INVESTIGATIONS IN THE DESIGN OF THE SUCTION SIDE OF A HORIZONTAL TRIPLEX PUMP

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TRIPLEX PUMP

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PREFACE

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

I wish to express my indebtedness to Professor Harry M. Wyatt, Jr. for his assistance and constructive criticism while acting as the Gaso Project leader and as my thesis adviser.

I would also like to thank the staff of the School of Civil Engineering for the valuable instruction they have given me. In particular, I would like to thank Professor Jan J. Tuma, Head of the School of Civil Engineering, and the others responsible for my being awarded a Graduate Assistantship and the Gaso Pump and Burner Manufacturing Company Fellowship. This financial assistance made this rewarding year of graduate study possible. I also wish to thank Professor Roger L. Flanders for his invaluable and cheerful advice while acting as my major adviser.

I express my most sincere gratitude to my parents, Floy and Orval Childs, and to two classmates, Pete Byers and Bob Hawk. Without their faith and encouragement I probably would never have returned to school for graduate study.

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S.B.C.

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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 Professor Wyatt consisted of approximately 150 tests on different installations in the States of Oklahoma, Texas, Louisiana, Kansas, Kentucky and Illinois. It was discovered by Professor 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, "rule of thumb" answers.

A survey of literature revealed that much more work has been done in controlling surges than eleminating the cause of the surges. Waller (1) and Wyatt (2) made notable contributions in this field. Lester (3) and Hicks (4) use the concept of net positive suction head to design the suction piping of the relatively steady flow centrifugal pumps. None of the references studied gave a good method of design of suction piping for a positive displacement pump. The Standards of Hydraulics Institute gives minimum suction lift for trade-pumps, but this can not be applied to a particular triplex pump.

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

The primary objective of this study is to determine the effect of different suction pressures on pump performance and operation for the above-mentioned pump with the suction manifold streamlined. The streamlined suction manifold is discussed in detail in Chapter II.

Tests were made at three different speeds, i.e., 250, 350 and 400 revolutions per minute. All tests were made at the maximum recommended discharge pressure of 800 pounds per square inch gage.

The secondary objective was the determination of flow rate, power required, and over-all efficiency as a function of pump speed.

The results are shown by plotting volumetric efficiency versus suction pressure for each speed. Flow rate, power, and over-all efficiency curves are also presented. All data and curves are then compared with similar data and curves for the standard pump. The data for the standard pump was taken by Watkins. (5). A detailed list of test equipment, a discussion of test procedure, and recommendations for futher study are included.

CHAPTER II

DISCUSSION OF SUCTION CONDITIONS

Pump users and designers have been aware of the need for focusing special attention on suction piping for a number of years. Manufacturers of centrifugal machines have adopted the concept of minimum net positive suction head and have employed this concept quite extensively. Reciprocating machines have not been without attention. William M. Barr, 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." Barr also stated that failure to have 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." Although many people have been aware for many years that much needs to be done in defining specific minimum conditions for safe and economical operation, very little has been done. The Standards of Hydraulics Institute provides a chart which enables one to obtain relative values of maximum permissable 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 and without a loss of efficiency?"

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

The object of this study is to answer some of these questions for a pump with a streamlined suction manifold and to determine if it is feasible to commercially manufacture a pump with a similar manifold. When the lowest limit of the permissible suction pressure is reached a phenomenon called cavitation occurs.

A. The Cause and Effects of Cavitation

The word cavitation implies a cavity or a void. If at a point in a fluid flow the existing pressure equals the vapor pressure of the fluid at the existing temperature, the fluid will vaporize forming a cavity or void. This is called cavitation.

In the pump tested, the highest velocity of the fluid is probably at the intake or suction valves. Since this is the area of the highest velocity it is also the area of the lowest pressure. If the suction pressure (the pressure forcing the fluid into the pump) is not high enough, the fluid will vaporize and part of this vapor will probably get into the cylinder. When the piston begins the discharge stroke, it will move relatively unopposed until the pressure in the cylinder is higher than the vapor pressure of the fluid. When this pressure is reached the piston will slap into the liquid and throw the discharge valve open. The result is noisy operation of the pump and a decrease in the volumetric efficiency of the pump. A similar effect upon the pump can be had by introducing air into the suction piping. This air can come from leaks in the suction piping. This statement justifies Mr. Barr's statement that suction piping must be "absolutely tight."

B. <u>Streamlining the Suction Manifold</u>

Many different methods for streamlining the suction manifold were considered. Some of these methods for filling in the cavity caused by the side suction ports were:

1) Use of plaster of paris
2) Use of leadite
3) Use of low melting temperature metal such as lead
4) Making a metal plug out of some soft metal
5) Making a wood plug.

After some investigation the first three methods were discarded. It was decided that the installation and removal of forms to make the material take the desired shape would be too difficult.

Because of its availability and ease of working, wood was selected over the soft metals. The plugs were made of four inch redwood stock. The final size and shape of the plugs are shown in Figure 2-1.

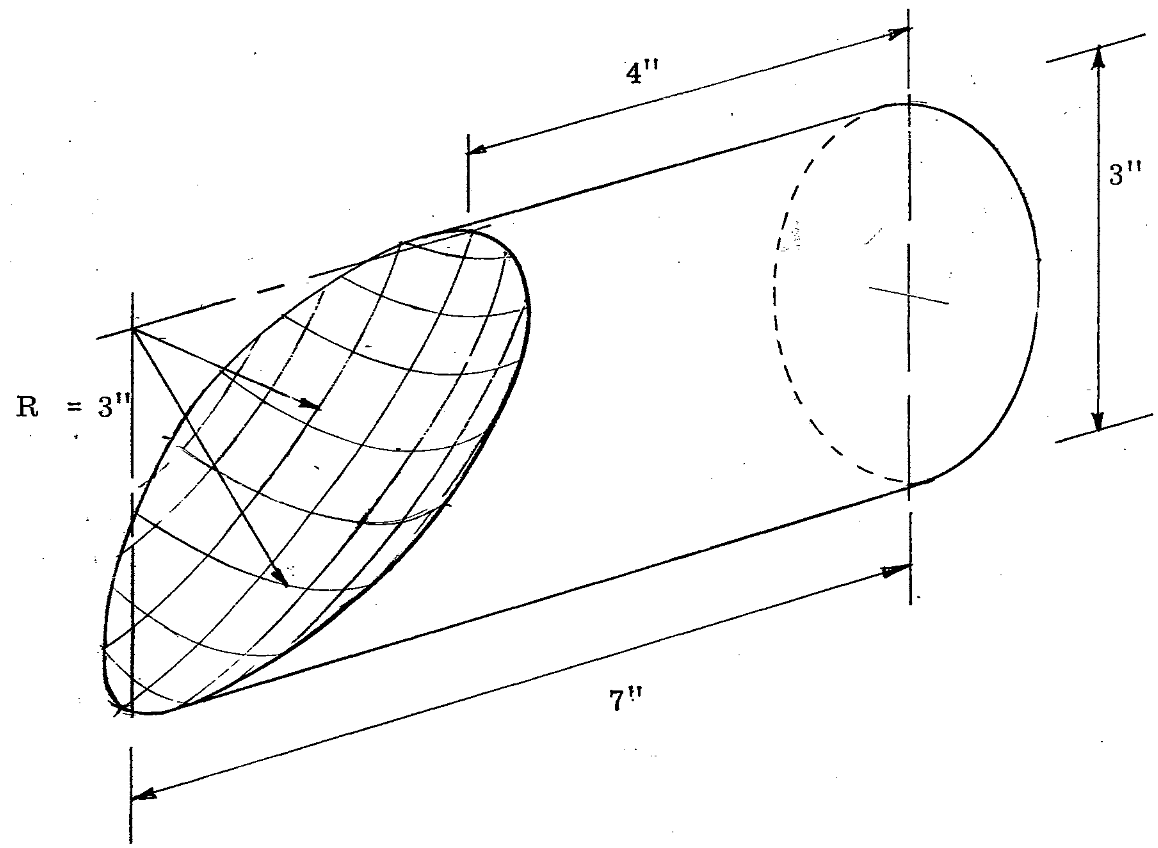


Fig. 2-1
REDWOOD PLUG

The plugs were installed and allowed to stand under a water pressure of approximately fifteen pounds per square inch for a period of three days before the positive suction pressure tests were run and four days before the negative suction pressure tests were run. This was done to make sure that all the air in the redwood plugs had been replaced by water. If the air had not been evacuated, it could have given cavitation effects at almost any negative suction pressure.

It was the writer's opinion that the streamlining of the suction manifold might reduce the turbulence in the area of the suction valves enough that it might make an appreciable difference in the allowable lower limits of suction pressure.

CHAPTER III

TEST EQUIPMENT

The pump tested was a single-acting Horizontal, Triplex Plunger Pump (Gaso Fig. 3365). The pump stroke was three inches and the bore was two and one-half inches. The pump was capable of handling approximately 93 barrels per hour at 350 rpm at a maximum operating pressure of 805 pounds per square inch gage.

For a more complete description of this pump see Page 13 of the Gaso catalogue.

The pump was mounted on portable skids and powered by a Buda Industrial Engine. The fuel for the engine was natural gas.

Equipment Used

I. Power

A. Oil Field Engine

Mfgr.	Buda Company, Harvey, Illinois
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 calibrated in order to predict the power input to the pump. (See Appendix A for details.)

II. FLOW CONTROL AND STABILIZATION

A. High Pressure Gate Valve

Mfgr:	Vogt Company, Louisville, Ky.
Size:	2"
Rating:	800 Wp at 750 $^{\circ}$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:	$2^{o}F$

B. Two Liquid in Glass Thermometers

Mfgr:	Fisher Instrument Co.
Range:	
Least Subdivision:	$10^{o}C$

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 (Item 1 and 2 under III - Flow Measurement)

An analytical investigation was made of the various methods for determining power input to the pump, measuring flow rate, determining pump speed, measuring a varying pressure and measuring increments of time. The instruments and methods selected and employed are outlined in a following paragraph. The power input to the pump was determined by testing the Buda engine and plotting manifold pressure versus load. A summary of the engine test is included in Appendix A. After the engine was tested, instrumentation selected, and necessary equipment fabricated, the test stand was constructed. The test group contained only two persons; therefore, ease of operation was of prime concern in building the test stand.

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 streamlined suction would be small; therefore, it was necessary to keep instrumentation error at a minium.

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;

exhaust pressure, fuel pressure, etc., could be matched 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 the suction pressure was varied from 5 pounds per square inch gage to that point at which a noticeable decrease in volumetric efficiency or excessive knocking occurred. This constituted a total of 49 runs.

All tests were run at a discharge pressure of approximately 800 pounds per square inch gage.

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.

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	^{o}F	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	^{o}F	Liquid-in-glass thermometer

CHAPTER IV

PROBABLE ERRORS IN TEST

After running a test one would like to be able to state the accuracy of the test. The results of this test do not lend themselves to a statistical analysis. Since never more than three points were taken at any one place, a statistical analysis would be meaningless.

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 Erroneous η is $\eta_{max} = \left(\frac{Q_a}{Q_t}\right)_{max}$

Maximum per cent error in η is $\frac{\eta_{max} - \eta}{\eta}$

where: $\eta_{max} = \frac{(Q_a) \text{ maximized}}{(Q_t) \text{ minimum}}$

$$Q_a = \frac{V_t - V_a}{T_a} \qquad (Q_a)_{max} = \frac{(V_t)_{max} - (V_a)_{min}}{(T_a)_{min.}}$$

$$Q_t = \frac{Rev}{T_t} \times D \qquad (Q_t)_{min} = \frac{Rev \times D}{(T_t)_{min}}$$

$$(V_t)_{max} = V_t + \Delta V_t = 34.144 + .036 = 34.180$$

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:

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

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.

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

$$T_a = \frac{1}{60} \text{ sec.} + 2 \times (1/2)(.01)$$

$$= .0167 + .0100 = .0267 \text{ sec.}$$

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

$$T_t = 1/60 \ (\frac{1}{60}) \text{ min} + 2 \times 1/2 \times .01 \text{ min.}$$

$$= .00028 + .01 = .01028 \text{ min.}$$

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:

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

Therefore, the maximum error in η is:

$$\Delta(\eta) = \frac{\eta_{max} - \eta}{\eta} = \frac{97.182 - 96.97}{96.97} \times 100$$

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

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, usually, 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 pounds per square inch gage. It is the opinion of the writer that pressure was determined correct to a plus or minus 10 pounds per square inch gage or $2\frac{1}{2}$ per cent. 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 per cent. The only other factor that affected the accuracy of power determination was the inability to read discharge pressure correctly.

An incorrect reading of 20 pounds per square inch gage in discharge pressure would result in a $2\frac{1}{2}$ per cent 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 $fr = \frac{f_1}{f}$. Electric timers, powered by the same current, were used to determine both the actual and theoretical flow rates. Volumetric efficiency is $= \frac{Q_a}{Q_t}$.

Each flow rate is multiplied by the same correction factor for frequency variation; $\frac{Q_a\ (fr)}{Q_t\ (fr)}$. 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 per cent.

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

Table V-I presents the observed data in condensed form. All runs that are not considered good are not included. Runs 29 through 152 constitute the runs on the standard suction manifold. Runs 153 through 200 were made on the streamlined suction manifold.

Tables V-II, V-III, V-IV and V-V present the tabular results of all good runs.

Curves 1, 2, 3 and 4 are the graphical results of Tables V-II, V-III, V-IV and V-V.

Curves 5, 6, 7 and 8 are the plots of capacity, horsepower required and per cent mechanical efficiency of the pump.

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	3.07	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	206.01	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	245.54	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

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)
154	28.93	730	4.3	8.6	98	13.8	800+	1500	6.071	58.40	59.40	332.97	86
155	28.94	730	4.3	8.6	97	10.0	800+	1400	5.678	59.40	55.75	332.98	86
156	28.94	730	4.3	8.6	97	5.10	800+	1500	6.098	55.75	52.40	333.99	86
157	28.95	730	4.3	8.6	97	3.90	800+	1500	6.095	57.80	56.50	334.08	87
158	28.95	725	4.25	8.6	97	2.90	800+	1400	5.699	56.50	57.85	335.00	87
159	28.95	725	4.25	8.6	97	1.98	800+	1400	5.702	57.85	56.95	334.40	87
160	28.95	725	4.25	8.6	97	1.10	800+	1400	5.708	56.95	49.75	334.34	87
162	28.94	1045	3.35	8.0	101	10.10	800+	1600	4.533	61.35	59.00	233.35	85
163	28.94	1040	3.30	8.0	101	5.10	800+	1500	4.259	59.00	58.40	234.05	85
164	28.93	1040	3.30	8.0	102	4.00	800+	1400	3.982	58.40	61.00	234.79	85
165	28.93	1035	3.30	8.0	102	3.00	800+	1500	4.274	61.00	61.00	235.02	85
166	28.93	1035	3.30	8.1	102	2.10	800+	1500	4.272	61.00	62.00	234.98	85
167	28.93	1035	3.30	8.1	102	0.90	800+	1500	4.272	62.00	58.10	234.49	86
174	28.85	1150	2.50	7.6	89	-0.95	850	1500	3.883	58.60	58.60	213.96	85
176	28.85	1150	2.50	7.6	90	-3.15	860	1500	3.887	59.70	56.20	214.32	85
178	28.85	1150	2.50	7.6	91	-5.20	860	1500	3.876	56.20	56.70	215.20	85
180	28.85	1200	2.65	7.5	92	-6.20	850	1600	4.012	56.70	50.90	214.25	85
181	28.85	1265	2.95	7.4	92	-6.55	840	1600	3.735	62.30	62.30	214.53	85
182	28.85	1040	2.40	8.0	94	-1.10	860	1500	4.267	62.30	61.90	234.82	85
183	28.86	1045	2.40	8.0	94	-4.20	880	1500	4.256	61.90	62.70	234.82	85
184	28.86	1055	2.50	7.9	94	-6.05	880	1400	3.916	62.70	60.10	233.07	86
186	28.86	1075	2.60	7.9	94	-7.00	880	1500	4.117	60.10	51.50	231.55	86
187	28.86	1095	2.63	7.9	95	-7.40	880	1500	4.082	60.45	58.85	231.97	86
188	28.88	750	4.50	8.7	103	-0.70	800	1600	6.287	58.75	61.50	323.51	86
189	28.88	750	4.45	8.7	104	-4.28	800	1400	5.514	61.50	65.25	324.76	86
190	28.88	790	4.90	8.7	104	-7.30	820	1500	5.558	65.25	63.80	322.84	86
192	28.88	865	5.55	8.6	105	-8.38	880	1600	5.379	62.80	61.40	310.04	86
193	28.87	945	6.05	8.5	106	-8.50	880	1700	5.313	61.40	61.00	309.74	86
194	28.88	1190	3.25	7.7	102	13.50	800	1500	3.727	61.00	59.85	205.24	86
195	28.88	1190	3.25	7.7	104	10.00	800	1400	3.474	59.85	57.00	204.68	87
196	28.88	1185	3.20	7.7	105	5.00	800	1400	3.471	57.00	55.00	204.75	87
197	28.88	1190	3.20	7.7	105	4.00	820	1400	3.477	55.00	52.95	205.05	87
198	28.88	1180	3.20	7.7	106	3.00	820	1400	3.485	58.50	54.50	205.48	87
199	28.88	1175	3.20	7.7	106	1.90	820	1500	3.742	54.50	54.00	206.01	87
200	28.88	1175	3.20	7.7	106	1.10	820	1400	3.491	54.00	52.00	205.87	87

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			N_{ip} $\overset{\circ}{=}$ 350 rpm			N_{ip} $\overset{\circ}{=}$ 400 rpm		
RUN NO	P_s	η	RUN NO	P_s	η	RUN NO	P_s	η
46	5.03	97.59	55	5.03	97.32	61	4.95	97.18
47	4.00	97.49	56	4.00	97.38	62	3.97	97.12
50	2.93	97.21	57	3.00	97.17	63	2.92	97.12
51	2.00	97.36	58	2.00	97.45	64	1.92	97.06
52	0.93	98.01	60	0.90	97.99	65	1.10	97.18
124	-0.80	97.48	129	-0.98	97.29	137	-0.95	96.94
125	-3.05	97.39	133	-4.83	97.10	138	-3.05	96.73
126	-6.15	96.56	134	-5.95	96.74	139	-4.05	96.78
127	-8.40	89.88	135	-7.00	94.30	140	-5.05	96.39
			136	-7.80	85.75	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

TABLE V - V TABULAR RESULTS

Streamlined Center Suction

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

RUN NO	P_s	η
154	13.80	97.44
155	10.00	97.43
156	5.10	97.42
157	3.90	97.33
158	2.90	97.42
159	2.00	97.61
160	1.10	97.42
188	-0.70	97.51
189	-4.20	97.32
190	-7.30	91.91
192	-8.35	86.90
193	-8.50	80.85

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

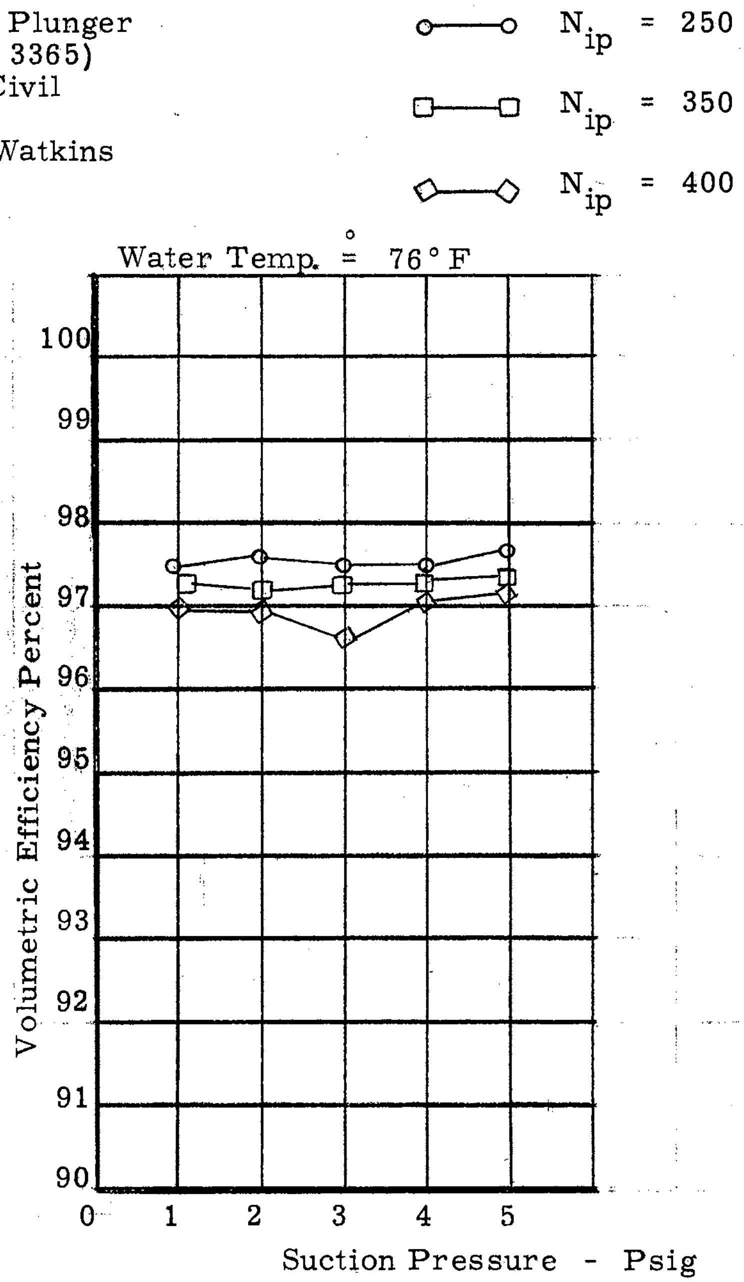
RUN NO	P_s	η
162	10.10	97.21
163	5.10	97.14
164	4.00	97.20
165	3.00	97.19
166	2.10	97.19
167	0.90	97.13
182	-1.10	97.06
183	-4.20	96.87
184	-6.05	96.13
186	-7.00	94.65
187	-7.40	93.93

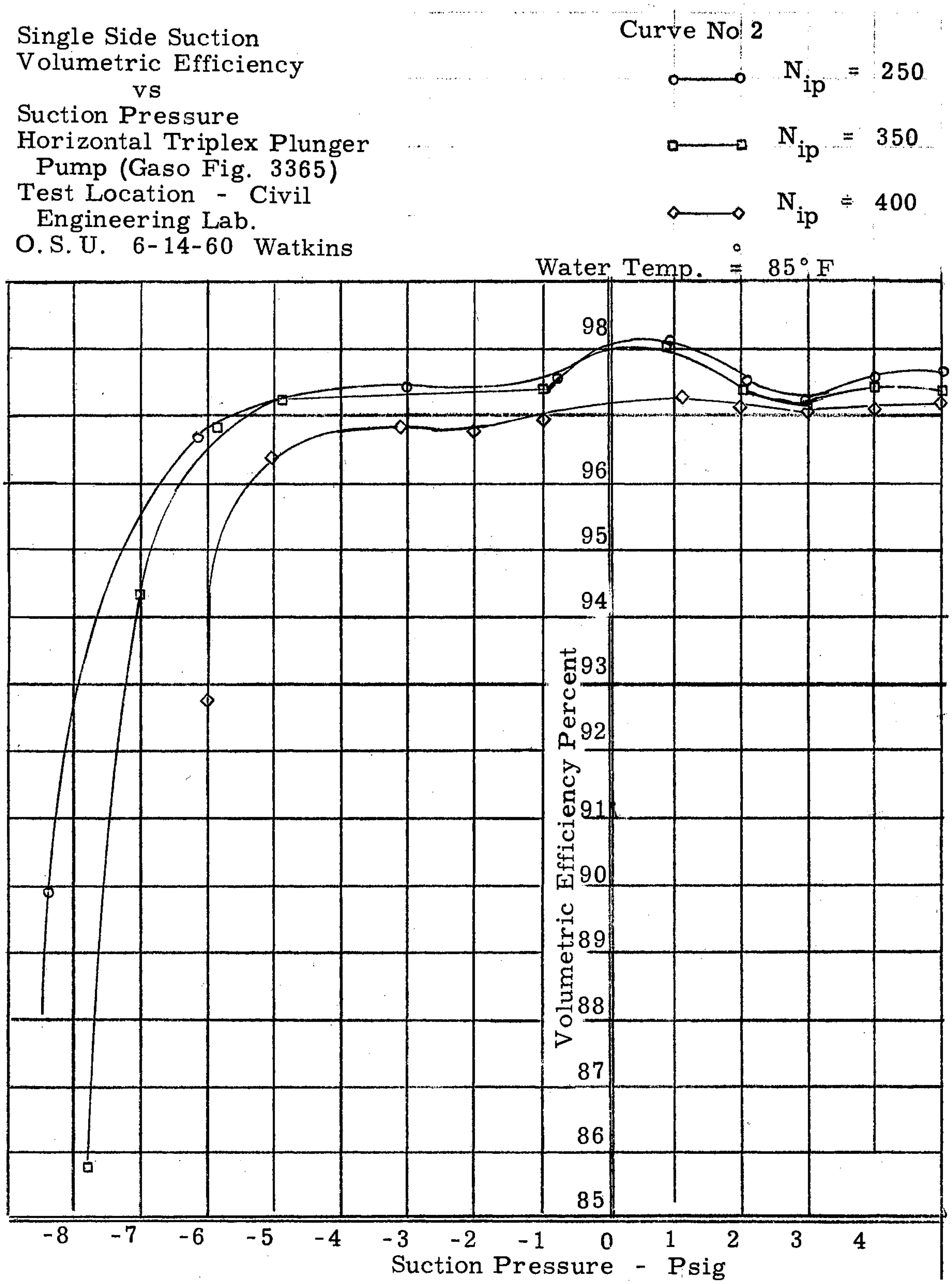
$N_{ip} \overset{\circ}{=} 400$

RUN NO	P_s	η
194	13.50	97.03
195	10.00	97.03
196	5.00	96.92
197	4.00	96.97
198	3.00	96.90
199	1.90	97.01
200	1.10	96.96
174	-0.95	96.96
176	-3.15	96.78
178	-5.20	96.24
180	-6.20	93.53
181	-6.50	87.23

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

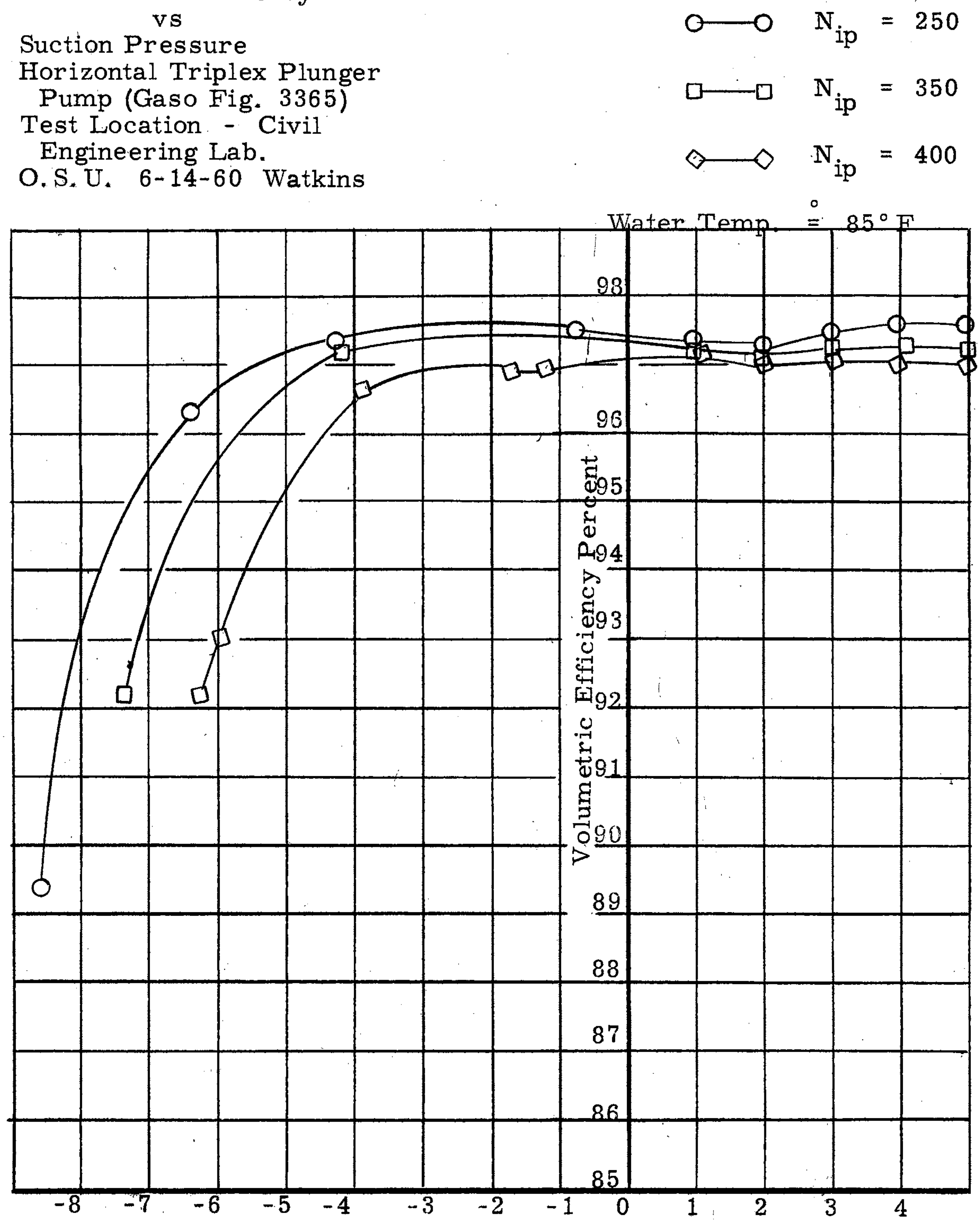
Curve No. 1




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

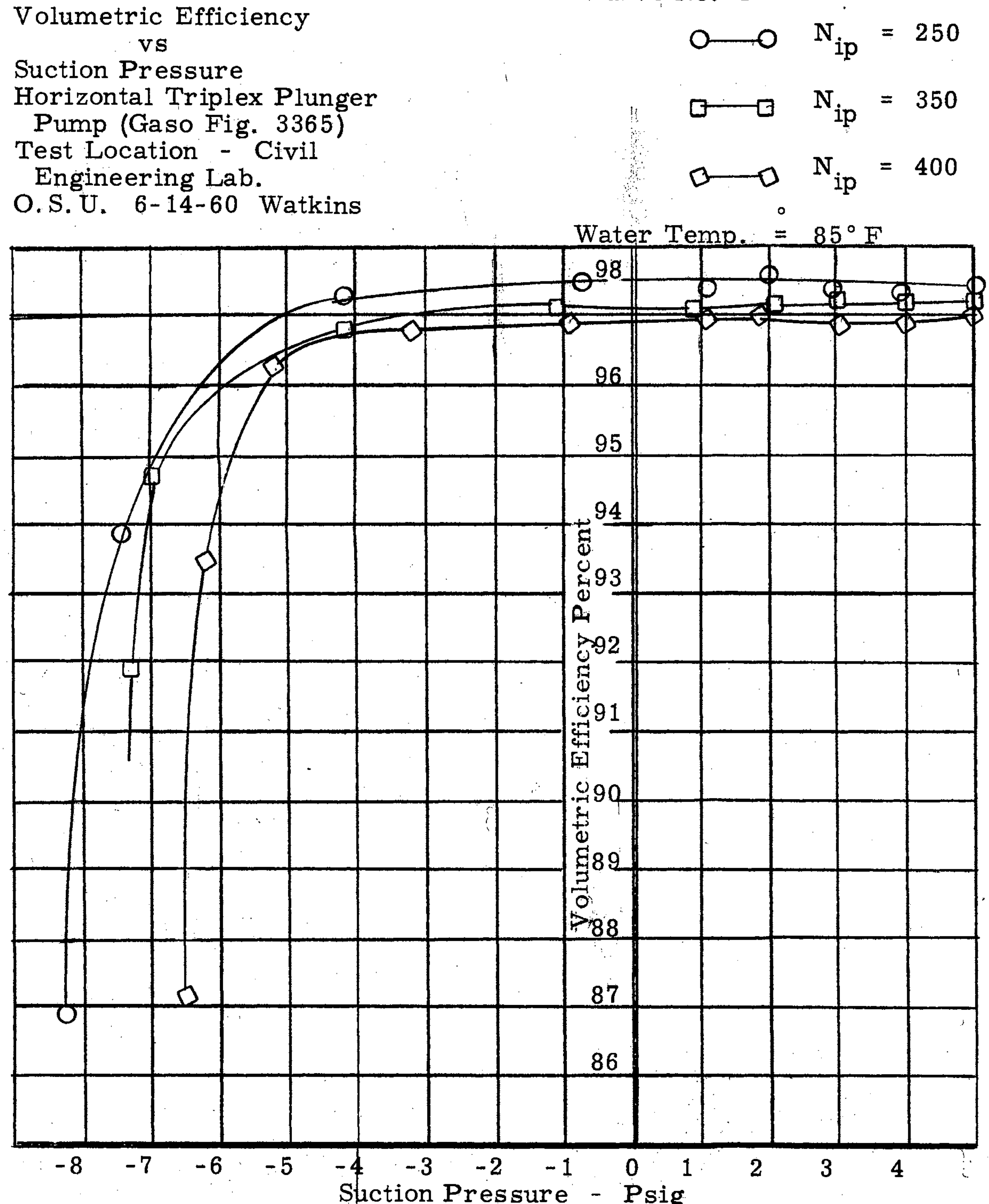
Single Center Suction
Volumetric Efficiency
vs
Suction Pressure
Horizontal Triplex Plunger
Pump (Gaso Fig. 3365)
Test Location - Civil
Engineering Lab.
O. S. U. 6-14-60 Watkins

Curve No. 3



Streamlined 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. 4



Curve No. 5

Power, Capacity and Efficiency
Pump vs Speed

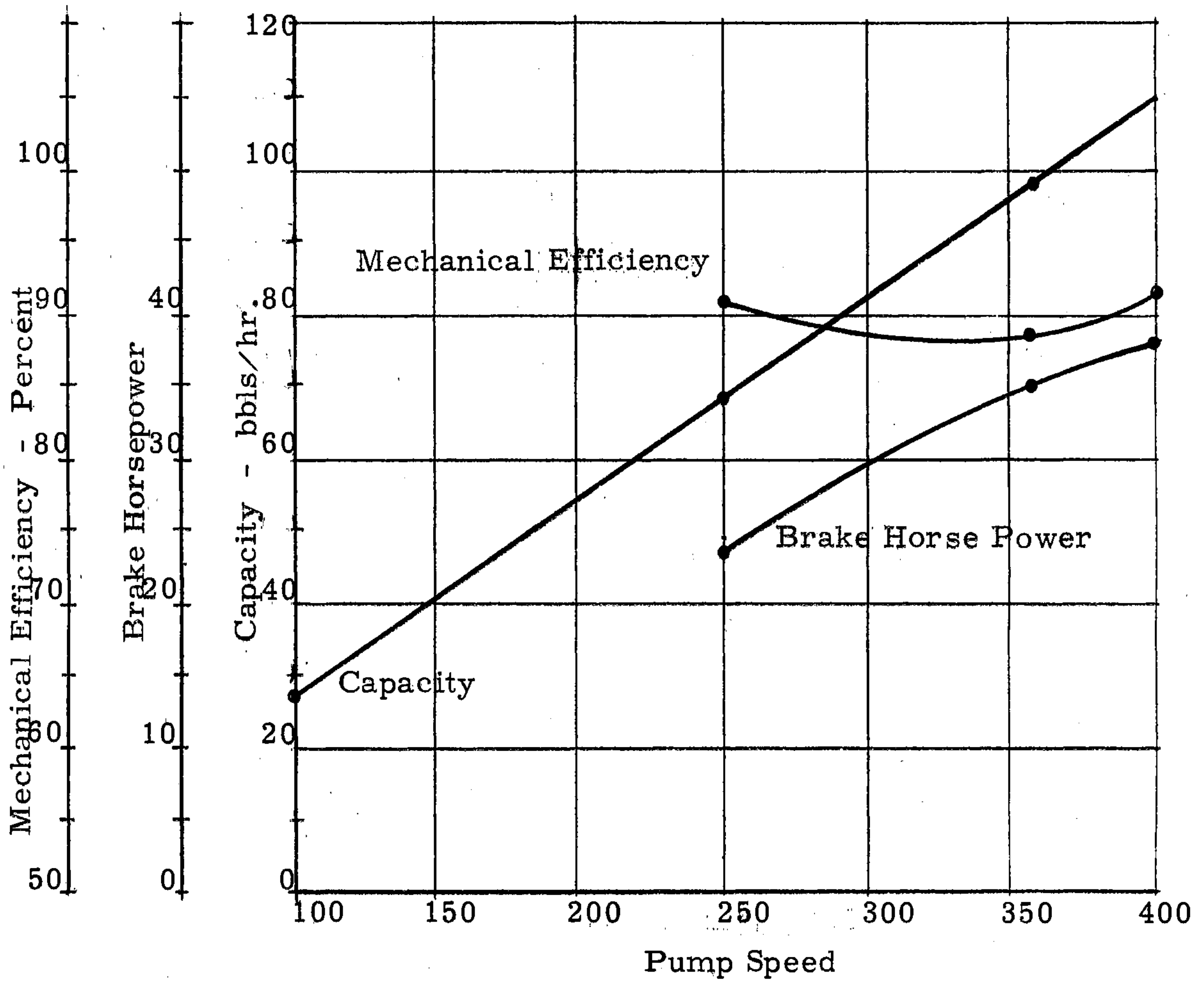
Horizontal Triplex Plunger Pump
(Gaso Fig. 3365)

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. 6

Power, Capacity and Efficiency
vs
Pump Speed

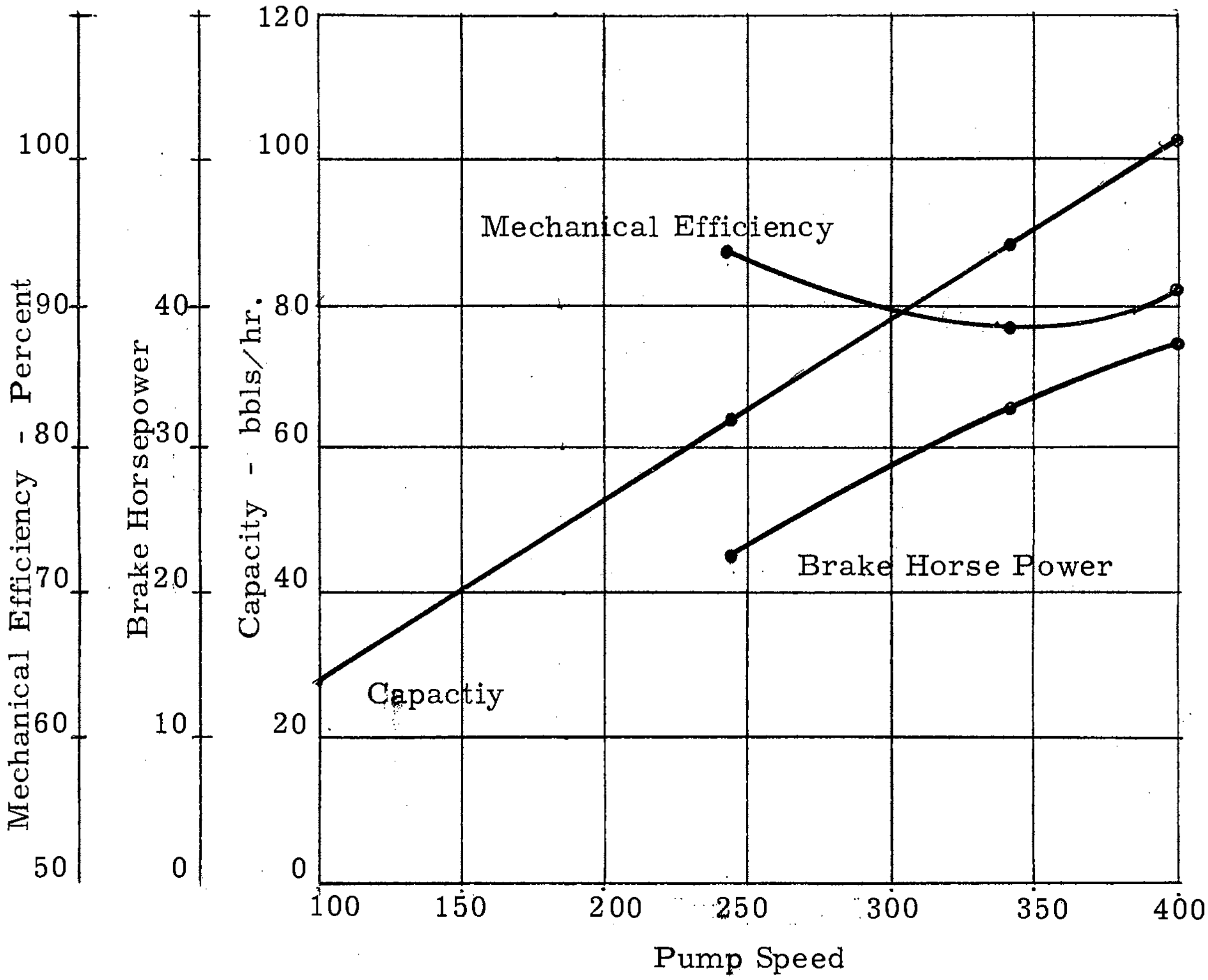
Horizontal Triplex Plumger Pump
(Gaso Fig. 3365)

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



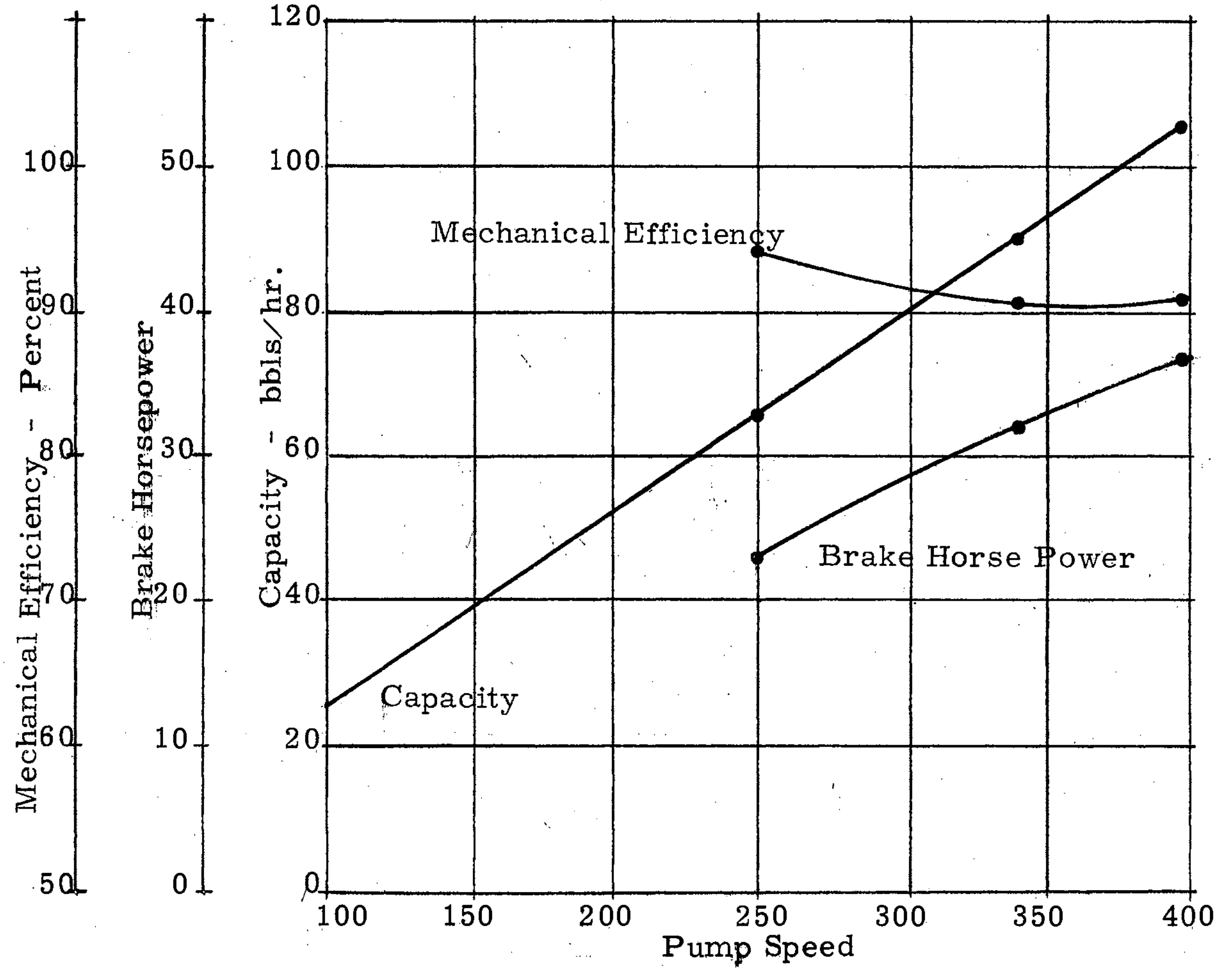
Curve No. 7

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



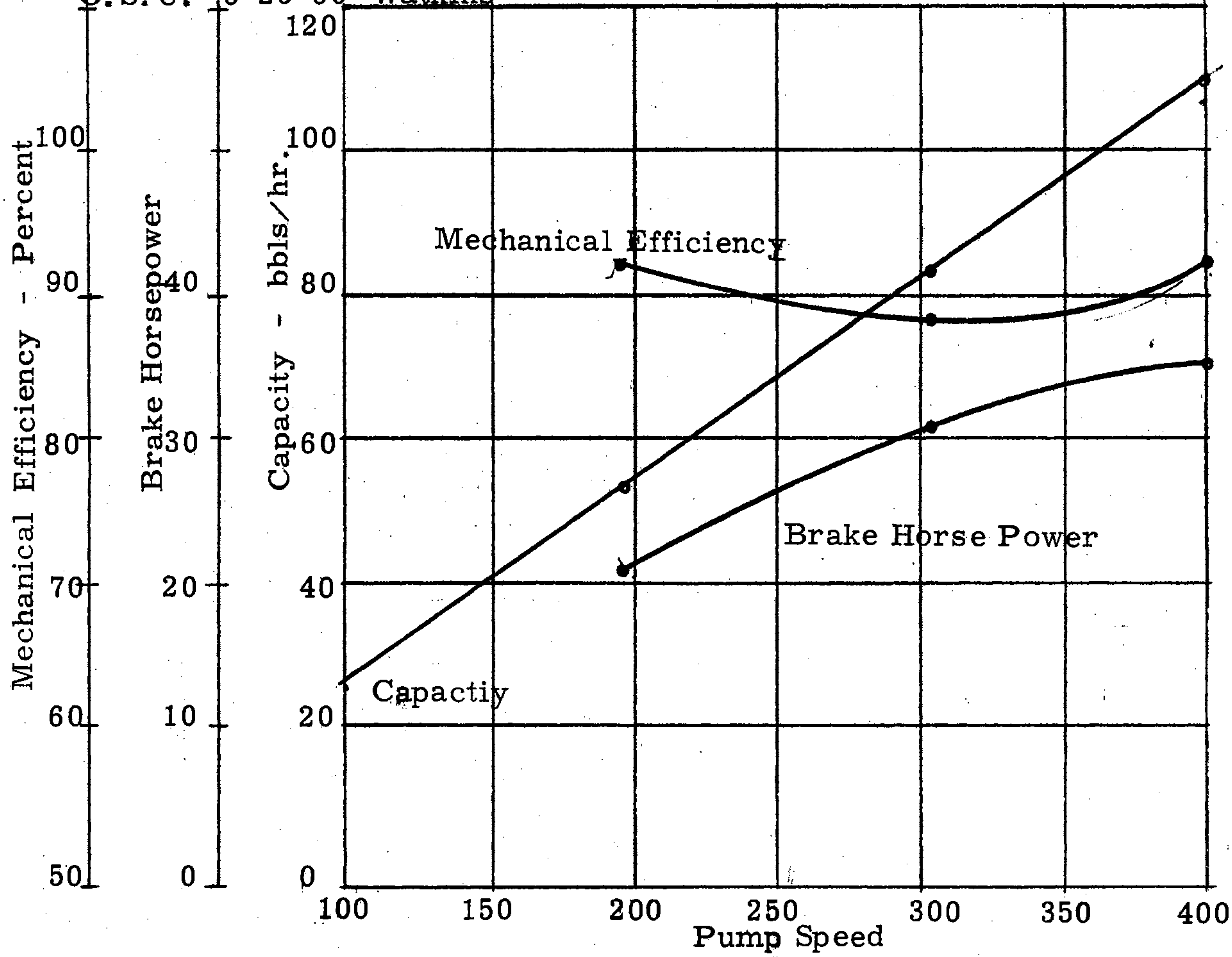
Curve No. 8

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: Streamlined Center

Discharge Pressure = 800 psig + 10 psig

Suction Pressure = 5.0 + 0.1



CHAPTER VI

CONCLUSIONS

The results of the tests are presented in graphical and tabular form in Chapter V.

If a similar unit is installed in the field, it should have a suction pressure of no less than a negative four pounds per square inch gage. If the pressure of vaporization of the fluid being pumped is higher than the pressure of vaporization of water at 90 degrees Fahrenheit, the minimum allowable suction pressure should be increased by the difference of these two vaporization pressures. Since most of the fluids pumped by this type of pump are lighter than water and have a higher pressure of vaporization, the minimum allowable suction pressure will usually be in the range of a negative four to eight feet of the fluid being pumped.

The streamlined center suction gives no appreciable increase in volumetric efficiency. However, the streamlined center suction did make the pump quieter and more stable.

The streamlined center suction should not be considered uneconomical to build until many further tests are run. The streamlined center suction could show an appreciable increase in volumetric efficiency if the pump were handling a compressible fluid.

It is the writer's opinion that a pump with only a center suction would give better results at negative suction pressures than a pump with two possible suction arrangements.

After studying the construction of the pump, the writer has decided that the design of the pump could probably be changed to good advantage. If the cylinder head of the pump were like that of an F-head engine, the suction manifold of this design could be much smoother than the existing pump and the voids in the cylinder heads and valve covers would be much smaller. The reduction of these voids would result in a much higher efficiency in compressible fluids. If valve maintenance were needed, the cylinder head could be removed and taken to a shop much easier than it can be done with the present pump.

There are many more things that need to be done in a similar series of tests. Determination of correct valve size and valve spring constants, for both suction and discharge valves, is needed. Also, there is much to be done in determining the power needed for positive displacement pumps with three or more cylinders operating under very high positive suction pressures.

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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

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

instead of the usual:

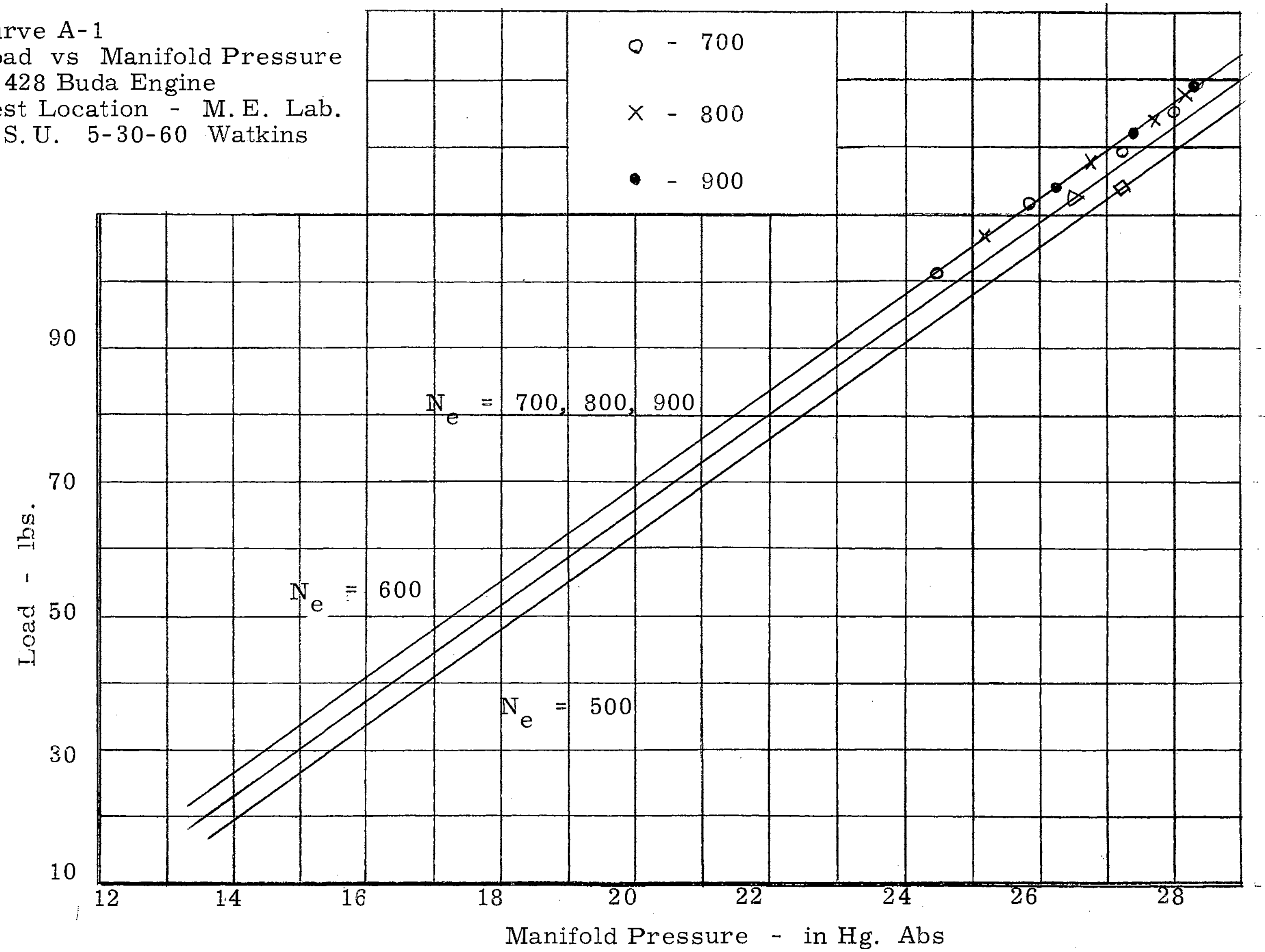
$$\frac{\text{Torque (ft/lbs) x Speed (rpm)}}{5252}$$

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

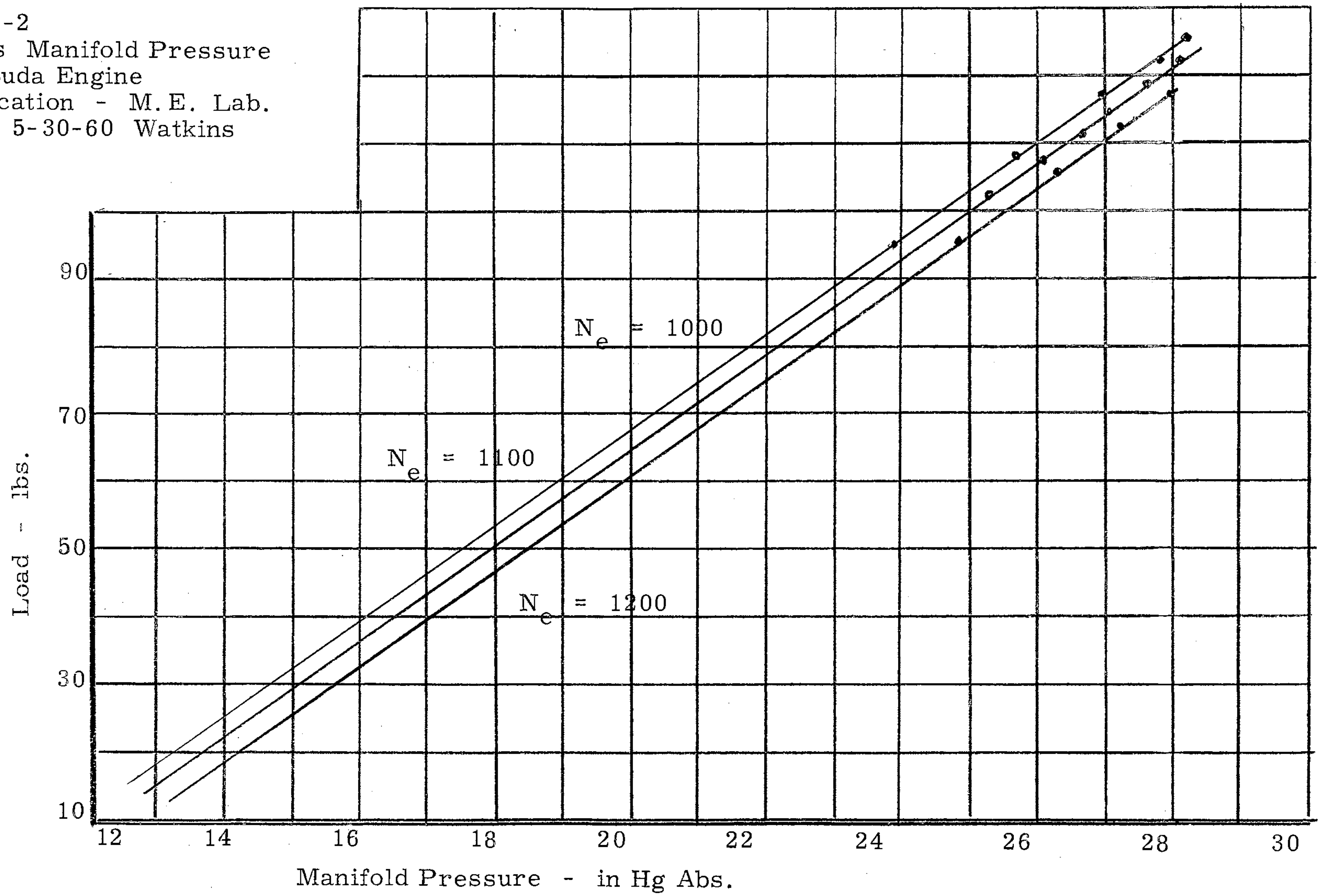
The results of the test plotted were nearly straight lines indicating good results.

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



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

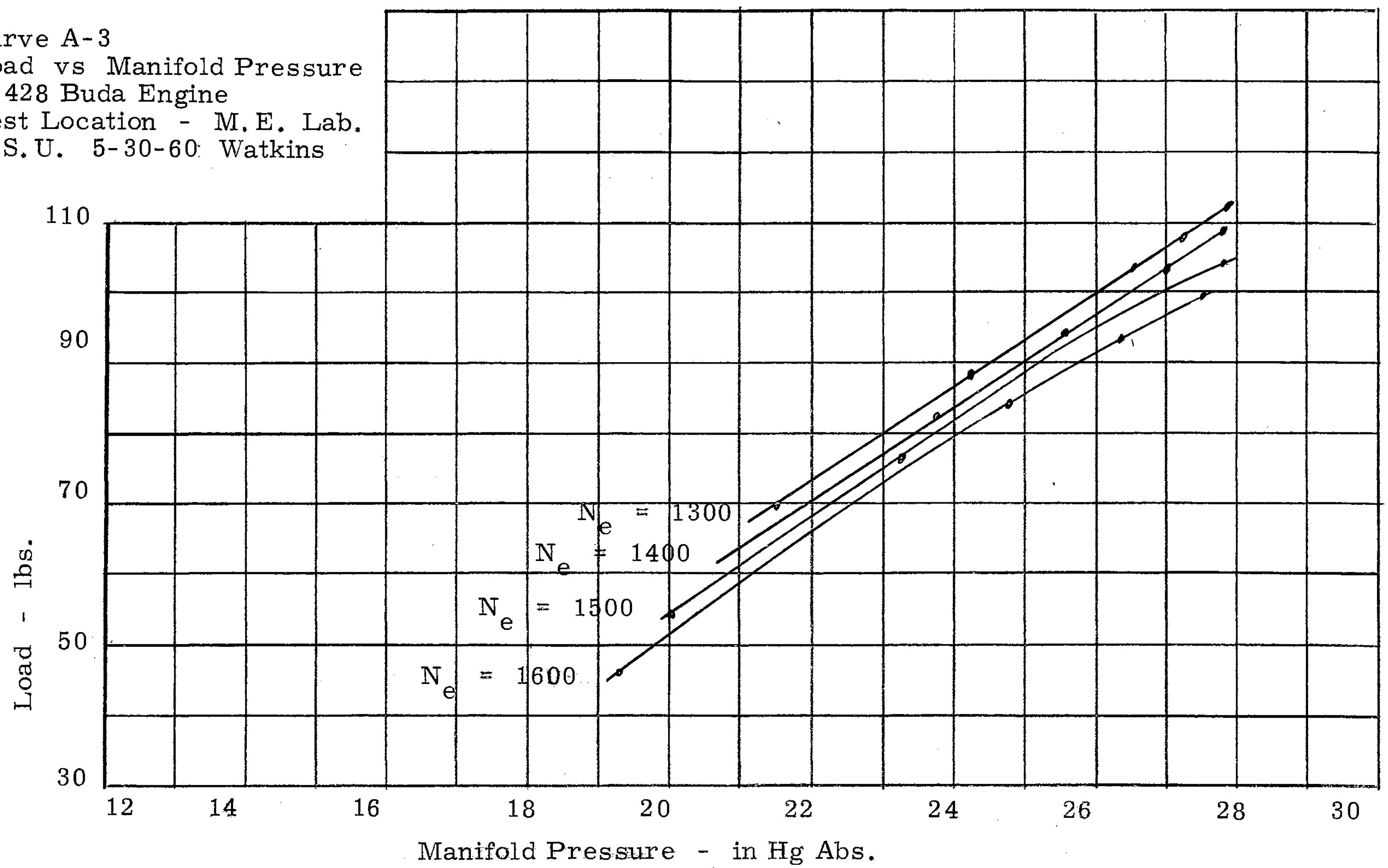


TABLE A-I

VERIFICATION OF ENGINE CALIBRATION DATA

Engine Speed (RPM)	Manifold Pressure (in Hg)	Estimated Load (lb)	Actual Load (lb)	Estimated 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

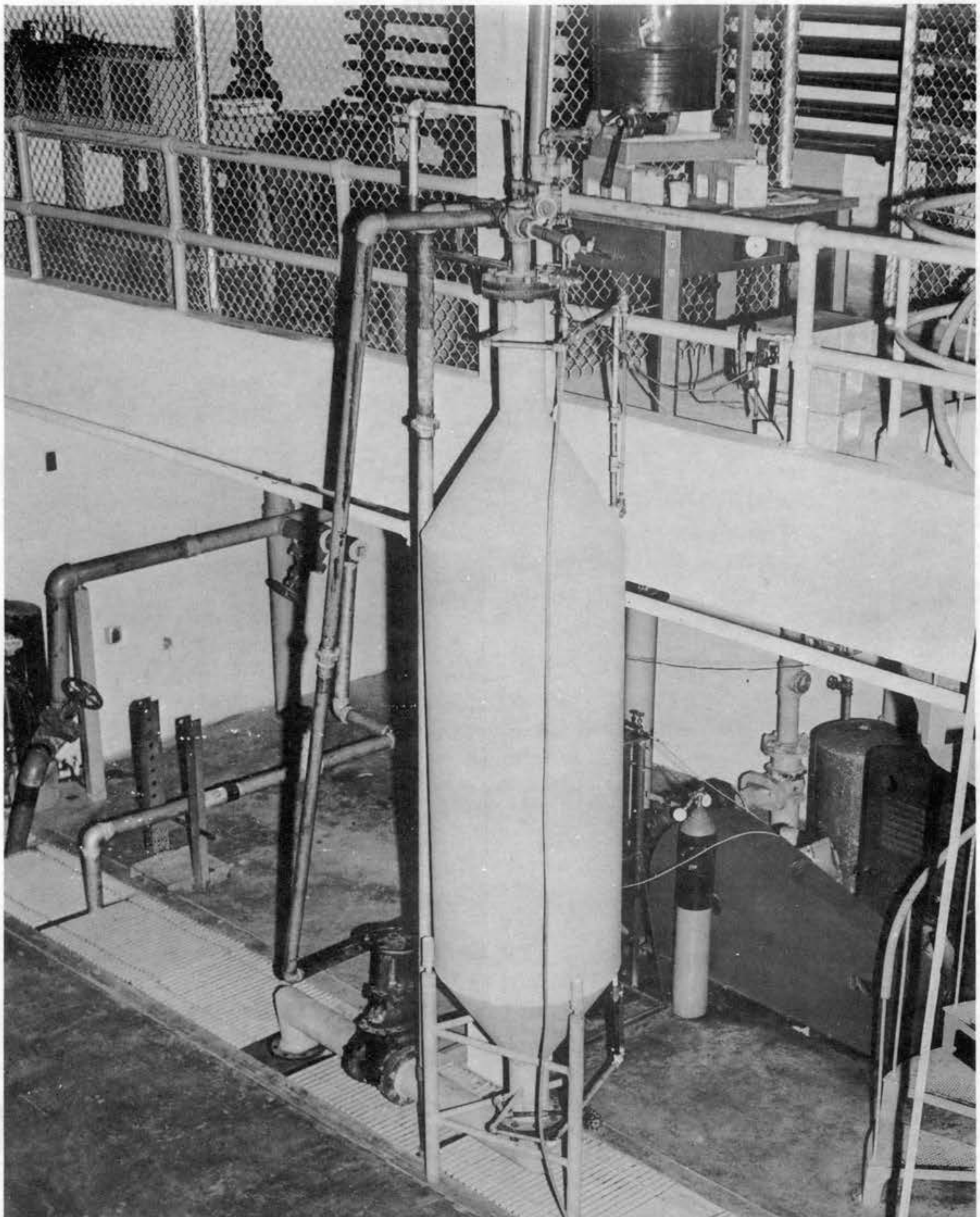
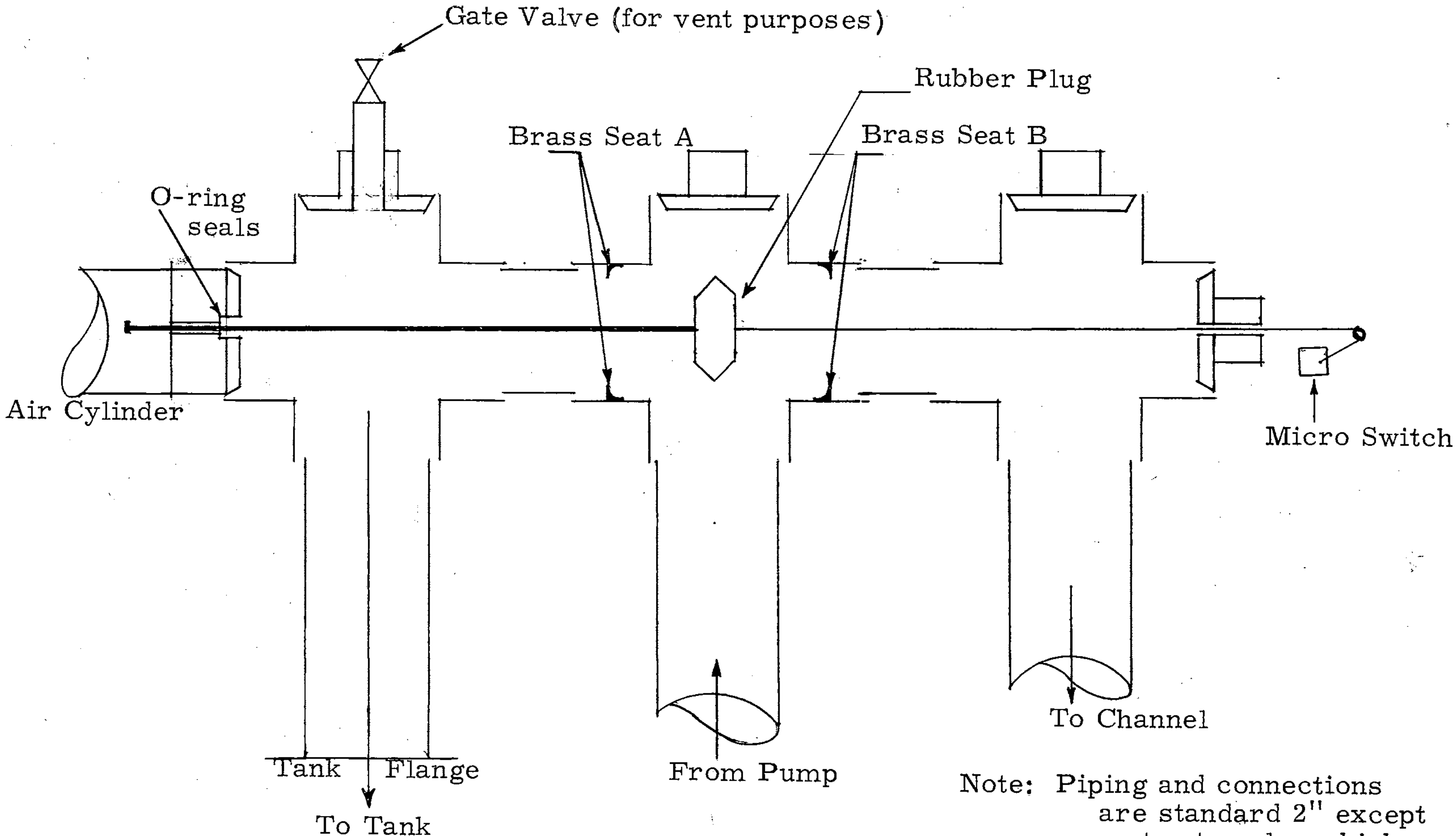


FIGURE B-1

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

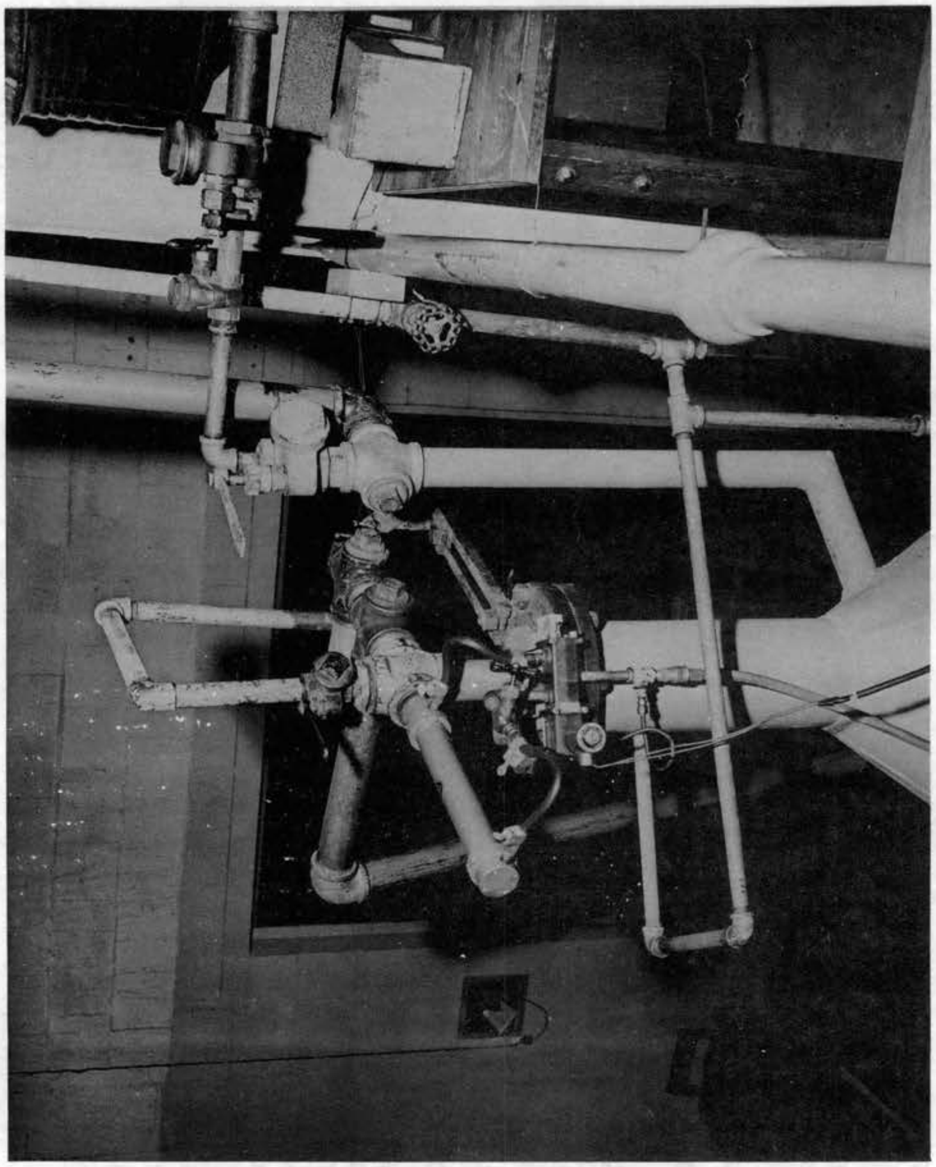


FIGURE B-3

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

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

$$\doteq 34.107$$

γ_t @ 63.5 = 62.315

γ_t @ 64.4 = 62.307

γ_t @ 65.3 = 62.300

$$V = \frac{328.51}{62.315} + \frac{1646.17}{62.300} + \frac{154.6\text{[illegible]}}{62.307}$$

$$\doteq 34.177$$

γ_t @ 65.3 = 62.300

γ_t @ 66.2 = 62.295

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

$$\doteq 34.148$$

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

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

<u>Brake Horsepower</u>

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

<u>Capacity</u>

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

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

Efficiency η_m –

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

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 $\eta_m = \frac{WHP}{BHP} = \underline{\underline{90.70}}$

VITA

Selma Bart Childs

Candidate for the Degree of

Master of Science

Thesis: INVESTIGATIONS IN THE DESIGN OF THE SUCTION SIDE OF OF A HORIZONTAL TRIPLEX PUMP

Major Field: Civil Engineering

Biographical:

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