but continually present a constant source of trouble for the operator. The pressure surges created by the reciprocating motion of the pump are not only responsible for costly damage to pipeline equipment but also impairs the proper operation of the pumping unit.

In the oil industry, the problem was initially confronted in the experiences with the first oil trunk line completed in Pennsylvania around 1865. The pump used on this pioneer pipeline was a simple single-cylinder pump connected to a pin on the flywheel of a single cylinder steam engine. This arrangement was soon outmoded by the direct-acting duplex pump (Fig. 1) which progressively became the standard for pipeline power. The poor economy of this pump was not of immediate importance at this early date since it was the only means available to pump the oil.

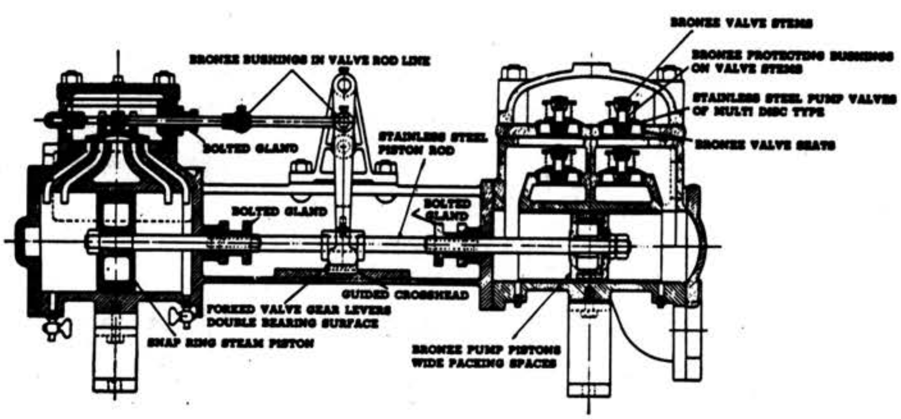

Fig. 1. Direct, double-acting, steam pump

The effects of pressure surges did not become a significant factor until the change-over from steam to diesel-driven pumps had taken place; the reason being that the steam cylinder provided a cushioning effect for the direct-acting pump by operating "short stroke" and therefore not giving a complete positive action to the pump. This effect permitted the steam to remain in the cylinder at the end of each stroke and thus absorb some of the shock of the high pressure surge much in the same manner as an automobile shock absorber smooths out the effect of an unexpected bump. In the diesel-driven pump, the full stroke has to be completed because of the obvious design of the crankshaft requiring a heavy flywheel that permits no damping effect for the surges of the pump.

The trend in recent years has been toward the high-speed multicylinder pump because of the demand for greater capacity, higher pressures, higher speeds, and a substantial decrease in pump weight per unit capacity. Although the surges produced by these high-speed pumps are comparatively small, they cannot be easily reduced since most desurging equipment is not effective because of its inertia when operating frequencies are above twenty cycles per second. Multicylinder pumps also present another source for trouble due to the tendency of high velocity waves to reflect in the pipe and thus build up high resonating pressures at strategic points.

It should be clear that the problem of pump pulsations is one that will become more important as changes in pipeline equipment take place, and will require necessary investigation to cope with changing conditions. The challenge before both manufacturer and operator of reciprocating pumps is that of recognizing the significance of the surges and attempting to reduce their unwanted effects. The material contained within this thesis is to verify some of the suspected effects of pressure surges and to be of value to future investigators by providing additional experimental information on the results that may be expected if surges are to be decreased or removed.

CHAPTER II

PREVIOUS INVESTIGATIONS

Pump and pipeline companies have always been hunting for new methods of improving the applicability, effectiveness, and efficiency of pumping facilitites. Their eagerness to learn of these methods has led to important contributions by interested investigators. Some of the information accumulated bears a definite relation to the aims of this report.

Before 1925, the literature on the subject of reciprocating pump surges was sketchy and scarce. The only work of significance which has come to the attention of the author was published in 1903 by Professor John Goodman of Leeds, England.[1] In this paper, Prof. Goodman clearly describes observations made on a plunger pump with and without an air vessel. The objects of his experiment were as follows:

> "1. How the "Slip" or the "Discharge Coefficient," of a pump, not fitted with a vacuum vessel, also the "Water Ram" pressure in the suction pipe are affected by:
>
> (a) A change of outlet or delivery pressure when the speed remains constant.
> (b) A change of speed when the delivery pressure remains constant.
> (c) A change in the length of the suction pipe with the other conditions remaining unchanged.
> (d) Running the pump without a suction valve.
>
> 2. The exact behavior as regards the opening and closing of the suction and delivery valves under various conditions of running.
>
> 3. The speed at which the plunger separates from the water during the early part of the stroke and catches it up later on, thereby producing a violent bang in the suction pipe.
>
> 4. The loss of pressure due to the friction of the water passing through the valves and passages of the pump.
>
> 5. The mechanical efficiency of the pump under various working pressures.
>
> 6. The effect of fitting a vacuum or air vessel to the suction pipe."

---

[1] Goodman, John, "Hydraulic Experiments on a Plunger Pump," Proceedings Institute of Mechanical Engineers, (Feb. 20, 1903), pp. 123-197.

The results of his experiments indicated that a pump could deliver up to 50 percent more volume than that displaced by the plunger. However, this does not indicate that the useful work done in delivering this quantity of water against the outlet pressure can be greater than the indicated work done in the pump barrel. His general conclusions were essentially the following:

> "1. That in a pump without a vacuum vessel, the quantity of water delivered depends on the speed of the pump and on a "Coefficient of Discharge".
>
> 2. Water-ram pressures on pumps not supplied with air or vacuum vessels may be serious and very much higher than theory gives.
>
> 3. Banging in suction-pipes of the pump is due to separation of the plunger from the water and their subsequent meeting.
>
> 4. That uncontrolled suction and delivery valves do not open and close in the simple manner they are supposed to do, except when the pump is running at slow speeds and when delivering against a moderate head."

Today, the significance of Prof. Goodman's paper is not in the results he obtained but rather in the interest he created in many of the engineers of that time. It was an attempt to explain some of the unusual phenomenon associated with reciprocating pumps which undoubtedly contributed greatly to the support or rejection of related theory.

In 1925, N. B. Delavan[2] reported on an investigation being conducted at Seneca Falls, New York, for the Goulds Manufacturing Company on the existence and effects of reciprocating pump surges. Although his interest was focused primarily on eliminating the costly damage to pipeline equipment by surges, they substantiated many basic facts that had up until that time been quite confusing. Mr. Delavan's article could be considered as an introduction or synopsis to the complete report published in 1929 by H. Diederichs and

---

[2] Delavan, N. B. "An Investigation Into the Cause of Breakage in Crude Oil Pipe-line Transportation Systems" Petroleum Development and Technology, A.I.M.E., Vol. 3, 1925, pp. 498-520.

P. D. Pomeroy.[3]

This final report was summarized exceptionally well by the editors of the American Society of Mechanical Engineers as dealing

> "First with the occurrence of surge or oscillating pressures in the field, in connection with the pumping of oil, resulting in some cases in serious damage. It next reports upon the experiments carried out at Seneca Falls in order to study the phenomenon while all the conditions of operation are under definite control. This is followed by a study of the theory which underlies the occurrence, and definite recommendations are given to eliminate it. It is pointed out that the means at hand for doing this are two fold: (a) the establishment of a proper relation between pump speed and length of discharge line, and (b) the use of air chambers. A diagram is given which shows the relation between length of line and the critical speed of duplex and triplex pumps, and the paper concludes with a discussion of air chamber design and operation."

J. W. Squire[4] of the Stanolind Pipe Line Company (now Service Pipeline Company) presented a valuable paper on pressure surges and vibration in reciprocating pump piping before the Petroleum Mechanical Engineering Conference of the A.S.M.E. at Amarillo, Texas, in 1948. His paper gave a brief background on what had been accomplished by other investigators and correlated their findings with what he had done. Mr. Squire pointed out the ineffectiveness of the surge chamber on high speed pumps and indicated that other means must be employed to eliminate the dangerous pulsations. His whole idea was to stop the surge at the source by redesigning the valves and other pump parts and also change the piping to avoid irregularities and thus give a more streamline effect to the flow. The paper was concerned primarily with multicylinder pumps and gave evidence that surges can be eliminated by properly designing the pump, manifold, and pipe layout.

---

[3] Diederichs, H. and Pomeroy, W. D. "The Occurrence and Elimination of Surge or Oscillating Pressures in Discharge Lines from Reciprocating Pumps" A.S.M.E. Transactions, Vol. 51, PET 51-2, 1929, pp. 9-49.

[4] Squire, J. W. "Pressure Surges and Vibration in Reciprocating Pump Piping", World Oil, Vol. 128 No. 12, March 1949, pp. 171-182.

W. E. Wilson[5] of Rapid City, South Dakota, has worked out an excellent method of comparing the performance characteristics of positive-displacement pumps and fluid motors on the basis of dimensionless-performance coefficients and dimensionless-efficiency curves. The advantage of this method is that the performance characteristics expressed in terms of coefficients permits a way to isolate the weaknesses of design and assists the redesigning to obtain better performance. The dimensionless-efficiency curves aid in presenting a wide range of data in compact form and provides a means for predicting the performance characteristics under almost any condition. Although this method of analysis is not employed in this thesis, its significance in presenting the performance data of positive displacement pumps is of great value in interpretating the results.

---

[5] Wilson, W. E. "Performance Criteria for Positive-Displacement Pumps and Fluid Motors", A.S.M.E. Transactions, Vol. 71, Feb. 1949, pp. 115-120.

CHAPTER III

STATEMENT OF PROBLEM

The primary purpose of this investigation is to determine the effect of pressure surges on the mechanical efficiency and general operation of a piston pump.

The mechanical efficiency of the pump is determined rather than the over-all efficiency because it has more practical significance since the efficiency of the driving motor is not included. The Test Codes of the Hydraulic Institute state that "The performance of pumps is to be based strictly on the actual mechanical power input to the pump and not on the arbitrary electrical measurements often used for determining the efficiency of the motor."[6] The mechanical efficiency which will be determined is the ratio of the water horsepower delivered, to the brake horsepower input.

Items which are included under the general operation of the pump are: (a) the instantaneous rate of flow and pressure in both suction and discharge lines, (b) horsepower input variation, and (c) the characteristic diagrams of pressure conditions as affected by the pulsating surges in both suction and discharge lines.

---

[6] Finch, V. C., Pump Handbook, National Press, 1948, pp. 145-159.

CHAPTER IV

ANALYSIS OF PROBLEM

Pressure surges are produced as a result of fluid being either accelerated or decelerated. In the piston pump, the pressure surge is caused by the sudden increase of velocity given to the fluid by the action of the piston. During the discharge stroke, the piston starts with a zero-forward velocity, increasing to a maximum at approximately midstroke, and decreasing to zero again at the end of the stroke. The velocity curve for the piston pump would be the top half of a true sine curve if it were not for the variation produced by the angularity effect of the crankshaft on the piston rod. In order to have a nonpulsating flow from a piston pump it is evident from Fig. 2 that an infinite number of cylinders would be necessary.

The characteristics of the discharge line will materially affect the pressure variations in the flow line. For a short line, where the inertia of the fluid column is small and the pipe friction is negligible, the volume entering the line at the pump will be equal to the volume discharged at the end of the line at the same instant since no compression of the liquid can take place. In a long line, there is a compression of the liquid during the middle of the discharge stroke owing to the inertia of the fluid column and the pipe friction, which produces a pressure at the start of the line in excess of the average discharge pressure. During the start and end of the discharge stroke, the pressure is lower than the average discharge pressure because of the moving column attempting to run away from the pump. The pressure curve will hence show alternate increases and decreases of pressure at the pump end of the line in a periodic manner. The pressure surges will travel at sonic velocity through the fluid from one end of the pipe to the other. The commonly accepted relationship for the celerity of a propagating pressure wave

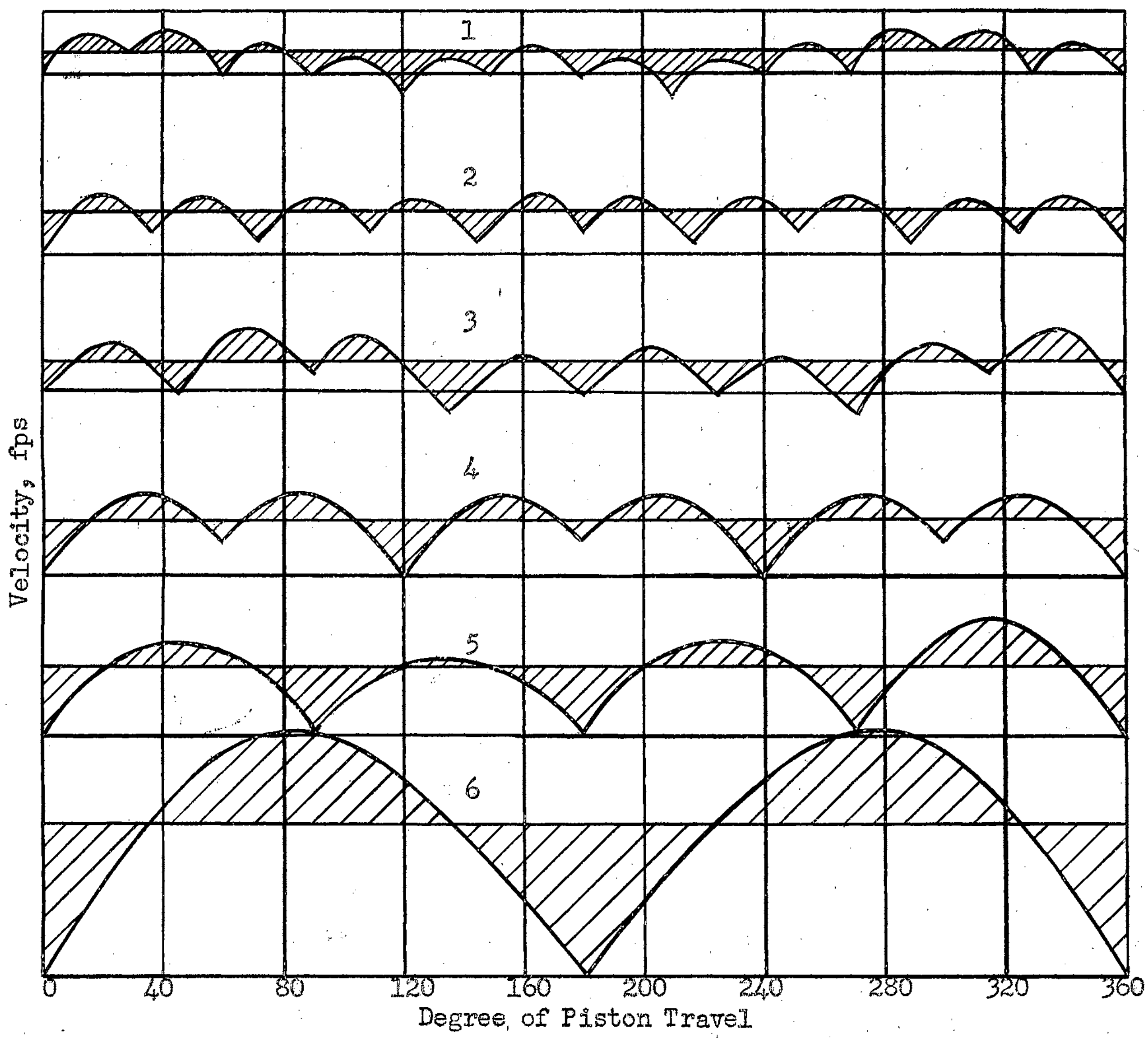

Fig. 2. Variation in Discharge Flow per Revolution of Single Acting Pumps. Number of cylinders in pump producing curve (1) 12, (2) 10, (3) 8, (4) 6, (5) 4, and (6) 2.

in a fluid contained in an elastic pipeline is

$$c = \frac{12}{\sqrt{\frac{w}{g}\left(\frac{1}{K} + \frac{d}{E\ t}\right)}} ,$$

in which c = celerity of pressure wave, ft per sec;
w = specific weight of fluid, lbs per cu ft;
g = acceleration due to gravity, ft per sec^2;
K = bulk modulus of elasticity of pipe, lbs per in.^{2};
E = linear modulus of elasticity of pipe, lbs per in.^{2};
d = nominal diameter of pipe, in.; and
t = wall thickness of pipe, in.

In the case of water at 70°F in a standard 2-inch galvanized steel pipe with a nominal diameter of 2.07 in., a bulk modulus of 300,000 lbs per in.^{2}, a linear modulus of elasticity of 30,000,000 lbs per in.^{2}, and a wall thickness of 0.154 in.; the celerity of the pressure wave would be

$$c = \frac{12}{\sqrt{\frac{62.3}{32.2}\left(\frac{1}{300{,}000} + \frac{2.07}{30{,}000{,}000 \times 0.154}\right)}}$$

$$c = 4{,}680 \text{ ft per sec.}$$

A definition of a long and short line has never been clearly defined to the author's knowledge but from necessity would be limited by the wave length of the pressure surge. A short line would hence be of such a length that

$$L = c/2f \quad \text{or less}$$

in which L = length of pipe, ft;
c = celerity of pressure wave, ft per sec; and
f = frequency of pump, cycles per sec.

For a long line the length would be

$$L = 2c/f \quad \text{or more.}$$

The pulsations produced by a piston pump can be substantially reduced by installing a surge alleviator adjacent to the discharge end of the pump. The action of such equipment tends to even out the surge by absorbing the high pressure and filling in the gap on the resulting low-pressure portion.

The common air chamber operates on the principle that during the middle part of the delivery stroke, while the fluid is being forced into the delivery pipe by the piston with a velocity above average, the additional fluid will flow into the air chamber. When the velocity of the fluid is below average at the ends of the stroke, the fluid will flow from the air chamber to make up the deficiency. The flow in the pipe beyond the air chamber is therefore maintained approximately constant.

The work that can theoretically be saved by attaching a desurger on the pump is accomplished by reducing the long fluid column that must be accelerated with each stroke. The only volume of fluid that is accelerated at each stroke will then be in the discharge line between the desurger and the pump. The acceleration head required for this length can be considered as insignificant.

Another way of understanding how work is saved by fitting a desurger to a pump is by graphical means. Consider the ideal basic curves for a single-cylinder double-acting pump. Without a desurger attached, the velocity-time curve would be similar to Fig. 3 which would approximate the pressure-time curve in the discharge line of the pump. The quantity-time curve for the pump will be the same as the velocity-time relationship shown in Fig. 3 since the rate of flow from the piston is equal to the area of the piston multiplied by the velocity of the piston less any losses from leakage.

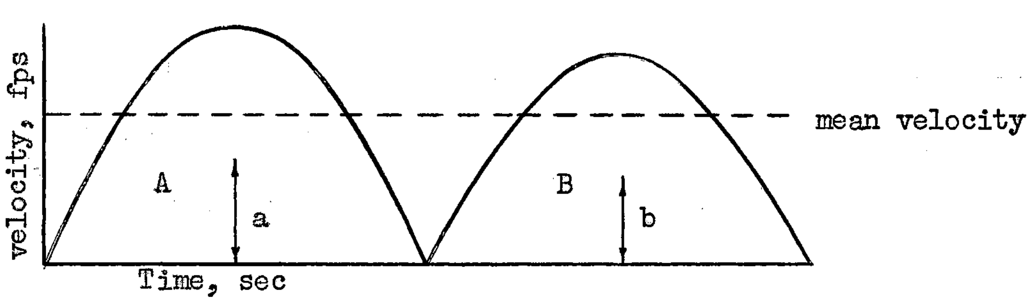

Fig. 3. Time-velocity curve without desurger

Part B of Fig. 3 will not be as high as part A because the piston rod will take up a portion of the volume normally filled with fluid. If the pump had a desurger attached, the ideal curve would resemble Fig. 4.

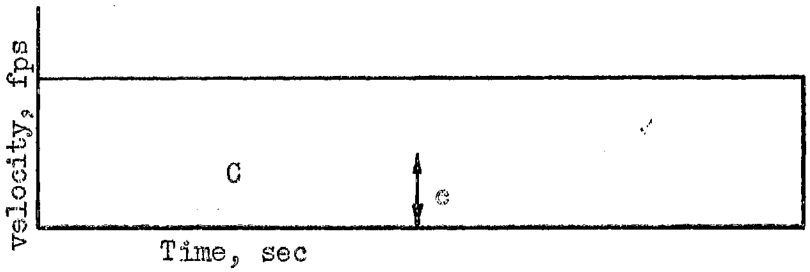

Fig. 4. Time-velocity curve with ideal desurger

Figure 4 appears as a rectangle because the portion of the curves above the average pressure of Fig. 3 have moved horizontally and down to fill in the hollows as illustrated in Fig. 5.

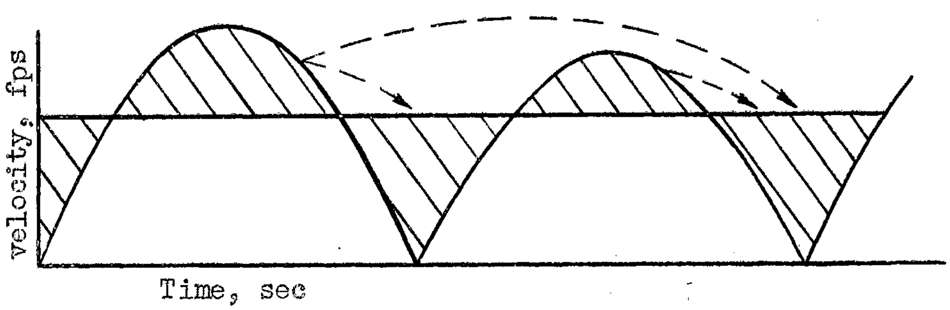

Fig. 5. Time-velocity curve illustrating the manner in which higher portions of a curve fills the lower portions to obtain an ideal desurger curve.

It is quite evident from Figs. 3 and 4 that area A multiplied by the distance from its centroid to the base plus area B multiplied by the distance from its centroid to the base is more than area C multiplied by the distance from its centroid to the base. This would indicate that less work is required to obtain the same discharge from a pump with a desurger than from one without a desurger.

The thought that gave the first incentive to this investigation was that it should take less power to move a load constantly than is required to start and stop it endlessly, making it reasonable to expect better economy, smoother action, and increased capacity from a given pump.

Very little has been mentioned concerning the suction side of the pump because this investigation has been concerned primarily with the reciprocating surges of the discharge side; however the theory of a piston pump would certainly be lacking if the characteristics of the suction side were not included.

The surge and water hammer produced on the suction side of the pump have little relation to that produced on the discharge side. The suction surges are dependent materially on the size and length of the suction pipe and on the average and instantaneous velocity of the fluid flowing. The water hammer on the suction side is attributed to the almost instantaneous closure of the pump valve against the fluid flowing toward it. In this case, the surge is due to the rapid deceleration of the fluid whereas on the discharge side it is due to the rapid acceleration of the fluid by the piston.

The suction side also affects other operating factors of considerable significance. The rapid deceleration of the fluid in the suction line will cause a water hammer pressure equal to

$$p = \frac{W}{g} \Delta V \quad c$$

in which
p = water hammer pressure, lbs per ft^2;
w = specific weight of fluid, lbs per cu ft;
g = acceleration due to gravity, ft per sec^2;
ΔV = difference in fluid velocity, ft per sec; and
c = celerity of pressure wave, ft per sec.

The water hammer pressure will last for a period of

$$T = 2L/c$$

in which
T = period of water hammer pressure, sec;
L = length of pipe, ft; and
c = celerity of pressure wave, ft per sec.

It is obvious that at low discharge pressures, the pump could deliver a greater volume of fluid than the volume displaced by the piston since the fluid would be driven through the valves. This condition would result in giving a volumetric efficiency greater than unity. This situation exists more frequently in long suction lines since the water hammer pressure lasts for a time interval of 2L/c which will allow more fluid to flow through the valves.

In the tests conducted for this investigation, the suction pressure and surge were maintained nearly constant by installing an airdome near the pump on the suction side and keeping the suction tank level unchanged. At the speeds the pump was tested, it is believed that the cylinder of the pump was full of fluid at each stroke.

Secondary surges produced by reflecting surfaces play an important part in confusing the results obtained by testing pumps. These surges can be reflected from elbows, valves, various connections, and discharge tanks. In long lines they are a source of resonating pressures which may build up and may cause the containing pipe to be stressed beyond the safe limit. Since this is a fluctuating load, fatigue failure may result in the weaker parts of the line.

Secondary surges have been minimized in this investigation by using short discharge lines as verified by an electro-pressure graph and Cathode-ray oscilloscope. The curves shown by this instrument connected to the test pump resemble Fig. 6.

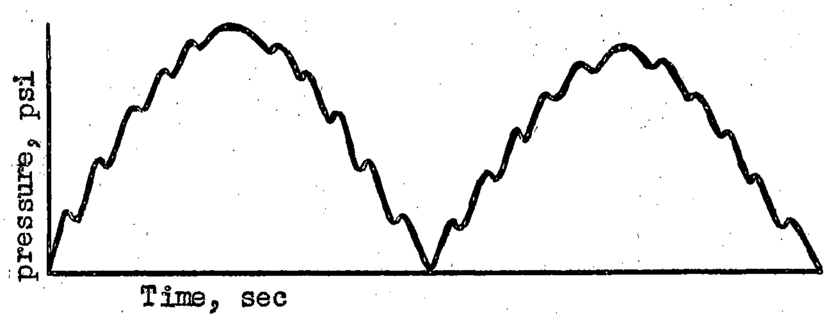

Fig. 6. Pressure-time curve. Effect of reflecting waves as obtained with a short discharge pipe.

On a pump with a long discharge line with reflecting surfaces the curves would be similar to Fig. 7.

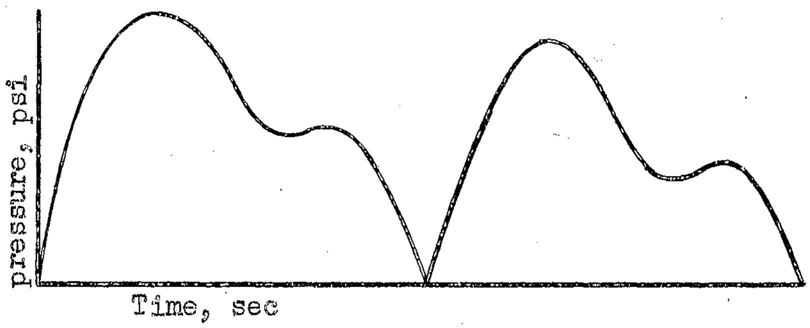

Fig. 7. Pressure-time curve showing effects of large reflecting surges on characteristic curve using a long discharge line.

CHAPTER V

APPARATUS AND EQUIPMENT

All tests were made using a $2\frac{1}{2}$" x 3" double-acting, single-cylinder piston pump manufactured by the F. E. Meyers Pump Company of Ashland, Ohio, which was generously loaned to the Hydraulics Laboratory of Oklahoma A. and M. College for experimental purposes. The power was furnished by a one-horsepower single-phase electric motor, for which accurate efficiency curves were available, and the power was transmitted to the pump by V-belts. The air domes which were on the original pump were either removed entirely or filled with water so that the surge could be closely controlled.

The suction pipe contained one bend 18" from the pump that allowed the suction pipe to be suspended in an open tank located about $8\frac{1}{2}$ feet below the center line of the pump. An air dome was connected at the top of the suction pipe with a gate valve to separate the chamber from the pipe. Bourdon pressure gauges were arranged between the air dome and the pump with one gauge throttled in order to obtain an average suction pressure and the other allowed to oscillate to indicate the magnitude of the surge. Outlets were provided for attaching an engine indicator and an electro-pressure graph pickup between the pump and the air dome.

The discharge line was designed to allow the operation of either an air dome or a Fluidynamic Desurger. Suitable valves permitted this equipment to be shut off and thus the water was pumped without any of the pressure surges being removed. Two Bourdon gauges were connected between the desurging equipment and the pump to provide an average discharge pressure and an instantaneous pressure reading. Outlets were again provided for indicators and electro-pressure graph pickup between the pump and the surge equipment. The valve for regulating the delivery pressure was placed beyond the air dome as indicated in Fig. 8. A spring loaded safety valve was installed to protect the equipment if mistakes were made in operation of the valves. Under normal operation this safety valve remained closed.

The rate of flow was determined by using the floating beam method with the water being discharged into a steel barrel mounted on a Fairbanks-Morse weighing scales. A valve at the bottom of the weighing tank permitted the water to be drained when not being weighed.

In order to measure the watt-hour input to the electric motor, a 110 volt kilowatt-hour meter was found necessary to integrate the surging current. The meter was calibrated with various constant loads to facilitate ease of reading for short test runs. A conventional watt meter was employed to measure the watt surge that was reflected on the power supply by the reciprocating motion of the pump.

Indicators used in connection with the experiments included a Chromatic Cantilever Spring Bachrach Indicator obtained from the University of Wisconsin, also two Coil-Spring Bachrach Engine Indicators belonging to the Oklahoma A. and M. Hydraulics Laboratory.

Electro-pressure graph pickups connected to a Cathode Ray Oscilloscope permitted the viewing of surge waves produced by the pump. Stop watches and revolution counters were also essential parts of the testing equipment.

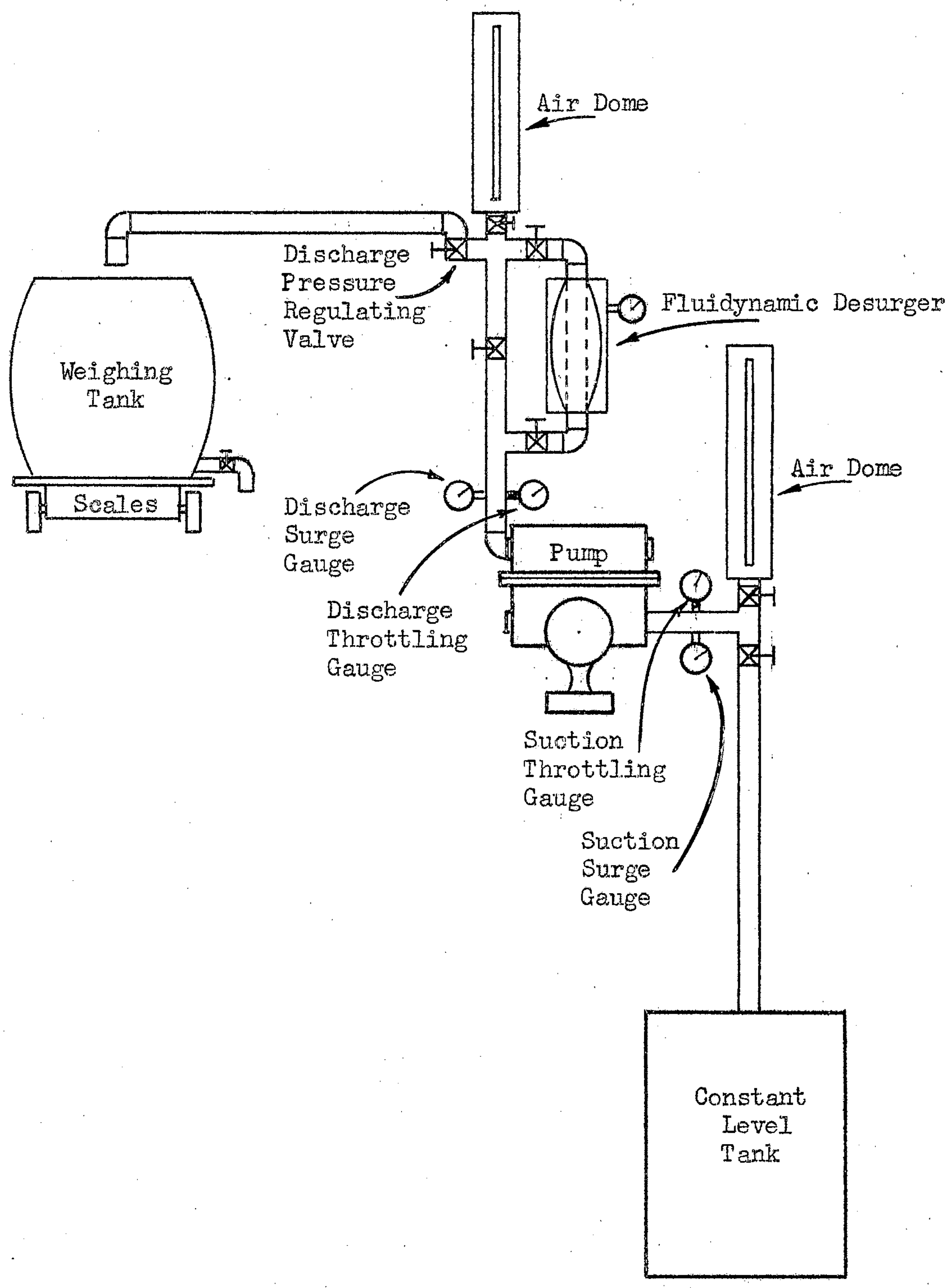

Fig. 8. Arrangement of Apparatus

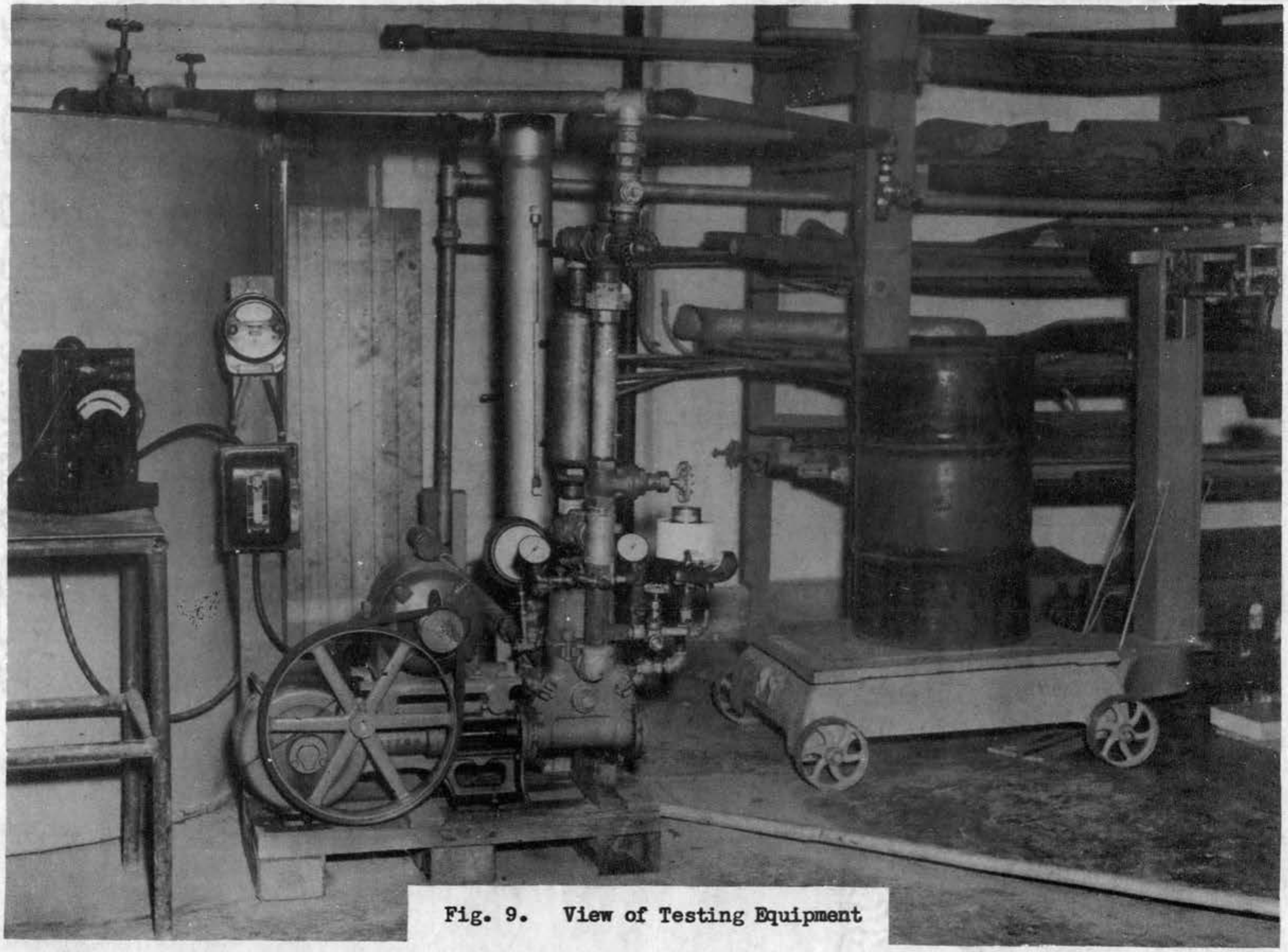

Fig. 9. View of Testing Equipment

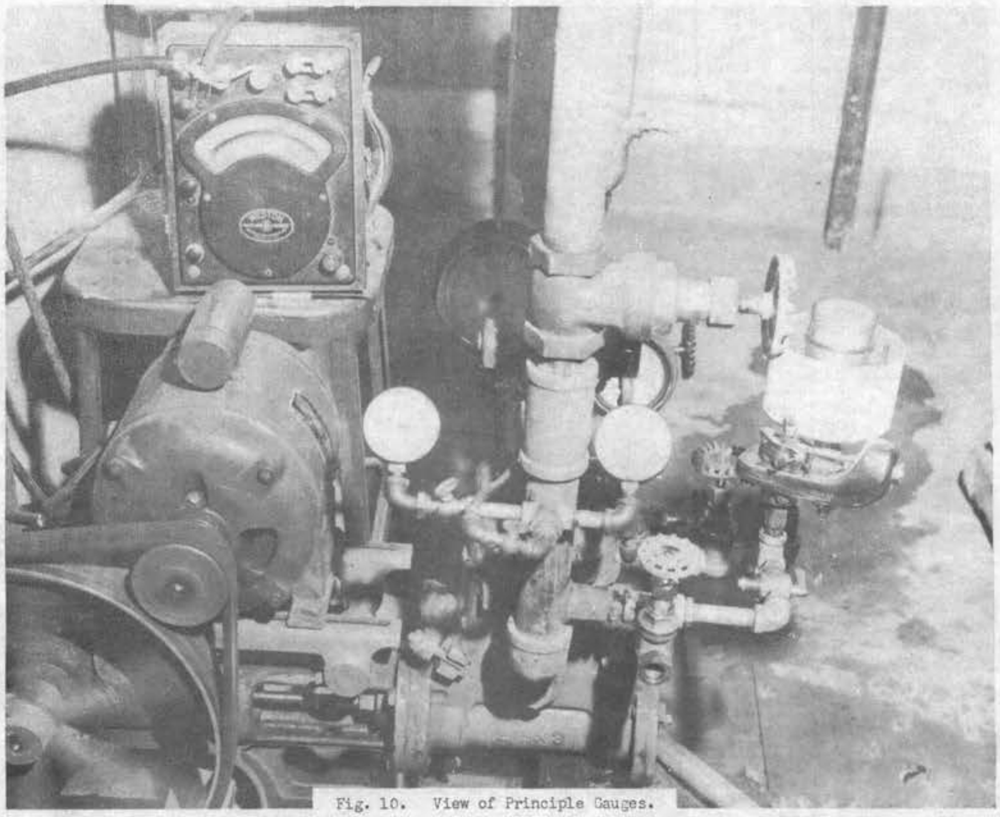

Fig. 10. View of Principle Gauges.

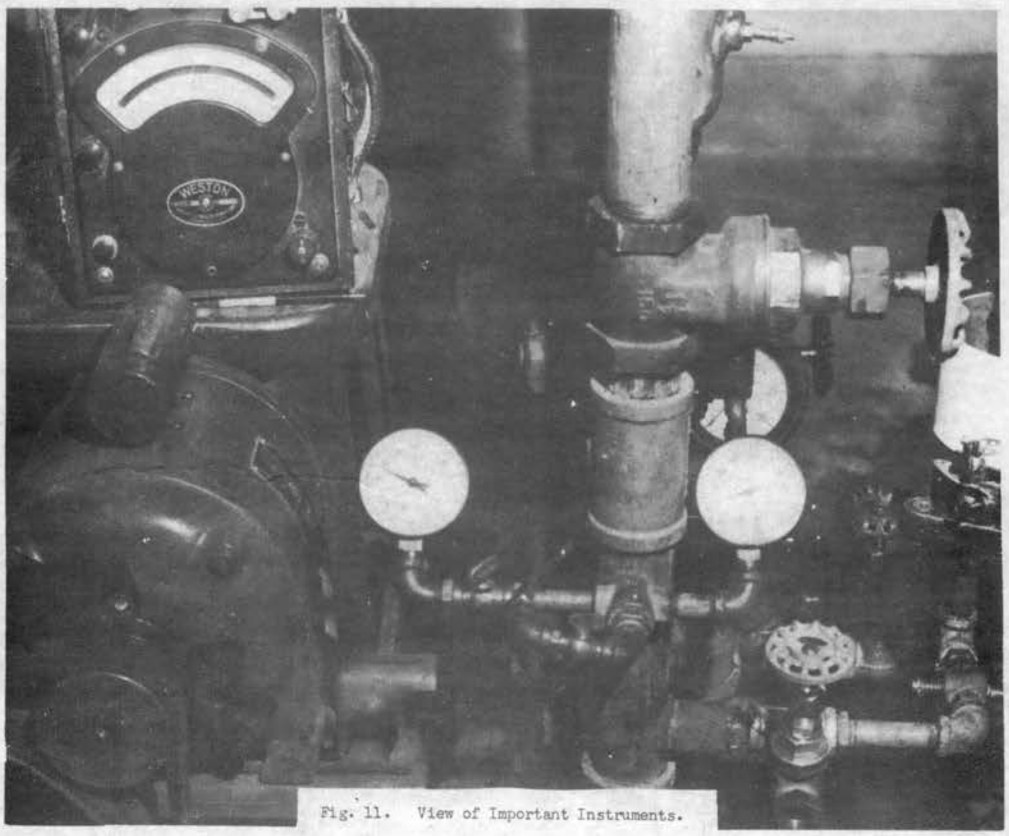

Fig. 11. View of Important Instruments.

CHAPTER VI

METHOD OF CARRYING OUT TESTS

In order to determine the effect of surges on the efficiency and operation of a piston pump, it was necessary to alter the amount of surge at various discharge pressures. The magnitude of the surge was varied by the adjustment of valves that changed the degree of operation of the desurgers. For a given magnitude of the discharge surge and average discharge pressure, the average pump speed, the quantity flowing per unit time, the average watts supplied to the motor, and the values of the average suction pressure and the surge on the suction side were obtained.

The magnitude of the surge and the average pressure for both suction and discharge sides of the pump were measured by Bourdon Gauges and engine indicators. The Bourdon gauges were calibrated with the conventional dead weight tester to insure accurate and consistant readings. Care was exercised in obtaining the total sweep of the surge gauges because it is very hard to know the exact limits of the hand swing. At the ends of the sweep, there appears a flicker or shadow that must be considered as part of the surge. A Bachrach Cantilever Spring type engine indicator was employed to check the magnitude of the surge by measuring the height of the characteristic curve and referring to the proper scale.

The average speed of the pump was measured by using a revolution counter and measuring the revolutions turned in one minute. Another instrument employed to measure the speed of the pump was a "Jagabi" hand tachoscope which consists of a stop watch and revolution counter meshed together in order that one stop lever would control both mechanisms. A tachometer was not applicable because the surge of the pump was reflected directly on the speed of the pump. No effort was made to maintain an exactly constant pump speed.

For the average watt input to the motor, it was necessary to clock the time required for 10 revolutions of the metal disk that revolved within the watt-hour meter. The calibration constant was found by connecting the meter with various constant loads and a regular watt meter. Knowing the watts pulled by the load and counting the disk revolutions for a certain period, the constant was calculated.

The quantity of water flowing was measured by the floating-beam method. This consists of starting the watch when the beam rises into balance, then adding a weight of 100 pounds to the scales and stopping the watch when enough water had emptied into the barrel to again balance the scales. This method is approved by the Hydraulics Institute and is accurate to one-fourth of one percent.

The same process of testing was used with each variation of average discharge pressure and surge pressure in order to obtain comparable data.

Original Experimental Data

Dates Tests Were Made:		
	Run #1	August 23, 1950
	Run #2	September 4, 1950
	Run #3	September 30, 1950
	Run #4	October 7, 1950
	Run #5	October 20, 1950

	Run				
Throttled P_d, psi		140		130	130
Discharge Surge, psi		20		20	40
Throttled P_s, psi		-3.7		-3.7	-3.7
Suction Surge, psi		4.9		4.9	4.9
Time-100# water, sec	3	87.6		87.6	87.2
	4	88.0		87.7	87.3
Pump Speed, rpm	3	344		347	345
	4	345		347	347
Time-10 KW rev, sec	3	9.7		10.0	9.8
	4	9.4		10.1	9.8
Power Surge, watts	3	190		190	190
	4	200		190	200

Throttled P_d, psi		120	120	120
Discharge Surge, psi		20	40	60
Throttled P_s, psi		-3.7	-3.7	-3.7
Suction Surge, psi		4.9	4.9	4.9
Time-100# water, sec	2	86.6	—	86.9
	3	86.8	86.8	87.2
	4	87.5	87.5	87.2
Pump Speed, rpm	2	345	—	347
	3	348	347	347
	4	350	348	346
	5	—	347	346
Time-10 KW rev, sec	2	10.5	—	10.6
	3	10.7	10.4	10.2
	4	10.5	10.4	10.2
Power Surge, watts	2	180	—	200
	3	180	190	200
	4	190	190	200

Throttled P_d, psi		110	110	110	110
Discharge Surge, psi		20	40	60	80
Throttled P_s, psi		-3.7	-3.7	-3.7	-3.7
Suction Surge, psi		4.9	4.9	4.9	4.9
Time-100# water, sec	2	88.1	86.8	86.8	87.8
	3	85.8	86.0	86.4	86.8
	4	86.0	86.0	86.4	86.8
Pump Speed, rpm	2	348	347	346	345
	3	348	350	348	346
	4	350	350	349	349
	5	—	349	348	348
Time-10 KW rev, sec	2	10.7	10.8	10.4	10.0
	3	11.0	10.9	10.8	10.6
	4	11.2	11.0	10.8	10.6
Power Surge, watts	2	180	180	190	200
	3	170	180	190	200
	4	180	180	190	210

Throttled P_d, psi		100	100	100	100	100
Discharge Surge, psi		20	40	60	80	100
Throttled P_s, psi		-3.7	-3.7	-.37	-3.7	-3.7
Suction Surge, psi		4.9	4.9	4.9	4.9	4.9
Time-100# water, sec	1	85.7	85.7	86.0	86.3	86.4
	2	86.2	87.0	86.8	—	—
	3	85.4	85.4	85.6	85.7	85.7
	4	86.2	86.2	86.0	86.2	—
Pump Speed, rpm	2	352	348	348	—	—
	3	351	351	350	352	351
	4	351	351	350	351	—
	5	352	351	350	350	349
Time-10 KW rev, sec	1	11.8	11.6	11.4	11.2	11.0
	2	11.2	11.0	10.8	—	—
	3	12.0	11.8	11.6	11.2	11.0
	4	11.8	11.8	11.6	11.4	—
Power Surge, watts	2	170	180	190	—	—
	3	160	170	180	190	190
	4	160	170	180	200	—

Throttled P_d, psi		90	90	90	90	90	90	90
Discharge Surge, psi		10	20	40	60	80	100	120
Throttled P_s, psi		-3.7	-3.7	-3.7	-3.7	-3.7	-3.7	-3.7
Suction Surge, psi		4.9	4.9	4.9	4.9	4.9	4.9	4.9
Time-100# water, sec	1	—	85.6	85.9	86.0	86.2	86.4	86.6
	2	86.4	86.1	86.6	—	—	—	—
	3	—	85.1	85.2	85.3	85.3	85.1	—
	4	85.3	85.9	85.6	85.6	86.0	86.2	—
Pump Speed, rpm	2	354	354	353	—	—	—	—
	3	—	352	352	351	352	351	—
	4	353	352	352	351	350	350	—
	5	—	353	352	352	351	351	—
Time-10 KW rev, sec	1	—	12.8	12.6	12.2	12.0	11.8	11.6
	2	12.0	12.0	12.2	—	—	—	—
	3	—	12.6	12.4	12.2	12.0	11.8	—
	4	12.5	12.5	12.4	12.2	12.0	11.8	—
Power Surge, watts	2	150	160	160	—	—	—	—
	3	—	150	160	180	180	190	—
	4	140	150	160	170	190	210	—

Throttled P_d, psi		80	80	80	80	80	80	80
Discharge Surge, psi		10	20	40	60	80	100	120
Throttled P_s, psi		-3.7	-3.7	-3.7	-3.7	-3.7	-3.7	-3.7
Suction Surge, psi		4.9	4.9	4.9	4.9	4.9	4.9	4.9
Time-100# water, sec	1	—	85.0	85.2	85.3	85.4	85.8	86.4
	2	85.5	86.2	—	—	—	—	—
	3	84.6	85.9	84.8	84.8	84.8	84.9	—
	4	85.1	85.6	85.4	85.4	85.3	86.1	86.8
Pump Speed, rpm	1	—	354	354	352	352	351	350
	2	357	353	—	—	—	—	—
	3	352	355	354	356	355	354	—
	4	355	354	353	353	353	353	352
	5	—	354	354	353	352	351	—
Time-10 KW rev, sec	1	—	13.4	13.2	13.0	12.8	12.6	12.2
	2	13.0	12.8	—	—	—	—	—
	3	13.8	13.6	13.2	13.0	12.8	12.6	—
	4	13.4	13.3	13.2	13.0	12.8	12.6	—
Power Surge, watts	2	130	140	—	—	—	—	—
	3	130	140	150	160	170	180	—
	4	130	140	150	160	180	190	190

Throttled P_d, psi		70	70	70	70	70	70	70
Discharge Surge, psi		10	20	40	60	80	100	120
Throttled P_s, psi		-3.7	-3.7	-3.7	-3.7	-3.7	-3.7	-3.7
Suction Surge, psi		4.9	4.9	4.9	4.9	4.9	4.9	4.9
Time-100# water, sec	1	85.2	85.6	85.8	86.0	86.1	86.4	86.7
	2	85.3	84.9	—	—	—	—	—
	3	86.1	84.6	84.8	85.5	85.7	85.0	—
	4	84.6	84.9	84.6	85.0	85.1	84.9	—
Pump Speed, rpm	1	358	356	355	354	353	353	352
	2	357	357	—	—	—	—	—
	3	355	354	352	353	353	353	—
	4	356	355	356	355	354	354	—
	5	—	354	354	353	353	353	—
Time-10 KW rev, sec	1	14.2	14.0	13.8	13.5	13.2	13.0	12.6
	2	13.8	13.8	—	—	—	—	—
	3	14.5	14.2	13.8	13.6	13.4	13.2	—
	4	14.6	14.4	14.0	13.7	13.6	13.5	—
Power Surge, watts	2	120	130	—	—	—	—	—
	3	130	130	140	150	160	180	—
	4	120	120	130	150	160	170	—

Throttled P_d, psi		60	60	60	60	60
Discharge Surge, psi		10	20	40	60	80
Throttled P_s, psi		-3.7	-3.7	-3.7	-3.7	-3.7
Suction Surge, psi		4.9	4.9	4.9	4.9	4.9
Time-100# water, sec	1	85.9	85.3	86.2	86.4	86.2
	2	85.8	86.0	—	—	—
	3	83.3	84.4	84.2	84.0	84.8
	4	84.3	85.2	84.9	85.0	84.2
Pump Speed, rpm	1	359	358	357	355	355
	2	355	356	—	—	—
	3	357	355	353	352	353
	4	356	356	356	356	356
	5	357	356	355	354	353
Time-10 KW rev, sec	1	15.4	15.2	15.0	15.0	14.2
	2	15.0	14.8	—	—	—
	3	15.2	15.0	14.8	14.4	14.2
	4	15.4	15.2	15.0	14.8	14.2
Power Surge, watts	1	110	110	120	130	150
	2	100	110	—	—	—
	3	110	120	130	140	150
	4	110	110	120	130	150

Throttled P_d, psi		50	50	50	50	50
Discharge Surge, psi		10	20	40	60	80
Throttled P_s, psi		-3.7	-3.7	-3.7	-3.7	-3.7
Suction Surge, psi		4.9	4.9	4.9	4.9	4.9
Time-100# water, sec	2	83.6	84.2	—	—	—
	3	83.7	83.5	83.6	84.8	83.5
	4	83.7	84.1	84.0	84.4	83.8
Pump Speed, rpm	2	360	360	—	—	—
	3	357	356	356	355	355
	4	357	357	357	355	356
	5	358	357	356	356	355
Time-10 KW rev, sec	2	16.0	16.2	—	—	—
	3	16.2	16.2	16.0	15.8	15.6
	4	16.4	16.3	16.2	15.8	15.8
Power Surge, watts	2	100	100	—	—	—
	3	90	100	110	120	140
	4	100	100	110	120	130

Throttled P_d, psi		40	40	40	40
Discharge Surge, psi		10	20	40	60
Throttled P_s, psi		-3.7	-3.7	-3.7	-3.7
Suction Surge, pai		4.9	4.9	4.9	4.9
Time-100# water, sec	2	83.4	84.9	—	—
	3	83.7	83.4	83.1	83.9
	4	83.7	83.6	83.4	83.2
Pump Speed, rpm	2	359	356	—	—
	3	358	358	355	355
	4	358	359	357	357
	5	358	358	357	356
Time-10 KW rev, sec	2	17.3	17.4	—	—
	3	17.8	17.2	17.0	17.0
	4	17.6	17.4	17.2	17.0
Power Surge, watts	2	80	90	—	—
	3	80	90	100	110
	4	80	80	90	100

Throttled P_d, psi		30	30	30
Discharge Surge, psi		10	20	40
Throttled P_s, psi		-3.7	-3.7	-3.7
Suction Surge, psi		4.9	4.9	4.9
Time-100# water, sec	2	84.5	84.0	—
	3	82.6	83.3	83.0
	4	82.8	83.4	83.0
Pump Speed, rpm	2	359	359	—
	3	358	358	357
	4	358	357	359
	5	359	358	358
Time-10 KW rev, sec	2	18.6	18.8	—
	3	19.2	18.8	18.6
	4	19.0	19.0	18.4
Power Surge, watts	2	60	70	—
	3	60	70	80
	4	70	70	80

Throttled P_d, psi		20	20
Discharge Surge, psi		10	20
Throttled P_s, psi		-3.7	-3.7
Suction Surge, psi		4.9	4.9
Time-100# water, sec	2	—	—
	3	82.3	82.2
	4	82.6	—
Pump Speed, rpm	2	—	—
	3	358	358
	4	360	—
	5	360	359
Time-10 KW rev, sec	2	—	—
	3	20.6	20.4
	4	20.4	—
Power Surge, watts	2	—	—
	3	50	60
	4	50	—

Calculated Data

Rate of Flow, lb/sec

Throttled P_d, psi	Run No.	Discharge Pressure Surge, psi: 10	20	40	60	80	100	120
20	3	1.215	1.218					
	4	1.210						
30	2	1.184	1.190					
	3	1.210	1.200	1.205				
	4	1.208	1.200	1.206				
40	2	1.200	1.180					
	3	1.195	1.200	1.220	1.202			
50	2	1.196	1.189					
	3	1.195	1.199	1.195	1.180	1.198		
	4	1.195	1.189	1.190	1.185			
60	1	1.165	1.172	1.161	1.159	1.161		
	2	1.165	1.164					
	3	1.200	1.185	1.184	1.190	1.179		
	4	1.188	1.175	1.180	1.176	1.175		
70	1	1.193	1.187	1.185	1.182	1.180	1.178	1.174
	2	1.172	1.179					
	3	1.161	1.182	1.180	1.170	1.168	1.177	
	4	1.182	1.179	1.182	1.150	1.174	1.180	
80	1		1.177	1.173	1.172	1.170	1.166	1.158
	2	1.170	1.161					
	3	1.182	1.165	1.180	1.180	1.180	1.179	
	4	1.175	1.169	1.172	1.173	1.172	1.161	1.152
90	1		1.169	1.164	1.162	1.160	1.158	1.155
	2	1.158	1.160	1.155				
	3		1.175	1.173	1.171	1.171	1.175	
	4	1.173	1.165	1.168	1.168	1.163	1.160	
100	1		1.172	1.168	1.162	1.159	1.158	
	2		1.161	1.150	1.151			
	3	1.171	1.171	1.168	1.166	1.166		
	4		1.160	1.160	1.164	1.160		
110	2		1.135	1.151	1.151	1.140		
	3		1.165	1.161	1.158	1.156		
	4		1.164	1.164	1.158	1.151		
120	2		1.155		1.151			
	3		1.151	1.151	1.148			
	4		1.143	1.143	1.148			
130	3		1.140	1.148				
	4		1.141	1.145				
140	3		1.141					
	4		1.136					

Pump Input, hp

Throttled P_d, psi	Run No.	Discharge Pressure Surge, psi 10	20	40	60	80	100	120
20	3	.425	.430					
	4	.430						
30	2	.490	.480					
	3	.470	.485	.505				
	4	.480	.480	.500				
40	2	.540	.538					
	3	.530	.545	.555	.555			
	4	.530	.540	.550	.555			
50	2	.595	.590					
	3	.590	.590	.595	.605	.615		
	4	.580	.585	.590	.605	.605		
60	1	.625	.635	.645	.645	.690	.710	.730
	2	.645	.655					
	3	.635	.645	.660	.680	.690		
	4	.625	.630	.635	.645	.655	.665	
70	1	.690	.700	.710	.730	.750	.760	.780
	2	.715	.715					
	3	.675	.690	.710	.730	.740	.750	
	4	.665	.680	.700	.720	.730	.735	
80	1		.740	.750	.765	.780	.795	.820
	2	.765	.780	.780		.795		
	3	.710	.730	.750	.770	.780	.795	
	4	.740	.755	.760	.770	.785	.795	.800
90	1		.780	.795	.820	.835	.850	.875
	2	.835	.835	.820				
	3		.795	.810	.820	.835	.850	
	4	.800	.800	.805	.820	.835	.850	
100	1		.850	.875	.885	.900	.920	.940
	2		.900	.920	.940			
	3		.835	.865	.865	.905	.920	
	4		.850	.850	.865	.880		
110	2		.950	.940	.965	.966		
	3		.920	.930	.940	.960		
	4		.900	.920	.935	.960		
120	2		1.050	1.030	1.040			
	3		.950	.970	1.000			
	4		.970	.975	.995			
130	3		1.150	1.350				
	4		1.050	1.350				
140	3		1.450					
	4		1.800					

Calculated Results

Throttled P_d, psi	20	20
Discharge Surge, psi	10	10
Throttled P_s, psi	-3.7	-3.7
Rate of Flow, lb/sec	1.214	1.213
Speed of Pump, rpm	359.1	358.5
Pump Input, hp	.427	.433
Work Output, ft	55.8	55.8
Water HP Output, hp	.1222	.1221
Mech. Efficiency, %	28.65	28.2
V_t Pumped, cuft/sec	.02042	.0204
V_a Pumped, cuft/sec	.0195	.01945
Vol. Efficiency, %	95.3	95.6

Throttled P_d, psi	30	30	30
Discharge Surge, psi	10	20	40
Throttled P_s, psi	-3.7	-3.7	-3.7
Rate of Flow, lb/sec	1.207	1.2065	1.206
Speed of Pump, rpm	358.9	358.35	357.4
Pump Input, hp	.478	.485	.494
Work Output, ft	78	78	78
Water HP Output, hp	.1696	.1695	.1694
Mech. Efficiency, %	35.45	34.95	34.32
V_t Pumped, cuft/sec	.0204	.02039	.0203
V_a Pumped, cuft/sec	.01937	.01935	.01932
Vol. Efficiency, %	95.0	95.1	95.3

Throttled P_d, psi	40	40	40	40
Discharge Surge, psi	10	20	40	60
Throttled P_s, psi	-3.7	-3.7	-3.7	-3.7
Rate of Flow, lb/sec	1.201	0.200	1.1995	1.990
Speed of Pump, rpm	358.4	357.9	357.1	356.4
Pump Input, hp	.529	.5375	.5475	.557
Work Output, ft	102	102	102	102
Water HP Output, hp	.2228	.2225	.2222	.222
Mech. Efficiency, %	42.15	41.4	40.6	39.82
V_t Pumped, cuft/sec	.02039	.02038	.02028	.02025
V_a Pumped, cuft/sec	.01929	.01926	.01923	.01920
Vol. Efficiency, %	94.6	94.7	94.8	95.0

Throttled P_d, psi	50	50	50	50	50
Discharge Surge, psi	10	20	40	60	80
Throttled P_s, osi	-3.7	-3.7	-3.7	-3.7	-3.7
Rate of Flow, lb/sec	1.195	1.194	1.193	1.192	1.190
Speed of Pump, rpm	357.8	357.3	356.4	355.8	355.5
Pump Input, hp	.580	.590	.603	.614	.625
Work Output, ft	125	125	125	125	125
Water HP Output, hp	.2718	.2715	.271	.2708	.2703
Mech. Efficiency, %	46.8	46.0	44.9	44.1	43.3
V_t Pumped, cuft/sec	.02038	.02033	.12022	.12021	.02020
V_a Pumped, cuft/sec	.01919	.01918	.01917	.01915	.01912
Vol. Efficiency, %	94.3	94.4	94.6	94.7	94.75

Throttled P_d, psi	60	60	60	60	60	60
Discharge Surge, psi	10	20	40	60	80	100
Throttled P_s, psi	-3.7	-3.7	-3.7	-3.7	-3.7	-3.7
Rate of Flow, lb/sec	1.189	1.188	1.187	1.186	1.184	1.183
Speed of Pump, rpm	356.9	356.4	355.6	355.0	354.6	354.5
Pump Input, hp	.632	.645	.657	.668	.680	.694
Work Output, ft	148	148	148	148	148	148
Water HP Output, hp	.320	.3197	.3193	.319	.3186	.3183
Mech. Efficiency, %	50.6	49.6	48.6	47.7	46.8	45.9
V_t Pumped, cuft/sec	.02035	.0203	.0202	.02019	.02018	.02017
V_a Pumped, cuft/sec	.01906	.019045	.01903	.01902	.01901	.01900
Vol. Efficiency, %	93.7	93.8	94.2	94.25	94.3	94.4

Throttled P_d, psi	70	70	70	70	70	70
Discharge Surge, psi	10	20	40	60	80	100
Throttled P_s, psi	-3.7	-3.7	-3.7	-3.7	-3.7	-3.7
Rate of Flow, lb/sec	1.182	1.181	1.1805	1.179	1.178	1.176
Speed of Pump, rpm	356.0	355.4	354.5	354	353.6	353.5
Pump Input, hp	.683	.6975	.712	.723	.737	.751
Work Output, ft	171	171	171	171	171	171
Water HP Output, hp	.3675	.3670	.3668	.3662	.3659	.3655
Mech. Efficiency, %	53.8	52.6	51.5	50.6	49.6	48.6
V_t Pumped, cuft/sec	.02025	.02020	.02019	.0201	.02008	.02007
V_a Pumped, cuft/sec	.01899	.01898	.01896	.08195	.01890	.08187
Vol. Efficiency, %	93.5	93.6	93.7	93.9	94.0	94.0

Throttled P_d, psi	80	80	80	80	80	80
Discharge Surge, psi	10	20	40	60	80	100
Throttled P_s, psi	-3.7	-3.7	-3.7	-3.7	-3.7	-3.7
Rate of Flow, lb/sec	1.176	1.175	1.174	1.173	1.172	1.170
Speed of Pump, rpm	354.9	354.2	353.4	352.8	352.5	352.4
Pump Input, hp	.734	.750	.766	.780	.793	.808
Work Output, ft	194.2	194.2	194.2	194.2	194.2	194.2
Water HP Output, hp	.4155	.415	.4147	.4145	.414	.413
Mech. Efficiency, %	56.7	55.3	54.05	53.1	52.2	51.2
V_t Pumped, cuft/sec	.02020	.02017	.02009	.02005	.02002	.0200
V_a Pumped, cuft/sec	.01887	.01885	.01883	.01888	.01880	.01878
Vol. Efficiency, %	93.4	93.5	93.6	93.8	93.9	93.9

Throttled P_d, psi	90	90	90	90	90	90
Discharge Surge, psi	10	20	40	60	80	100
Throttled P_s, psi	-3.7	-3.7	-3.7	-3.7	-3.7	-3.7
Rate of Flow, lb/sec	1.170	1.169	1.168	1.166	1.165	1.162
Speed of Pump, rpm	353.6	352.9	352.0	351.5	351.2	351.1
Pump Input, hp	.785	.802	.820	.835	.850	.865
Work Output, ft	217	217	217	217	217	217
Water HP Output, hp	.462	.4613	.4607	.460	.4591	.4582
Mech. Efficiency, %	58.8	57.5	56.1	55.1	54.0	53.0
V_t Pumped, cuft/sec	.02008	.02006	.0200	.01995	.0199	.01986
V_a Pumped, cuft/sec	.01878	.01875	.01870	.01868	.01865	.01863
Vol. Efficiency, %	93.3	93.4	93.5	93.5	93.6	93.8

Throttled P_d, psi	100	100	100	100	100
Discharge Surge, psi	20	40	60	80	100
Throttled P_s, psi	-3.7	-3.7	-3.7	-3.7	-3.7
Rate of Flow, lb/sec	1.162	1.161	1.160	1.159	1.158
Speed of Pump, rpm	351.5	350.5	350.1	349.9	349.7
Pump Input, hp	.855	.875	.890	.906	.922
Work Output, ft	241	241	241	241	241
Water HP Output, hp	.510	.5093	.5086	.5078	.507
Mech. Efficiency, %	59.6	58.2	57.2	56.0	55.0
V_t Pumped, cuft/sec	.0200	.01994	.0199	.01985	.0198
V_a Pumped, cuft/sec	.01861	.01860	.01859	.01855	.01853
Vol. Efficiency, %	93.2	93.25	93.25	93.4	93.7

Throttled P_d, psi	110	110	110	110
Discharge Surge, psi	20	40	60	80
Throttled P_s, psi	-3.7	-3.7	-3.7	-3.7
Rate of Flow, lb/sec	1.156	1.155	1.154	1.153
Speed of Pump, rpm	350	349	348.5	348.4
Pump Input, hp	.907	.928	.945	.962
Work Output, ft	264	264	264	264
Water HP Output, hp	.555	.5548	.5544	.554
Mech. Efficiency, %	61.2	59.7	58.6	57.5
V_t Pumped, cuft/sec	.01994	.01990	.019875	.01984
V_a Pumped, cuft/sec	.01858	.01855	.01852	.01850
Vol. Efficiency, %	93.1	93.2	93.25	93.3

Throttled P_d, psi	120	120	120
Discharge Surge, psi	20	40	60
Throttled P_s, psi	-3.7	-3.7	-3.7
Rate of Flow, lb/sec	1.150	1.448	1.147
Speed of Pump, rpm	348.3	347.4	347
Pump Input, hp	.960	.983	1.000
Work Output, ft	287	287	287
Water HP Output, hp	.5790	.5780	.5770
Mech. Efficiency, %	60.3	58.8	57.7
V_t Pumped, cuft/sec	.01982	.01977	.01975
V_a Pumped, cuft/sec	.01849	.01840	.01838
Vol. Efficiency, %	93.0	93.1	93.1

Throttled P_d, psi	130	130		140
Discharge Surge, psi	20	40		20
Throttled P_s, psi	-3.7	-3.7		-3.7
Rate of Flow, lb/sec	1.143	1.142		1.137
Speed of Pump, rpm	346.6	345.7		344.8
Pump Input, hp	1.012	1.037		1.065
Work Output, ft	309	309		332
Water HP Output, hp	.6425	.6418		.6860
Mech Efficiency, %	63.5	62.0		64.4
V_t Pumped, cuft/sec	.01971	.01970		.01963
V_a Pumped, cuft/sec	.01831	.01830		.01825
Vol. Efficiency, %	92.9	92.9		92.9

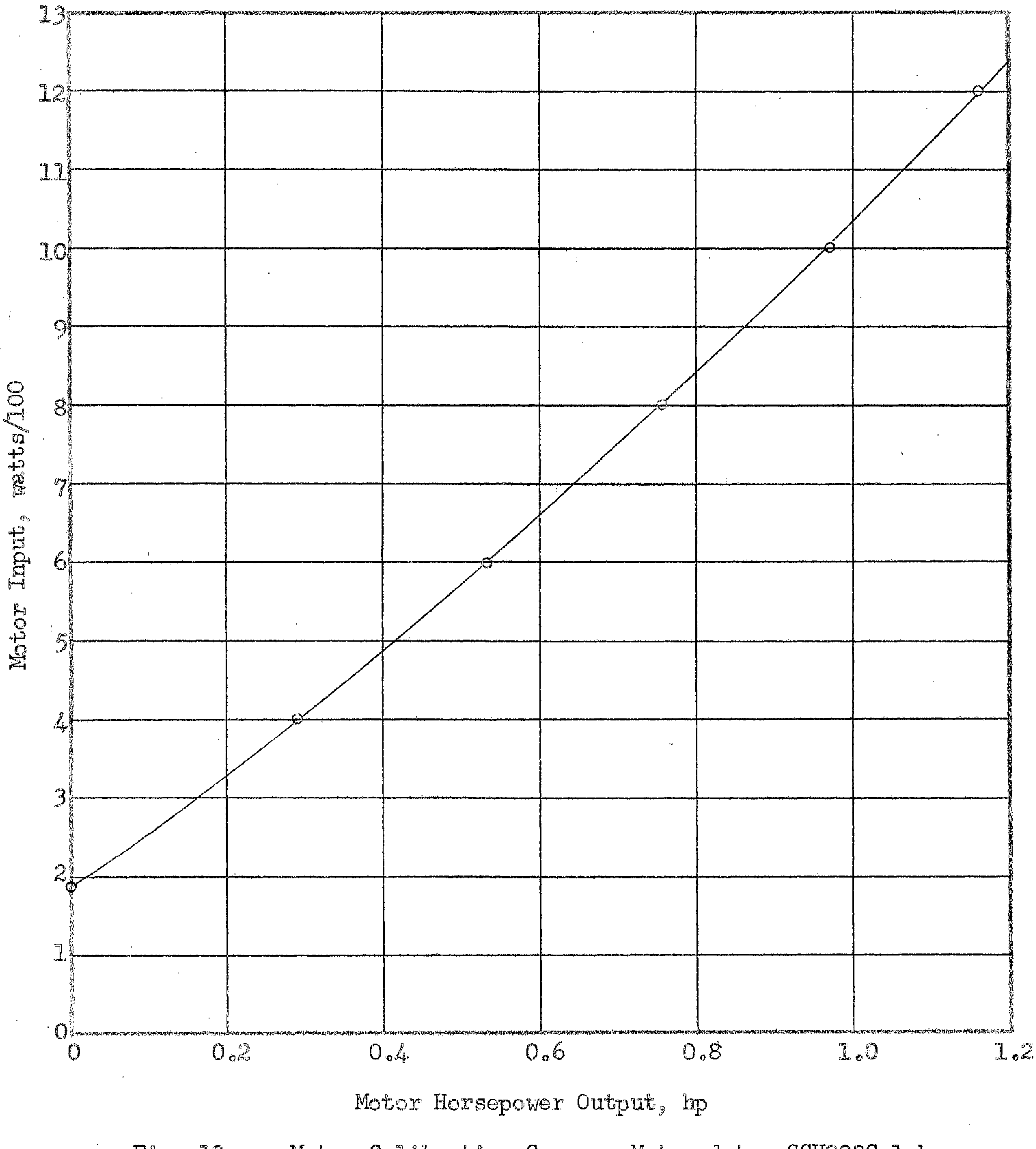

Fig. 12. Motor Calibration Curve. Motor data: CSH203C-1 hp, 1750 rpm, 4 pole, 60 cycles, 115 volts, breakdown torque 6.80 ft lb, lock rotor torque 10.5 ft lb, and lock rotor amperes (115 v) 66. Courtesy of Century Electric Co., St. Louis, Missouri.

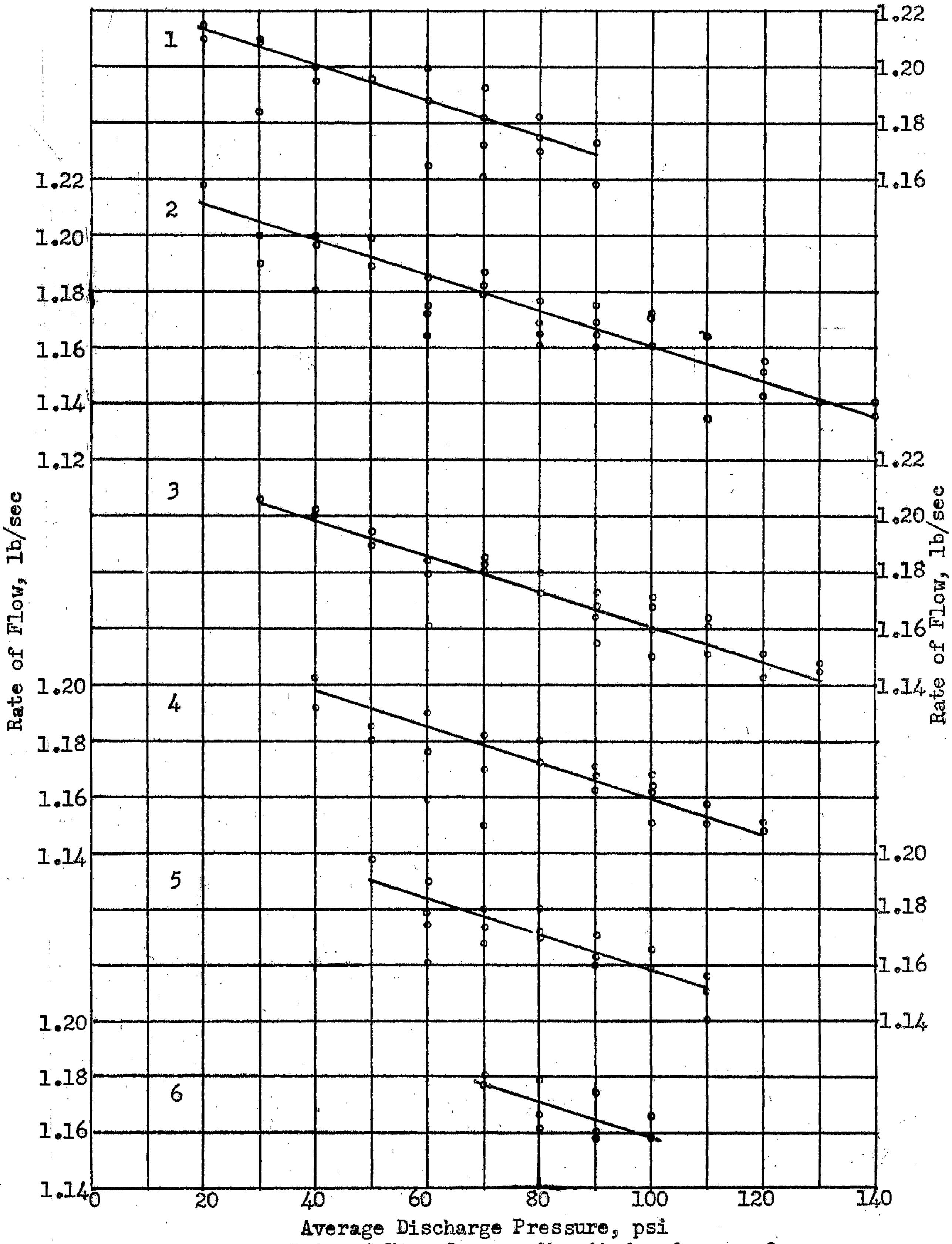

Fig. 13. Rate of Flow Curve. Magnitude of surge for curve (1) 10 psi, (2) 20 psi, (3) 40 psi, (4) 60 psi, (5) 80 psi, and (6) 100 psi.

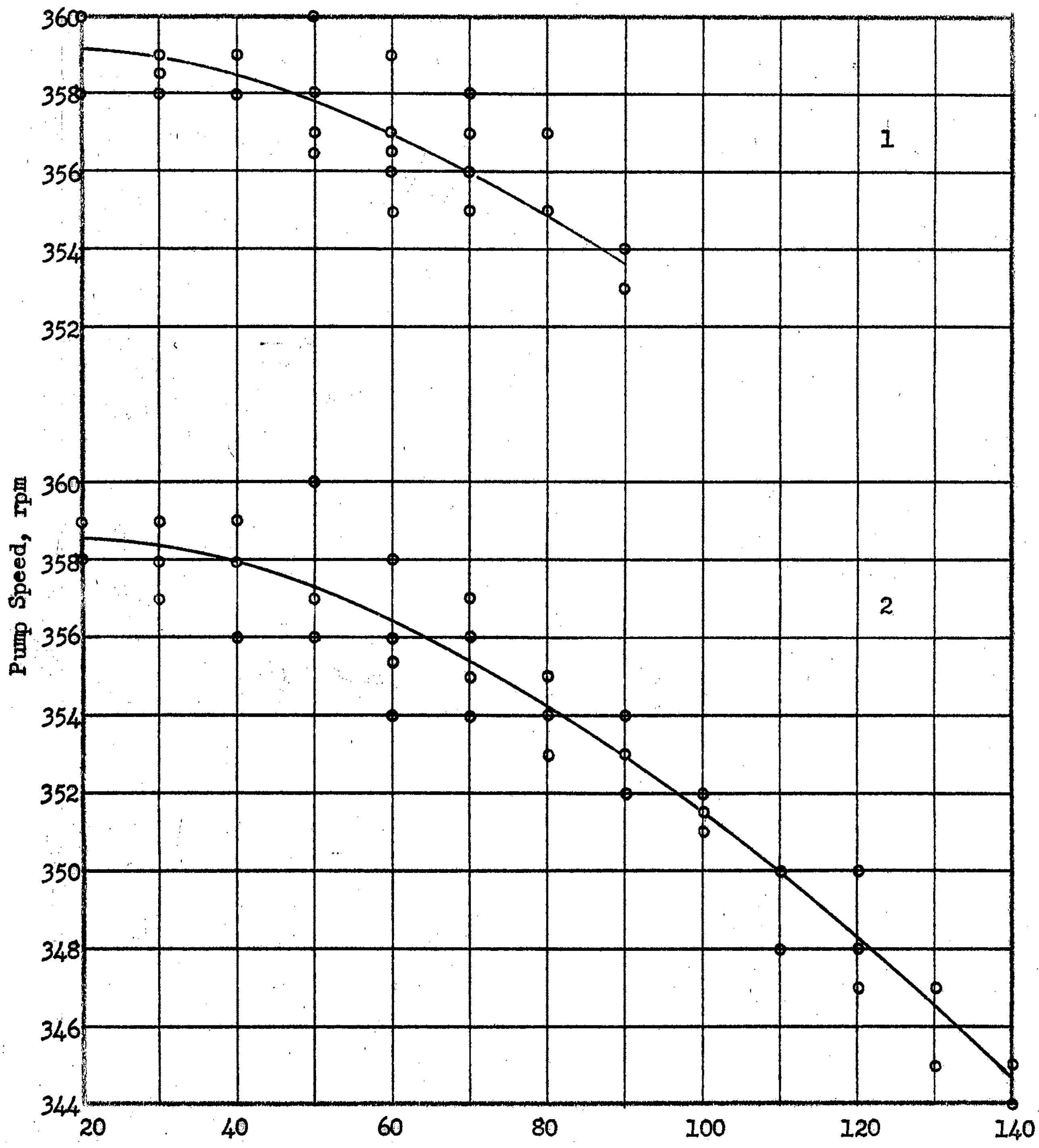

Fig. 14a. Average Discharge Pressure Versus Pump Speed. Magnitude of surge for curve (1) 10 psi, and (2) 20 psi.

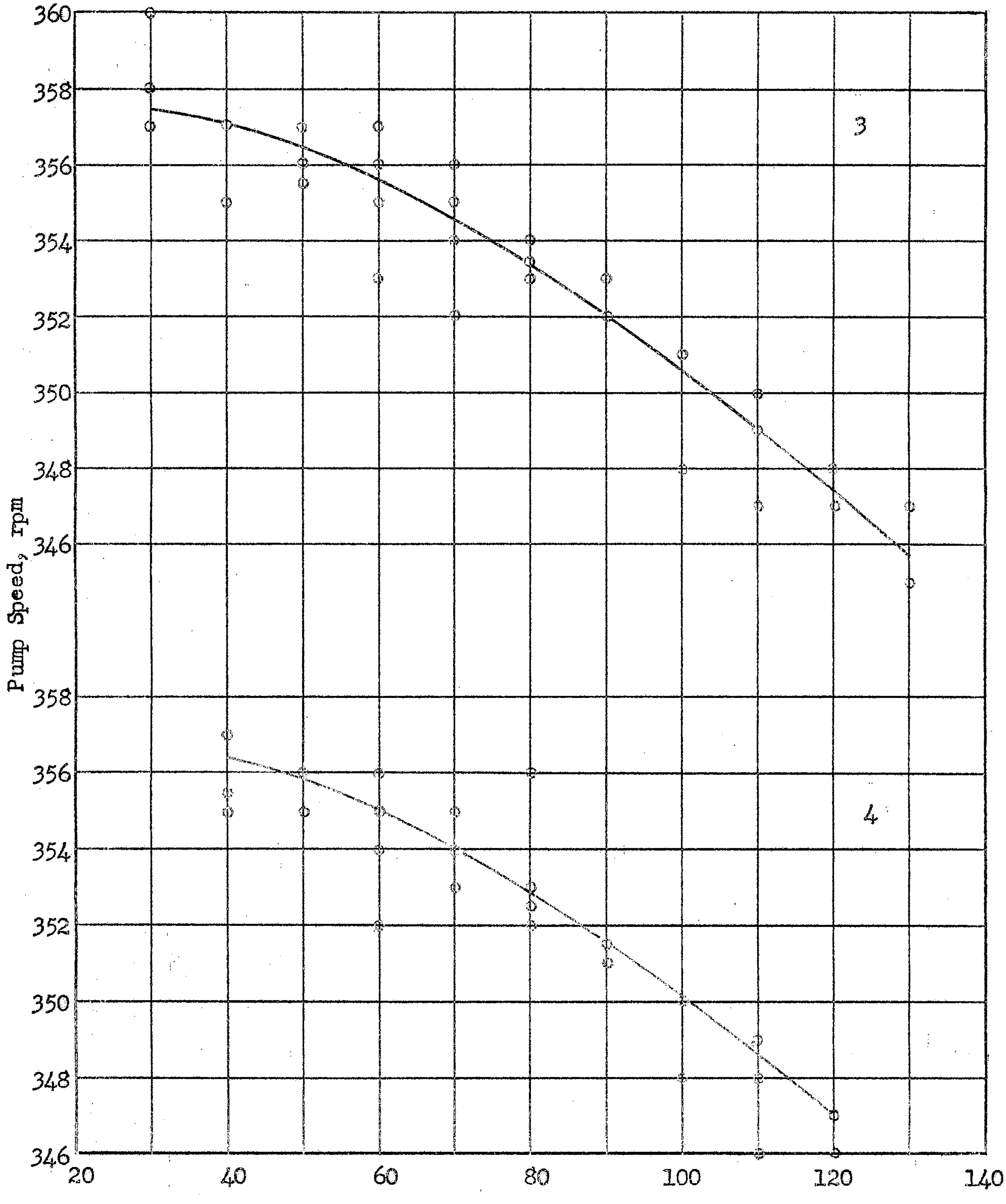

Fig. 14b. Average Discharge Pressure Versus Pump Speed.
Magnitude of surge for curve (3) 40 psi, and (4) 60 psi.

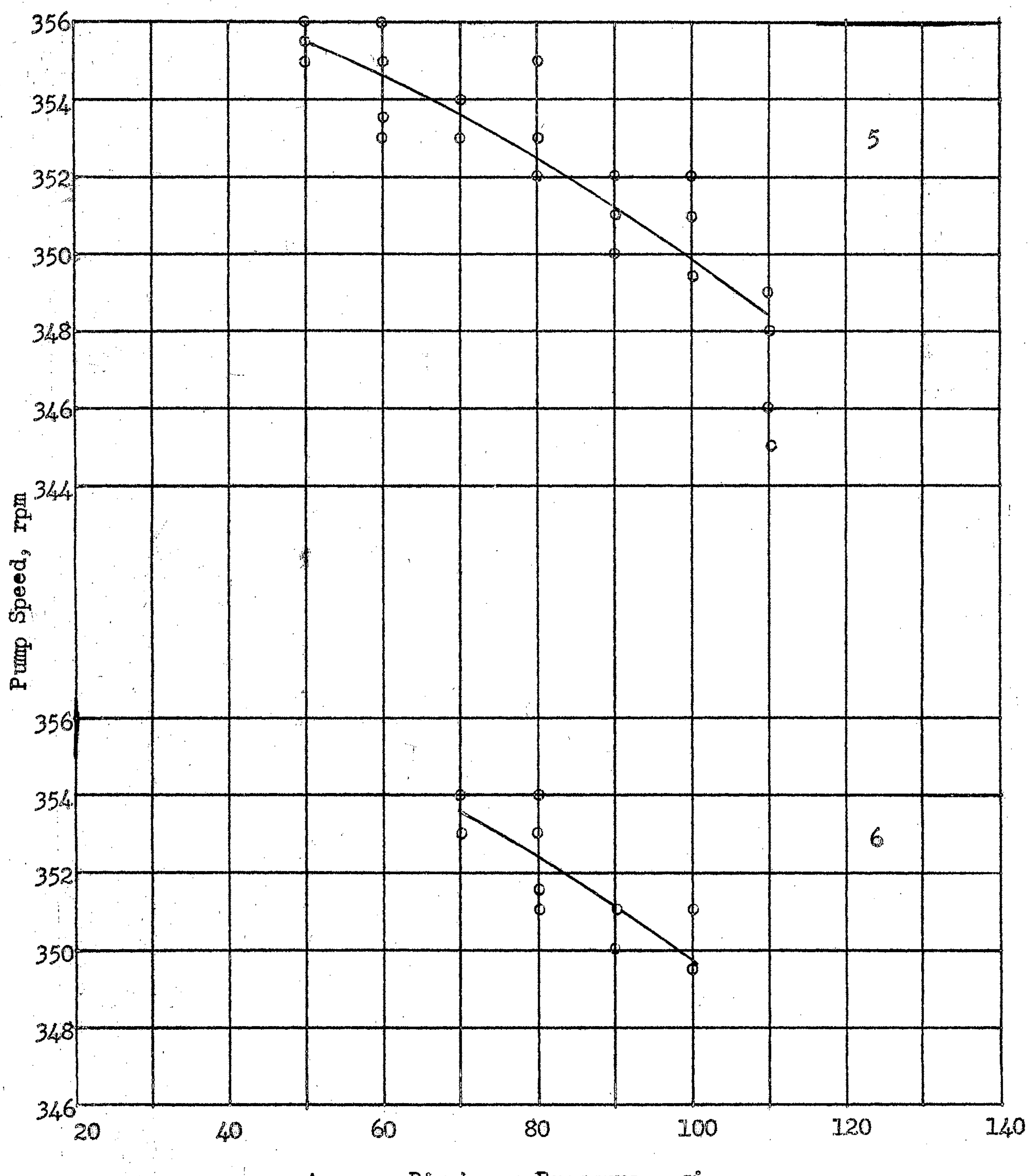

Fig. 14e. Average Discharge Pressure Versus Pump Speed. Magnitude of surge for curve (5) 80 psi, and (6) 100 psi.

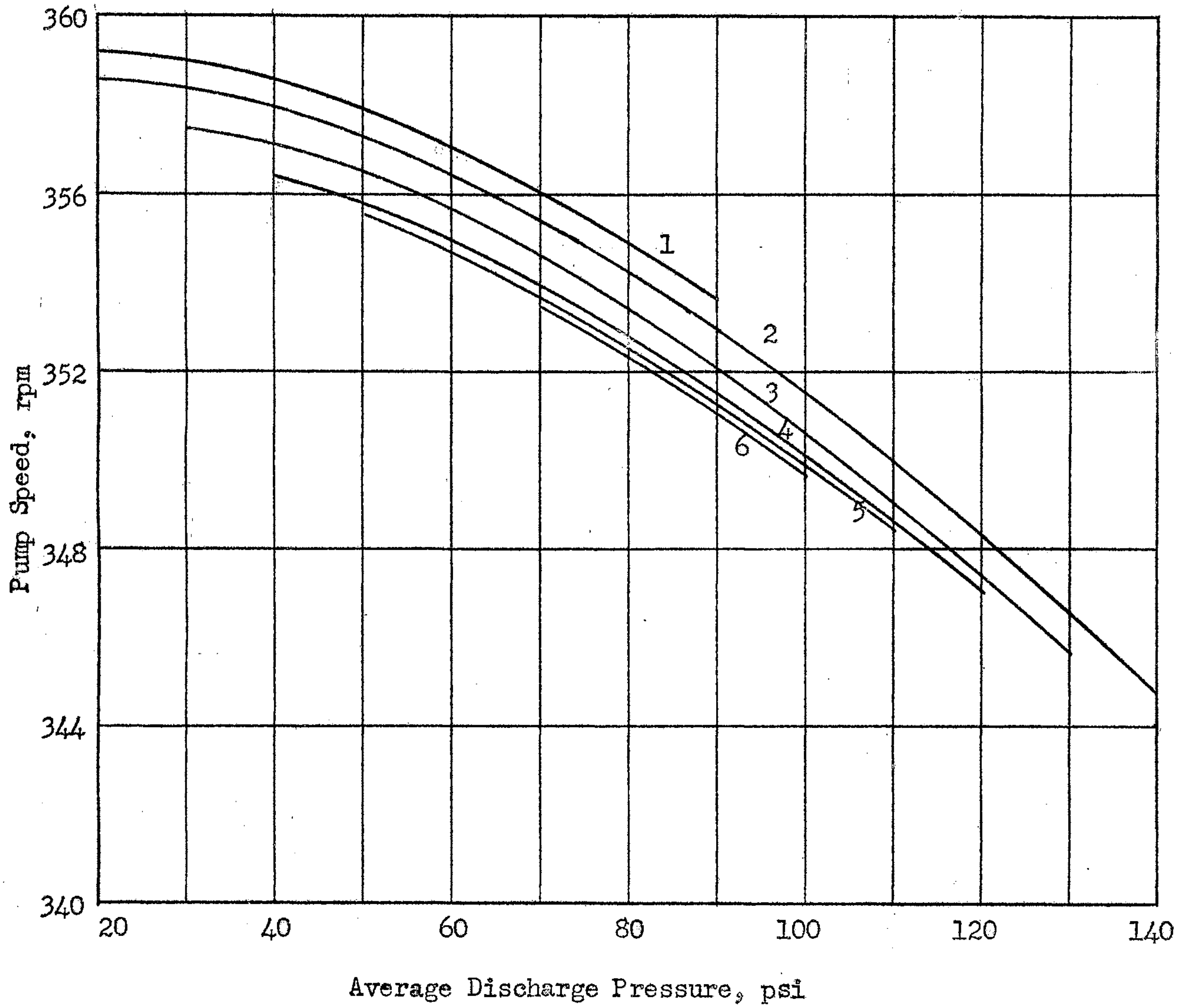

Fig. 14. Average Discharge Pressure Versus Pump Speed. Magnitude of surge for curve (1) 10 psi, (2) 20 psi, (3) 40 psi, (4) 60 psi, (5) 80 psi, and (6) 100 psi.

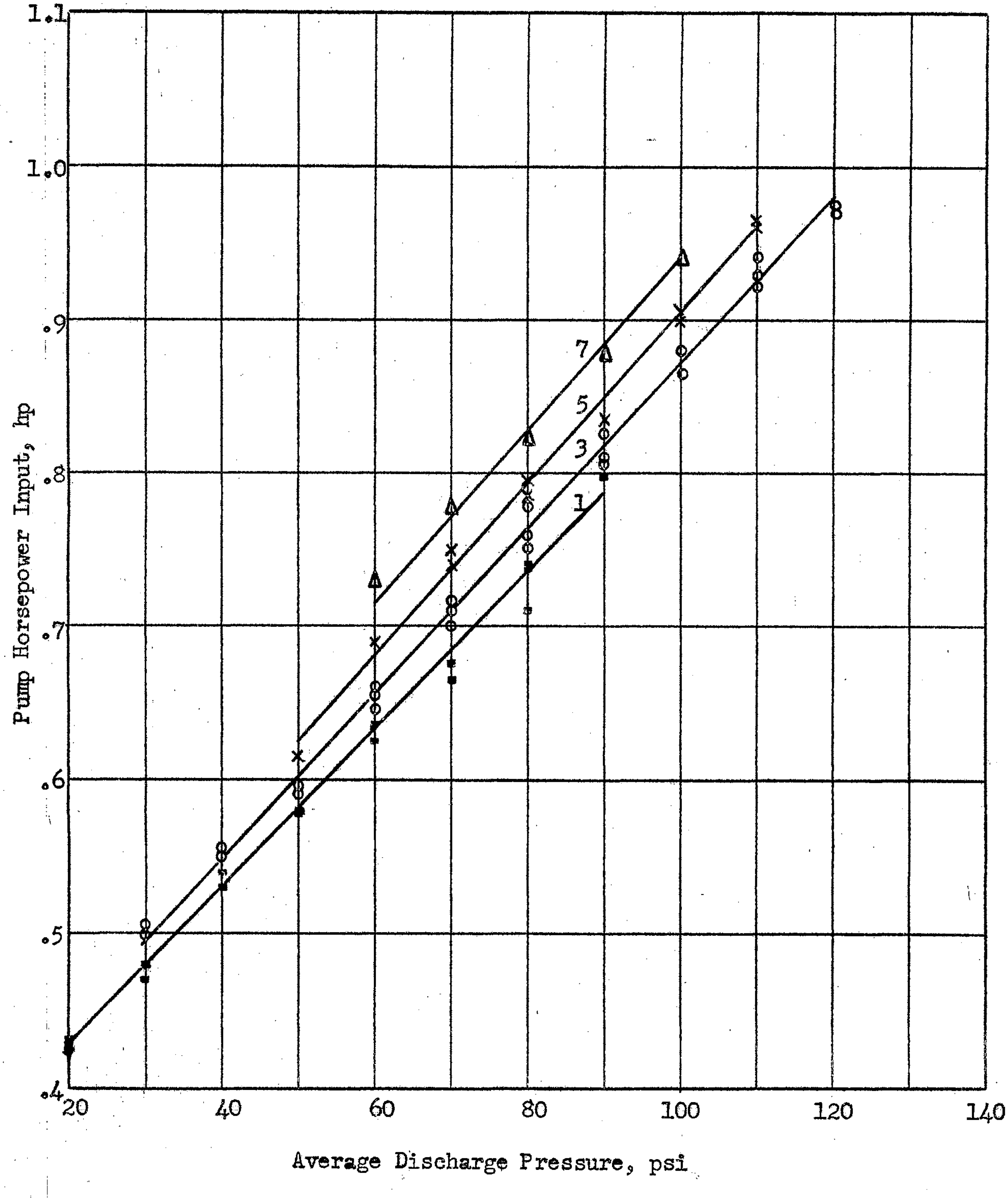

Fig. 15a. Average Discharge Pressure Versus Pump Horsepower Input. Magnitude of surge for curve (1) 10 psi, (3) 40 psi, (5) 80 psi, (7) 120 psi.

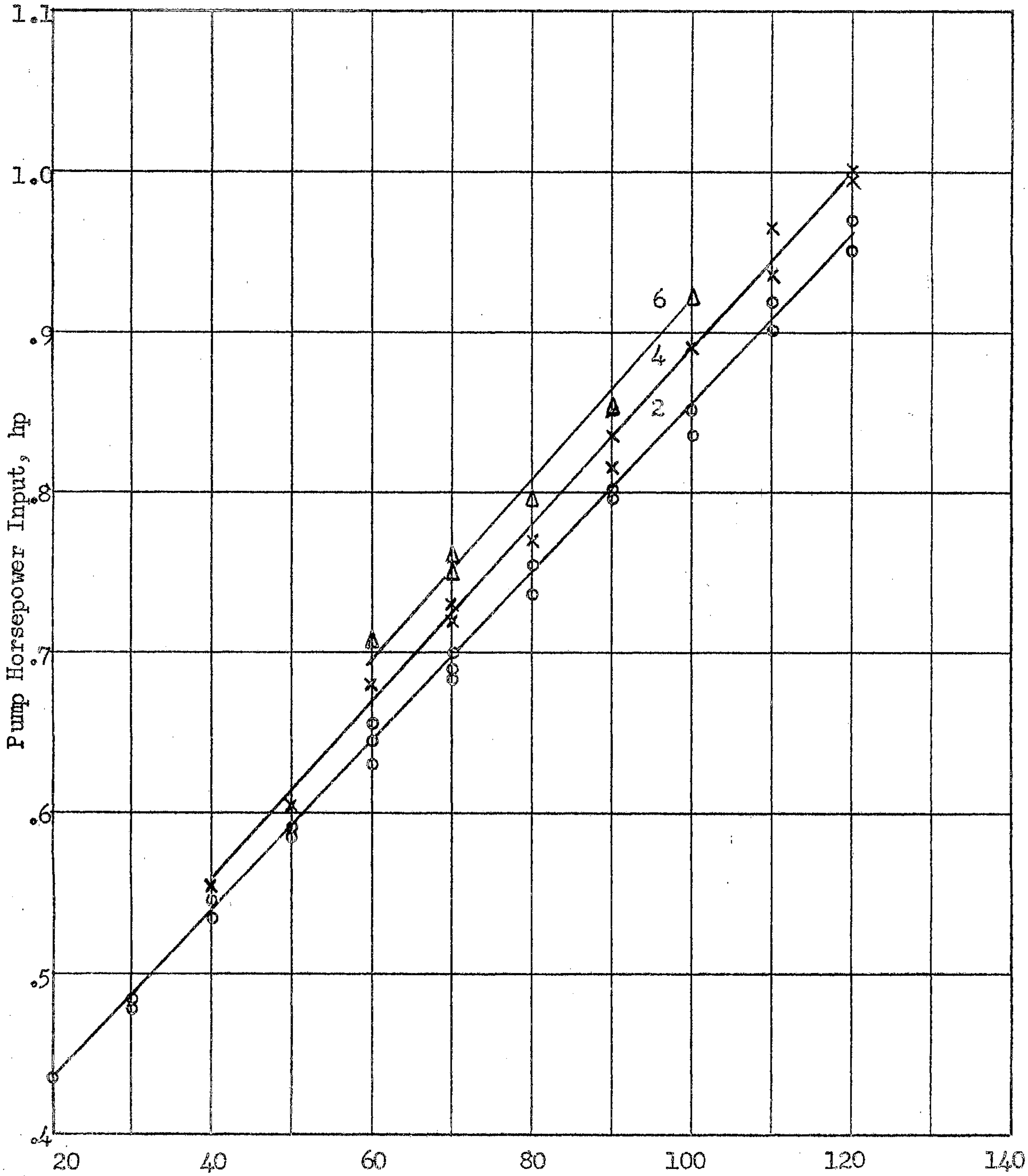

Fig. 15b. Average Discharge Pressure Versus Pump Input.
Magnitude of surge for curve (2) 20 psi, (4) 60 psi, and (6) 100 psi.

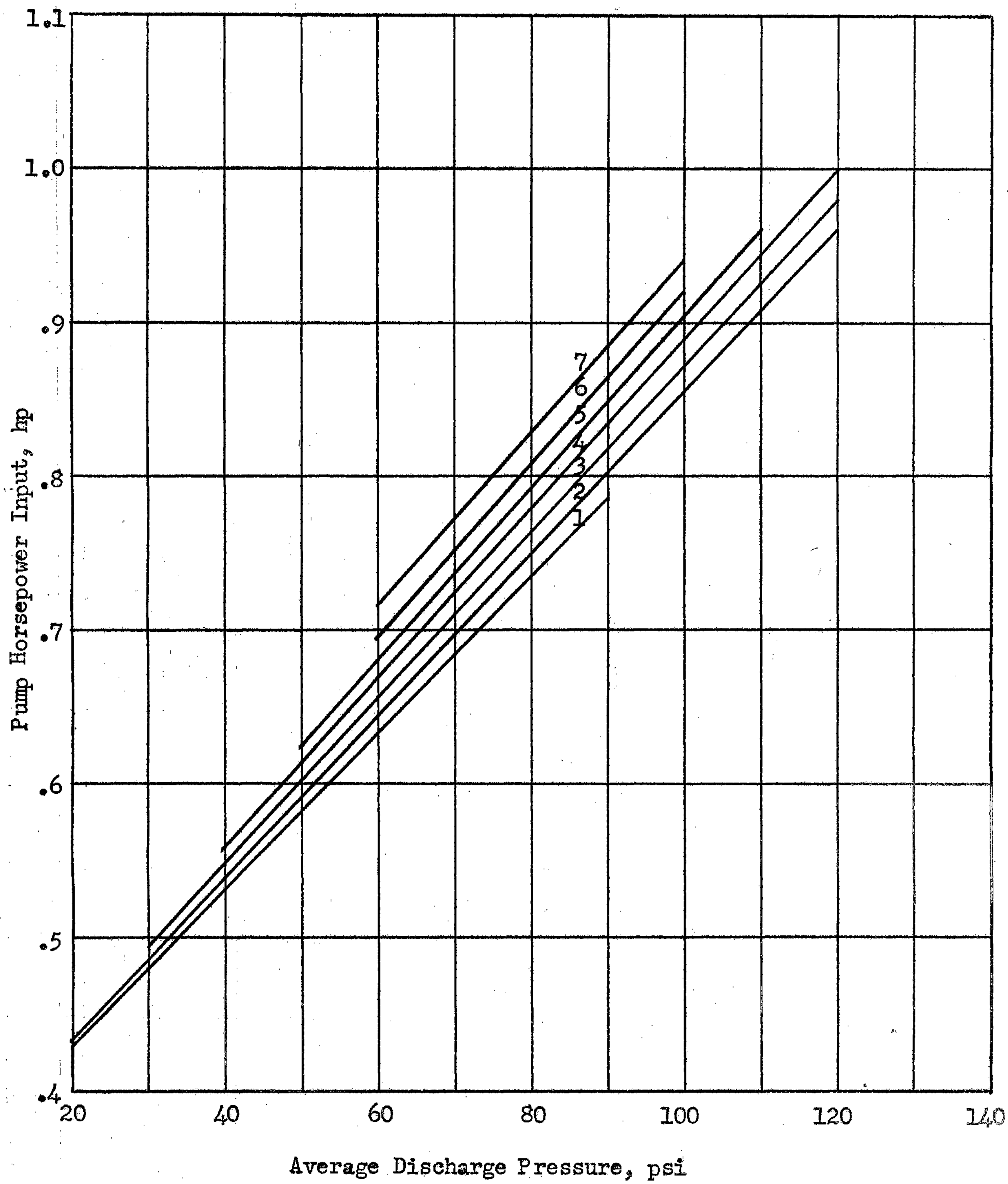

Fig. 15. Average Discharge Pressure Versus Pump Horsepower Input. Magnitude of surge for curve (1) 10 psi, (2) 20 psi, (3) 40 psi, (4) 60 psi, (5) 80 psi, (6) 100 psi, and (7) 120 psi.

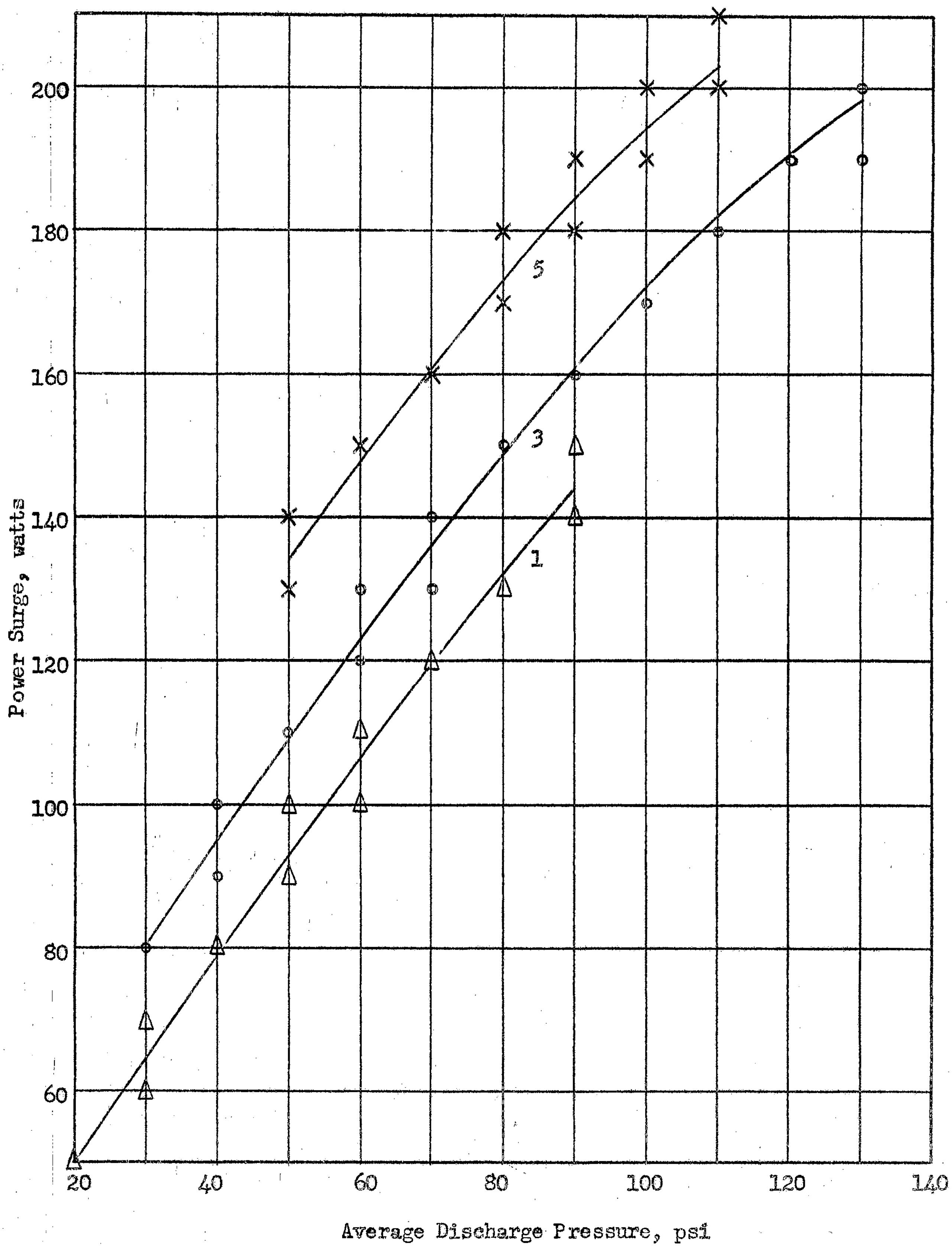

Fig. 16a. Average Discharge Pressure Versus Power Surge. Magnitude of surge for curve (1) 10 psi, (3) 40 psi, and (5) 80 psi.

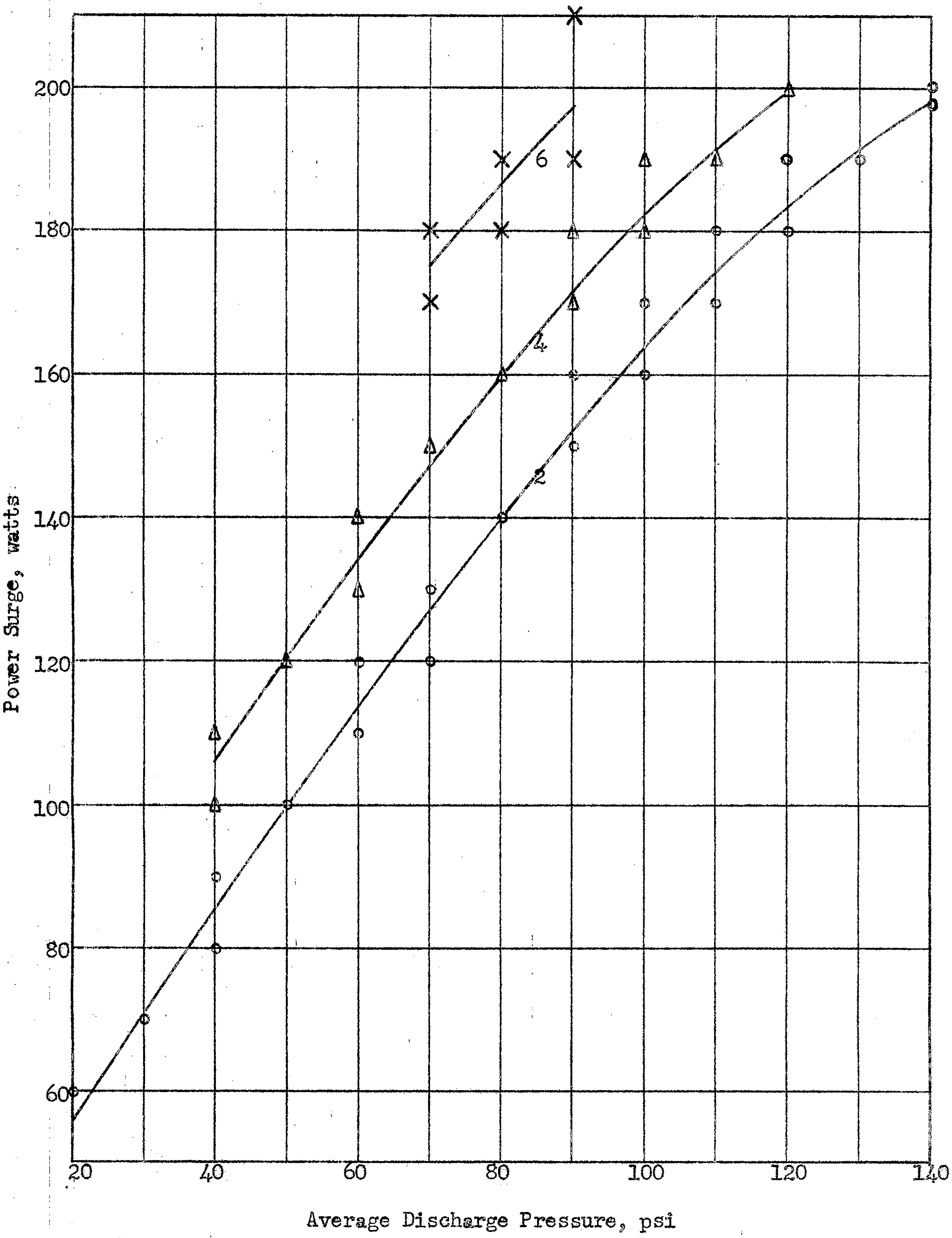

Fig. 16b. Average Discharge Pressure Versus Power Surge. Magnitude of surge for curve (2) 20 psi, (4) 60 psi, and (6) 100 psi.

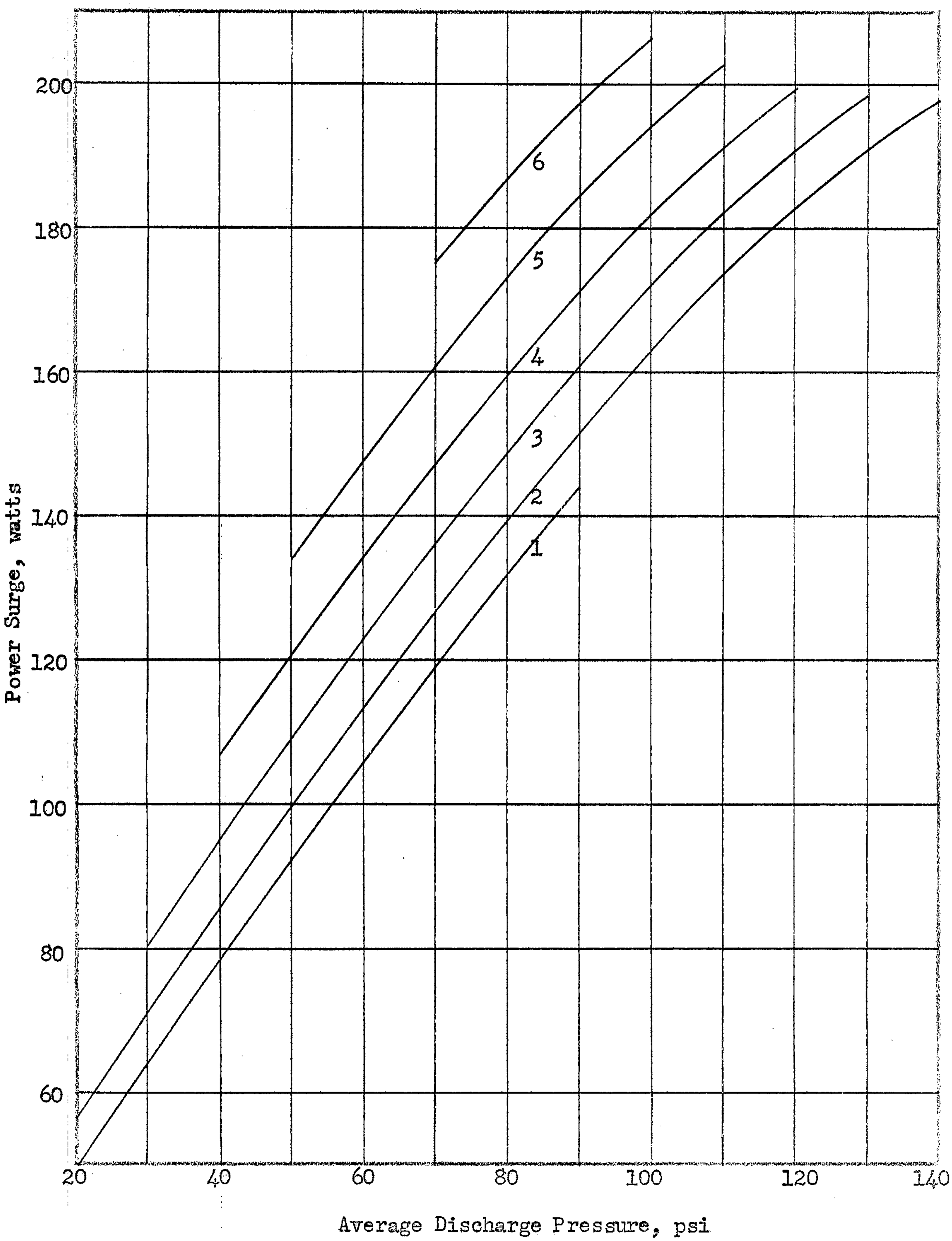

Fig. 16. Average Discharge Pressure Versus Power Surge. Magnitude of surge for curve (1) 10 psi, (2) 20 psi, (3) 40 psi, (4) 60 psi, (5) 80 psi, and (6) 100 psi.

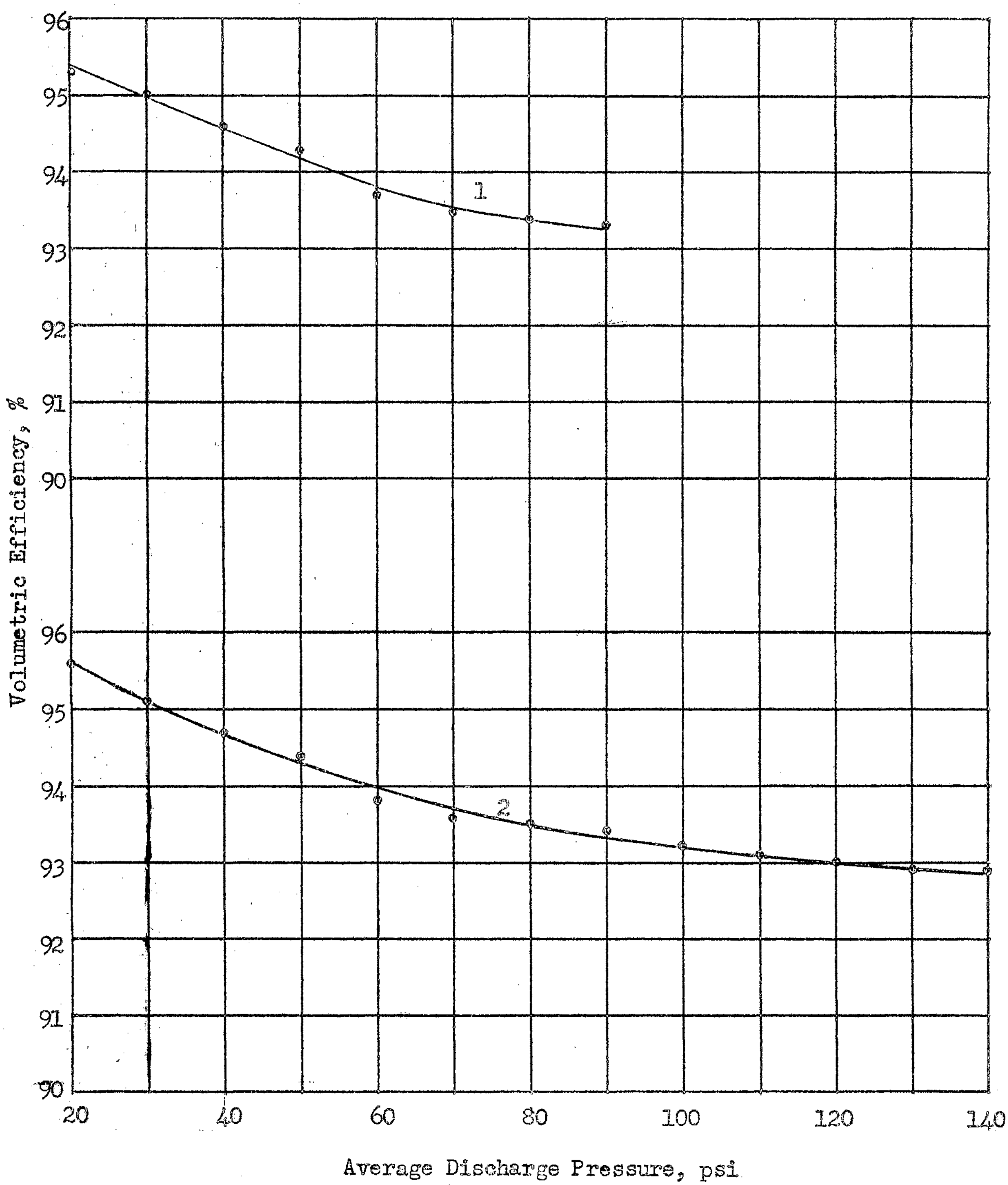

Fig. 17a. Average Discharge Pressure Versus Volumetric Efficiency. Magnitude of surge for curve (1) 10 psi, and (2) 20 psi.

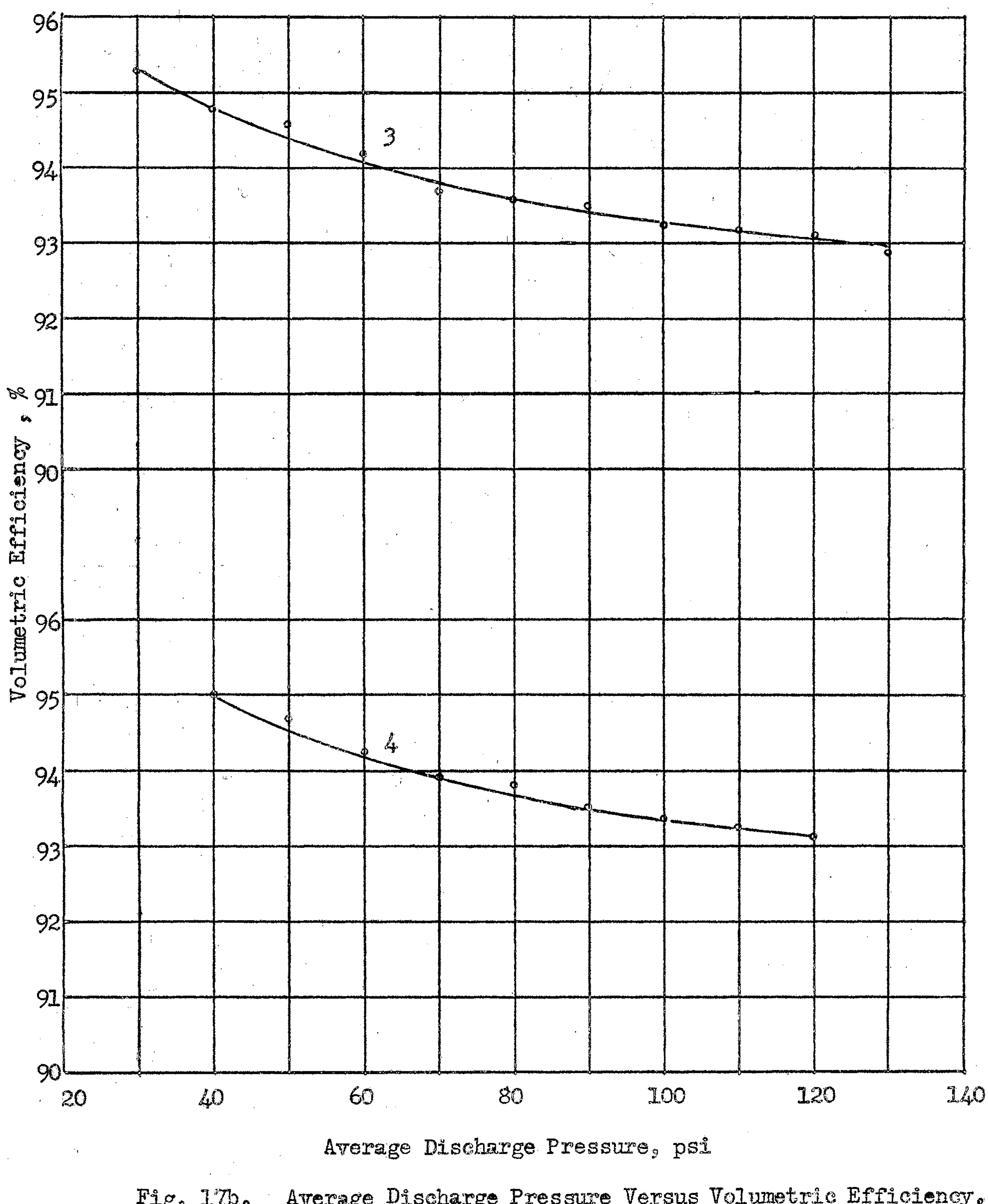

Fig. 17b. Average Discharge Pressure Versus Volumetric Efficiency. Magnitude of surge for curve (3) 40 psi, and (4) 60 psi.

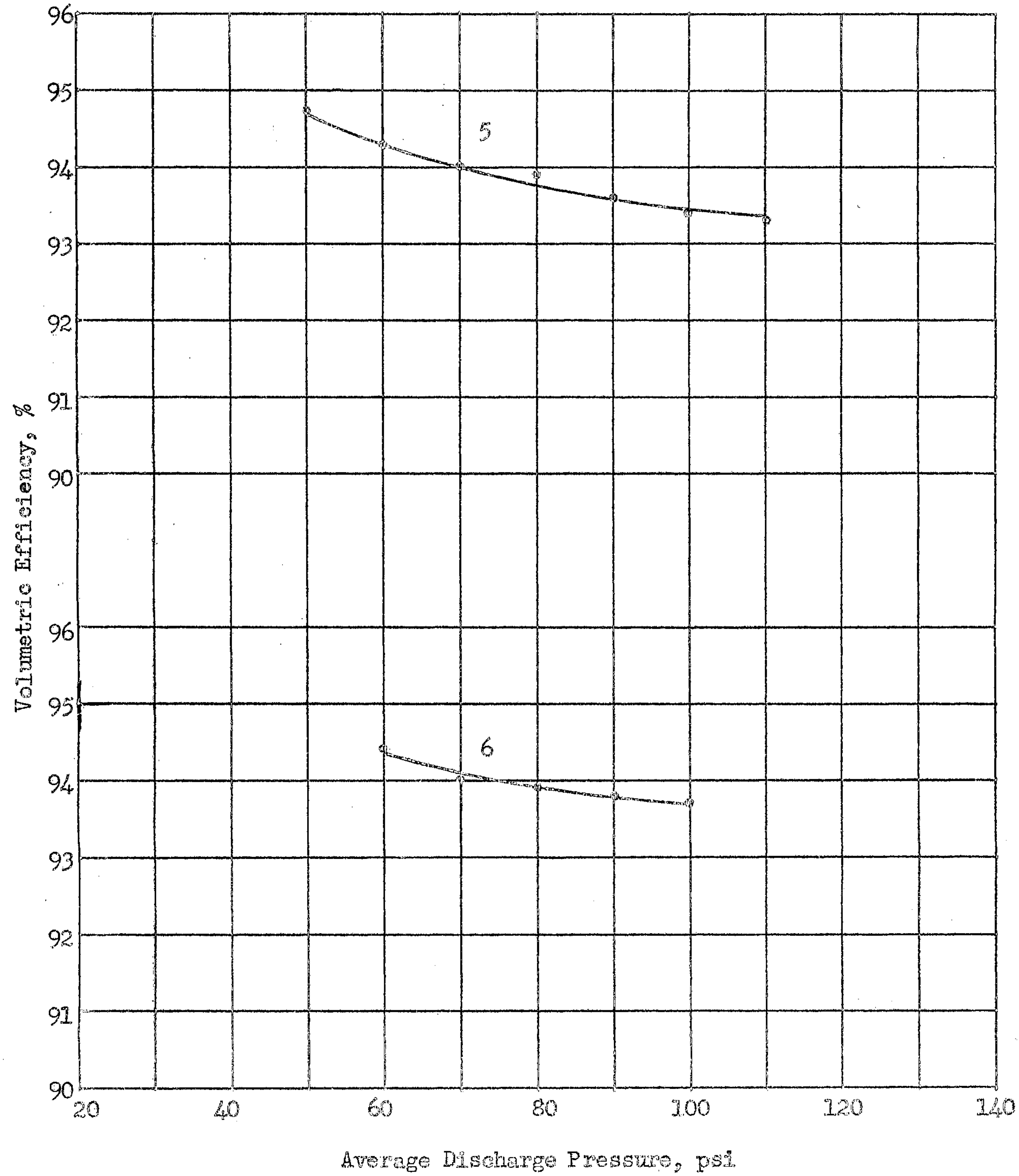

Fig. 17e. Average Discharge Pressure Versus Volumetric Efficiency. Magnitude of surge for curve (5) 80 psi, and (6) 100 psi.

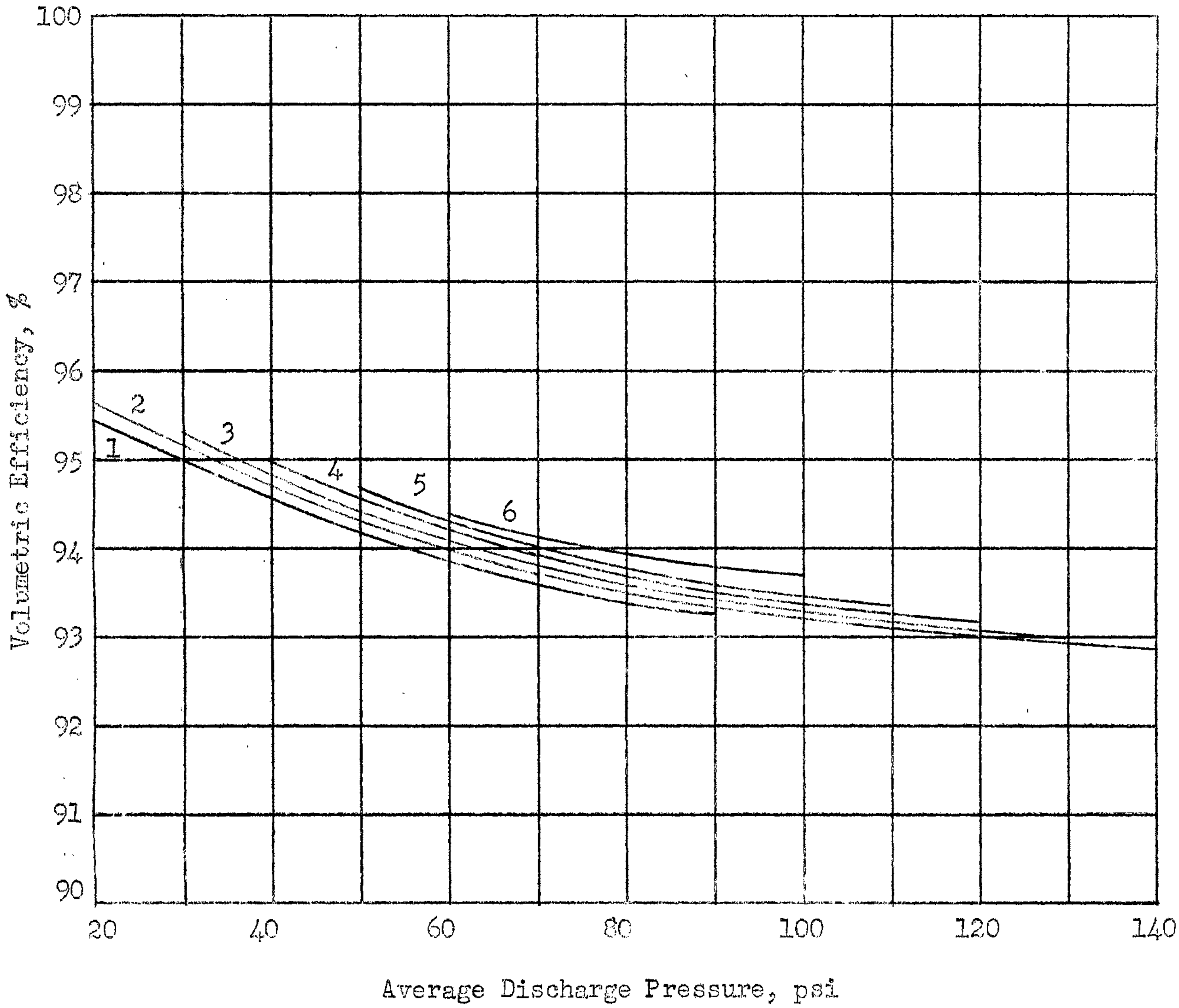

Fig. 17. Average Discharge Pressure Versus Volumetric Efficiency. Magnitude of surge for curve (1) 10 psi, (2) 20 psi, (3) 40 psi, (4) 60 psi, (5) 80 psi, and (6) 100 psi.

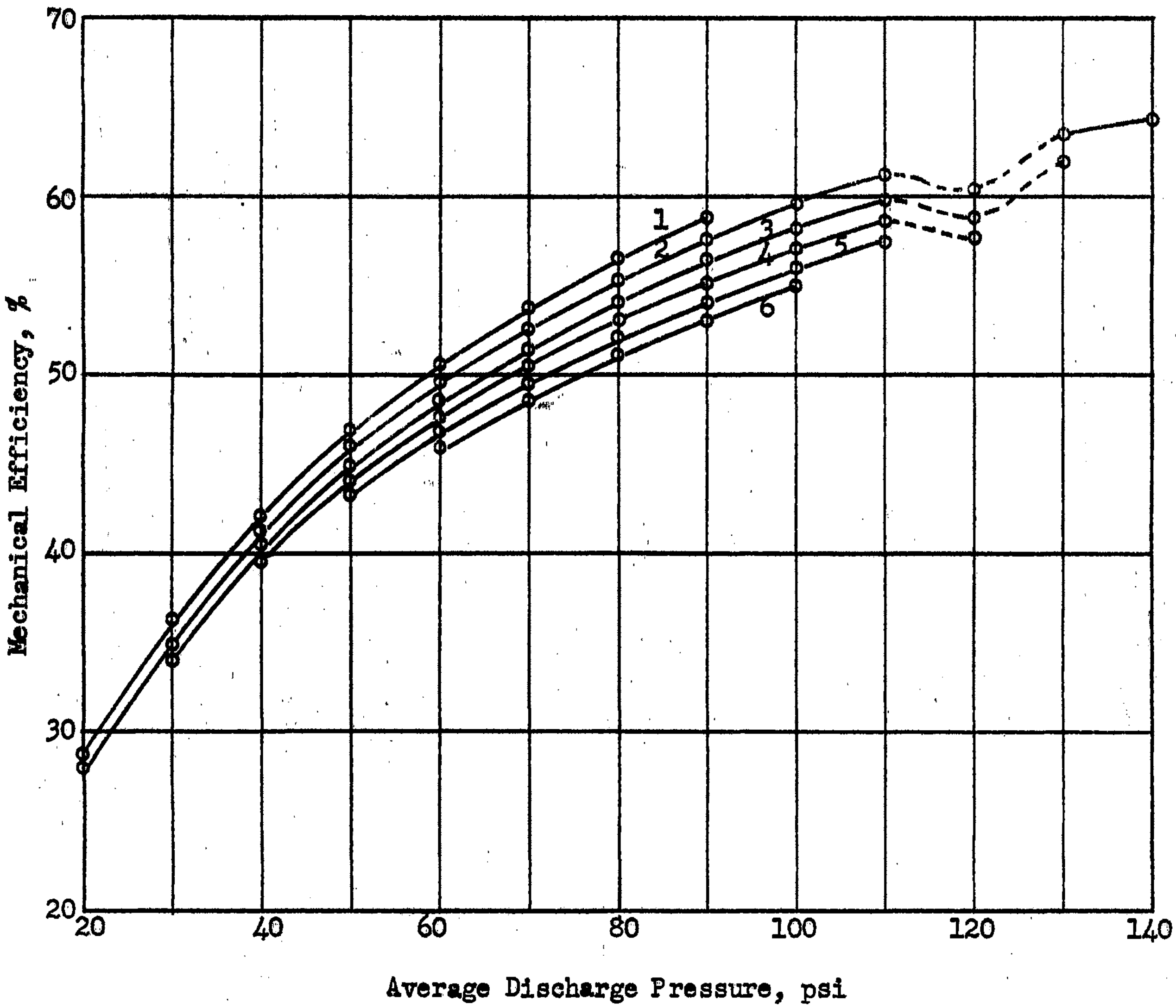

Fig. 18. Average Discharge Pressure Versus Mechanical Efficiency. Magnitude of surge for curve (1) 10 psi, (2) 20 psi, (3) 40 psi, (4) 60 psi, (5) 80 psi, and (6) 100 psi.

CHAPTER VII

SAMPLE CALCULATIONS

All sample calculations are made for a discharge pressure of 60 psi, a discharge surge with a magnitude of 60 psi, and a suction pressure of -3.7 psi.

Work Done by the Pump in Feet of Water:

Using Bernoulli's Equation as applied to a pump,

$$\text{Work} = (P_d'/w + V_d^2/2g + Z_d) - (P_s'/w + V_s^2/2g + Z_s).$$

Since the average velocity entering the pump is essentially the same as the velocity leaving the pump, then

$$\text{Work} = (P_d' - P_s')/w + (Z_d - Z_s)$$

where:

w = specific weight of fluid or 62.3 lb/ cu ft @ 75°F;
P_d' = throttled discharge pressure, psf;
P_s' = throttled suction pressure, psf;
Z_d = elevation of discharge gauge, ft; and
Z_s = elevation of suction gauge, ft.

$$\text{Work} = 144\, P_t/62.3 + Z_t = 2.31\, P_t + Z_t = \text{ft lb/lb}$$

where:

$P_t = P_d - P_s$ or total pressure, psi; and
$Z_t = Z_d - Z_s$ or difference in elevation, ft.

At given conditions, $Z_t = 1$ ft and $P_t = 63.7$ psi.

$$\text{Work} = 2.31(63.7) + 1 = 148 \text{ ft lb/lb}.$$

Rate of Flow from Pump:

In the original data, the time in seconds for 100 lbs of water to trip the measuring scales was obtained; therefore,

$$Q = 100/\text{time} = \text{lb/sec}.$$

At given conditions, time was 84.5 sec; therefore,

$$Q = 100/84.5 = 1.186 \text{ lb/sec}.$$

Water Horsepower Output of Pump:

$HP_o = \text{Work} \times Q/550$

At given conditions Work was 148 and $Q = 1.186$ lb/sec; therefore,

$HP_o = 148 \times 1.186/550 = 0.3190$ hp.

Calibration of Watt Hour Meter:

W = watts pulled by a constant load as indicated by a conventional watt meter, watts;
T_W = time required for 10 revolutions of watt-hour-meter disk, sec;
N = revolutions of disk; and
C = calibration constant for the watt-hour-meter.

$C = (W/N) \times T_W.$

Time required for 10 revolutions of disk for one load of 1060 watts was 9.9 seconds; therefore,

$C = (1060/10) \times 9.9 = 1050$ watt-sec/rev.

This value of the calibration constant was the average of 8 different resistances.

Horsepower Input to the Pump:

W_m = watt input to motor

$W_m = 1050\ N/T_W.$

At given conditions N was 10 rev and T_W was 14.5 sec; therefore,

$W_m = 1050 \times 10/14.5 = 725$ watts.

From the efficiency curve of the electric motor, Fig. 12, the horsepower output of the motor can be determined. By neglecting belt slip, the motor output can be assumed equal to pump horsepower input. For 725 watts, the pump horsepower input would be 0.67 hp.

Mechanical Efficiency of the Pump:

$ME = HP_o/HP_i \times 100$

Using the previous calculated values for HP_o and HP_i,

$ME = 0.3190/0.67 \times 100 = 47.7\ \%.$

Theoretical Volume Pumped per Second:

A = area of piston, sq ft;
L = length of stroke, ft; and
n_s = strokes of pump per sec.

$V_t = ALn_s$.

For the double-acting pump used,

Diameter of cylinder is $2\frac{1}{2}$ inches,
Length of stroke is 3 inches,
Volumes pumped per stroke is 2, and
Revolutions per stroke is 5.

Expressed in terms of revolutions per minute of the pump,

$$V_t = \frac{(2\frac{1}{2})^2 \times \pi \times 3 \times 2 \times RPM_p}{1728 \times 4 \times 5 \times 60}$$

$$V_t = 0.0000569 \times RPM_p.$$

At the given conditions, pump speed was 355 rpm; therefore,

$$V_t = 0.0000569 \times 355 = .02019 \text{ cu ft/sec.}$$

Actual Volume Pumped per Second:

$V_a = Q/w$ cu ft/sec.

At given conditions rate of flow was 1.186 lb/sec; therefore,

$V_a = 1.186/62.3 = 0.01902$ cu ft/sec.

Volumetric Efficiency:

$VE = V_a/V_t \times 100.$

Using the previous calculated values for V_a and V_t,

$VE = .01902/02019 = 94.25$ %.

CHAPTER VIII

SUMMARY

The task of testing a reciprocating pump and interpreting the data involves the study and practice of fundamental engineering principles. The nature and effects of the different surges developed in operating a piston pump have always been of great concern to pump and pipeline companies and their interest has prompted the investigation of some of the accompanying phenomena.

The equipment for the series of tests performed was arranged in order to maintain the variables of the suction side of the pump nearly constant and the discharge side was designed to allow both surge pressure and average discharge pressure to be varied over a large range. The design of the apparatus not only permitted obtaining comparable data that would give a trend of the effects of reciprocating surges but also helped to control other factors that could influence the results.

The series of tests made, proved that the mechanical efficiency of a piston pump can be substantially increased by removing the reciprocating surges. The mechanical efficiency of the test pump was improved six percent by decreasing the magnitude of the discharge surge from 100 psi to 10 psi. Improved performance can also be realized by reducing the magnitude of the surge in maintaining an even pump speed, a steady flow of fluid, reducing unnecessary and dangerous vibrations from the line, and establishing a constant power requirement.

There can be little doubt as to the importance of removing surges from pumping units even if the only incentive is economy. The results of this investigation substantiated the evidence that reciprocating surges effect all operational characteristics of a piston pump and should be removed by installing an effective desurger.

The material contained in this report should warrant the attention of persons connected with the following interests: (1) pump maintenance, because of excessive stresses produced by vibrations and high pressures; (2) pump designers, because of the change in pump characteristics due to varying the surge; (3) pump installation crews, because efficient pipe layout is important in reducing effects of secondary surges; (4) electric companies, because of the extra problems involved in a surging power requirement; and (5) pump operators, because high surges reduce pump efficiency and hence the unit economy.

CHAPTER IX

RESULTS AND CONCLUSION

The several graphs presented in this report illustrate very effectively the manner in which the operation of a piston pump is affected by reciprocating surges. It can be noted from Fig. 15 that an increase in the magnitude of the discharge surge will cause a decided increase in the amount of power required for a given average discharge pressure. Figure 13 is a series of curves that represent the rate of flow plotted against average discharge pressure for all measured surges. These curves show that the rate of flow from the pump will decrease with an increase in surge pressure. Since the mechanical efficiency of the pump varies directly with input power and the rate of flow, it is evident that these two important factors contribute to decrease the mechanical efficiency (Fig. 18) of the unit with an increase of surge pressure.

The accurate measurement of the rate of flow for this investigation was difficult to achieve. This would be unimportant for a large capacity pump but with a small pump the percent error is increased for small quantities of fluid flowing.

Figure 17 shows the variation in volumetric efficiency as a result of increasing the average discharge pressure or the magnitude of the surge. The effect realized in this case is characteristic of a particular pump, suction pipe, pump speed, and flywheel effect. The reason proposed for the increase in volumetric efficiency with an increase in surge for a particular test pump, is that the higher surge on the discharge side will help close the discharge valve tight and thus prevent any loss of fluid back into the cylinder. Where the high surge is not experienced, the valve will not be closed as fast and some fluid will be lost because of the differential pressure between the discharge line and the pump cylinder. If this reasoning is plausible, it would be possible to install a spring that would reverse the trend obtained. Due to the fact that the change in volumetric efficiency was so small and there were so many factors involved, it is felt that the result of this phase of the investigation is inconclusive. A factor neglected in calculating the volumetric efficiency was the volume occupied by the piston rod inside the cylinder on the return stroke. This has the effect of lowering the volumetric efficiency.

Figure 16 shows that as the pressure surge increases the surge of the power supply also increases. At high surge pressures this power variation becomes an important aspect. On large power pumps a condition of a surging power requirement would certainly be significant. If the test pump had been equipped with a heavy flywheel this power surge would not have been as large but such a flywheel would give a more positive action to the pump and would change the character of the pressure surges by increasing the magnitude of the surge.

Figure 14 shows pump speed plotted against discharge pressure. As expected, the increase in surge pressure causes a definite decrease in pump speed; also, pump speed is considerably reduced with an increase in average discharge pressure. The omission of the belt slip in the calculations of pump speed, would not affect the power transmitted by more than two percent according to reliable authorities and in hydraulic calculations can be considered negligible.

In order to simulate conditions that would be found in actual field practice, some parts of the tests were not conducted in the usual testing manner; for example, instead of maintaining the customary constant pump speed, it was allowed to change with the other variable factors. Consequently, some of the results may appear to be exaggerated when compared with values not obtained in this manner.

The results of this investigation in the estimation of the author have been very successful and should help to substantiate in the minds of concerned individuals the range of factors that are unmistakably effected by both reciprocating and reflecting surges. It is true that the values obtained using the small test pump in this investigation would not be comparable with data obtained from large multi-cylinder high speed pumps; but there can be no question as to the effect of surges on such items as variation in power supply, rate of flow, pump speed, and the efficiency of the pumping unit.

It would be infeasible to predict an accurate improvement in every pump based on the results of this report. The importance of tests of this nature is to obtain trends of variable factors and the magnitude of operational data for proving the existence of the phases investigated. There is, however, every indication that substantial savings and improved performance would justify installing equipment to reduce surges to a minimum.

BIBLIOGRAPHY

1. Delevan, D. B. "An Investigation into the Cause of Breakage in Crude Oil Pipe-line Transportation Systems." A.I.M.E.-Petroleum Development and Technology, III (1925), 498-520.

2. Diederichs, H. and Pomeroy, W. D. "The Occurrence and Elimination of Surge or Oscillating Pressures in Discharge Lines from Reciprocating Pumps." A.S.M.E. Transactions, PET 51-2 (1929), 9-49.

3. Durand, W. F. Hydraulics of Pipe Lines. New York: D. Van Nostrand Company, 1921.

4. Editors. "Pulsations in Duplex Pumps." Petroleum Times, XIX (June, 1928), 1054.

5. Finch, V. C. Pump Handbook. Millbrae, California: National Press, 1948.

6. Goodman, John. "Hydraulic Experiments on a Plunger Pump." Proceedings Institute of Mechanical Engineers, (Feb. 20, 1903), 123-197.

7. Graham, F. D. Audels Pumps. New York: J. Theo. Audel and Company, 1949.

8. Hydraulic Institute. Standards of Hydraulic Institute, Reciprocating Pump Section. New York: Hydraulic Institute.

9. Kristal, F. A. and Annett, F. A. Pumps. New York: McGraw-Hill Book Company, 1940.

10. Lewitt, E. H. Hydraulics. London: Sir Isaac Pitman and Sons, 1947.

11. Thilenius, Fred. "Oil Engines as a Drive for Pipe Line Pumps." Supplement Issue of Mechanical Engineering, XLVIII (June, 1926), 663-670.

12. Wilson, W. E. "Performance Criteria for Positive-Displacement Pumps and Fluid Motors." A.S.M.E. Transactions, LXXI (Feb., 1949), 115-120.

THESIS TITLE: THE EFFECT OF PRESSURE SURGES ON THE EFFICIENCY AND OPERATION OF A PISTON PUMP

NAME OF AUTHOR: ERNEST CHESTER FITCH, JR.

THESIS ADVISER: J. P. EVERETT

The content and form have been checked and approved by the author and thesis adviser. "Instructions for Typing and Arranging the Thesis" are available in the Graduate School office. Changes or corrections in the thesis are not made by the Graduate School office or by any committee. The copies are sent to the bindery just as they are approved by the author and faculty adviser.

NAME OF TYPIST: FERN B. HALL