INVESTIGATION OF SUCTION CONDITIONS FOR A HORIZONTAL TRIPLEX PLUNGER PUMP

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INVESTIGATION OF SUCTION CONDITIONS
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PLUNGER PUMP

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PREFACE

This study is a result of a research project sponsored by Gaso Pump and Burner Company, Tulsa, Oklahoma, and the School of Civil Engineering at Oklahoma State University.

I wish to express my sincere appreciation to Jan J. Tuma, Head of the School of Civil Engineering, and to those people responsible for selecting me as recipient of C. E. Bretz Fellowship. I could not have continued my education without this financial assistance. Mr. Bretz has my deepest admiration and appreciation for providing this fund.

I am deeply indebted to Mr. Harry M. Wyatt for his invaluable assistance and constructive criticism while acting as Gaso Project Leader and my Adviser.

I wish also to express my gratitude to Professor Roger Flanders for his timely advice and cheerful remarks. Preston Wilson and Arlin Harris, of the RAD Laboratory, Professor Bert Davenport, John McCandles and George Cooper, of the Mechanical Engineering Laboratory, gave valuable assistance, which I sincerely appreciate.

My parents deserve special thanks for inciting in me the realization of the need for continued education.

Most of all, I am grateful to my wife, Sue Elizabeth, for her unselfishness and for her understanding toleration of me throughout this trying year.

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NOMENCLATURE

t_g	Temperature of gas (fuel)
t_w	Temperature of water
BHP	Brake horsepower
D	Volumetric pump displacement in one revolution of pump
HP	Horsepower
N	Angular speed
N_e	Angular speed of engine
N_{ip}	Initial angular speed of pump
N_p	Angular speed of pump
P	Pressure
P_b	Barometric pressure
P_d	Discharge pressure of pump
P_g	Gas pressure (fuel)
P_m	Absolute pressure in engine intake manifold
P_s	Suction pressure of pump
P_v	Vacuum on engine intake manifold
Q	Flow rate
Q_a	Actual flow rate
Q_t	Theoretical flow rate

T	. . . . .	Time
T_a	. . . . .	Time recorded in measurement of actual flow rate
T_t	. . . . .	Time recorded in measurement of theoretical flow rate
V	. . . . .	Volume
V_a	. . . . .	Volume of water added to volumetric tank
V_t	. . . . .	Volume of volumetric tank
W	. . . . .	Weight
W_1	. . . . .	Weight on scales before adding water to volumetric tank
W_2	. . . . .	Weight on scales after adding water to volumetric tank
WHP	. . . .	Water horsepower at pump discharge
γ_t	. . . . .	Specific weight of water at given temperature
$\Delta(\ )$	. . . .	Change in ( )
η_m	. . . . .	Per cent efficiency (Mechanical)
η_v	. . . . .	Per cent efficiency (Volumetric)

CHAPTER I

INTRODUCTION

The need for investigation into suction requirements for piston or plunger type pumps was recognized by Professor Harry M. Wyatt, Jr., after an extensive field study. The research conducted by Mr. Wyatt consisted of approximately 150 tests of different installations in the states of Oklahoma, Texas, Louisana, Kansas, Kentucky and Illinois. It was discovered by Mr. Wyatt that there are no consistent methods for designing suction piping and that the values of minimum suction pressure quoted by men in the field, pump manufacturers and distributors were not standard for any given pump. They were, in general, simply "rule of thumb" answers. Extensive research in this field is obviously needed.

A survey of the literature was conducted. It was found that much work has been done in the field of pressure surge elimination (Wyatt [1] and Waller [2]), design of discharge piping, centrifugal pump suction design (Lester [3] and Hicks [4]), etc., but no references were found that indicated where research had been done on suction requirements for a particular Triplex pump. The <u>Standards of Hydraulics Institute</u> (6) gives minimum suction lift for trade pumps, but this cannot be applied to a particular triplex pump.

This study is the result of a research project financed by Gaso Pump and Burner Company, Tulsa, Oklahoma. The pump investigated was a Horizontal Triplex Plunger Pump (Gaso Fig. 3365).

The primary object of this study was to determine the effect of different suction pressures on pump performance and operation for each of three suction piping arrangements. These suction conditions were: double suction, single-side suction, single-center suction. (See Figure 2-1.)

Tests were made at three different speeds: 250, 350, and 400 rpm, and at maximum recommended discharge pressure of 800 psig.

The secondary object was the determination of capacity, horsepower required, and over-all efficiency as a function of pump speed.

The results of the study are shown by plotting suction pressure versus efficiency for each speed and suction piping arrangement. Capacity, power, and efficiency curves are also presented. A detailed list of test equipment, a discussion of test procedure, an estimation of the accuracy of plotted results, and recommendations for further study are included.

CHAPTER II

DISCUSSION OF SUCTION REQUIREMENTS

For a number of years, pump users and designers have been aware of the need to focus special attention on suction piping. Manufacturers of centrifugal machines adopted the concept of minimum net positive suction head and have employed this concept extensively. Reciprocating machines have not been completely neglected either. William M. Barr (5), in a volume published in 1899, stated that suction piping for piston and plunger pumps "should be as short and direct as possible" and that "they must be tight". He stated that failure to have the suction pipes "absolutely tight" would mean "uncertainty and loss of efficiency, if not complete failure of the pump to perform the service for which it was intended". It is pathetic that, even though piston and plunger pump designers were aware of suction requirements many years ago, very little was actually done by the manufacturers to define the specific conditions that are necessary for efficient and smooth operation. <u>The Standards of Hydraulic Institute</u> (6) provides a chart that enables one to obtain relative values of maximum permissible lift, but as stated in the "Standards", "The suction lift obtainable with a reciprocating pump is affected by the type of pump as well as the design of suction valves, pistons, and suction passages"; therefore, values taken from the "Standards" may not be applicable to a particular pump. The question still remains: What is the minimum value of suction pressure at which the pump can be operated smoothly, without loss of efficiency?

In all fairness to the customer and distributor, the pump manufacturer should be able to provide a definite, intelligent answer to this question and to many others which arise.

Another question of immediate concern to pump manufacturers is: Does location of suction piping affect efficiency?

The objective of this study was to answer these questions.

It is an established fact that suction pressure does affect pump efficiency and operation. The major factor affecting these is a phenomenon referred to as cavitation. A discussion of cavitation and its effects, as well as a discussion of location of suction piping, follows.

Cavitation and its Effects

If a liquid is being pumped, and, if at the same point in the liquid flow the existing fluid pressure equals the vapor pressure of the liquid pumped at the existing temperature, the liquid will vaporize forming a cavity or void. In the particular type of pump tested, the most probable places for cavitation to occur are in the passages around the intake valves. Those are the smallest flow areas; therefore, the velocity of flow is greatest, resulting in the least pressure.

The result of the fluid vaporizing is that the cylinder is filled on the suction stroke with a mixture of vapor and liquid. When the piston passes top dead center and begins to compress the mixture, it moves relatively unopposed until the vapor pressure of the liquid is reached. The discharge valve will not have opened at this point. When the vapor pressure of the liquid is reached, the vapor suddenly condenses, thus, the piston encounters a large resistance to motion. The discharge valve is, thus, slammed open, resulting in noisy, inefficient operation. The forces developed by this action are great enough to cause serious damage to pump parts; therefore, the pressure on the suction side of a reciprocating pump must be kept above the vapor pressure of the liquid being pumped in order to prevent cavitation.

It should also be noted that, when operating a pump under a condition of suction lift, leaks in the suction system can cause the same effects as vaporization of the liquid being pumped. When a leak is present in the suction piping, air is drawn into the system and the entrained air produces effects similar to those produced by the fluid vaporizing, hence, Mr. Barr's statement that suction piping must be "absolutely tight".

Effect of Location of Suction Piping

The designers of the particular pump tested recommend that the pump should be run with either single side suction or double suction depending upon the speed of the pump. A proposal was made, by this group, to investigate the feasibility of operating the pump with a single suction entering at the center of the intake manifold. These three arrangement are shown in Figure 2-1.

The connections necessary to obtain the various arrangements are outlined in the following Table. (Please refer to Figure 2-1.)

TABLE II-I

CONNECTIONS NECESSARY FOR SUCTION PIPING ARRANGEMENTS

Type of Flange	Type of Suction		
	Double	Single Side	Center
Open pipe flange at	Ports 1 and 3	Port 1	Port 2
Blind flange at	Port 2	Ports 2 and 3	Ports 1 and 3

The pump tested was designed in 1949. This series of triplex pumps was first built with a 2-inch suction and

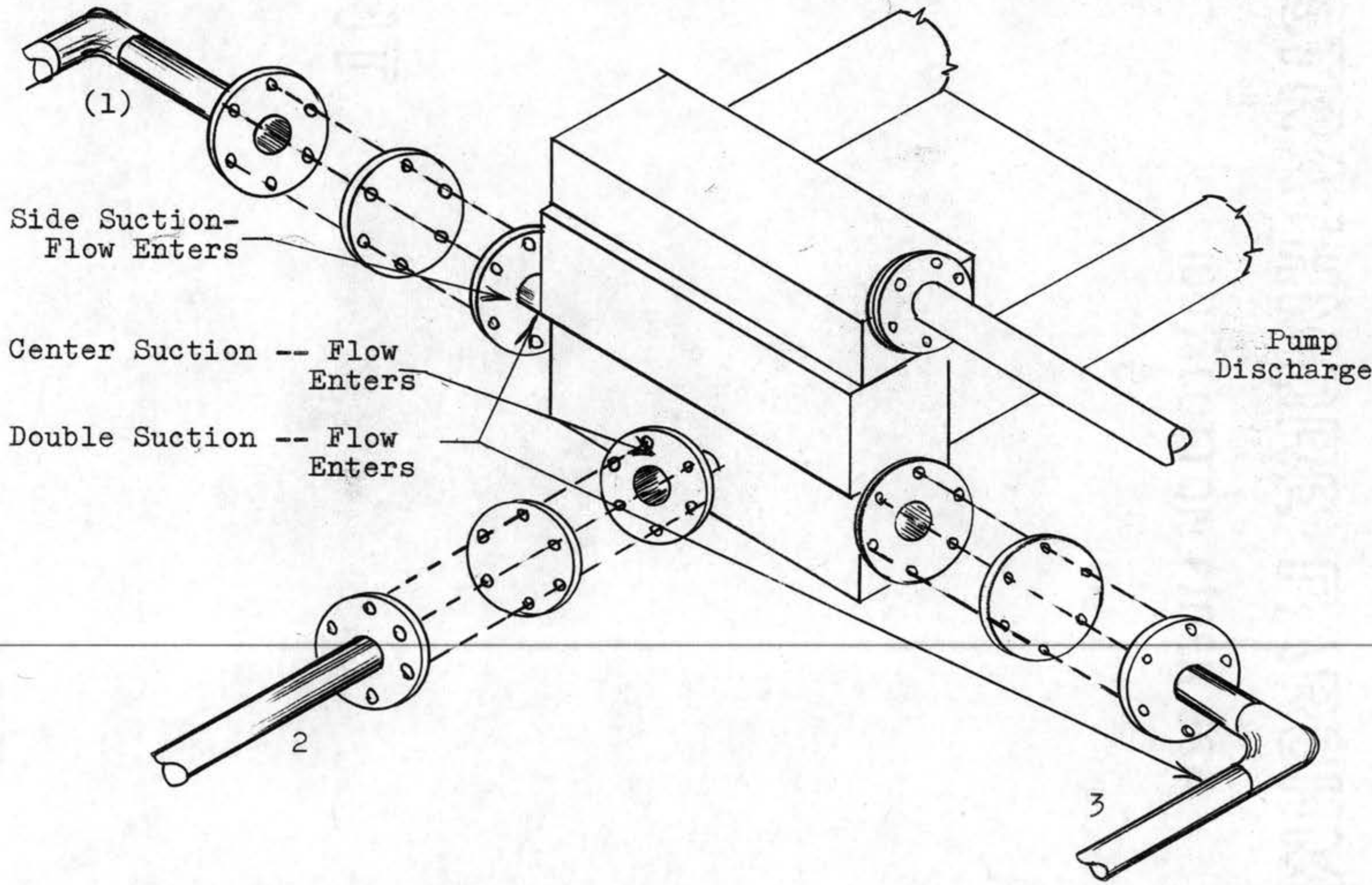


Fig. 2-1

Sketch of Pump Manifold and Suction Piping manufacturers recommendations were to operate with double suction at speeds greater than 200 rpm. This recommendation, according to Mr. H. A. Wienecke, Chief Design Engineer, Gaso Pump and Burner Company, was the result of a limited number of tests conducted on the original pump. The tests indicated smoother operation with double suction. A model of this pump with 3-inch suction was built in 1957. Again, double suction was suggested for speeds above 200 rpm. This recommendation followed a series of tests run on the pump with noise level and smoothness of operation as the criteria for optimum performance. It is now an established fact that the side suction is the noiser, however, this does not mean necessarily that it is less efficient. Some leading pump designers believe that the surging action against the blind flange could possibly supercharge one or more of the cylinders, causing an increase in efficiency. With this in mind, efficiency and other factors were considered in this study. Each of the suction conditions will now be considered separately.

Single Side Suction

Field tests have shown this arrangement to be very noisy. It is thought that the noise is due to the fluid surging against the blind flange. A noted design engineer related instances where the replacement of the blind flange with a bull plug eliminated some of the noise. The bull plug partially eliminated the surges by changing the length of the chamber to avoid resonance or by absorbing the pressure waves. The dampening of the surge by using the bull plug varified that the blind flange is a cause of the knocking.

Double Suction

This condition, while yielding smoother operation with the elimination of the flanges, causes a condition of extreme turbulence since two fluid streams are trying to flow in opposite directions. A loss of efficiency could result from this increased turbulence.

Single Center Suction

This arrangement has inherent advantages over the other two piping systems. There are no opposing streams, as in double suction, therefore, the flow should be less turbulent. The center suction system also has several advantages over the side suction arrangement. The initial direction of the fluid motion is not perpendicular to the flanges, therefore, less vibration should occur. The center suction should be more efficient than the side system because there occurs less bending of the stream lines with center suction.

CHAPTER III

TEST APPARATUS

Description of Apparatus Tested

The apparatus tested was a single-acting Horizontal, Triplex Plunger Pump (Gaso Fig. 3365) manufactured by Gaso Pump and Burner Company, Tulsa, Oklahoma. The pump had a stroke of three inches with a two and one-half inch bore. The capacity of the pump was approximately 93 bbls/hr. at 350 rpm with a maximum operating pressure of 805 psi.

For complete description of this pump, see page 13, Gaso catalogue.

Instruments and Apparatus Employed

I. POWER

A. Oil Field Engine

Mfgr:	Buda Company, Harvey, Ill.
Model:	K-428
Type:	Vertical "L" Head, Four cycle
Bore:	4 3/8"
Stroke:	4 3/4"
Displacement:	428 Cu. In.

Note: This engine was precisely calibrated by the test group in order to predict the power input to the pump. (See Appendix A, page 45 for details.)

II. FLOW CONTROL AND STABILIZATION

A. High Pressure Gate Valve

Mfgr:	Vogt Company, Louisville, Ky.
Size:	2"
Rating:	800 Wp at 750 °F
Cat. No:	5-9538 SW

B. Laboratory Equipment: This consisted of a centrifugal pump pumping from a large sump to a tank so piped to deliver a constant head to test apparatus.

C. Several Low Pressure Gate Valves of Various Sizes

D. Stabilizer

Mfgr:	Pulsating Engineering Co. Tulsa, Oklahoma

E. Desurger

Mfgr:	Westinghouse Air Brake Co. Wilmerding, Pa.

III. FLOW MEASUREMENT

A. Volumetric Tank: See Appendix B, page for calibration and fabrication details.

B. Platform Scale

Mfgr:	Howe Scale Co., Rutland, Vermont
Serial No:	6005053
Capacity:	250 Lbs.
Least sub-division:	.01 Lb.

Note: These scales were recently purchased and calibrated.

C. Tank Equipped With Gate Valves

D. Diverter

See Appendix B for fabrication details.

IV. TIMING

A. Electric Timer

Mfgr:	Standard Electric Time Co. Springfield, Mass.
Type:	S-1
No:	42671
Least Sub-division:	.01 Sec.

B. Two Electric Timers

Mfgr:	Standard Electric Springfield, Mass.
Least Subdi-vision, 1:	.01 Min.
Least Subdi-vision, 2:	.001 Min.

C. Stopwatch

Mfgr:	Sonex
Least Subdi-vision:	0.1 Sec.

V. PRESSURE

A. Well-Type Manometer

Mfgr:	Trimount Inst. Co. Chicago, Ill.
Type:	30 W
Serial No:	2033
Range:	0-30"
Least Subdi-vision:	0.1 Inch
Fluid used:	Oil - s.p. = 1

B. Well-Type Manometer

Mfgr:	Meriam Instrument Co. Cleveland, Ohio
Type:	W
Model:	A-324
Serial No:	None
Least Subdi-vision:	0.1 Inches
Fluid used:	Mercury

C. Well-Type Manometer - Multiple-Scale Selection

Mfgr:	Meriam Instrument Cleveland, Ohio
Type:	A-338 A
Serial No:	8280
Range:	61 Inches
Scale Used:	Pounds per Square Inch
Range:	0-30 psi
Least Subdivision:	0.1 psi

D. Pressure Gage

Mfgr:	Jas. P. Marsh Corp. Skokie, Ill.
Type:	Bourdon tube
Range:	0-1000 psi
Least Subdivision:	20 psi

E. Aneroid Barometer

Least Subdivision:	.02

VI. SPEED

A. Bristol Counter

Mfgr:	G. J. Root Co. Bristol, Conn.
Range:	4 digits

B. Strobotac

Mfgr:	General Radio Co.
Type:	631-BL
Serial No.	27952
Range:	600-3600 rpm
Least Subdivision:	10 rpm

C. Revolutions Counter

Mfgr:	Miller and Falls Co.
No.	73310
Type:	J52
Least Subdivision:	1 Revolution

VII. TEMPERATURE

A. Thermometer and Well

Mfgr: Refinery
Range:
Least Subdivision: 2°F

B. Two Liquid in Glass Thermometers

Mfgr: Fisher Instrument Co.
Range:
Least Subdivision: 10°C

VIII. CALIBRATION INSTRUMENTS

A. Dead Weight Tester

Mfgr: Manning, Maxwell and Moore, Inc.
Type: 1300
Serial No: 1-57-10
Range: 10 to 2500 psi

B. Dynamometer

Mfgr: General Electric
No: 7278658
Type: TLC 2556H
Model: 26-6-439

690 amp; 250 volts

Capacity:

As motor - delivers 200 hp
As generator - absorbs 250 hp
Torque Arm = 21.008 In.

C. Dynamometer Scales

Mfgr: Toledo Scale Co.
Model: 9704
Serial No: 773976
Capacity: 500 Lb.

D. Tank and Scales (Items 1 and 2 under III - Flow Measurement)

Test Method

Preliminary Procedure

Preparation for the conduction of the actual experiment was considerably more extensive and required a great deal more time than did the test proper. Analytical investigation was made of the various methods for determining power input to the pump, measuring flow rate, determining pump speed, measuring a surging pressure and measuring increments of time. Suitable instruments and methods were selected and employed. These are outlined in a following paragraph. The power input to the pump was determined by testing the Buda Engine and plotting suitable curves. A summary of the engine test is included in Appendix A, page 45. After the engine was tested, instrumentation selected, and necessary equipment fabricated, the test stand was constructed. Seemingly, time was spent unnecessarily in building the test setup, but the test group contained only two persons; therefore, ease of operation was of prime concern.

The test stand as constructed was quite versatile. The suction piping was arranged so that it could be connected in any one of the three arrangements stated previously and altered for negative suction with a minimum of effort. Figure 3-1 shows the pump with center suction (Positive or negative).

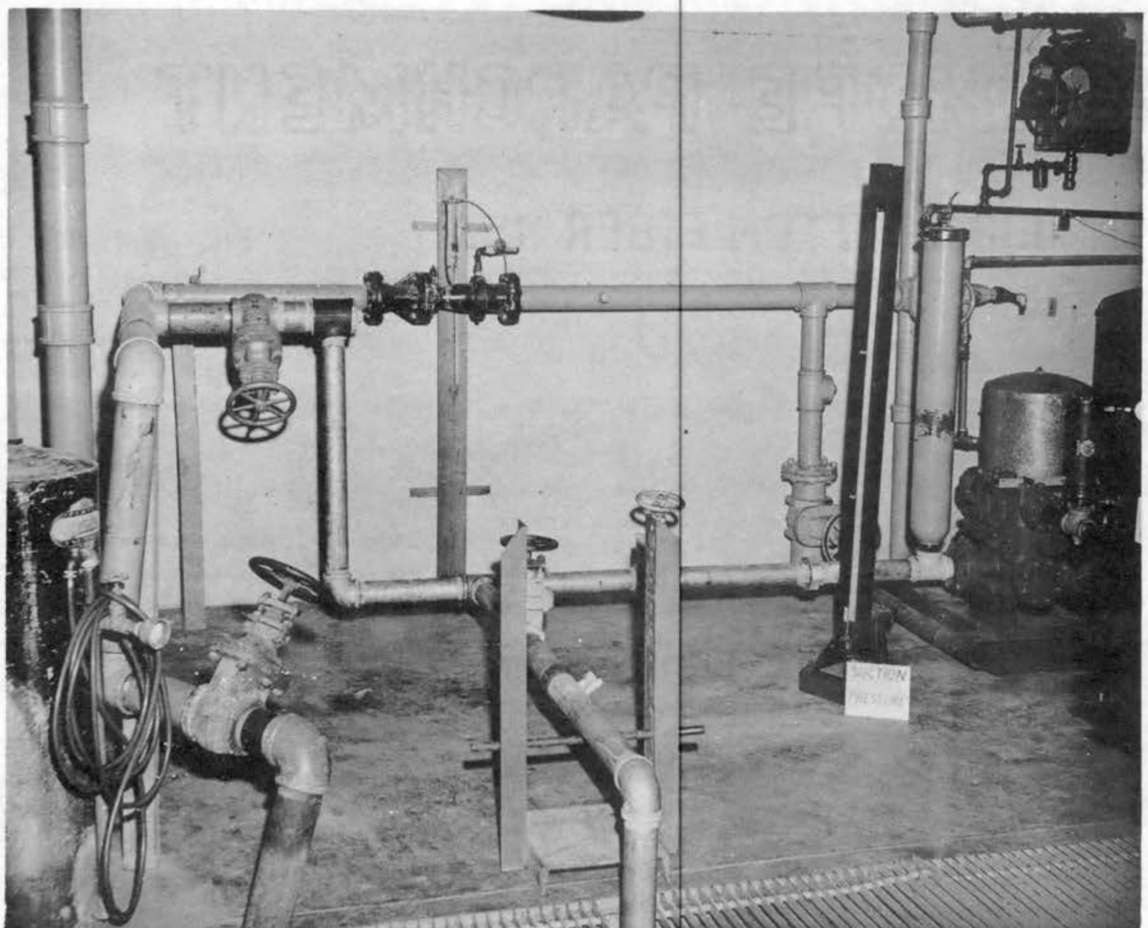


FIGURE 3-1

C E N T E R   S U C T I O N   P I P I N G   A R R A N G E M E N T

It should be noted that the pump was never run with an extraneous suction pipe connected. Blind flanges were always installed on the pump for any suction port that was not being used.

Procedure

Every effort was made to attain simplicity in the test procedure. Some of the instrumentation may seem unduly complicated and too precise for experiments of this nature, but preliminary tests indicated that the difference in results obtained from the three suction piping arrangements would be small; therefore, it was necessary to keep instrumentation error at a minimum. The instruments as they were used by this group gave sufficient accuracy.

Before discussing the test procedure, it should be noted that care was exercised in the installation of the Buda Engine on the pump to insure that conditions affecting power output; namely, exhaust pressure, fuel pressure, etc., could be altered to agree exactly with calibration conditions.

Test runs were made at initial pump speeds of approximately 250, 350, and 400 rpm, for each of the three suction piping arrangements. With the pump running at a constant speed and with a given piping system, the suction pressure was varied from 5 psig to that point at which a noticeable decrease in volumetric efficiency or excessive knocking occurred. This constituted a total of (151) runs.

Conducting so many runs seems an extravagant waste of time, but it was deemed necessary to obtain realistic and reliable results.

All tests were run at a discharge pressure of approximately 800 psig.

For positive suction pressure readings, the test procedure was basically as follows: Suction pressure, discharge pressure, fuel pressure, and engine speed (i. e., pump speed) were set to predetermined values and held relatively constant. Records were kept of these values to indicate that they did remain constant. Flow rate, manifold vacuum and gas temperature were measured and recorded.

The procedure was basically the same for the negative suction runs, but it should be noted that when cavitation began to develop, the engine speed increased while discharge pressure remained constant. This was due to a decrease in flow rate. Before beginning a run, effort was made to allow the engine speed to stabilize, however, in some instances this was not possible. Flow rate and engine speed were averaged for relatively the same time interval; thus, results obtained were reasonably realistic.

Methods of Measurement

To satisfy the stated objective of the test, observed data included those quantities listed in the following Table. The Table also indicates the units of the item as read from the instrument and the method employed to obtain the measurement of the quantity.

TABLE III-I

METHODS OF MEASUREMENT

Observed Data	Units	Method of Measurement
1. Approximate speed	rpm	Strobotac
2. Manifold vacuum	In.-Hg.	Meriam well-type manometer. Fluid - Hg.
3. Fuel pressure	In.-H_2O	Tri-mount well-type manometer. Fluid-oil sp.gr. = 1
4. Fuel temperatures	^oF	Liquid-in-glass thermometer
5. Suction pressure	psi	Special Merriam well-type manometer. Scale read directly in psi. Fluid - Hg.
6. Discharge pressure	psi	Bourdon pressure-gage
7. Pump revolutions	rev.	Bristol Mechanical Counter
8. Time	Minutes	Electric timer
9. Weight on scales before and after addition of fluid to volumetric tank	Lbs.	Platform scales
10. Time to till tank	Seconds	Electric timer
11. Barometric pressure	In.-Hg.	Aneroid Barometer and Meteorology Department of O. S. U.
12. Water temperature	^oF	Liquid-in-glass thermometer

Figure 3-2 shows the pressure measuring instruments and location of surge dampeners.

The procedure and methods listed on the preceding pages made possible the attainment of the stated objective of the test, and, therefore, conclude this description of the test methods.

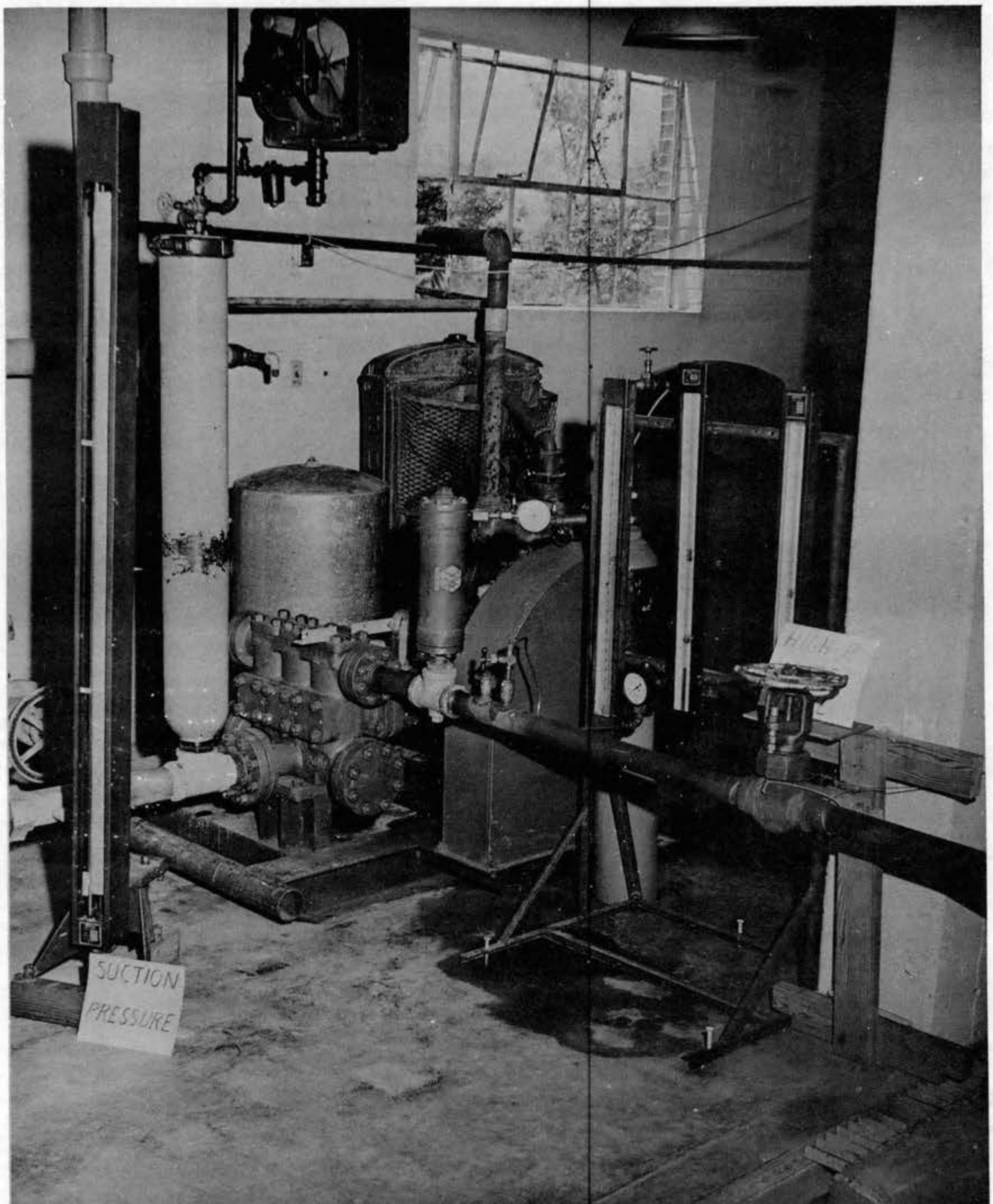


FIGURE 3-2

F L U I D E N D O F T H E P U M P

Note - Pressure measuring instruments and
location of surge dampeners.

CHAPTER IV

DISCUSSION OF TEST ERROR

It is desirable to know the over-all accuracy of this test. In other words, one would like to be able to state that the plotted results are accurate within a certain percentage. The estimation of probable error in testing, determination of the accuracy, or establishment of the validity of a group of points, however, requires a statistical approach. The results of this test did not lend themselves to a statistical analysis. A statistical analysis has no meaning or significance unless a large number of values are studied. In some cases in this study, only one and never more than two or three test runs were made at a particular value of suction pressure; therefore, a statistical study of the curves obtained in this test could not be conducted. Although a statistical analysis was impractical, a discussion of probable errors based on instrument accuracy is presented.

Volumetric Tank Calibration

The calibration procedure is outlined in Appendix B. The scales used to calibrate the tank were new and they were calibrated with standard weights. The tank was filled only three times. The readings were very consistent; therefore, it was considered unnecessary to fill the tank more times.

The three values are:

$$34.107 \text{ ft.}^3,\ 34.177 \text{ ft.}^3 \text{ and } 34.148 \text{ ft.}^3 \ .$$

The arithmetic average of these volumes was found to be 34.144 ft.^{3}. The root mean square of these numbers is 34.14408. The volume of the tank was taken to be 34.144 ft.^{3}. Only three values of the tank volume were available; therefore, a statistical analysis to obtain standard deviation, confidence level, etc., has no meaning.

A crude value of the error involved in using 34.144 ft.^{3} as the true tank volume is:

$$\text{Error} = \frac{\text{maximum deviation from mean}}{\text{mean}}$$

$$= \frac{34.144 - 34.107}{34.144} \times 100 = 0.105\% \ .$$

Maximum Error in η_v

$$\eta_v = \frac{Q_A}{Q_T}$$

$$\text{maximum error in } \eta_v = \frac{\eta_{v\,max} - \eta_v}{\eta_v}$$

$$\text{where: } \eta_{v\,max} = \frac{(Q_A) \text{ maximized}}{(Q_T) \text{ minimized}}$$

$$Q_A = \frac{V_T - V_A}{T_A} \qquad (Q_A)_{max} = \frac{(V_T)_{max.} - (V_A)_{min.}}{(T_A)_{min.}}$$

$$Q_T = \frac{Rev}{T_T} \times D \qquad (Q_T)_{min} = \frac{Rev \times D}{(T_T)_{min.}}$$

$$(V_T)_{max} = V_T + \Delta V_T$$

$$= 34.144 + .036 = 34.180$$

where: V_T is volume of tank

ΔV_T is maximum deviation = .036

$$(V_A)_{min} = (V_A) - \Delta V_{meniscus} - \Delta V_{scales}$$

Assuming that the error in reading each meniscus was no greater than 1/8 inch, the maximum error would be:

$$\Delta V_{meniscus} = 2 \times (1/8)(1/12) \frac{\pi 6^2}{(4)(144)} = \underline{\underline{0.00102 \text{ ft.}^3}}$$

The scales were sensitive to 0.05 Lb., therefore, the maximum error in weighing the volume added to fill to the reference is:

$$\Delta V_{scales} = \frac{2 \times .05}{62.3} = \underline{\underline{.00161 \text{ ft.}^3}} .$$

Then:

$$(V_A)_{min.} = V_A - .00263 \text{ ft.}^3 .$$

The maximum error in timing was the reactive time of the clutch, plus twice of one-half the least subdivision on the timer.

$(T_A)_{min} = T_1 - \Delta T_1$

$\Delta T_A = \frac{1}{60}$ sec. + 2 x(1/2)(.01)

= .0167 + .0100 = .0267 sec.

$(T_T)_{max} = T_2 + \Delta T_2$

ΔT_T = 1/60 ($\frac{1}{60}$) min. + 2 x 1/2 x .01 min.

= .00028 + .01 = .01028 min.

A run was chosen that would give the highest percentage of error. Run 44 was chosen. From the data, it may be obtained that:

V_A = 0.135 ft.³ Rev. = 4300

T_A = 204.44 sec. η = 96.97

T_T = 10.68 min

$V_{T\ max}$ = 34.180 ft.³

$(V_A)_{min}$ = 0.135 ft.³ - .00263 = .1324 ft.³

$(T_A)_{min} = T_1 - \Delta T_1$ = 204.44 - .0267 = 204.413 sec.

$(T_T)max = T_2 + \Delta T_2$ = 10.68 + .01038 = 10.6903 .

Then:

$(Q_A)_{max} = \frac{34.180 - .1324}{204.413}$ = .16656 cfs.

$(Q_T)min = \frac{4300}{10.6903} \times \frac{2.5566 \times 10^{-3} ft.^3}{60}$ = .17139 cfs.

Then:

$$\eta_{v\,max} = \frac{.16656}{.17139} = 97.182 .$$

Maximum error in $\eta_v = \frac{\eta_{v\,max} - \eta_v}{\eta_v}$

$$= \frac{97.182 - 96.97}{96.97} \times 100$$

$$= \frac{.212 \times 100}{96.97} = .219\% .$$

The writer considers the above to be an indication of the accuracy of this test.

Errors in Pressure Measurement

All pressure measurements, with the exception of discharge, were made with well-type manometers. The manometer is an accurate means of measuring pressure and, does not require calibration. Discharge pressure was determined by a bourdon-tube pressure gage. This gage was calibrated several different times with a dead weight tester. The accuracy of measuring pressure with this gage was limited by the readability of the gage. The least subdivision on the gage was 20 psig. It is the opinion of the writer that pressure was determined correct to a plus or minus 10 psig or 2 1/2%. To use the type of instrumentation described here, one had to first reduce the pressure surges. This was accomplished by placing a suction stabilizer in the suction piping and a desurger in the discharge piping.

Errors in Determination of Horsepower Required by the Pump

Table A-I shows that the calibration of the Buda engine was correct to 2.0%. The only other factor that affected the accuracy of power determination was the inability to read discharge pressure correctly.

An incorrect reading of 20 psig in discharge pressure would result in a 2 1/2% error in power determination.

Other Factors Affecting Accuracy of Results

The determination of volumetric efficiency was the prime objective of this study. It was considered to be the most indicative characteristic of the pump. There were several factors that did not produce error in determining volumetric efficiency, but may have caused small errors in other quantities. One such factor was the variation in frequency of the A.C. current used to power the electric timers. The frequency of the University power station output varied during this test from 59.7 to 60.3 cycles per second. To correct for this variation in frequency, one would have to multiply the time obtained with a timer by the ratio of frequencies $f_r = f_c/f_a$ Electric timers, powered by the same current, were used to determine both the actual and theoretical flow rates. Volumetric efficiency is $\eta = \frac{Q_A}{Q_T}$. Each flow rate is multiplied by the same correction factor for frequency variation; $\eta = \frac{Q_A\,(f_r)}{Q_T\,(f_r)}$.

The variation of line frequency has no effect on determination of volumetric efficiency. The effect of this variation on accuracy of power determination was small. The effect on either flow rate was less than 0.5%.

Another possible source of error was the result of water collecting on the sides of the volumetric tank. This error was eliminated by setting the lower reference level immediately after draining the tank, when calibrating the tank and for all of the test readings.

CHAPTER V

OBSERVED DATA AND RESULTS

The results of this study are presented in Tables and Graphs on the following pages.

Run No	P_b (in. Hg)	N_e (App) (RPM)	P_v (in. Hg)	P_g (in H_2O)	T_g (°F)	P_s (psig)	P_d (psig)	Rev	Time (min.)	$Wt._1$ (lbs.)	$Wt._2$ (lbs.)	Time (sec.)	T_w (°F)
29	28.70	735	5.0	8.6	94	5.0	800	4750	19.051	43.81	34.42	328.10	74
30	28.71	735	5.0	8.6	94	3.99	800	4000	16.025	63.60	55.25	329.14	75
31	28.72	730	5.05	8.6	94	3.02	800	4300	17.288	95.40	92.98	330.36	74
32	28.73	740	4.9	8.6	93	2.03	800	4900	19.539	90.39	86.63	327.82	74
33	28.74	735	5.3	8.7	93	0.94	795	5400	21.610	80.57	73.73	328.17	74
35	28.85	1050	4.0	7.7	94	4.96	800	6300	17.556	97.01	85.80	228.63	75
36	28.83	1040	4.25	7.8	94	4.07	795	4100	11.512	89.00	79.69	230.47	76
37	28.80	1040	4.3	7.8	93	3.03	795	4200	11.823	71.52	63.42	231.25	76
38	28.80	1040	4.3	7.8	96	1.97	795	4900	13.804	92.02	87.81	231.84	76
39	28.80	1040	4.3	7.8	99	1.05	795	3500	9.910	83.82	78.82	233.09	76
40	28.75	1190	3.88	7.5	90	5.0	800	4500	11.180	83.26	67.57	203.54	76
41	28.75	1180	3.85	7.6	90	4.0	805	4400	10.968	80.26	75.05	205.49	76
42	28.75	1175	3.83	7.6	90	2.98	805	4500	11.225	93.94	89.20	206.66	76
44	28.75	1180	3.85	7.5	90	2.0	800	4300	10.680	67.33	58.90	204.44	76
45	28.75	1180	3.85	7.5	90	0.97	800	4700	11.675	92.75	89.24	205.07	77
46	28.75	720	5.3	8.7	86	5.03	800	3600	14.800	86.64	83.79	337.21	78
47	28.75	720	5.3	8.7	87	4.00	800	3600	14.810	80.49	78.11	337.68	78
50	28.72	720	5.02	8.7	85	2.93	800	3600	14.780	92.53	88.18	336.61	80
51	28.70	730	5.1	8.7	89	2.00	800	3700	14.890	95.26	89.69	329.73	80
53	28.67	720	5.1	8.7	94	0.93	795	3500	14.152	94.13	89.17	331.57	80
55	28.65	1000	4.2	7.7	94	5.03	800	4300	12.610	74.45	71.37	241.37	80
56	28.64	1000	4.2	8.0	94	4.00	795	3900	11.470	79.33	72.70	241.47	80
57	28.63	1000	4.2	8.1	96	3.00	800	3800	11.160	82.17	75.19	241.54	80
58	28.63	1000	4.2	8.1	94	2.00	800	3500	10.290	68.44	62.74	241.42	80
60	28.63	1000	4.2	8.0	90	0.90	800	4100	10.010	78.88	74.03	240.72	81
61	28.55	1175	3.7	7.6	90	4.95	800	4700	11.750	82.51	72.47	205.46	81
62	28.55	1170	3.7	7.6	90	3.97	800	4300	10.770	80.47	74.00	206.20	82
63	28.56	1170	3.73	7.6	88	2.92	805	5100	12.790	69.12	56.50	206.12	82
64	28.57	1170	3.70	7.5	87	1.92	805	4100	10.270	91.42	85.00	206.35	82
65	28.58	1175	3.70	7.6	87	1.10	805	4200	10.520	78.87	72.57	206.13	82
66	28.61	740	5.15	8.8	82	4.93	800	3600	14.40	92.70	88.18	328.18	82
72	28.60	1010	4.40	8.0	80	5.05	795	3900	11.578	83.45	74.89	243.29	82
77	28.65	1175	3.90	7.6	80	4.95	800	4100	10.340	84.46	62.90	205.34	82
82	29.06	735	4.75	8.6	94	4.98	800	3600	14.444	96.28	92.16	328.96	79
83	29.06	735	4.85	8.6	96	3.98	810	3400	13.643	86.50	77.44	328.30	79
84	29.06	735	4.85	8.6	96	2.90	810	3400	13.667	65.15	56.38	328.90	79
85	29.06	735	4.85	8.6	96	2.00	810	3200	12.842	97.75	96.60	329.48	79
86	29.06	750	5.00	8.6	96	1.0	800	3600	14.139	96.60	89,80	321.76	79
94	29.12	1030	3.80	8.0	90	-2.65	810	1700	4.908	74.60	74.60	238.48	81
95	29.12	1030	3.80	8.0	90	-5.15	810	3700	10.675	74.20	71.23	237.83	82
96	29.12	1070	4.10	7.9	90	-7.45	800	4400	12.142	61.03	53.36	238.86	82
97	29.2	1170	3.33	7.6	90	-1.20	810	3500	8.923	67.28	61.00	210.14	81
105	29.06	1040	3.80	8.0	103	-4.20	810	1500	4.248	61.00	61.00	233.81	83

TABLE V - I OBSERVED DATA

Run No	P_b (in. Hg)	N_e(App) (RPM)	P_v (in. Hg)	P_g (in H_2O)	T_g (°F)	P_s (psig)	P_d (psig)	Rev	Time (min.)	$Wt._1$ (lbs.)	$Wt._2$ (lbs.)	Time (sec.)	T_w (°F)
107	29.06	1040	3.80	8.1	104	-6.35	810	1500	4.240	59.60	58.85	234.90	83
108	29.07	1165	3.70	7.6	102	-1.70	800	1600	4.030	58.85	55.20	207.83	84
113	29.07	1065	3.90	7.9	95	4.95	800	1500	4.209	63.00	53.68	230.25	84
114	29.07	1060	3.90	7.9	95	4.05	800	1500	4.220	61.20	60.45	231.78	85
115	29.07	1055	3.90	7.9	94	2.90	805	1500	4.204	60.45	58.15	230.74	86
116	29.07	1050	3.90	7.9	94	1.80	810	1600	4.505	58.15	54.90	231.93	86
117	29.07	1050	3.85	7.9	94	0.93	810	1400	3.946	54.90	53.15	232.17	87
118	29.07	1180	3.80	7.5	94	4.9	800	1500	3.747	62.70	49.55	205.01	87
120	29.07	1150	3.70	7.6	94	3.95	810	1500	3.819	58.10	52.55	209.70	87
121	29.07	1150	3.70	7.6	94	3.03	820	1500	3.822	59.30	48.45	209.45	87
122	29.08	1150	3.65	7.6	94	1.9	820	1400	3.578	58.80	48.05	209.92	87
123	29.08	1150	3.65	7.6	94	0.98	820	1400	3.576	62.35	59.70	210.55	87
124	29.09	750	4.50	8.6	86	-0.80	820	1800	6.756	63.25	62.25	326.49	83
125	29.09	750	4.50	8.6	86	-3.05	820	1400	5.571	62.25	58.55	326.81	83
126	29.09	750	4.50	8.6	86	-6.15	820	1500	5.948	58.55	56.15	328.61	83
127	29.09	800	4.95	8.5	86	-8.40	820	1600	5.897	56.15	55.00	328.32	84
129	29.05	1010	3.50	8.1	92	-0.98	810	1700	5.033	60.90	60.55	243.76	85
133	29.05	1000	3.45	8.2	105	-4.83	820	1400	4.164	58.05	58.05	245.64	85
134	29.05	1000	3.45	8.2	101	-5.95	820	1500	4.439	58.05	57.95	245.15	85
135	29.05	1015	3.60	8.2	100	-7.00	820	1600	4.627	58.50	57.10	245.79	85
136	29.02	1130	4.2	8.0	100	-7.80	820	1600	4.171	57.10	54.30	243.30	86
137	29.02	1170	2.9	7.5	100	- .95	810	1600	4.013	58.95	55.00	206.95	86
138	29.02	1180	3.0	7.5	99	-3.05	820	1500	3.733	55.00	48.70	205.40	86
139	29.02	1180	3.0	7.5	98	-4.05	820	1400	3.479	60.20	55.30	205.19	86
140	29.02	1180	3.0	7.5	98	-5.05	820	1600	3.968	55.30	49.80	205.61	87
141	29.02	1230	3.2	7.3	98	-6.03	830	1700	4.053	59.00	47.90	205.02	87
142	*	730	4.2	8.5	90	-0.7	820	1500	6.094	57.60	55.10	333.46	85
143		710	4.1	8.5	90	-4.3	820	1400	5.826	55.10	51.75	342.07	85
144		700	4.0	8.5	90	-6.38	840	1500	6.250	58.20	58.20	346.22	85
146		710	4.0	8.5	89	-8.15	850	1500	6.234	61.70	56.00	360.40	85
148			4.8	8.4	89	-8.53	850	1500	5.563	56.00	54.55	331.92	85
149		1170	3.2	7.5	89	-3.90	820	1500	3.791	54.55	51.90	209.38	86
150		1190	3.3	7.5	89	-4.95	820	1500	3.731	59.45	52.50	208.90	86
151		1210	3.4	7.4	88	-6.00	820	1500	3.659	59.00	52.20	209.41	86
152		1235	3.5	7.4	88	-6.30	800+	1700	4.053	59.40	53.30	206.84	86

TABLE V - I-CONT'D OBSERVED DATA

TABLE V - II TABULAR RESULTS

Double Suction

$N_{ip} \overset{\circ}{=} 250$

RUN NO	P_s	η
29	5.00	97.65
30	3.99	97.46
31	3.02	97.45
32	2.03	97.57
33	0.94	97.46

$N_{ip} \overset{\circ}{=} 350$

RUN NO	P_s	η
35	4.96	97.25
36	4.07	97.23
37	3.03	97.23
38	1.97	97.16
39	1.05	97.21

$N_{ip} \overset{\circ}{=} 400$

RUN NO	P_s	η
40	5.00	97.20
41	3.98	97.07
42	2.98	96.60
44	2.00	97.03
45	0.97	97.03

TABLE V - III TABULAR RESULTS

SIDE SUCTION

initial speed $\overset{\circ}{=}$ 250 rpm

RUN NO	P_s	η
46	5.03	97.59
47	4.00	97.49
50	2.93	97.21
51	2.00	97.36
52	0.93	98.01
124	-0.80	97.48
125	-3.05	97.39
126	-6.15	96.56
127	-8.40	89.88

$N_{ip} \overset{\circ}{=}$ 350 rpm

RUN NO	P_s	η
55	5.03	97.32
56	4.00	97.38
57	3.00	97.17
58	2.00	97.45
60	0.90	97.99
129	-0.98	97.29
133	-4.83	97.10
134	-5.95	96.74
135	-7.00	94.30
136	-7.80	85.75

$N_{ip} \overset{\circ}{=}$ 400 rpm

RUN NO	P_s	η
61	4.95	97.18
62	3.97	97.12
63	2.92	97.12
64	1.92	97.06
65	1.10	97.18
137	-0.95	96.94
138	-3.05	96.73
139	-4.05	96.78
140	-5.05	96.39
141	-6.03	92.73

TABLE V - IV TABULAR RESULTS

CENTER SUCTION

$N_{ip} \overset{\circ}{=} 250$ rpm

RUN NO	P_s	η
82	4.93	97.56
83	3.90	97.56
84	2.97	97.39
86	1.97	97.27
85	0.93	97.36
142	-0.70	97.52
143	-4.30	97.36
144	-6.40	96.38
146	-8.15	92.20
148	-8.57	89.47

$N_{ip} \overset{\circ}{=} 350$ rpm

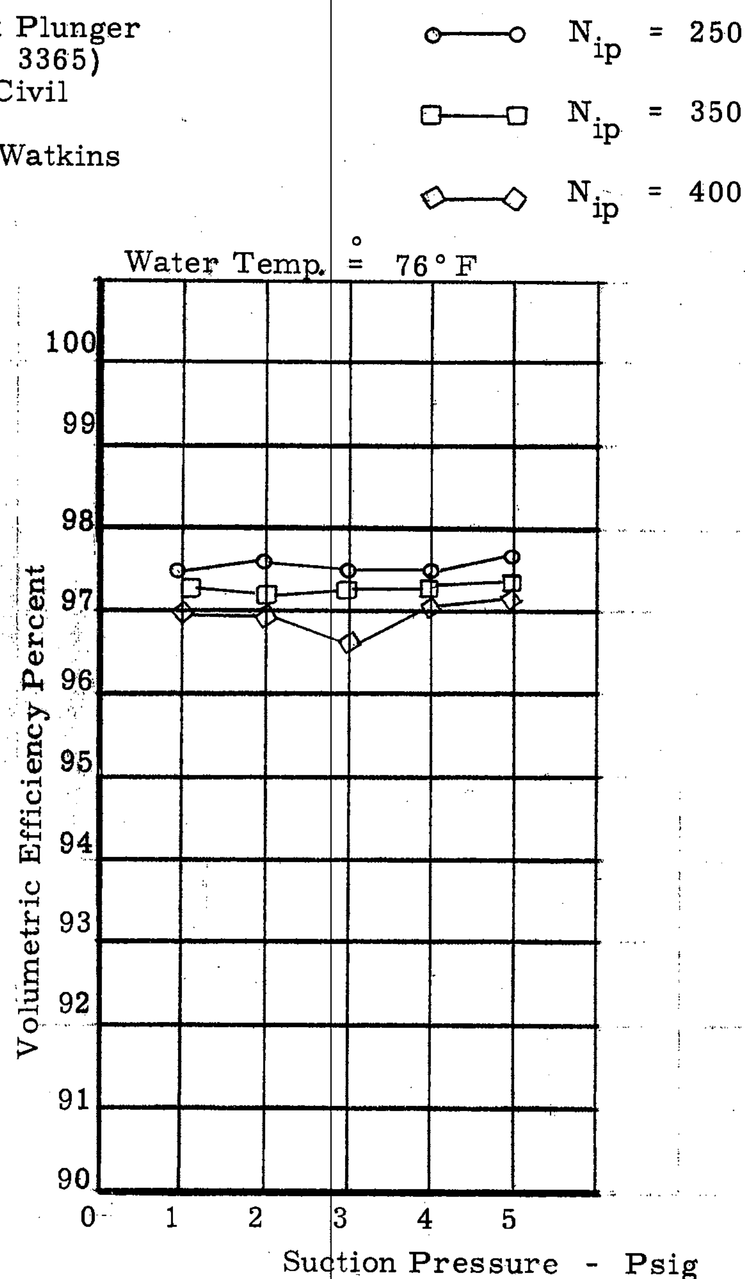
RUN NO	P_s	η
113	4.95	97.17
114	4.08	97.23
115	2.90	97.24
116	1.80	97.16
117	0.93	97.16
94	-2.65	97.02
105	-4.20	97.14
95	-5.15	97.09
107	-6.35	97.01
96	-7.35	92.23

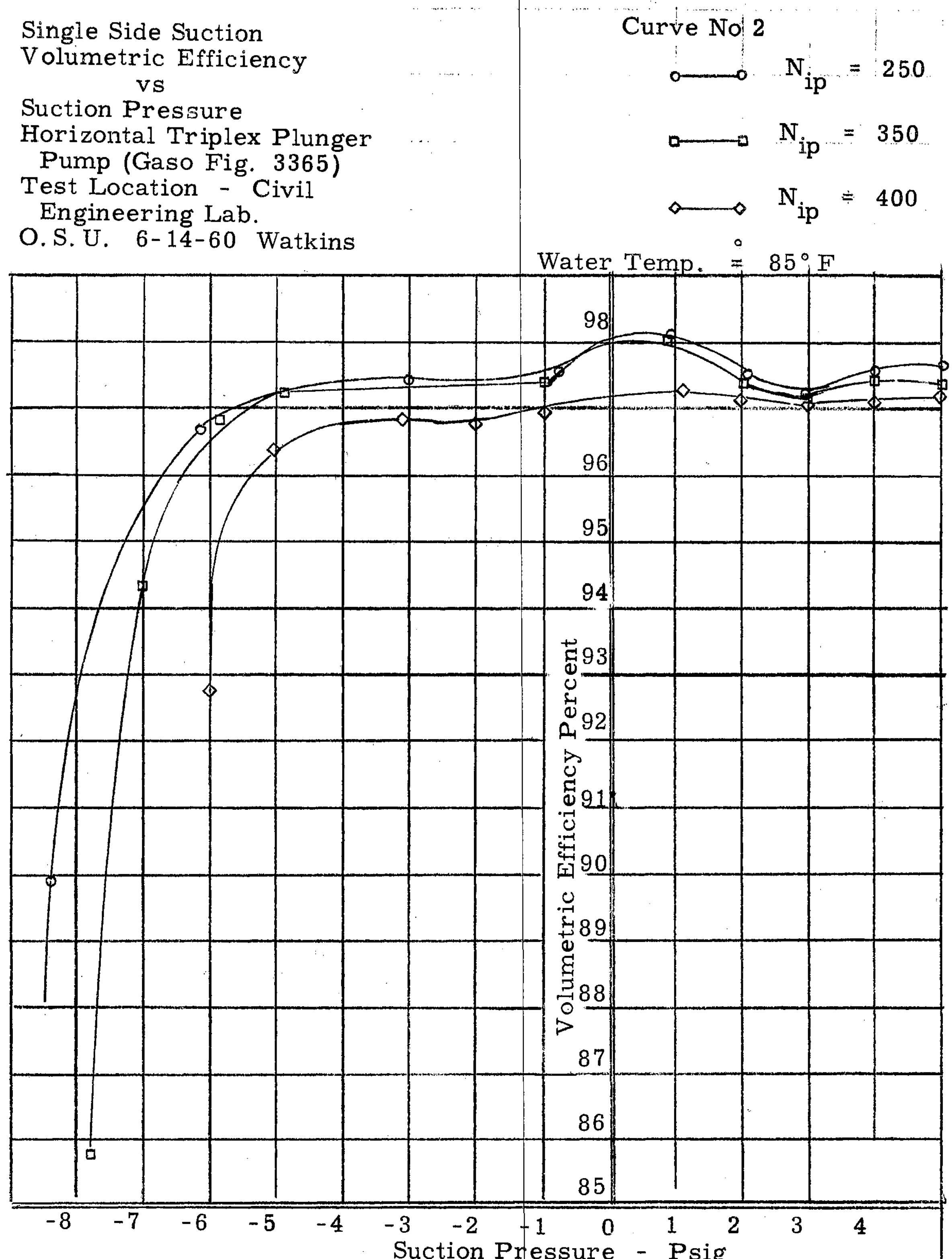
$N_{ip} \overset{\circ}{=} 400$ rpm

RUN NO	P_s	η
118	4.95	97.01
120	3.95	97.01
121	3.03	97.01
122	1.90	97.06
123	0.98	97.12
97	-1.20	96.95
108	-1.70	96.93
149	-3.90	96.62
150	-4.95	95.10
151	-6.00	93.02
152	-6.30	92.11

Double Suction
Volumetric Efficiency
vs
Suction Pressure
Horizontal Triplex Plunger
Pump (Gaso Fig. 3365)
Test Location - Civil
Engineering Lab.
O.S.U. 6-14-60 Watkins

Curve No. 1

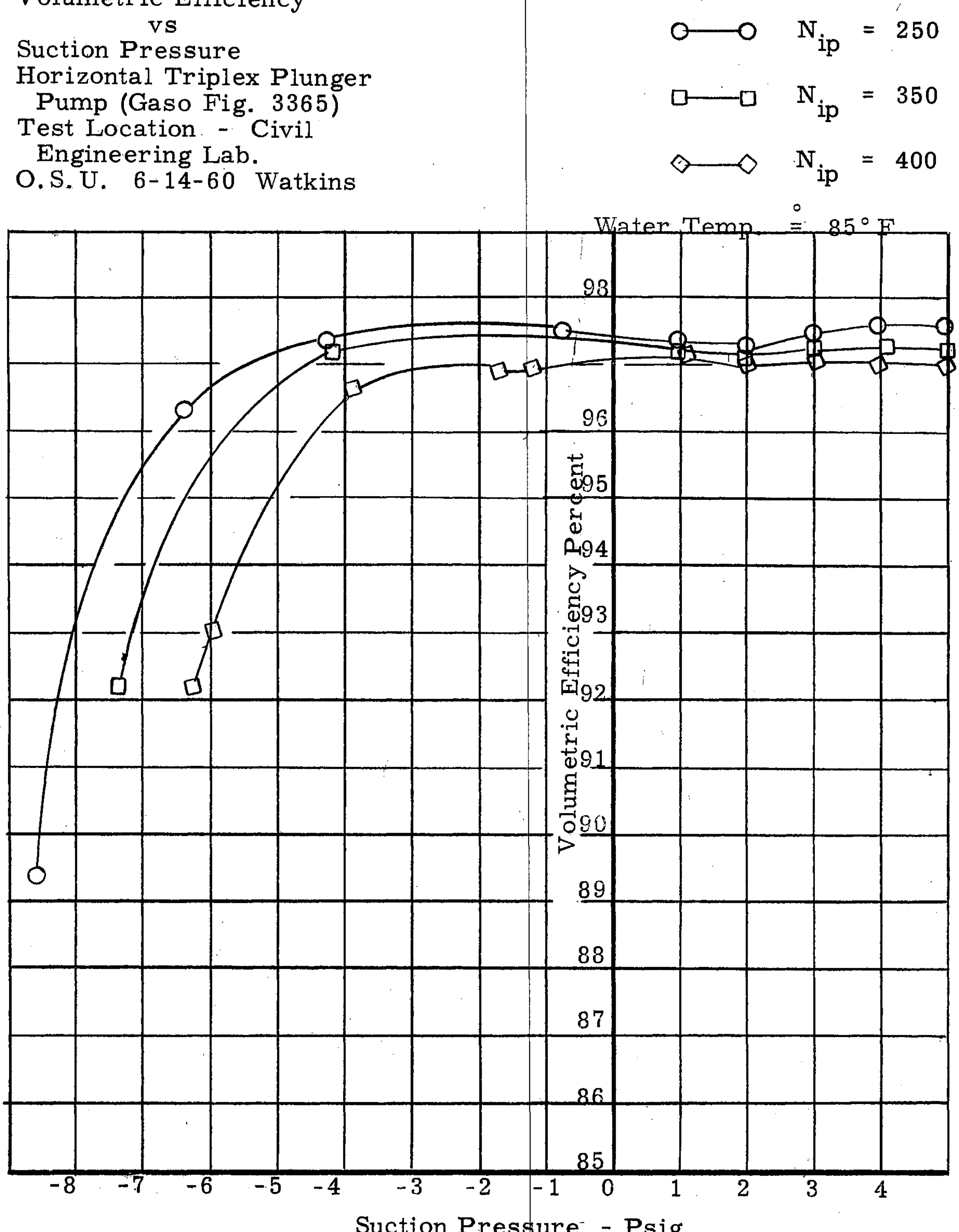




Single Side Suction
Volumetric Efficiency
vs
Suction Pressure
Horizontal Triplex Plunger
Pump (Gaso Fig. 3365)
Test Location - Civil
Engineering Lab.
O.S.U. 6-14-60 Watkins
Curve No 2
N_{ip} = 250
N_{ip} = 350
N_{ip} = 400
Water Temp. = 85° F
98
96
95
94
93
92
91
90
89
88
87
86
85
Volumetric Efficiency Percent
-8
-7
-6
-5
-4
-3
-2
-1
0
1
2
3
4
Suction Pressure - Psig

Single Center Suction
Volumetric Efficiency
vs
Suction Pressure
Horizontal Triplex Plunger
Pump (Gaso Fig. 3365)
Test Location - Civil
Engineering Lab.
O.S.U. 6-14-60 Watkins

Curve No. 3



Curve No. 4

Power, Capacity and Efficiency
Pump ^{vs} Speed

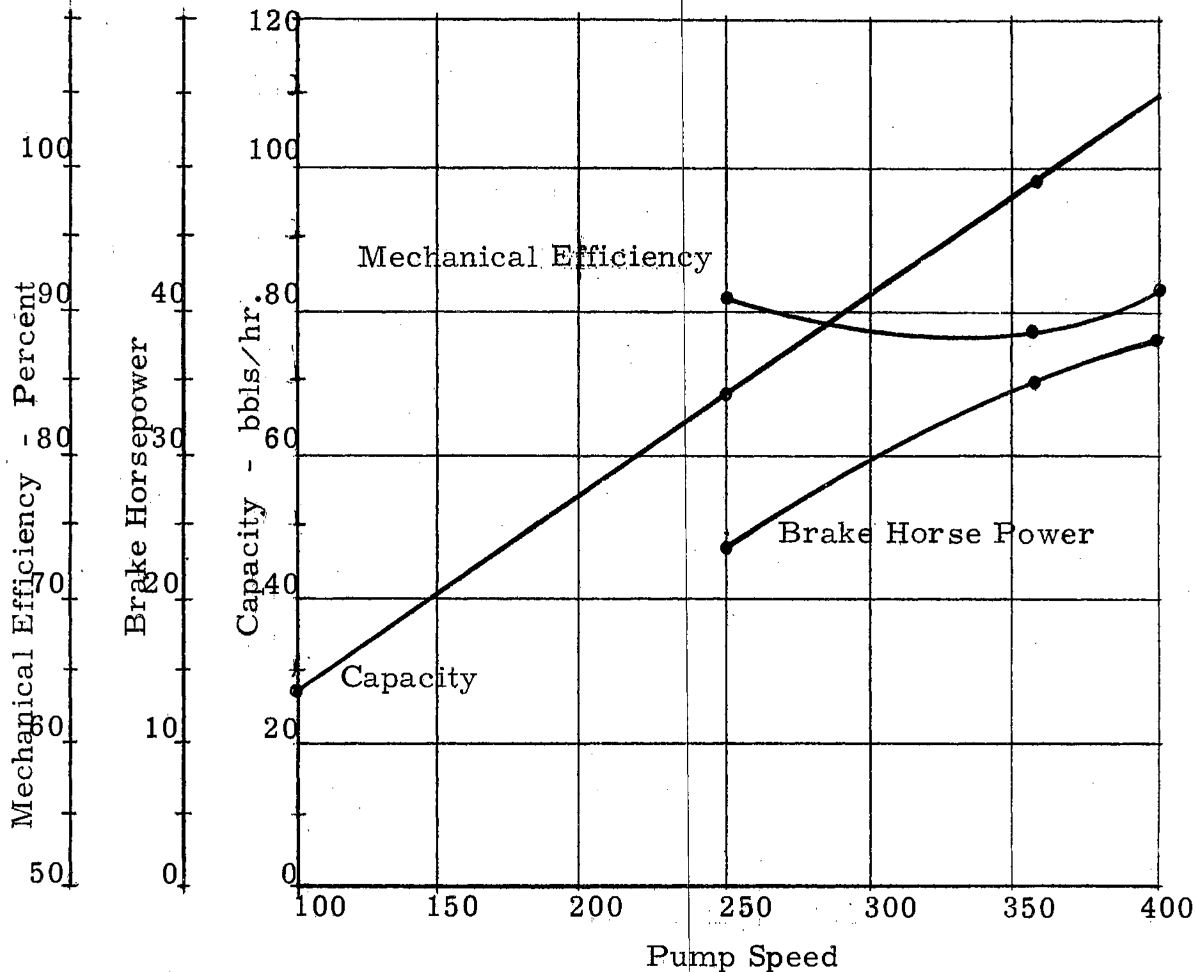
Horizontal Triplex Plunger Pump
(Gaso Fig. 3365)

Type of Suction - Double

Discharge Pressure = 800 psig + 10 psig

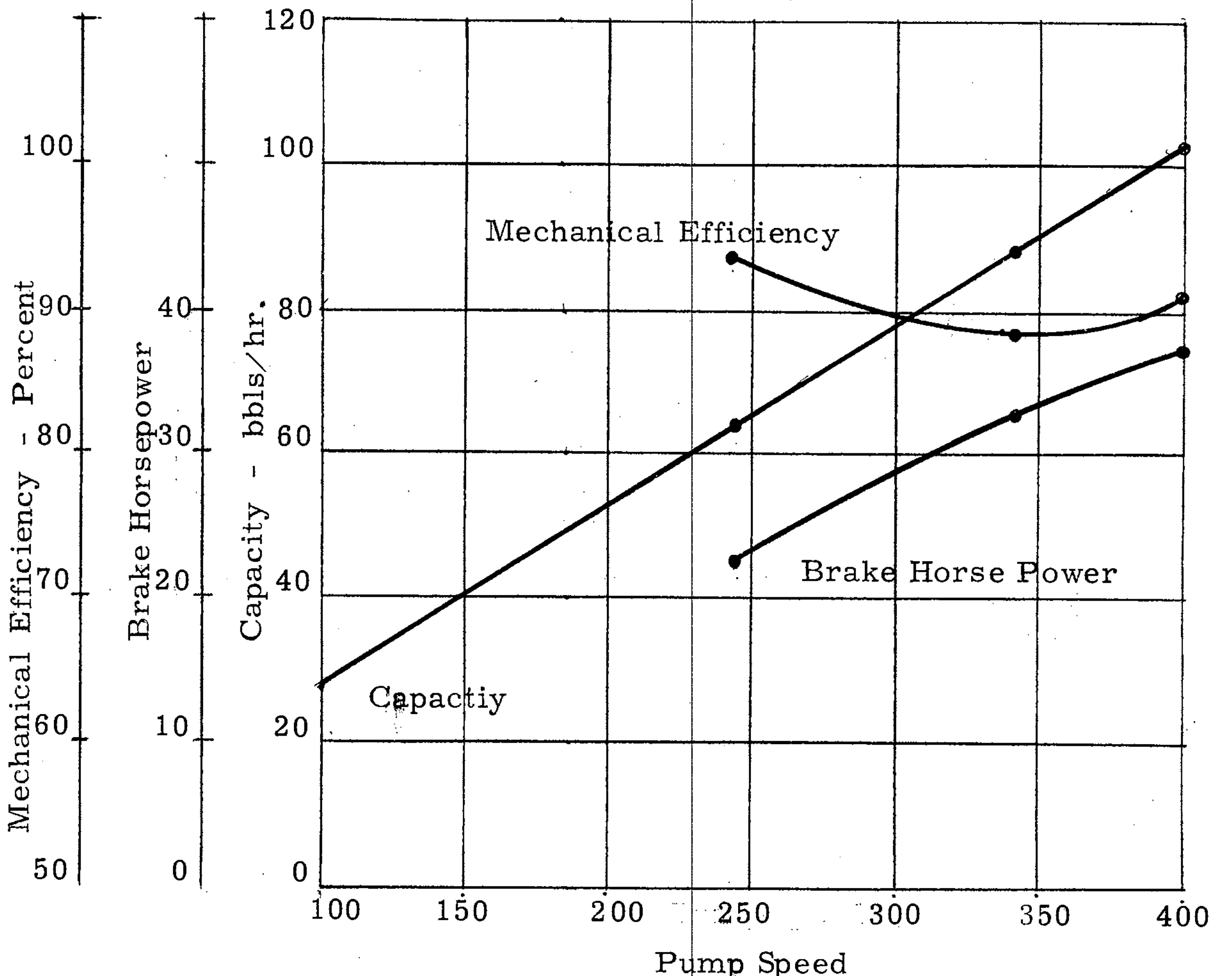
Suction Pressure = 5.0 + 0.1

Test Location - Civil Engineering Lab. O.S.U. 6-25-60 Watkins



Curve No. 5
Power, Capacity and Efficiency
vs
Pump Speed
Horizontal Triplex Plumger Pump
(Gaso Fig. 3365)
Test Location - Civil Engineering Lab.
O.S.U. 6-25-60 Watkins

Type of Suction - Single Side
Discharge Pressure - 800 psig + 10 psig
Suction Pressure = 5.0 + 0.1



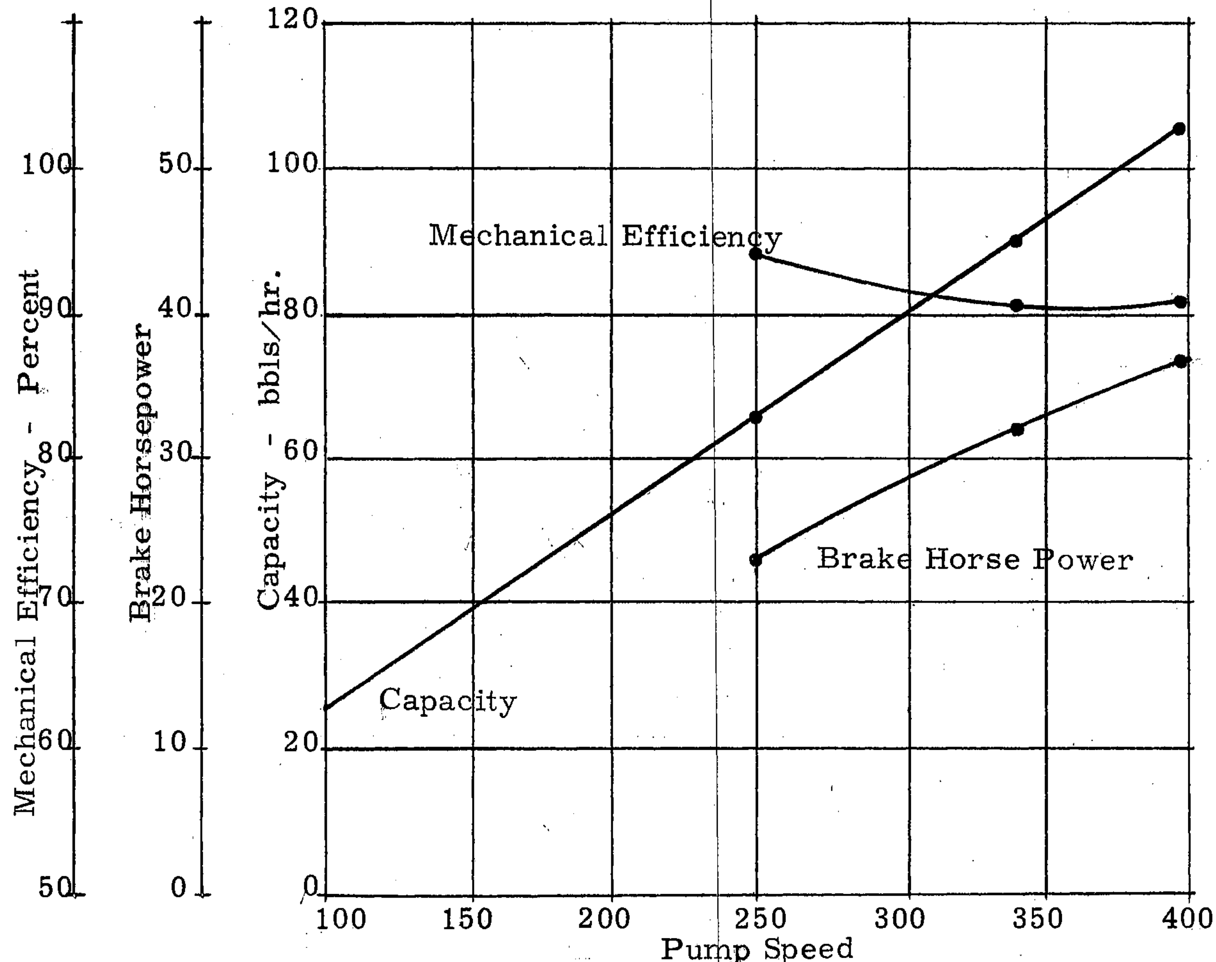
Curve No. 6

Power, Capacity and Efficiency
vs
Pump Speed
Horizontal Triplex Plunger Pump
(Gaso Fig. 3365)
Test Location-Civil Engineering Lab.
O.S.U. 6-25-60 Watkins

Type of Suction: Single Center

Discharge Pressure = 800 psig + 10 psig

Suction Pressure = 5.0 + 0.1



CHAPTER VI

CONCLUSIONS

The purposes of this study, as stated in Chapter I, were:

1. To determine the effect of suction pressure on volumetric efficiency and pump performance for each of three suction piping arrangements and each of three speeds.
2. To determine capacity, efficiency and power as functions of pump speed.

It may be concluded that these objectives were satisfied.

The first startling results of this test, to one unfamiliar with the test apparatus, are the high volumetric efficiencies obtained. Actually, the values are realistic because the pump tested was a new pump and laboratory facilities used in the test were very good. A point to be noted is that volumetric efficiency decreased with increase in pump speed.

Several interesting facts may be discovered by examining the various curves shown in Chapter V. One such fact is the effect of valve spring sizing. This is indicated by the increase in volumetric efficiency at low positive pressures. All the volumetric efficiency curves show the same trend. Volumetric efficiency tends to decrease slightly until about +2 psig is reached, then the curves begin to rise. Possibly, at the higher pressures, the suction pressure is great enough to overcome the forces of the valve springs. The intake valve is held open for a short period of time as the plunger begins its discharge stroke allowing some fluid to flow back into the suction chamber. In a similar manner, some loss in efficiency occurs because of the finite time required for the discharge valves to close. At low pressures of +2 psig and below, the intake valves are not opened by the suction pressure. It may be noted that this effect is more pronounced at low speeds. Another interesting point is that the single side suction shows a greater increase for the low speeds and low pressures than does either of the other two conditions, (Curve No. 2). Some engineers attribute this effect to a supercharging action resulting from fluid vibrating against the blind flange, (Figure No. 2-1.)

At +3 psig, double suction, 400 rpm, the volumetric efficiency dropped approximately 0.57%. A thorough check of data and calculations was made and no grounds were found to question the validity of this reading. No obvious reason offered itself to explain this unexpected occurrence. Noted design engineers concluded, after considerable discussion, that the natural frequency of the valve spring must have been reached for an instant.

Results of this study indicated no significant difference in volumetric efficiency, power required, capacity or over-all efficiency, while operating the pump with any of the three piping arrangements considered. Cavitation accompanied by excessive vibration occurred in the range of pressures from + 5 psig to + 8.5 psig, depending upon pump speed for both center and side suction. Although results indicated little difference in pump performance for the three conditions, the pump seems to run more stably with center suction. This may have been due to the location of the suction stabilizer.

Attention should be drawn to the fact that a suction stabilizer was used in the suction piping and a desurger in discharge piping to dampen surges, thus facilitating pressure measurements. An attempt was made to run the pump without a suction stabilizer, but damaging vibrations developed in the test apparatus. The writer suggests using a suction stabilizer of some nature in suction piping for speeds greater than 300 rpm.

It is the writer's opinion that study should be made of the possibility of streamlining the intake manifold while employing a center suction arrangement. The writer suggests that elimination of the pockets at each end of the manifold would result in smoother operation and slightly higher efficiency.

Results of this study also indicated that investigation should be made to provide a method for correctly sizing both intake and discharge valve springs.

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APPENDIX A

DETERMINATION OF POWER OUTPUT OF A BUDA INDUSTRIAL ENGINE

One of the most difficult problems encountered in this study was the determination of power input to the triplex pump. Consideration was given to several methods. These included:

1) Determining torque by mounting strain gages on the engine shaft between the clutch and belt pulley.
2) A dynamometer arrangement.
3) Cradling the engine and measuring torque directly.
4) Determining power output of the engine as a function of manifold pressure and engine speed.

The last method was suggested by an engineer with the Mid-Continent Pipeline Company. It required less alteration of equipment and was most desirable from an economical standpoint since facilities were available at the Mechanical Engineering Laboratories for performing the required test. This method was chosen.

The engine was moved to the Mechanical Engineering Laboratories, installed in the G.E. dynamometer test stand and tested.

A summary of the complete test is presented in this appendix. The following items were included in this presentation:

1) Stated objective of the test
2) A summary and conclusions
3) Results of test
4) Estimation of test accuracy.

Objective

The objective of this test was to determine the power output of a Buda Industrial Engine, Model K-428, as a function of manifold pressure and speed.

Summary and Conclusions

The ultimate purpose of this test was as stated in the test objective. Of primary interest was the determination of power output. The extent of this test was limited to a range of speeds between 1600 rpm and 500 rpm and values of torque from 80.53 ft./lbs. to 225.84 ft./lbs. In order to obtain sufficient data to complete the test satisfactorily, approximately 70 runs were made. Some of these were thrown out because of mechanical failures of the engine and errors in test procedure.

Several items usually found in a report of this type were omitted in this appendix. Among the omitted items was a complete description of instrumentation employed. The writer would like to emphasize, however, that the facilities at the Mechanical Engineering Laboratories were elaborate, and that all instrumentation used in obtaining those quantities pertinent to the fulfillment of the stated objective was satisfactory under American Society of Mechanical Engineers codes; hence, quire sufficient to yield acceptable results.

Results of this test are presented as three curves of load versus manifold pressure. The term load refers to the actual scale reading read directly from the dynamometer scales. Results were left in this form because power is easily calculated with values of load and speed. The length of dynamometer torque arm is 21.008 inches or 1.7507 feet. With this torque arm, horsepower is

$$\frac{\text{Load (lbs.)} \times \text{Speed (rpm)}}{3000}$$

instead of the usual:

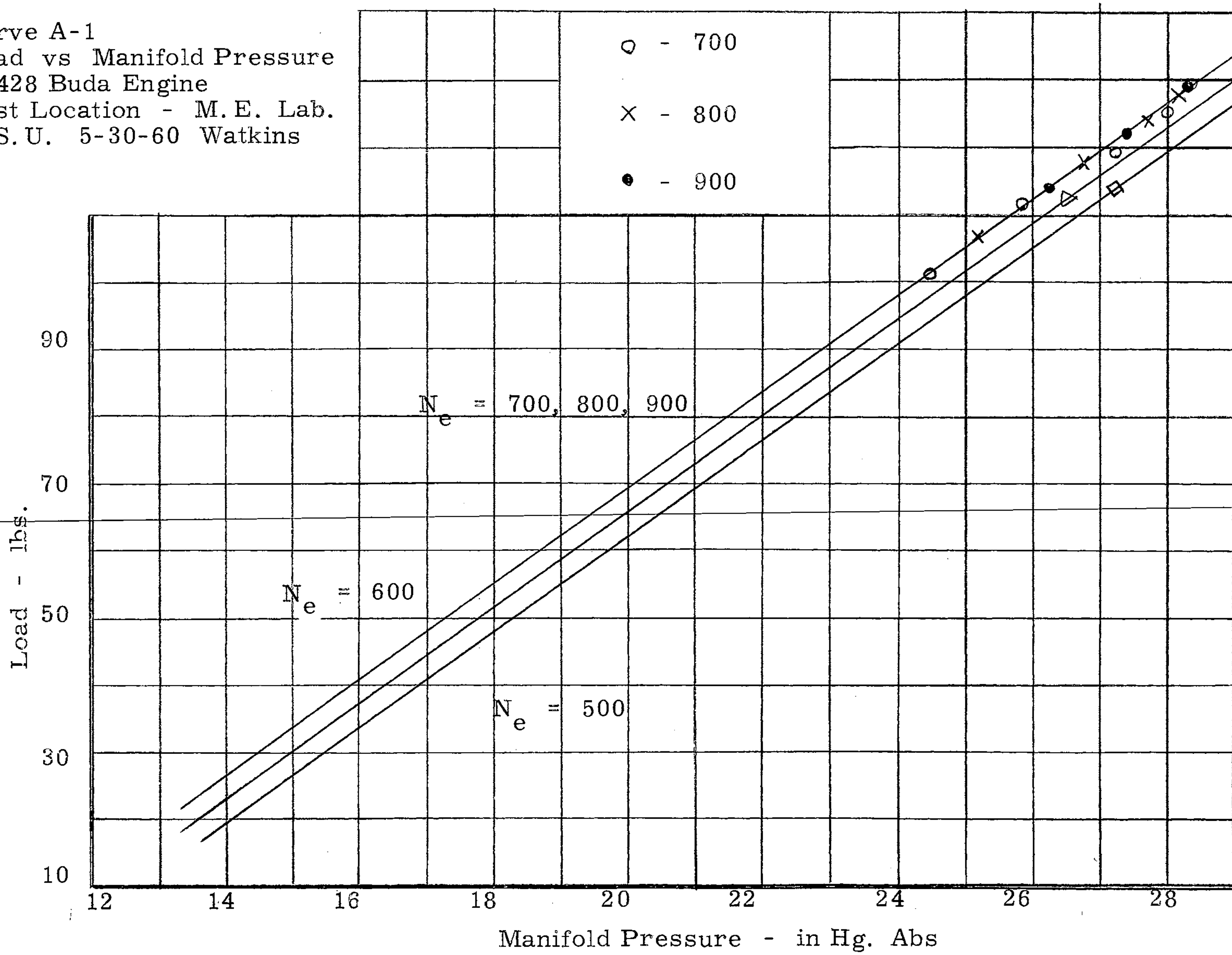
$$\frac{\text{Torque (ft/lbs)} \times \text{Speed (rpm)}}{5252}$$

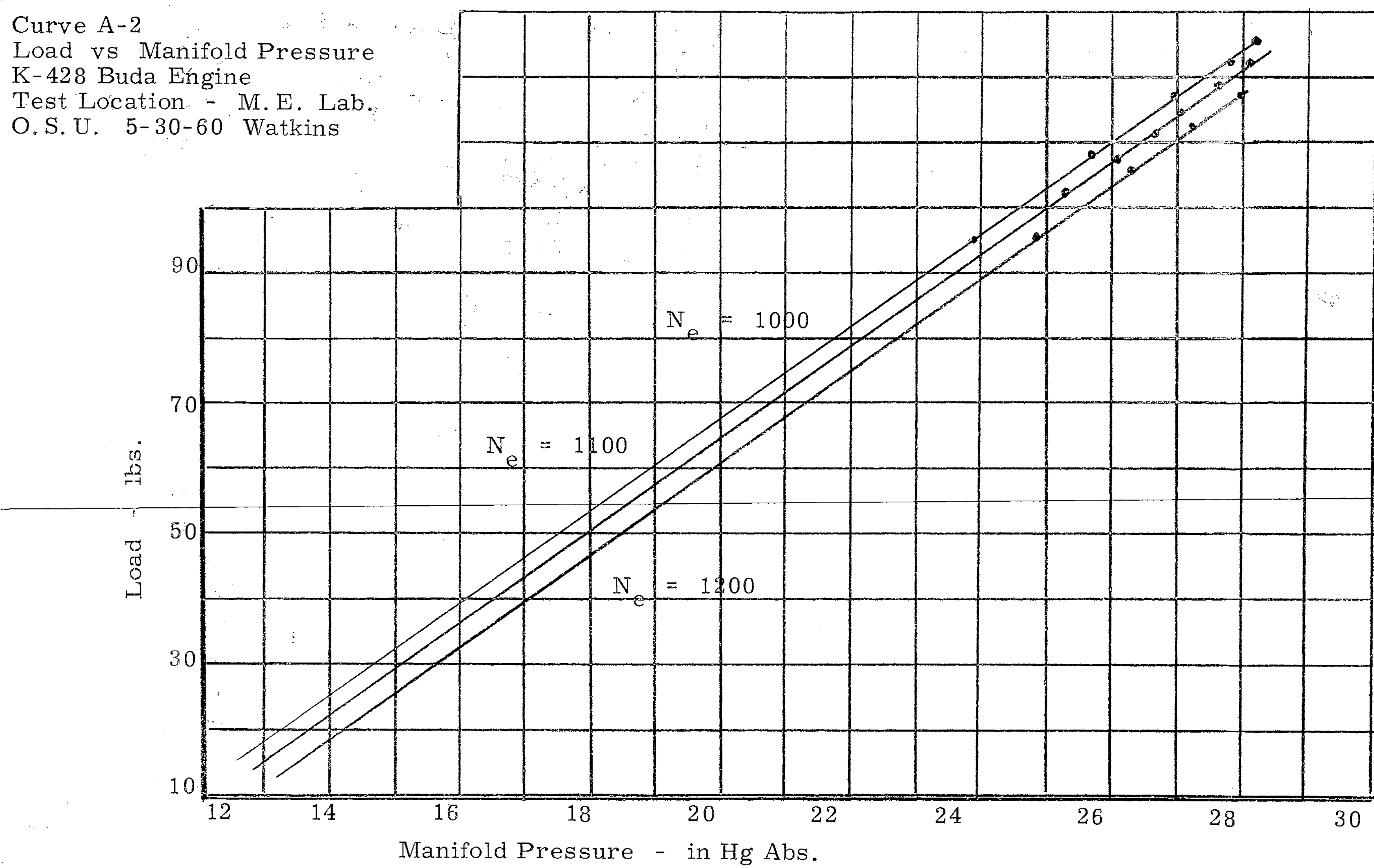
The value of 1.7507 is $\frac{5252}{3000}$.

The results of the test plotted were nearly straight lines indicating good results.

The accuracy and validity of the test were excellent within the range of speeds and loads considered. This is shown in Table A-1 which consists of a comparison of power as obtained from the curves and actual test values. The comparison was made for random throttle settings. It indicates that within the test range, power output of the engine can be predicted within 2.0 per cent.

Curve A-1
Load vs Manifold Pressure
K-428 Buda Engine
Test Location - M. E. Lab.
O. S. U. 5-30-60 Watkins




Curve A-2
Load vs Manifold Pressure
K-428 Buda Engine
Test Location - M. E. Lab.
O. S. U. 5-30-60 Watkins
N_e = 1000
N_e = 1100
N_e = 1200
90
70
50
30
10
Load - lbs.
12
14
16
18
20
22
24
26
28
30
Manifold Pressure - in Hg Abs.

Curve A-3
Load vs Manifold Pressure
K-428 Buda Engine
Test Location - M.E. Lab.
O.S.U. 5-30-60 Watkins

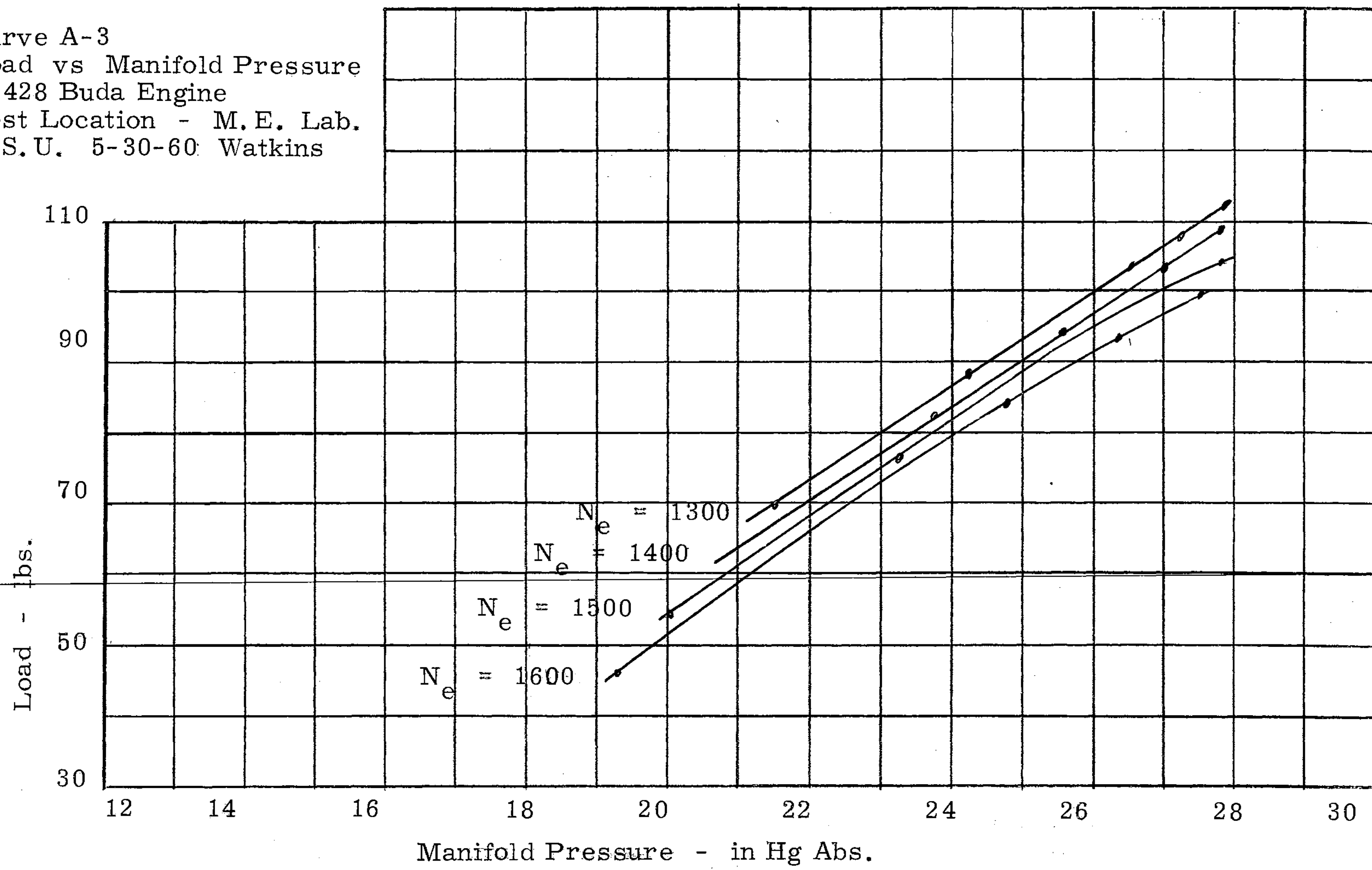


TABLE A-I

VERIFICATION OF ENGINE CALIBRATION DATA

Engine Speed (RPM)	Manifold Pressure (in Hg)	Esti-mated Load (lb)	Actual Load (lb)	Esti-mated HP	Actual HP	Error %
1404	28.0	110.0	108.5	51.5	50.8	+1.38
1598	27.7	101.2	101.5	53.9	54.1	-0.37
1503	27.8	105.2	105.0	52.7	52.6	+0.19
995	28.5	127.0	126.0	42.1	41.8	+0.72
900	25.3	106.5	105.0	32.0	31.5	+1.59
1199	23.3	85.0	83.5	34.0	33.4	+1.80
1197	23.3	85.0	84.0	33.9	33.5	+1.19
1507	21.2	62.2	61.0	31.2	30.6	+1.96

APPENDIX B

THE VOLUMETRIC TANK AND DIVERTER

The flow rate was measured by the diverter method. In the method used all readings were made at a static condition which gave a very high accuracy.

The apparatus used consisted of three main parts:

1) Diverter (Figs. B-1, B-2, B-3)
2) Tank (Fig. B-1)
3) Electric timer

The tank and diverter were fabricated for the purpose of high-accuracy determination of flow rate and ease of use.

The large part of the tank was approximately thirty inches in diameter and six feet high. The small ends were six inch pipe. The upper and lower references were ten feet and one inch apart.

The upper and lower references are on the same level as the small diameter parts of the tank. The references were placed at this level so that an erroneous meniscus reading would make a small volumetric error in proportion to the total volume of the tank. This was the primary reason for the volumetric tank being built as shown in Figure B-1.

The diverter was powered by an air cylinder which was controlled by a solenoid. The cylinder operated under an air

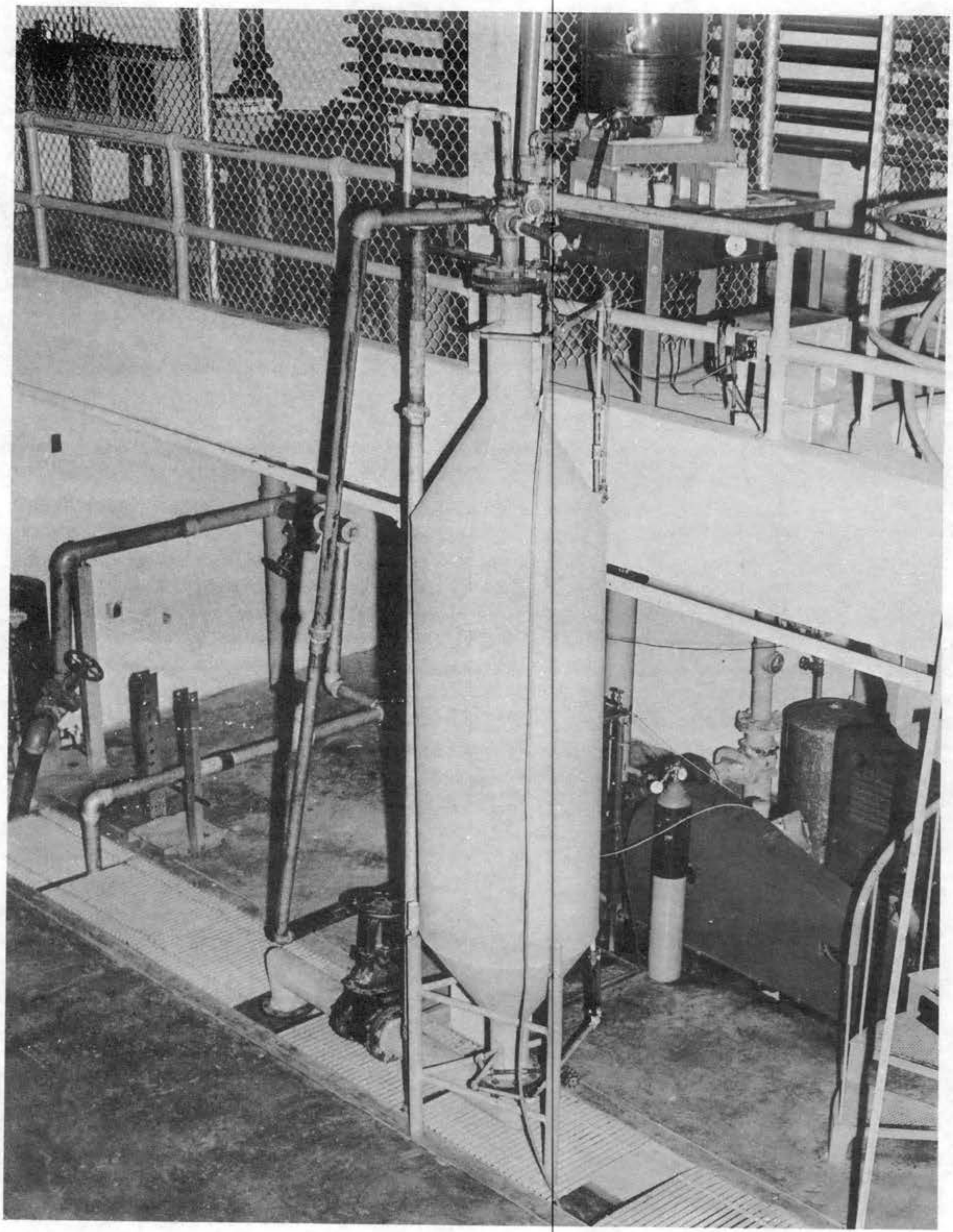
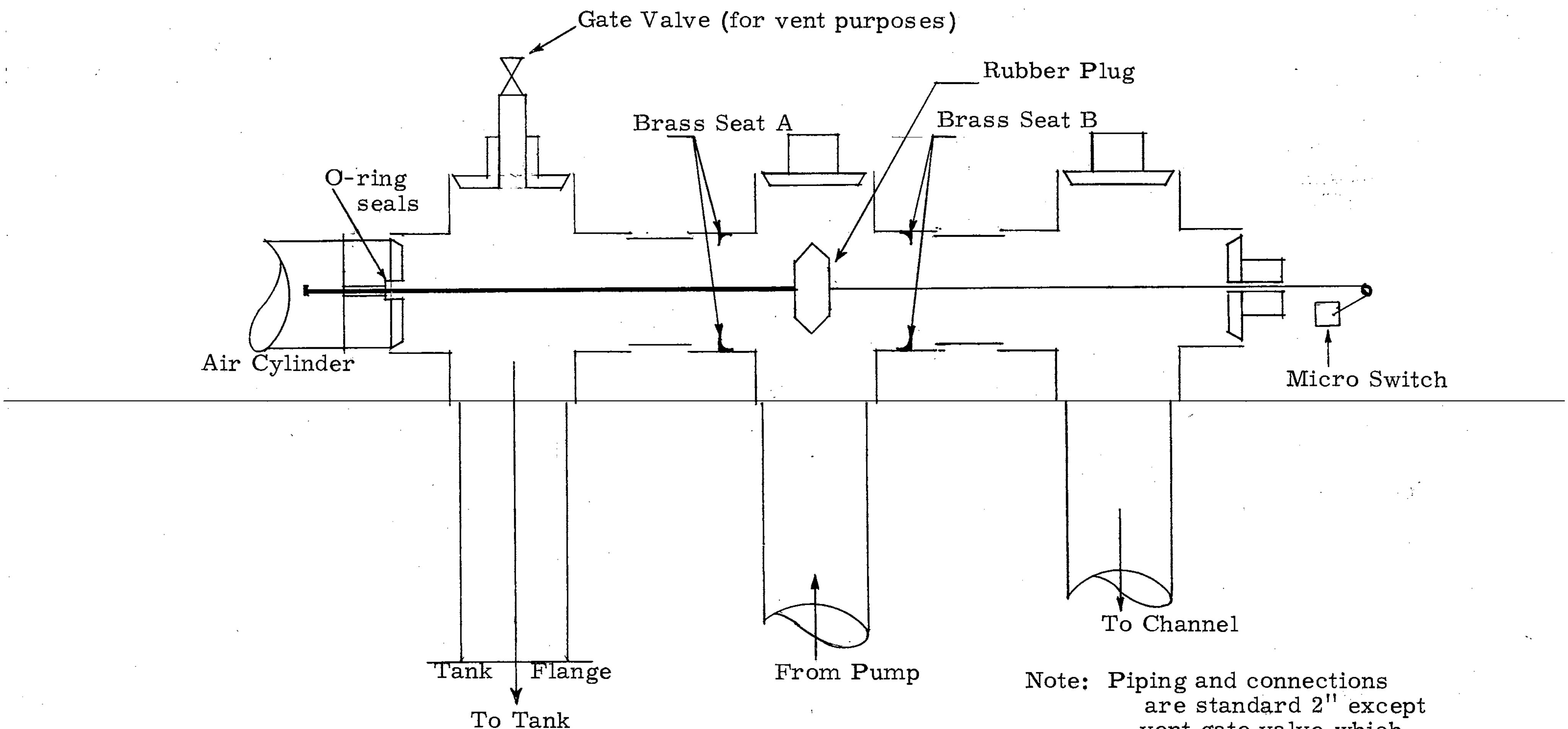


FIGURE B-1

VOLUMETRIC TANK

Fig. B-2 - CROSS-SECTION SKETCH OF DIVERTER



Note: Piping and connections
are standard 2" except
vent gate valve which
is 3/4 " standard

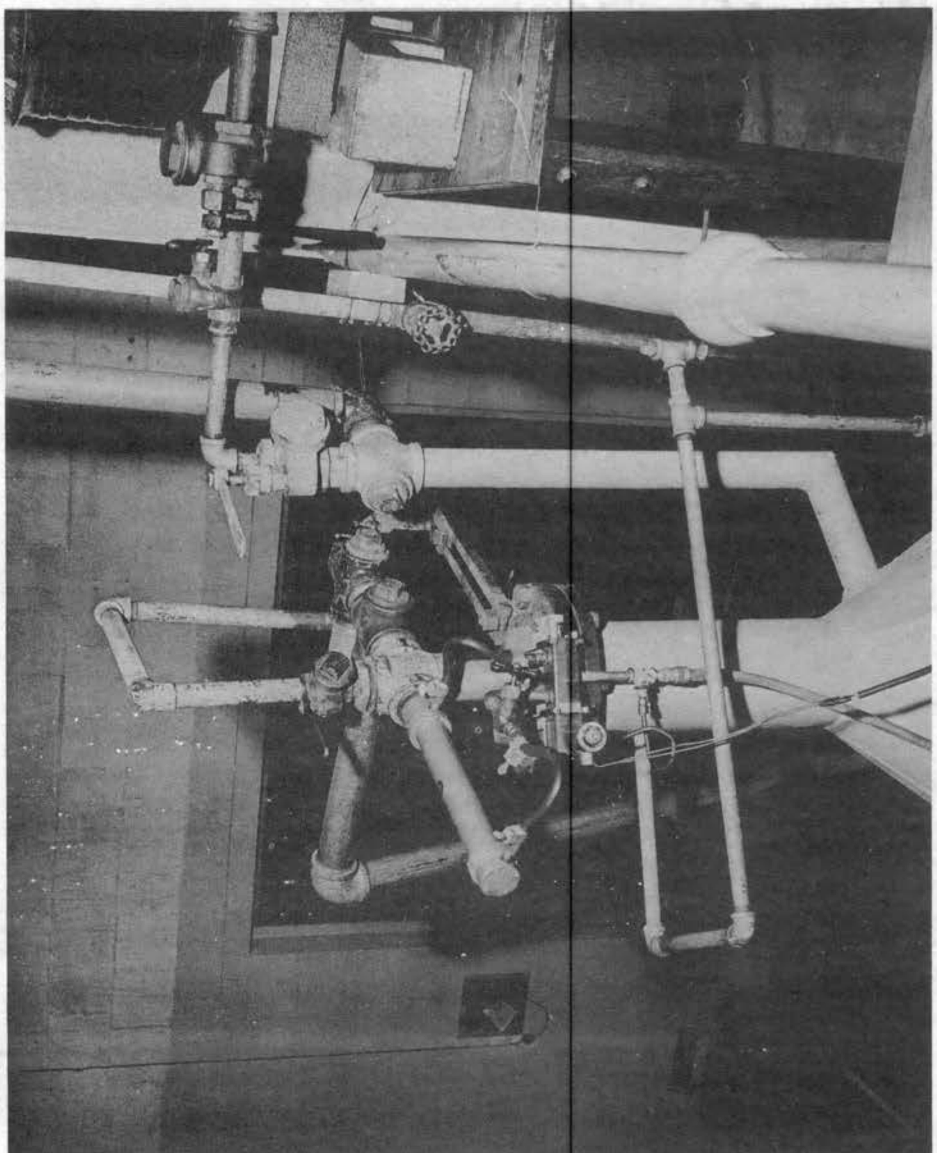


FIGURE B-3

D I V E R T E R

pressure of 125 pounds per square inch gage furnished by the laboratory air compressor.

The timer used measured to 1/100 of a second and was powered by conventional 60 cycle alternating current.

A. Tank Calibration

The upper and lower references were marked on the sight glasses. The volume between the references was found by first running the tank full and then draining it to the lower reference. Then approximately 200 pounds of water was run into the barrel on the scales. The barrel of water was weighed before and after being drained. This process was repeated until the level of the water was at the upper reference. The weight of water added each time was then divided by the specific weight of the water for its temperature prior to being drained. These quotients were summed to get the volume of the tank. The entire procedure was then repeated adding approximately 150 pounds each time and then again adding approximately 100 pounds each time. This was done to help eliminate inherent inaccuracies in the scales. The data recorded in calibrating the tank is included in this appendix.

B. Measurement of Flow Rate

The procedure used in measuring the flow rate is presented in step form.

1) The tank was filled and drained. The level was brought to the lower reference immediately.

2) After steady state conditions were reached, switch "C" was closed which energized the solenoid and caused the cylinder to move the rubber plug from seat "A" to seat "B." Halfway between "A" and "B" the brass rod connected to the piston tripped the microswitch "D" and started the timer.
3) After the level had risen such that it was within twelve inches of the upper reference, switch "A" was opened. This diverted the flow around the tank and stopped the timer when the piston was halfway from "B" to "A."
4) The tank was vented and the time was recorded from the timer.
5) The weight of the barrel and water was recorded.
6) The water level was brought to the upper reference.
7) The barrel and water were reweighed and this weight was subtracted from the weight in step 5. This remainder was divided by the specific weight of water and this quotient was subtracted from the volume of the tank.
8) The flow rate was then calculated by dividing the last remainder in step 7 by the time recorded in step 4.

TABLE B-I

NO	Gross Wt. - lbs	Tare Wt - lbs	Net Wt. - lbs	t_w °F
1	250.00	51.85	198.15	63.5
2	236.75	51.99	185.26	64.4
3	235.80	52.00	183.80	64.4
4	244.12	52.20	191.92	64.4
5	249.00	52.20	196.80	64.4
6	250.00	52.20	197.80	64.4
7	250.00	50.80	199.20	64.4
8	248.20	51.10	197.10	64.4
9	248.50	51.50	197.00	64.4
10	249.00	50.60	198.40	64.4
11	248.15	68.45	179.70	64.4
12	249.31	51.63	197.68	63.5
13	182.11	51.28	130.83	63.5
14	206.10	51.46	154.64	64.4
15	191.13	51.42	139.71	65.3
16	194.03	51.85	142.18	65.3
17	193.74	51.37	142.37	65.3
18	205.16	51.77	153.39	65.3
19	202.34	53.71	148.63	65.3
20	203.56	52.71	150.85	65.3
21	212.17	52.31	159.86	65.3
22	207.21	53.25	153.96	65.3
23	206.50	53.04	153.46	65.3
24	211.00	52.77	158.23	65.3
25	203.72	60.19	143.53	65.3
26	159.97	51.33	108.64	65.3
27	156.02	51.57	104.45	65.3
28	156.18	51.51	104.67	65.3
29	153.50	51.95	101.55	65.3
30	156.00	50.75	105.25	66.2
31	160.96	51.07	109.89	66.2
32	156.34	51.34	105.00	66.2
33	160.08	50.51	109.57	66.2
34	158.17	52.04	106.13	66.2
35	158.61	50.75	107.86	66.2
36	162.01	52.11	109.90	66.2
37	155.26	52.26	103.00	66.2
38	230.81	52.37	178.44	66.2
39	192.45	50.76	141.69	66.2
40	157.81	51.87	105.94	66.2
41	157.60	52.39	105.21	66.2
42	158.14	52.43	105.71	66.2
43	158.00	52.89	105.11	66.2
44	158.11	51.76	106.35	66.2
45	79.51	76.56	2.95	66.2

γ_t @ 63.5 °F = 62.316

γ_t @ 64.4 °F = 62.307

$$V = \frac{198.15}{62.315} + \frac{1926.08}{62.303}$$

$$= 34.107$$

γ_t @ 63.5 = 62.315

γ_t @ 64.4 = 62.307

γ_t @ 65.3 = 62.300

$$V = \frac{328.51}{62.315} + \frac{1646.17}{62.300} + \frac{154.6}{62.307}$$

$$= 34.177$$

γ_t @ 65.3 = 62.300

γ_t @ 66.2 = 62.295

$$V = \frac{419.31}{62.300} + \frac{1708.00}{62.295}$$

$$= 34.148$$

Average V = 34.144

APPENDIX C

SAMPLE CALCULATIONS

This appendix includes a summary of the calculations necessary to obtain the desired results. All calculations are made for run #44.

Quantities of interest in this study were:

<u>Volumetric Efficiency η_v</u>

$$\eta_v = \frac{Q_a}{Q_t}$$

where $$Q_a = \frac{V_t - V_a}{\text{Time}}$$

The tank volume V_t was determined as outlined in Appendix B and found to be: $\underline{V_t = 34.144 \text{ ft.}^3}$

$$V_a = \frac{W_1 - W_2}{\gamma} = \frac{67.33 - 58.90}{62.2} = 0.1360$$

then $$Q_a = \frac{34.144 - .1360}{204.44} = \underline{0.1663 \text{ cfs}}$$

consider now: $Q_t = N_p \times D$

where $$D = \frac{(\text{Area of Cylinder})\ (\text{Stroke})\ (\text{No. of Cylinder})}{1728 \text{ in}^3/\text{ ft}^3}$$

then $$D = \frac{(4.9087 \text{ in}^2)\ (3 \text{ in})\ (3)}{1728 \text{ in}^3/\text{ft}^3} = \underline{2.5566 \times 10^{-2}} \frac{\text{ft}^3}{\text{rev}}$$

and $$N_p = \frac{\text{Rev}}{\text{Time}} = \frac{4300 \text{ rev}}{10.680 \text{ min}} = \underline{402.62 \text{ rpm}}$$

therefore, $Q_t = (N)(D) = \underline{.1715}$

$$\eta_v = \frac{Q_a}{Q_t} = \underline{\underline{96.97}}$$

Brake Horsepower

BHP was calculated according to the following formula:

$$BHP = \frac{Load \times N}{3000}$$

This formula is a variation of the ordinary relationship

$$BHP = \frac{TN}{5252} \qquad T = Load \times \frac{5252}{3000}$$

Load was obtained from engine calibration data. (See Appendix A) All that is needed is manifold pressure in in. Hg. abs. and engine speed, N_e

$$P_m = P_b - P_v = 28.75 - 3.85 = 24.90$$

$$N_e = N_p \times \frac{dp}{de} = N_p \times 2.95 = 402.6 \times 2.95 = 1187.6$$

From curve A-2 $\qquad$ Load $= 97.0$

then $BHP = \dfrac{Load \times N}{3000} = \underline{\underline{38.40}}$

Capacity

$$Q_{\frac{bbls}{hr}} = Q_a\,(cfs) \times 641.14 \; \frac{bbls/hr}{cfs} = (.1663)(641.14)$$

$$= \underline{\underline{106.62}}$$

<u>Efficiency η_m –</u>

$$\eta_m = \frac{WHP}{BHP}$$

where WHP is the usable power available from the water and is

$$WHP = \frac{Q\gamma H}{550}$$

where H, the energy of the fluid, is defined by the equation

$$H = \frac{v_2^2 - v_1^2}{2g} + \frac{p_2 - p_1}{\gamma} + z_2 - z_1 + u_2 - u_1 + \text{Losses}$$

The first three terms of the right side of the above equation are from the very common Bernoulli equation. The first and third terms are small compared to the second. The fifth term of the right side is also small compared to the second term. The fourth term can be written

$$u_2 - u_1 = 778 \ (\Delta t_w)$$

where Δt_w is the change in temperature of the water while in the pump. Although Δt_w is small, it could be significant in some cases. Because of a lack of equipment, Δt_w was not measured in this case. H now reduces to

$$H = \frac{p_2 - p_1}{\gamma}$$

Since $p_2 = p_d$ and is accurate to I 10 psi, p_1, which was never this large in absolute value, can be neglected. Now,

$$WHP = \frac{Q\gamma \frac{p_d}{\gamma}}{550} = \frac{Q\,p_d}{550} \qquad \text{(where } p_d \text{ is in psfg)}$$

$$= \frac{Q\,p_d}{3.819} \qquad \text{(where } p_d \text{ is in psig)}$$

$$WHP = 34.83 \quad \text{for run \#44}$$

then $\eta_m = \dfrac{WHP}{BHP} = \underline{\underline{90.70}}$

VITA

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Master of Science

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