UNITED STATES
DEPARTMENT OF THE INTERIOR
BUREAU OF RECLAMATION

GLENDIVE PUMPING PLANT ACCEPTANCE TEST
BUFFALO RAPIDS PROJECT
(October 12-18, 1940)

HYDRAULIC MACHINERY LABORATORY
REPORT NO. HM-16

Denver, Colorado
December 12, 1940

Bureau of Reclamation
Hydraulic Machinery Laboratory

Denver, Colorado
December 5, 1940

Laboratory Report No. HM-1[illegible]
Glendive Pumping Plant
Buffalo Rapids Project
Test conducted by: G.J.Hornsby
Checked by: D. J. Hebert
Submitted by: G.J.Hornsby

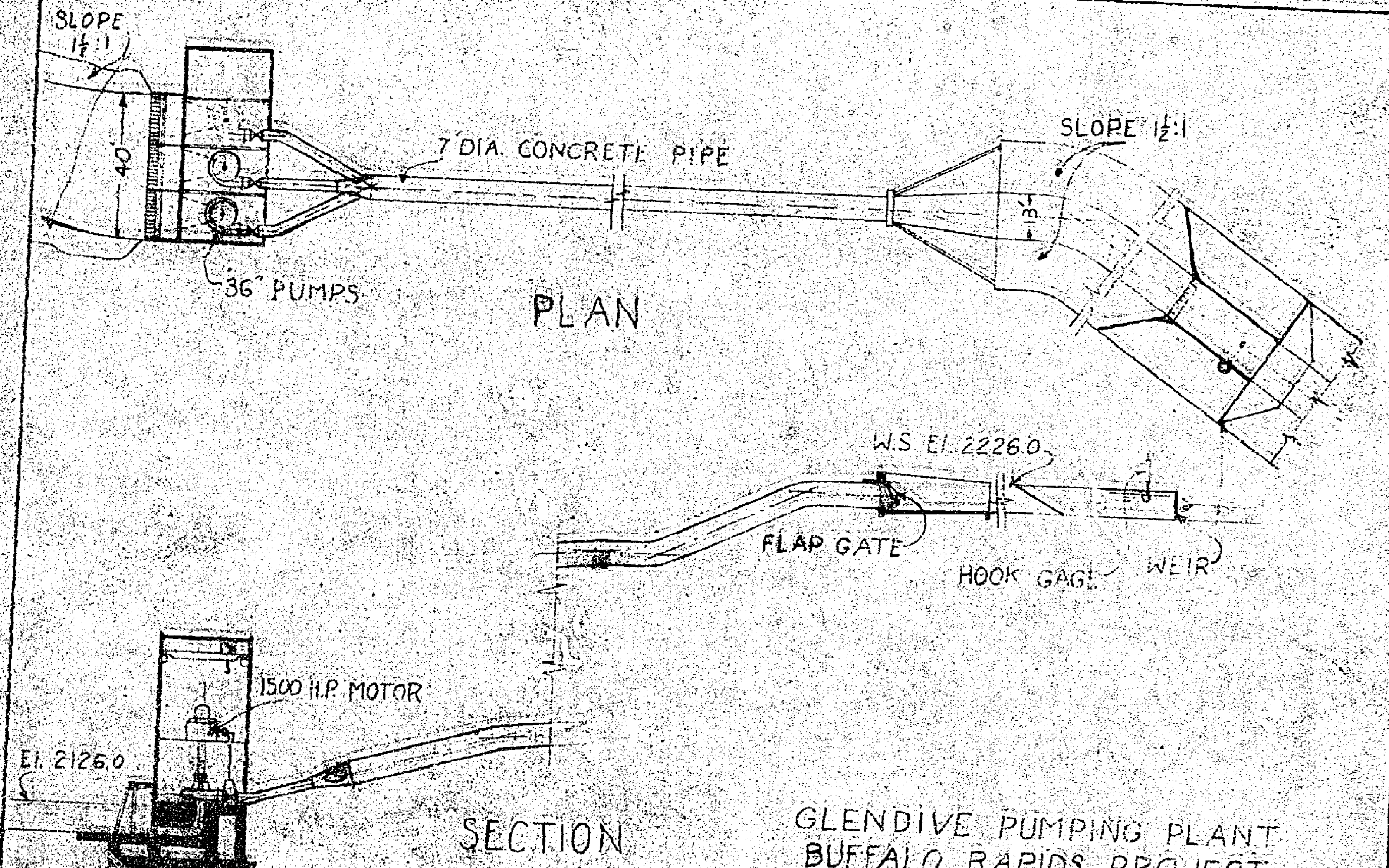
Subject: Field acceptance test on pumping units Nos. 1 and 2 of the Glendive pumping plant, Buffalo Rapids project, Glendive, Montana - Specifications No. 1018-D, October 12-18, 1940.

I. THE TEST

1. Purpose of test. These pumps were first tested in September 1939 to determine whether or not they met the conditions specified in the contract as to head, discharge, efficiency, cavitation, and general performance. (See laboratory report No. HM-1[illegible]) The results of that test indicated that certain changes in the impeller would have to be made by the manufacturer before the pumps would meet the specified conditions. The changes were made and a preliminary test made by the field office indicated that the pumps were ready for the official acceptance tests. The acceptance tests are covered in this report.

2. The pumping plant. The plant is located on the north bank of the Yellowstone River about thirty miles upstream from Glendive and about two miles downstream from the village of Fallon, in Montana. It is a typical irrigation pumping plant consisting, at the time of tests, of two installed units with provision for a third similar unit. The units are of the vertical type consisting of a 1,500-horsepower, 2,200-volt, 60-cycle, 400-r.p.m., synchronous motor direct-connected to a 36-inch single-suction centrifugal volute pump. The pumps are rated at 110-cubic-feet-per-second discharge at a dynamic head of 103 feet. The motors are rated at 95.7 per cent efficiency at 100 per cent power factor and full load, and the pumps are rated at 89 per cent efficiency at rated dynamic head and full discharge. The rated over-all efficiency of the units is, therefore, 85 per cent.

The pumps have individual inlets drawing through rack structures from a common inlet canal. Each pump discharges through a 36-inch gate valve and a short length of 36-inch steel pipe into a leg of the three-way connection, then through a 7-foot diameter concrete common delivery pipe leading into the main canal at the top of the hill. At the point where the delivery pipe discharges into the warped-wing transition to the canal there is an 84-inch swing check, or flap valve, to prevent reversal of flow in case of emergency. The complete arrangement is shown on figure 1.



GLENDIVE PUMPING PLANT
BUFFALO RAPIDS PROJECT

Fig. 1

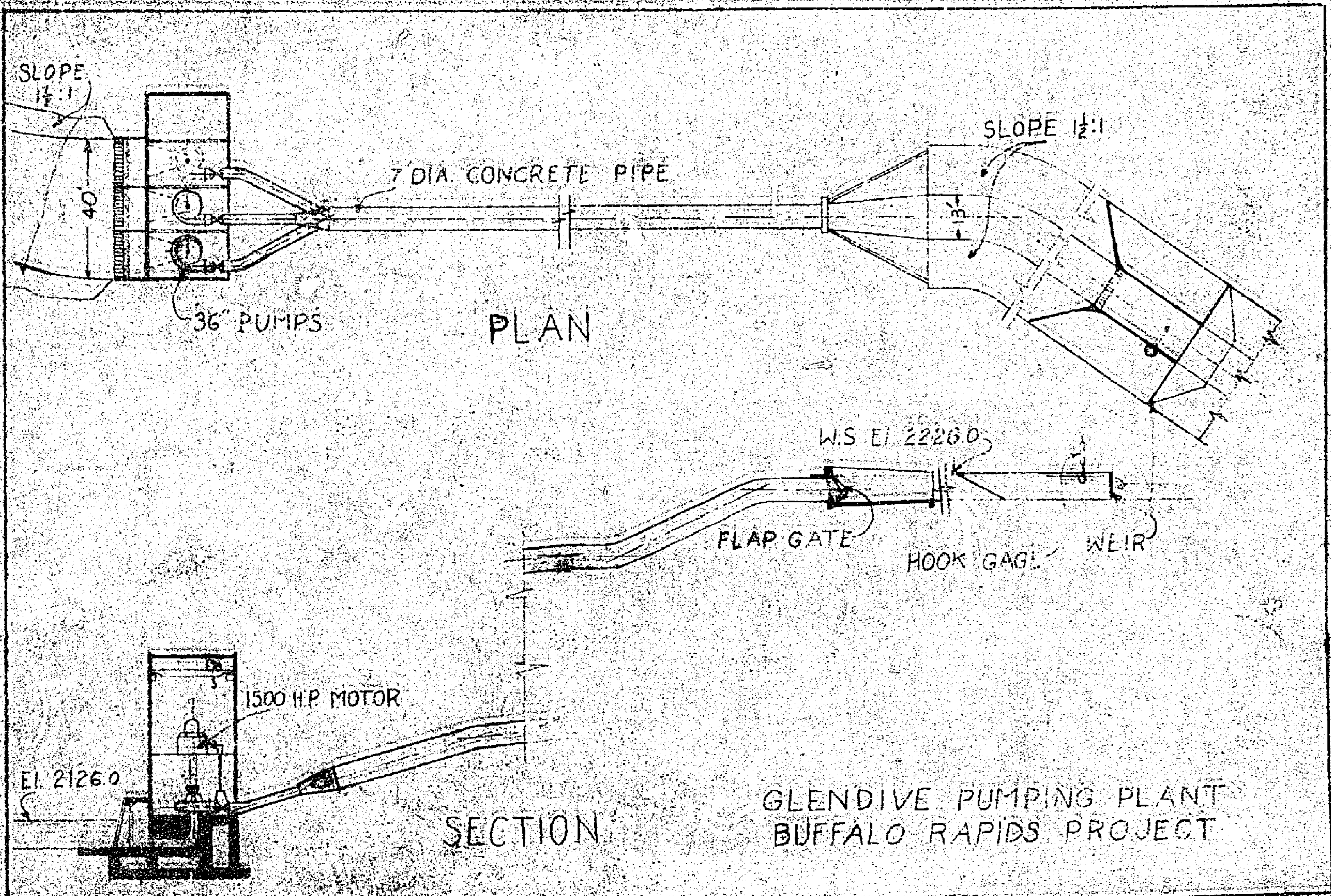

SLOPE 1½:1
40'
36" PUMPS
7' DIA. CONCRETE PIPE
SLOPE 1½:1
13'
PLAN
W.S. El. 2226.0
FLAP GATE
HOOK GAGE
WEIR
1500 H.P. MOTOR
El. 2126.0
SECTION
GLENDIVE PUMPING PLANT
BUFFALO RAPIDS PROJECT

Fig. 1

3. <u>Test procedure.</u> The horsepower input to the motors was measured by means of electrical test instruments connected as shown in the wiring diagram, figure 2. The motor field rheostat was adjusted so as to bring the power factor up to 100 per cent as indicated by balanced readings on the two wattmeters. The power factor meter on the switchboard was not used (see "Notes on Test"). The power input as measured by the wattmeters was checked by means of an ammeter and a voltmeter, each of which could be switched from phase to phase of the motor leads. As a further check, the rotations of the watthour meter disk were counted for each run. The electrical input to the motors thus measured was translated by means of curve, figure 5, into shaft horsepower output to give the horsepower input to the pump. The dynamic head on the pump was measured by means of a mercury column on the discharge side and a water column on the suction side. As the Bourdon-type test gage proved to be of no value in the previous test, it was not used in this test. As the discharge pressure was measured from the center line of the pump and the suction was measured from a point 2.75 feet below this line, the total dynamic head, H, was determined as follows:

$$H = 13.6\ h_d + h_s + 2.75 + \frac{(V_d^2 - V_s^2)}{2g},$$

where,

h_d = discharge pressure in feet of mercury,
h_s = suction pressure in feet of water,
V_d = discharge velocity in feet per second, and
V_s = suction velocity in feet per second.

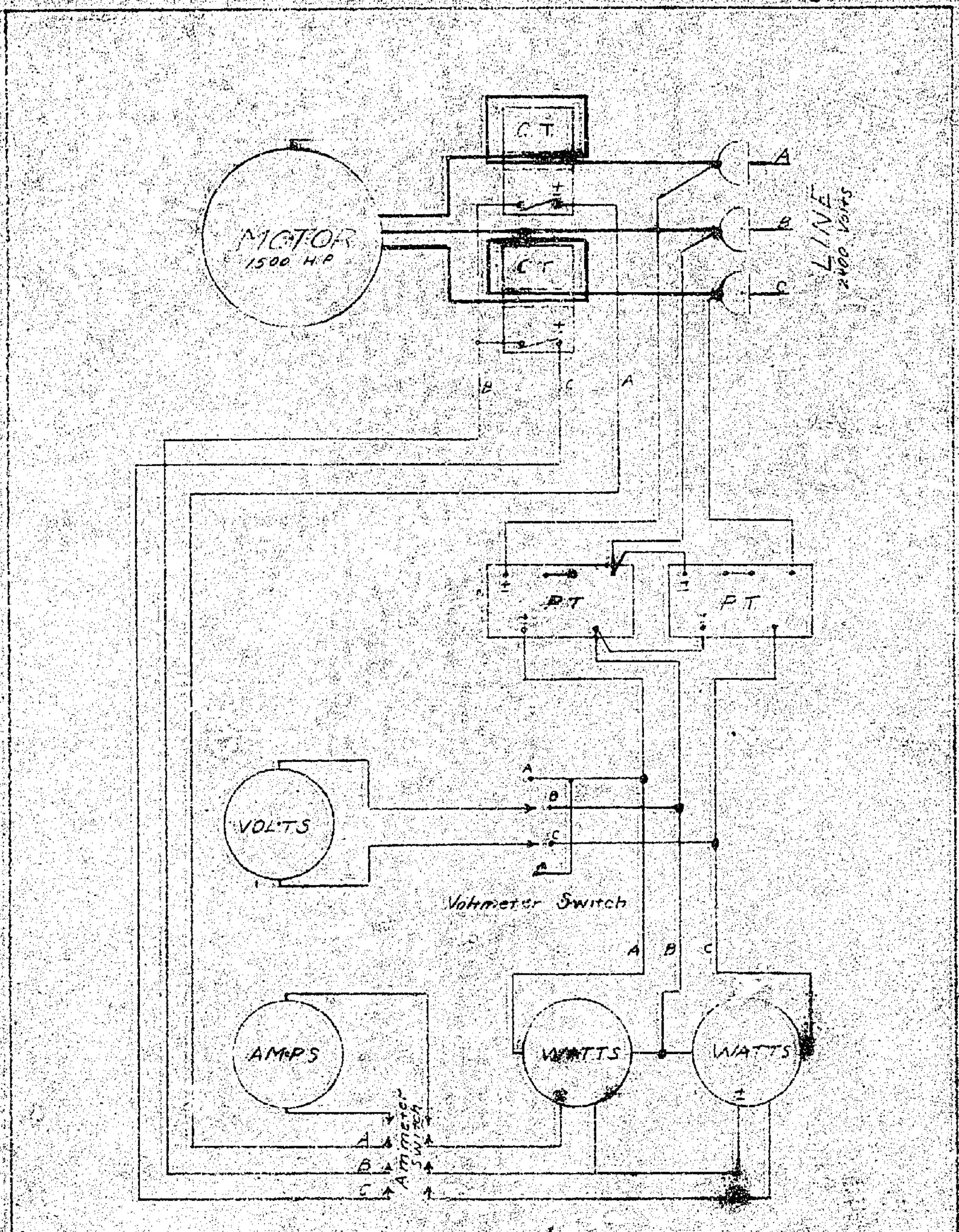
The pipe areas in the plane of the piezometers at both the suction and discharge were carefully measured. The set-up is shown diagramatically in figure 3.

The discharge was measured over a weir designed in strict accordance with Standards of Hydraulic Institute Code. The head over the weir was measured by means of a hook gage reading to thousandths of a foot. The hook gage zero was determined before and after each test by the method shown in figure 3. A discharge curve, figure 4, was plotted according to the Hamilton Smith formula, as shown on the curve sheet. The photograph, plate I, shows the measuring weir in operation.

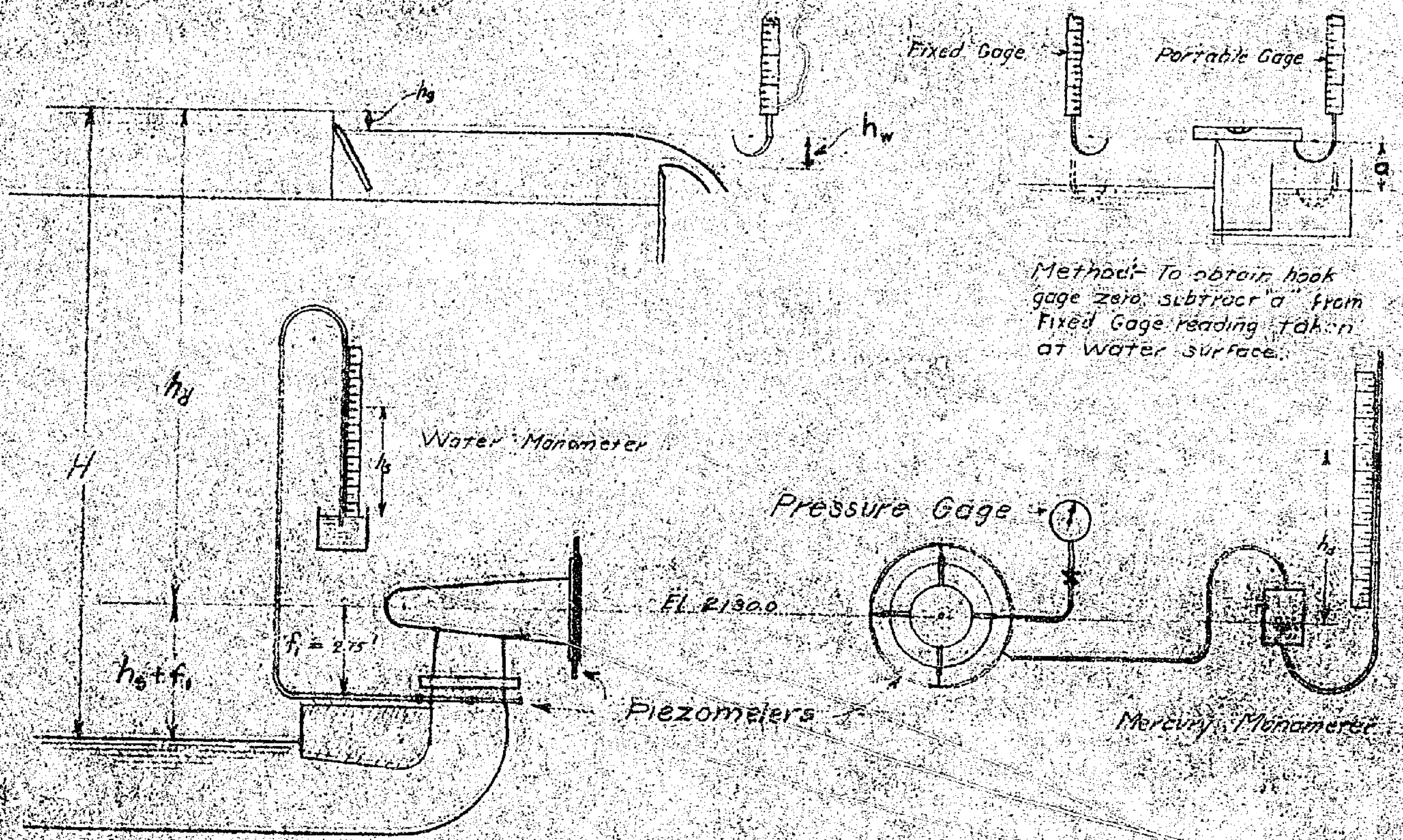
The canal seepage and weir leakage were found to be insignificant.

Each test consisted of five runs. The first run was with the

Fig. 2



WIRING DIAGRAM
Glendive Pumping Plant
Buffalo Rapids Project



HEAD MEASUREMENTS
Glendive Pumping Plant
Buffalo Rapids Project

Fig. 3

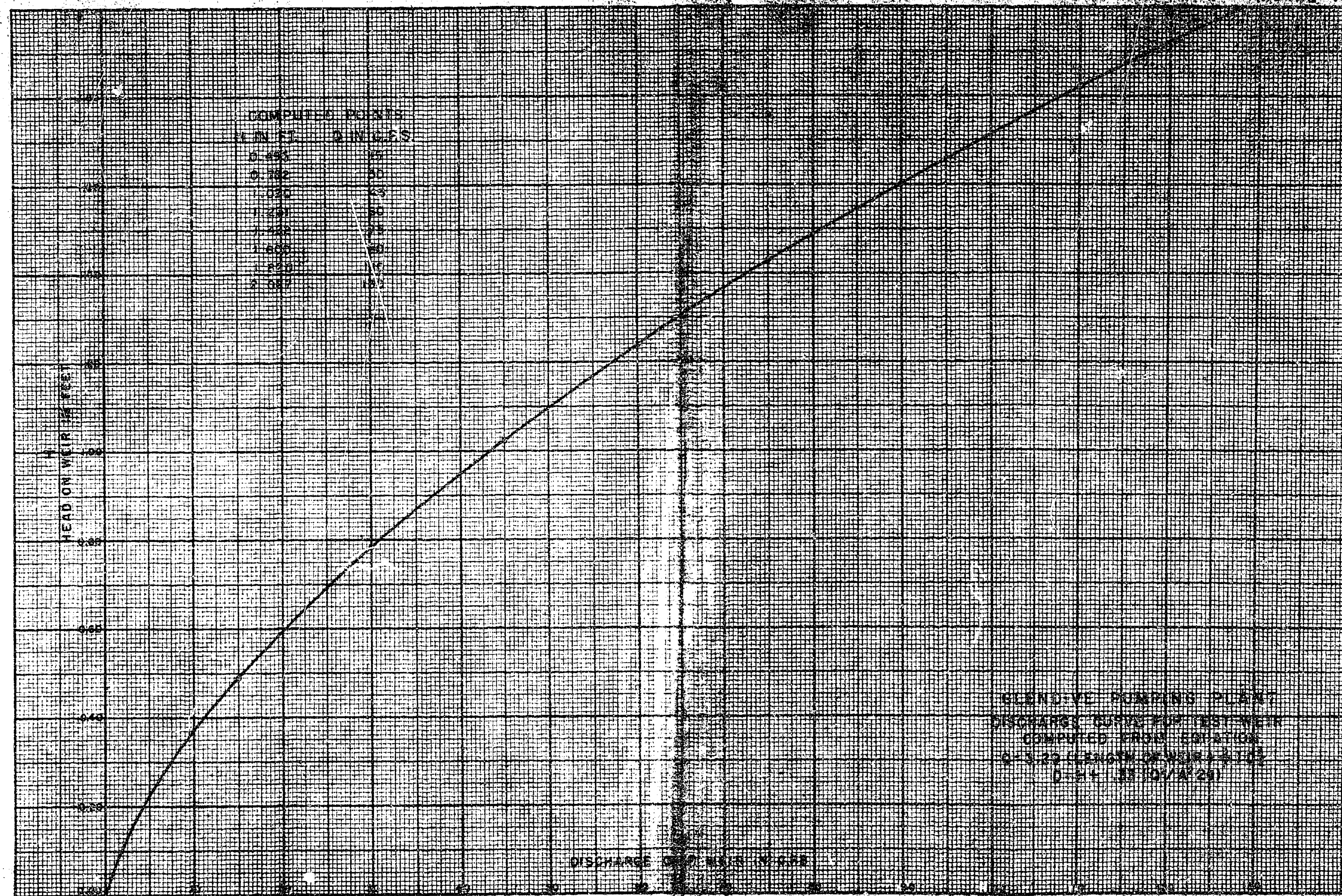
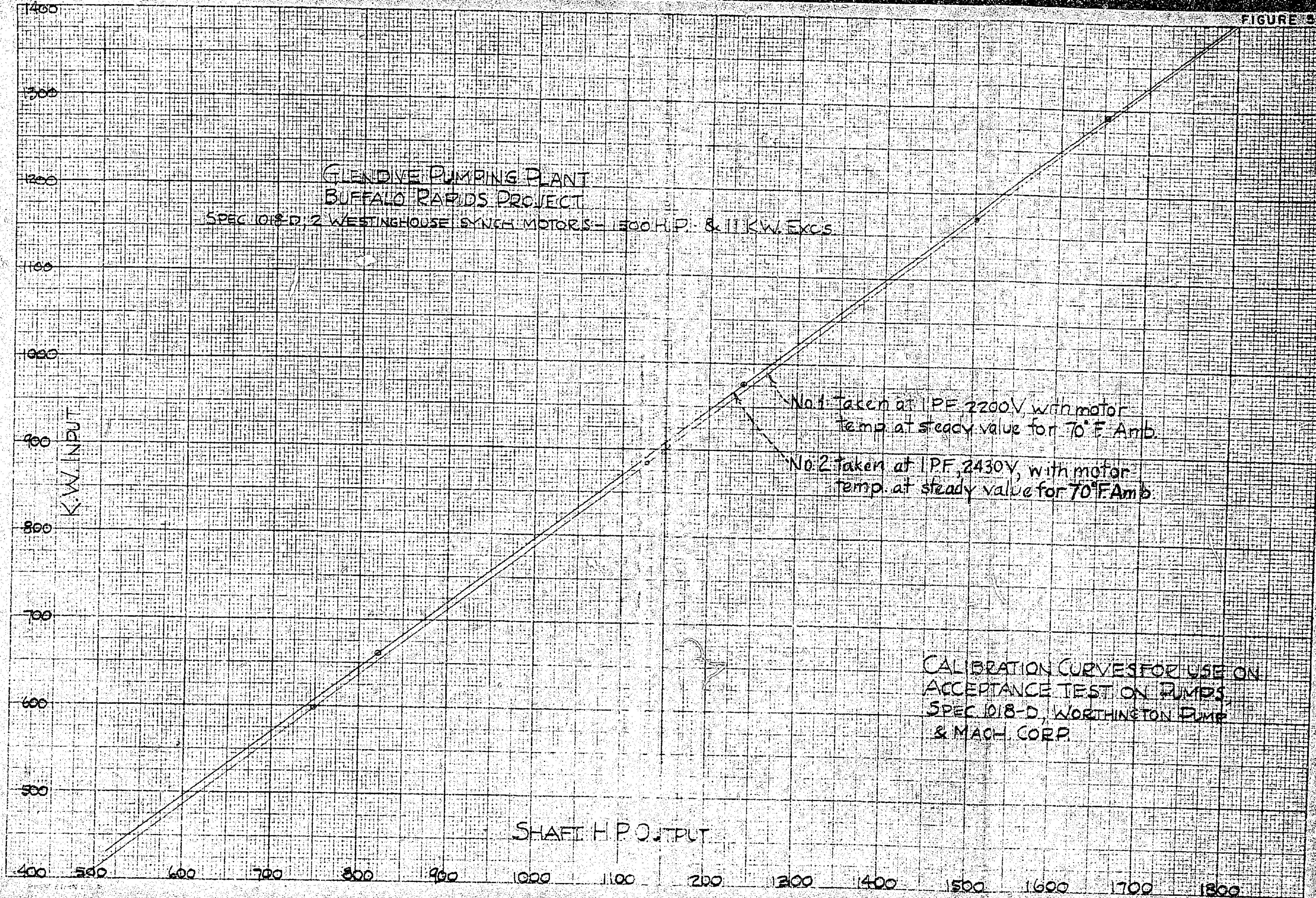

COMPUTED POINTS
H IN FT. Q IN C.F.S.
HEAD ON WEIR IN FEET
1.00
0.80
0.60
0.40
DISCHARGE OVER WEIR IN C.F.S.
GLENDIVE PUMPING PLANT
DISCHARGE CURVE FOR TEST WEIR
COMPUTED FROM EQUATION

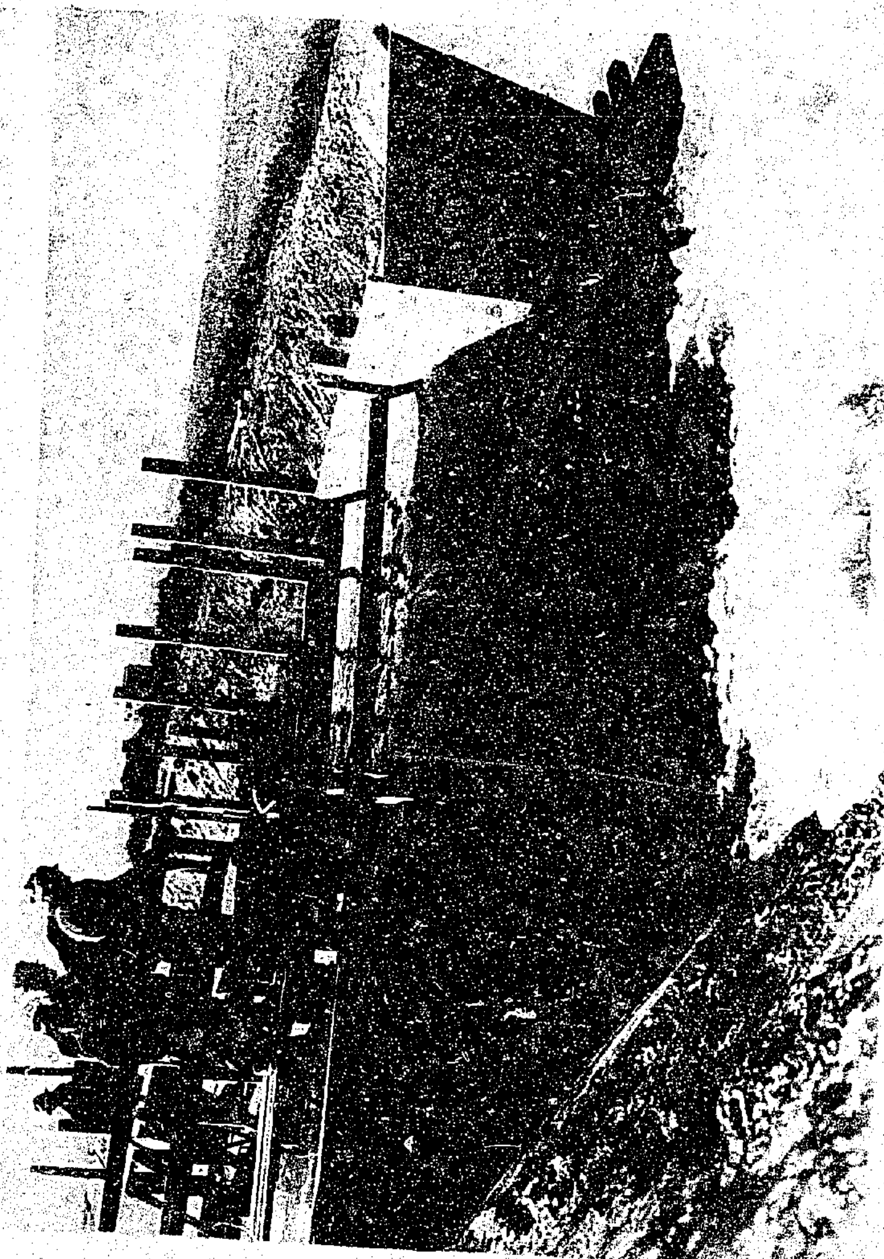
FIGURE 9



EUGENE DIETZGEN CO. NO. 346-D

9-22-39, 243-D-447

Plate I



Measuring Weir Glendine Pumping Plant

valve fully open for maximum discharge. For succeeding runs the valve was set for discharges of approximately 95, 80, 60, and 105 cubic feet per second in order. The shut-off head was also determined. The reason for going back to the 105-cubic-feet-per-second discharge was to check for any "drift" or gradual change in performance of the unit or accuracy of test instruments which may have taken place during the test. Each run consisted of the average of ten readings taken at one-minute intervals, except the motor r.p.m. which was taken at the end of each test only. The following readings were taken for each run:

A. Electrical:-

Ammeter, phases A, B, and C.
Voltmeter, phases A-B, B-C, and C-A.
Wattmeters, 1 and 2.
Watthour meter, disk turns.

B. Hydraulic:-

Discharge manometer, mercury.
Suction manometer, water.
Weir hook gage.
Inlet staff gage.

C. Temperatures:-

Pump room.
Inlet to motor.
Motor stator windings.
Motor, or operating, room.
Water at pump inlet.

It is fully realized that some of these data are not essential to the test, but they are on file in the laboratory in case any question arises requiring their use.

4. Results of tests. Due to the fact that the motors were designed to operate on 2,200 volts and the transformers supplied the power at 2,400 volts, it was decided to make a test of three runs on one of the pumps at full discharge and 2,200 volts. This test was made on unit No. 1 and the results are plotted on performance curve figure 6; test is noted as "2,200-volt test." This test will be discussed later in this report.

The results of test on unit No. 1 are tabulated in table 1 and plotted on performance curve figure 6. The results of test on unit No. 2 are tabulated in table 2 and plotted on performance curve figure 7. The water inlet temperature for test on unit No. 1 ranged from 9.5° C. to 10.5° C., and for test on unit No. 2 ranged from 10.8° C. to 11.5° C.

II. NOTES ON THE TESTS

1. Performance of the pumps. The pump performance showed a vast improvement over that observed in the previous tests. The cavitation noises appeared to be completely cleared up and the operation was smooth and quiet. The water seal in the stuffing glands works satisfactorily when the pumps are operating. The pumps are sometimes slow in priming, especially after being idle for some time. This is due to the fact that sealing water is supplied to the stuffing gland only when the pumps are operating. This condition could be improved by connecting in the spring water supply for priming purposes only.

2. Pump wear rings. The wear rings on unit No. 1 had been damaged and later repaired by building up with babbitt and machining. In order to ascertain whether or not this condition had again developed, or that the repaired rings had been improperly fitted causing the difference in performance between the units tested, a careful check of the runner clearances was made on both units. The measurements were made at the quarter points both top and bottom on both units. In plan, point A is upstream, point B is left, point C is downstream, and point D is right. These measurements in inches are shown below

	Points	A	B	C	D
Unit No. 1	Top	0.033	0.040	0.050	0.042
	Bottom	0.036	0.053	0.040	0.020
Unit No. 2	Top	0.030	0.038	0.031	0.026
	Bottom	0.069	0.036	0.028	0.055

3. The outlet transition. This structure has continued to settle somewhat but no serious breaks have occurred. A cut-off collar has been built around the conduit between the elbow and the transition, and the wings of the transition will be built up to the original elevation.

4. The motors. Both motors start, come up to speed, and synchronize normally. The collector rings are smooth except in spots which appear to be burned. There is a definite line of demarkation between the smooth and the rough portions of the periphery. The temperature rise in the motor windings appears to be excessive. The operator states that during the operating season the motors became so hot that it was necessary to reduce the pump discharge materially, sometimes as much as 85 percent. The observed temperatures during the tests are shown in tables 3 and 4.

TABLE 1

Unit No. 1 - Test No. 1

Run No.	Q	V_s	V_d	$\frac{V_d^2 - V_s^2}{2g}$	$H_s + f_1$	H_d	Total Head	WHP	BHP	Eff.
1	108.4	14.69	15.79	0.515	5.48	96.97	102.97	1265	1450	87.2
2	95.0	12.87	13.85	0.387	4.70	106.53	111.61	1203	1408	85.5
3	81.0	10.97	11.81	0.281	4.19	113.61	118.09	1085	1319	82.4
4	59.9	8.12	8.73	0.154	3.11	121.43	124.70	848	1147	73.9
5	104.4	14.15	15.22	0.478	5.25	100.0	105.72	1252	1445	86.7
6	shut-off	-	-	-	2.20	135.12	137.32	-	631	-

TABLE 2

Unit No. 1 - Test No. 1

Run No.	Q	V_s	V_d	$\frac{V_d^2 - V_s^2}{2g}$	$H_s + f_1$	H_d	Total Head	WHP	BHP	Eff.
1	114.0	15.49	16.56	0.528	5.71	95.44	101.68	1315	1483	88.7
2	101.7	13.82	14.88	0.428	4.97	104.28	109.68	1265	1446	87.5
3	88.5	12.02	12.86	0.241	4.26	111.90	116.40	1167	1365	85.6
4	64.5	8.76	9.38	0.194	3.20	119.46	122.86	899	1197	75.1
5	110.0	14.94	15.98	0.435	5.47	97.69	103.60	1304	1474	88.5
6	shut-off	-	-	-	2.03	135.91	137.94	-	644	-

TABLE 3

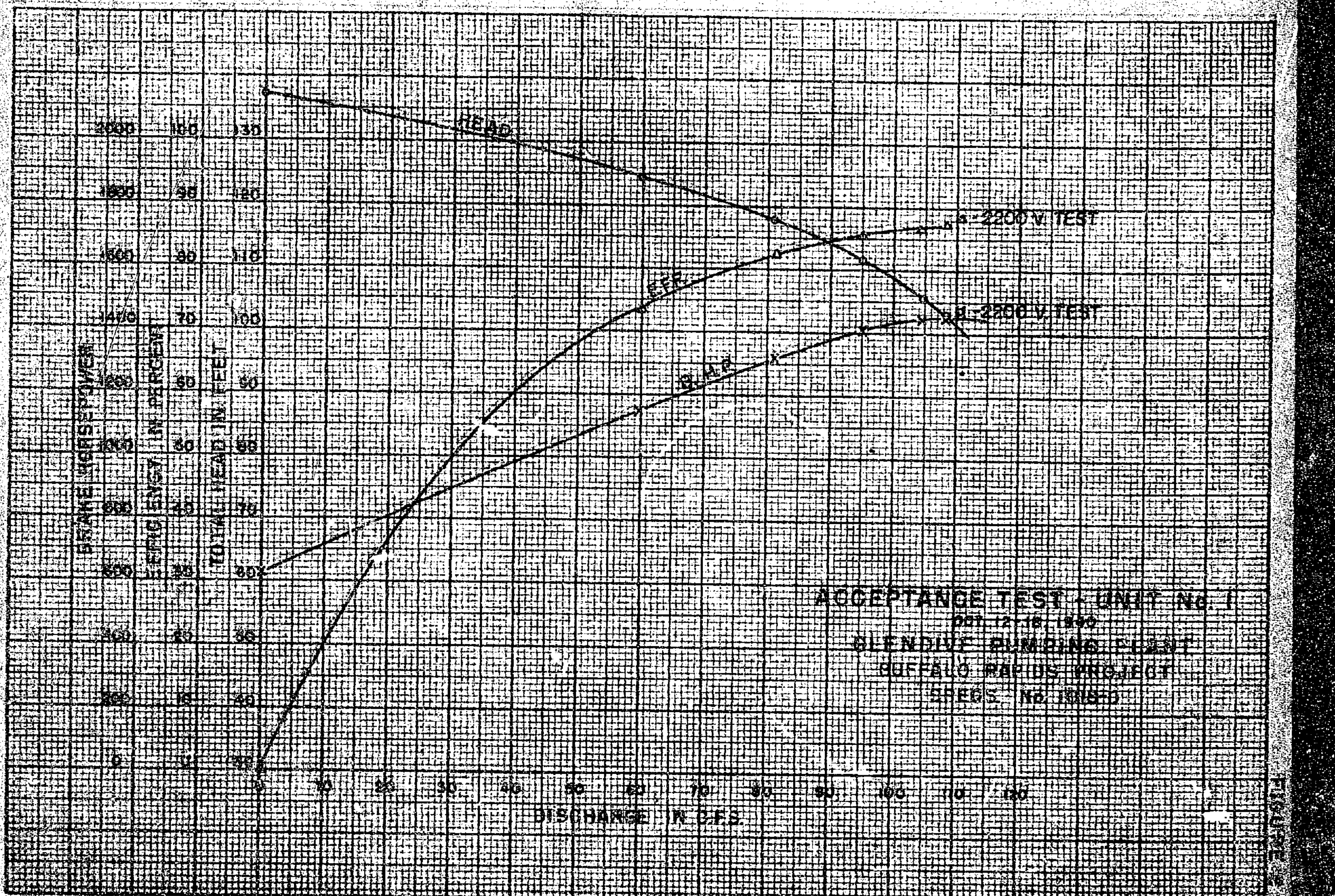
MOTOR TEMPERATURES
Unit No. 1 - Test No. 1

Run No.	Kw. Input	Temperatures, Degrees C. Pump Room	Under Motor	Stator Windings	Operating Room	Remarks
1	1137.4	16.0	-	64.5	23.9	Temperatures taken near the completion of each run
2	1101.5	16.0	-	65.0	25.6	
3	1037.6	16.0	-	67.0	26.4	
4	906.7	16.0	-	67.5	26.7	
5	1131.9	16.0	-	70.5	27.8	

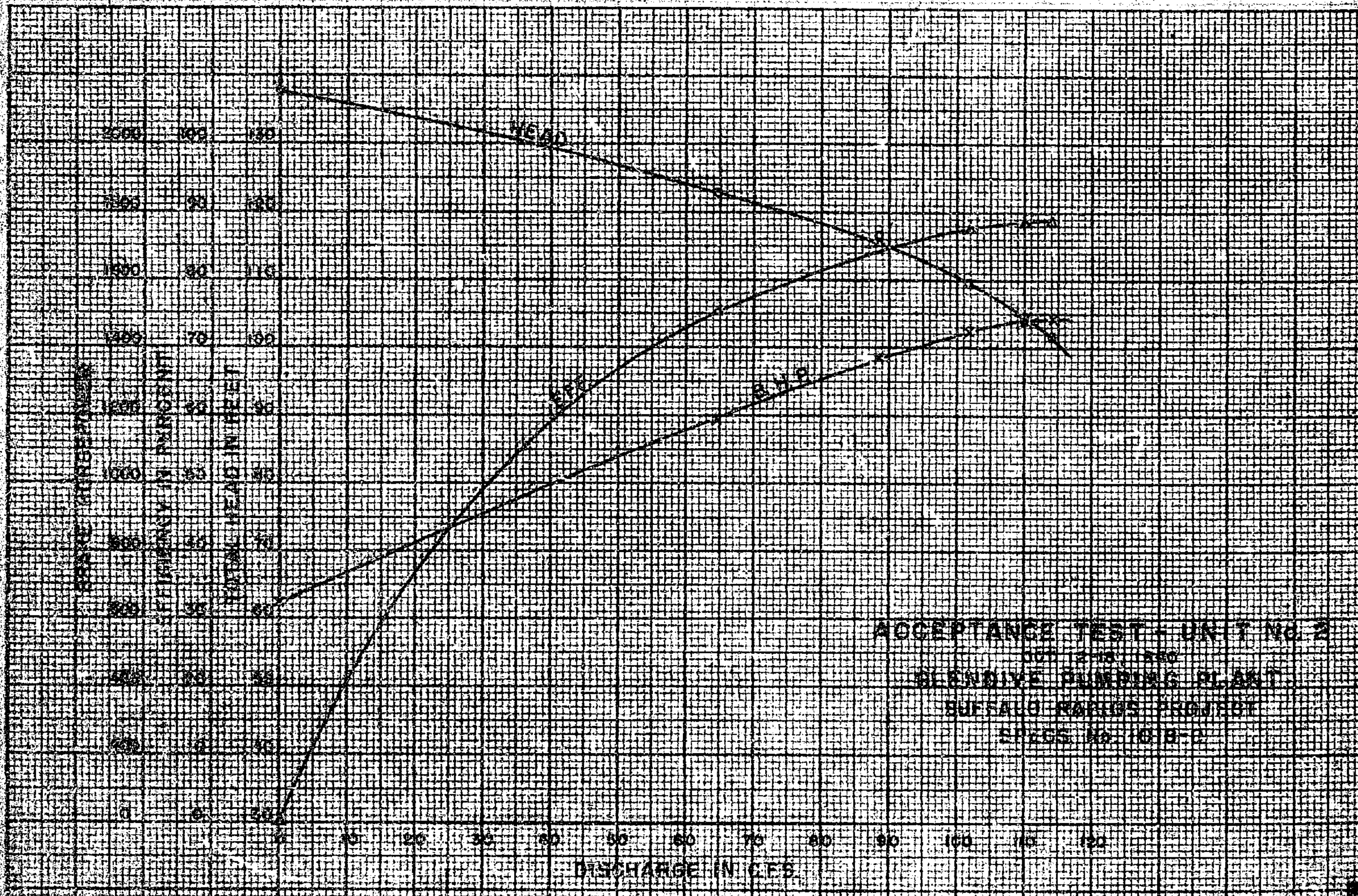
TABLE 4

MOTOR TEMPERATURES
Unit No. 2 - Test No. 1

Run No.	Kw. Input	Temperatures, Degrees C. Pump Room	Under Motor	Stator Windings	Operating Room	Remarks
1	1162.5	15.0	18.0	50.0	22.8	Temperatures taken near the completion of each run
2	1132.4	15.0	18.7	62.0	25.6	
3	1071.8	15.0	19.5	68.0	27.8	
4	943.6	15.4	19.5	72.0	28.9	
5	1154.4	15.1	21.0	76.0	30.0	


HEAD
EFF.
B.H.P.
2200 V. TEST
2200 V. TEST
BRAKE HORSEPOWER
EFFICIENCY IN PERCENT
TOTAL HEAD IN FEET
DISCHARGE IN C.F.S.
ACCEPTANCE TEST - UNIT No. 1
GLENDIVE PUMPING PLANT
BUFFALO RAPIDS PROJECT

FIGURE



ACCEPTANCE TEST - UNIT No. 2
GLENDIVE PUMPING PLANT
BUFFALO RAPIDS PROJECT
SPECS. No. 1018-D
HEAD
EFF
B.H.P.
DISCHARGE IN C.F.S.
EFFICIENCY IN PERCENT
TOTAL HEAD IN FEET

FIGU

The air circulation through the motor seems to be rather sluggish, due to lack of fan vanes on the rotor and to the large air inlet space blocked off by the guide bearing and supporting structure under the motor.

4. The "2,200-volt test." This test was run on unit No. 1. The motor was started at 2:45 a.m., October 14 and allowed to run until 7:00 a.m. to bring the temperature up to normal. At 6:30 a.m. the voltage was reduced to 2,200 and the test started. Three complete runs were made in succession at full discharge. The results were averaged and plotted on curve in figure 5. Comparison shows that the discharge for this test was three cubic feet per second more than for test No. 1 on this same unit. The discharge of the pump is a function of the frequency of the power circuits, such that a difference of about 0.75 of a cycle would be sufficient to make this difference in discharge. Unfortunately no means of measuring the frequency was available and the speed counter and stop-watch method of determining the speed was not sufficiently accurate to measure this small difference. Therefore it is suggested that in future tests a portable frequency meter be used to determine the speed of the unit. Comparing the 2,200-volt test with the test on unit No. 2, curve figure 6, it will be seen that the points fall almost exactly together. Since the pumps are supposed to be identical, being made off the same patterns, it may be concluded that the performance will be identical. This, however, is not always true due to variations in handling in the foundry, but the difference seldom amounts to as much as one per cent plus or minus in the performance of the finished product.

The power-factor meter would not operate satisfactorily. On the previous test it was found that the hair spring was broken (probably burned out). The instrument was subsequently sent to the factory for repairs. Upon being reinstalled on the switchboard it would not function though the factory insists that it was in good condition when it was returned to the project.

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