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DEVELOPMENT OF PUMP CHARACTERISTICS FROM FIELD TESTS

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16. ABSTRACT

Field tests were conducted to determine the pump characteristics at four pumping plants on the Southern Nevada Water project. Each pump tested was instrumented to measure speed, torque, and pressure head as functions of time. Records were obtained for pump startup, normal operation, and power failure. The Southern Nevada system was modeled for these same operations on a master computer program, TAHS. Using TAHS, the pump characteristics were adjusted by the iterative process until the computer results were in agreement with the field test records.

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UNITED STATES DEPARTMENT OF THE INTERIOR * BUREAU OF RECLAMATION

ACKNOWLEDGMENTS

The authors appreciate the efforts of Richard Arbour and Eugene Campbell of the Electric Power Branch, Engineering and Research Center who assisted with the field tests for this study. Their contributions included the development of some of the instrumentation used in the tests.

CONTENTS

FIGURES

LETTER SYMBOLS AND QUANTITIES

Symbol	Quantity	Metric unit	U.S. customary unit
A, B, C	Dimensionless coefficients in pump startup equation		
g	Gravitational acceleration	m/s^2	ft/s^2
H	Head produced by pump	m	ft
h	Head ratio = H/h_r		
h_r	Rated head	m	ft
M_m	Motor torque	N·m	lb·ft
M_p	Pump shaft torque	N·m	lb·ft
M_r	Rated torque	N·m	lb·ft
n	Rotational speed	r/min	r/min
n_r	Rated speed	r/min	r/min
n_s	Pump specific speed	$\frac{(r/min)\sqrt{m^3/s}}{m^{0.75}}$	$\frac{(r/min)\sqrt{gal/min}}{(ft)^{0.75}}$
Q	Capacity (discharge)	m^3/s	ft^3/s
Q_r	Rated capacity	m^3/s	ft^3/s
q	Capacity ratio = Q/Q_r		
t	Time	s	s
WR^2	Product of weight of revolving parts and the square of the radius of gyration	$kg{\cdot}m^2$	$lb{\cdot}ft^2$
α	Speed ratio = n/n_r		
β_m	Motor torque ratio = M_m/M_r		
β_p	Pump shaft torque ratio = M_p/M_r		
θ	Angle for characteristic curves = $\tan^{-1} \alpha/q$	degree	degree
π	3.141 59 . . .		
ω	Angular velocity	rad/s	rad/s

INTRODUCTION

Southern Nevada Water Project—Stage I is shown on figure 1. Stage II is currently under development. Pumping Plant I consists of 10 identical units which pump raw water from Lake Mead to the Alfred Merritt Smith Water Treatment Plant. Each unit is a vertical, two-stage pump with a specific speed n_S of 63.2 (r/min) $(m^3/s)^{0.5}/m^{-0.75}$, [3265 (r/min) $(gal/min)^{0.5}/(ft)^{-0.75}$].

From the water treatment plant clearwell (Forebay 1A), treated water flows by gravity to Pumping Plant 1A. It is then pumped into two pipelines, the Main Aqueduct and the Boulder City Lateral. Seven large identical units pump most of the flow through the Main Aqueduct toward Henderson and Las Vegas, Nev. Three smaller pumps presently discharge into the Boulder City Lateral. The specific speed of the large pumps is 30.0 (1550) and that of the smaller pumps is 14.5 (765).

Pumping Plant 3 is located in Henderson, Nev. It consists of four identical pumps with a specific speed of 27.2 (1405). Pumping Plant 6, with four identical units, is located in Las Vegas. Each pump has a specific speed of 28.9 (1490).

AUTOMATION

The Southern Nevada Water Project will be automated in the near future. The automation will consist of flow control valves at all rate-of-flow control stations regulated by minicomputers at the valve locations, ultrasonic water level devices in the surge tanks, pressure transducers in the forebay and regulating tanks, and controls on the pumps. Information from each of these components will be transmitted to real-time computers located at a master station in the water treatment plant. Decisions will have to be made at the master station to control the startup and shutdown of all pumps in the system during normal and emergency operation. For all pumps in the system, it then becomes a matter of extreme importance that the *actual* pump characteristics are known. This study was undertaken to satisfy that need by developing the actual pump characteristics.

FIELD TESTS

Five separate pumps were instrumented, one representative for each of the five specific speed units installed in the system to date. The pumps were instrumented so that speed n, torque M, and head H could be measured instantaneously, and the time histories recorded on four-channel oscillograph paper. Rotational speed was measured with a counter which recorded banded markings on the pump shaft. Strain gages mounted on the pump shaft registered torque. Pressure head was measured with pressure transducers located on each side of the pump, except at Pumping Plant 1 where the vertical shaft prevented access to the submerged intake line, therefore four sets of pump characteristics were developed. Test setups are shown on figures 2 through 8.

Tests were performed to produce time histories for pump startup and power failure at several valve openings. For each test, the pump was brought up to rated speed against a closed discharge valve. The valve was opened to the desired position. Then power was cutoff and after the resultant hydraulic transient had subsided the valve was closed slowly. In each pumping plant all pumps were shutdown except for the unit being tested. Before each test, the forebay tank water surface elevation was recorded. This water surface did not change significantly by the short interval operation of only one pump so it was assumed to remain constant. Similarly, it was assumed the closest afterbay tank downstream from the test pump maintained initial water surface elevation. Figure 9 shows a typical test setup.

Test results recorded by the oscillograph are shown on figures 10 and 11. The torque and head curves were difficult to read accurately due to the fluctuations caused by cavitation. It was assumed that the fluctuations were of equal magnitude in the positive and negative directions, thus placing the actual curve at the center of the wide band of fluctuation. The scale of the graphs also limited the accuracy for which they could be read.

The graphical data on figures 10 to 11 are from a test with a 100-percent valve opening. The partial valve opening results were consistent with that for the full opening — varying in magnitude — but not in the general pattern of the transients. The 100-percent opening test results were used to develop the pump characteristics because the oscillograph curves registered at the larger valve openings could be read with greater accuracy.

As mentioned previously, the head upstream of the pumps in Pumping Plant 1 could not be measured. This prevented determining the differential head across the pumps that is needed to derive their characteristics. Since Pumping Plant 1 units discharge only a short distance into a forebay tank at the water treatment plant, these characteristics were not as important as units in the other pumping plants.

DEVELOPMENT OF CHARACTERISTICS

To develop the characteristics for a given pump four variables are needed as functions of time. They are the pump rotational speed n, the pump shaft torque M, differential pressure head across the pump H, and the pump flow Q. Given a complete record of all four variables, the pump characteristics can be calculated directly. In this case, however, there was no available means to measure the rapid flow variation in the pump. Whereas the other three variables could be determined, a complete record of the closed pipeline flow could not be obtained during the transients being studied. It was decided for this reason to model the field tests on a computer using program TAHS, that would yield Q as a function of time.

Program TAHS simulates various operating criteria for any system described to it by the user. In this study, each pump under consideration was modeled in a small system which had the forebay tank for that pumping plant as the upstream boundary, and the first afterbay tank downstream from the pump as the terminating downstream boundary. To attain agreement and verification with field test results, computer runs were made simulating field operations.

Given a set of characteristics for a pump, program TAHS uses that data as the base for calculating any pump operation. Characteristics are stored in the program for different pump specific speeds. When exact characteristics for a pump are not known, the user may select one of the sets stored in TAHS that will be approximately the correct characteristics. Prior to this study, it was customary to assume that the best approximation was the set of characteristics associated with the specific speed closest to that of the pump being considered.

For each test pump the first computer run used known characteristics for a pump with slightly different properties from the test pump, which yielded approximate values for the flow-time history. Correlating this flow data with speed, torque, and pressure head measured in the field, a new set of characteristics was developed to be used for a second computer iterative run. The values of predicted flow became increasingly accurate with each set of revised characteristics, and in turn lead to a more refined set of characteristics for the test pump. The iterations converge on the correct characteristics.

For this process to be accurate, correct WR^2 values must be used. The WR^2 was verified by comparing the drop in pump speed (following power failure) in the TAHS run with the speed recorded in field tests. The WR^2 input value to TAHS was varied until the *speed* versus *time* curve from the computer output matched the field test data. Because pump characteristics also affect the speed-time curve, the WR^2 value had to be continually readjusted when the characteristics were changed.

For all test pumps, the motor torque ratio β_m and pump speed ratio α relationship was developed. All units approximate a quadratic startup equation of the form:

$$\beta_m = A\alpha^2 + B\alpha + C$$

Coefficients A, B, and C were determined (for each pump) by comparing the TAHS pump startup speed with the field test data. Initially, the coefficients were estimated using the pump shaft torque M_p and rotational speed n as measured in the field. If the motor torque M'_m could be measured, a direct solution would have been possible. Since a direct solution was not possible, several computer iterations were made to zero in on the correct startup equation.

RESULTS

As the characteristics were developed for each pump, the WR^2 values were revised from the "as built" values used initially. The following is a table of "as built" values provided by the pump contractor and actual computer iterated values. Significant inertia variation is noted between the WR^2 columns.

When the actual WR^2 values were used in the computer model, the TAHS results matched the field tests quite closely. Figures 12 through 15 show the drop in pump speed after power failure

Pump-ing plant	Unit No.	Rated speed, r/min	"As built" WR^2 kg·m²	"As built" WR^2 lb·ft²	Actual WR^2 kg·m²	Actual WR^2 lb·ft²
1A	6-large	1200	527	12 500	536	12 800
1A	8-small	1800	73.7	1 750	56.9	1 350
3	1	1800	10.5	250	23.6	560
6	1	900	160	3 800	177	4 200

as calculated by TAHS compared with the measured field test data. The similarity between the two curves is markedly better for all pumps than it was when using the "as built" WR^2 values in the computer model.

Since the power failure field test results were used to develop the characteristics, only a partial range of the characteristics could be calculated. The range of values is within the zone $45° \leq \theta \leq 225°$, where $\tan \theta = \alpha/q$. This zone covers pump operations of most concern. The characteristics developed for each of the four pumps are shown on figure 16. These characteristics developed for the Southern Nevada pumps can be compared to eight additional sets shown on figures 17 and 18. Until recently the three characteristics on figure 17, for specific speeds 24.6, 147, and 261 (1270, 7600, and 13 500), were the only complete sets of pump characteristics available. The five remaining sets were acquired recently to broaden the range of specific speed for which there are known characteristics. Twelve sets of characteristics are currently stored in TAHS (figs. 16-18). On figures 16-18 it should be noted that the four 90°-quadrants are approximate zones. The divisions between these zones of operation varies from one pump to the next and will not necessarily occur at defined 90° intervals of θ.

The four test pump startup relationships resulted as follows:

Pumping plant	Pump startup equations
1A - large unit	$\beta_m = -0.76\alpha^2 + 2.0\alpha + 0.60$
1A - small unit	$\beta_m = -1.70\alpha^2 + 2.0\alpha + 1.10$
3	$\beta_m = -3.10\alpha^2 + 4.2\alpha + 0.80$
6	$\beta_m = -4.00\alpha^2 + 4.0\alpha + 0.44$

The startup equations were stored in TAHS and then used to model startup situations for the four pumps. Figures 19-22 compare the predicted startup rate with that measured in the field.

DISCUSSION

An observation made in developing the pump characteristics was the close interrelation between the WR^2 value and the characteristics for each pump. In the process of computing a set of characteristics, successive iterations would not converge on the proper values if the WR^2 used in TAHS were not the actual value for the test pump. Similarly — following a power failure — the rate at which the pump rotational speed decreased was dependent upon the characteristics input (TAHS) for that pump, and this rate was used to calculate the actual WR^2 value. The WR^2 value could be estimated before developing the actual pump characteristics from speed reduction in the first few seconds after the power was cutoff, since this initial speed reduction was least affected by the characteristics. But if this initial WR^2 estimate was too far off, it would lead to divergence in the attempt to converge on the characteristic curve. Therefore, the WR^2 value for each pump had to be monitored closely to assure that successive iterations of the characteristics would converge correctly.

As noted previously (see RESULTS — table) the actual WR^2 values differed from the values provided by the contractors. The unit tested at Pumping Plant 3 evidenced an extreme differential from the "as built" value claimed by the contractor. The actual value is more than twice the "as built" value, which is conservative for providing greater surge protection. Unit 8, at Pumping Plant 1A, was the only test pump for which the actual WR^2 was less than the required WR^2 in the specifications. Whereas the "as built" values provided by the pump contractors satisfied the specifications requirements, it appears that the contractors' values were not always accurate.

The most significant conclusion which can be drawn from this study concerns the relationship between the pump specific speed and the characteristics. The four sets of characteristics that were developed for the Southern Nevada system pumps showed much less of a determinate pattern than had been expected. Many have assumed that pump characteristics vary proportionate to specific speed; however, it is indicative that this assumption is incorrect. The characteristic curves on figures 16-18 show very little pattern related to their corresponding specific speed.

Some generalizations can be made about pump performance based on specific speed. Pump shutoff head, which occurs at $\theta = 90$ degrees on the head characteristic curves, generally increases in magnitude as the specific speed increases. There are only a few minor exceptions to this among the pumps under study. There also appears to be some pattern related to the type of flow through pumps. Although most pumps in this investigation are radial-flow pumps ($n_S < 70$ (r/min) $(m^3/s)^{0.5}/m^{-0.75}$), pumps in the mixed-flow ($70 < n_S < 145$) and axial-flow regimes ($145 < n_S$) exhibit more consistent properties. Mixed- and axial-flow pumps show much more of a pattern of characteristics with specific speed than do radial-flow pumps. In the higher specific speed regions, the amount of surge protection supplied by a pumping unit appears to vary proportionally to n_s. However, the data accumulated in the radial-flow regime indicate no correlation between characteristics and specific speed.

After developing characteristics for the four Southern Nevada system pumps, they were included with eight other sets of characteristics stored in TAHS. A computerized problem (TAHS) was developed similar to the Southern Nevada system field tests for modeling the power failure. The problem was then run 12 times, each time using a different set of pump characteristics while keeping all other test parameters constant. The results of the problem are shown on figure 23. The figure compares the maximum and minimum pressure heads developed at the discharge side of the pump for each set of characteristics programmed. It should be noted that the static head at that location in the system was 76 meters (250 ft), so the tests in the mixed- and axial-flow regimes resulted in zero head rise. For specific speeds in the radial-flow regime there was no apparent relation between the pump specific speed and the pressures developed in this test.

The Southern Nevada system pumps are not the first which have exhibited discrepancies in the relationship of characteristics to specific speed. Recently there have been other situations at Bureau of Reclamation projects where inconsistencies have developed regarding the specific speed of pumping machinery. In one case three different manufacturers submitted data on pumps to be supplied for a future pumping plant. The pumps from all three manufacturers had the same specific speed, yet all three had markedly different characteristics. Another case involved pump-turbine units installed at a pumped-storage powerplant under two separate contracts. The units had identical specific speeds, yet the pump characteristics varied significantly between the two contracts. Comparing these pump-turbine units with regular pumps is valid, since this comparison is based on field tests of the pump-turbines operating in the pump mode. It is apparent that impeller design can have a significant effect on pump performance—independent of specific speed. The authors believe that pumps with the same specific speeds can have a large variance in their respective characteristic values.

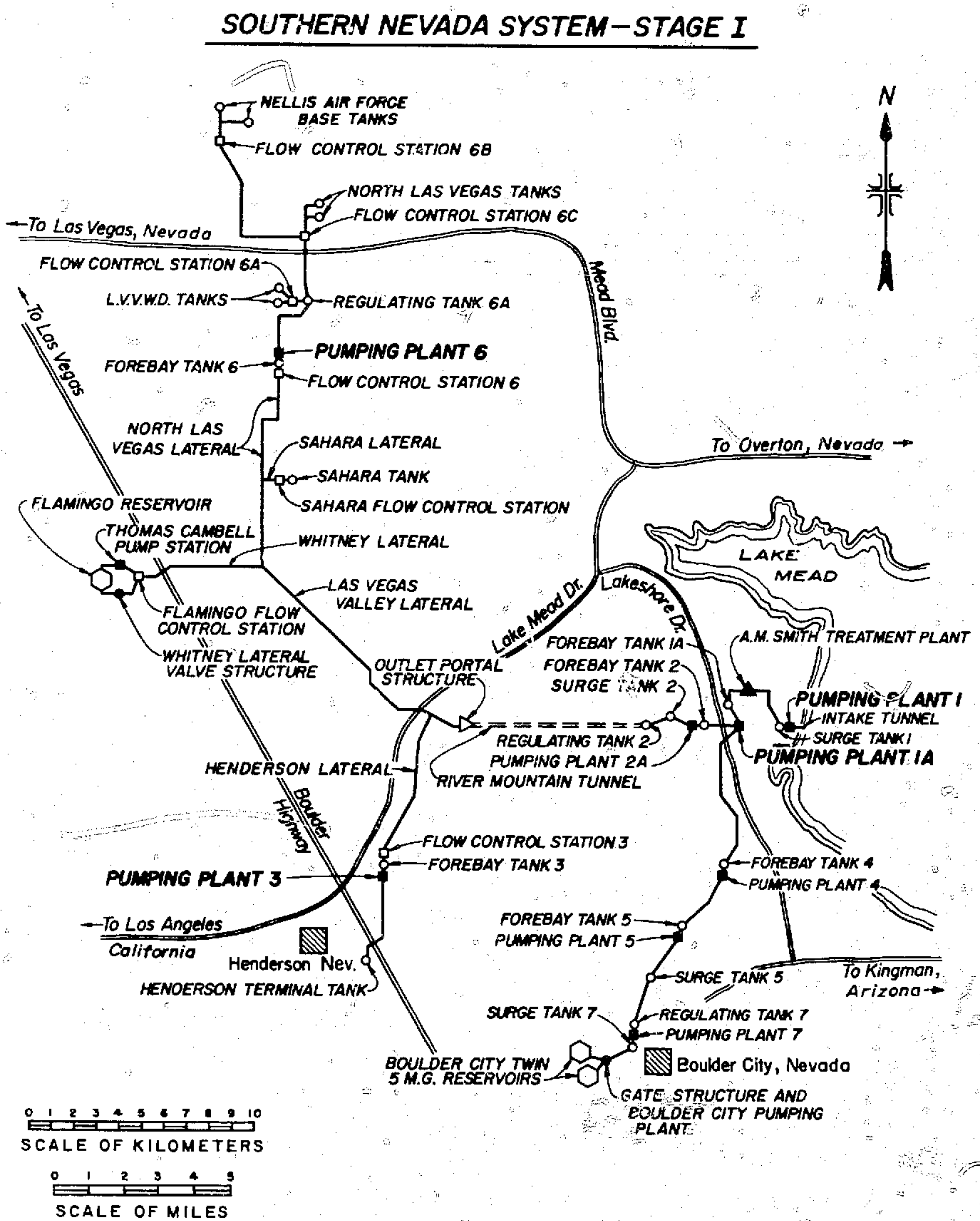


Figure 1. — Principal features in the Southern Nevada Water Project — Stage I.

Figure 2. — Forebay tank at Pumping Plant 3. Photo C801-D-78992

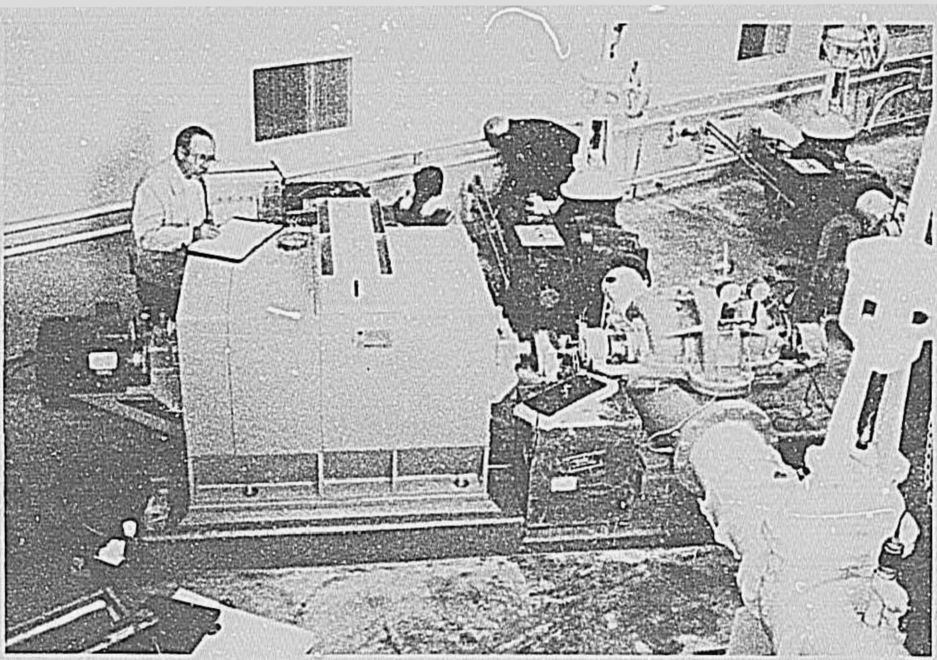


Figure 3. — Pump, motor, and valves at Pumping Plant 3. Photo C801-D-78993

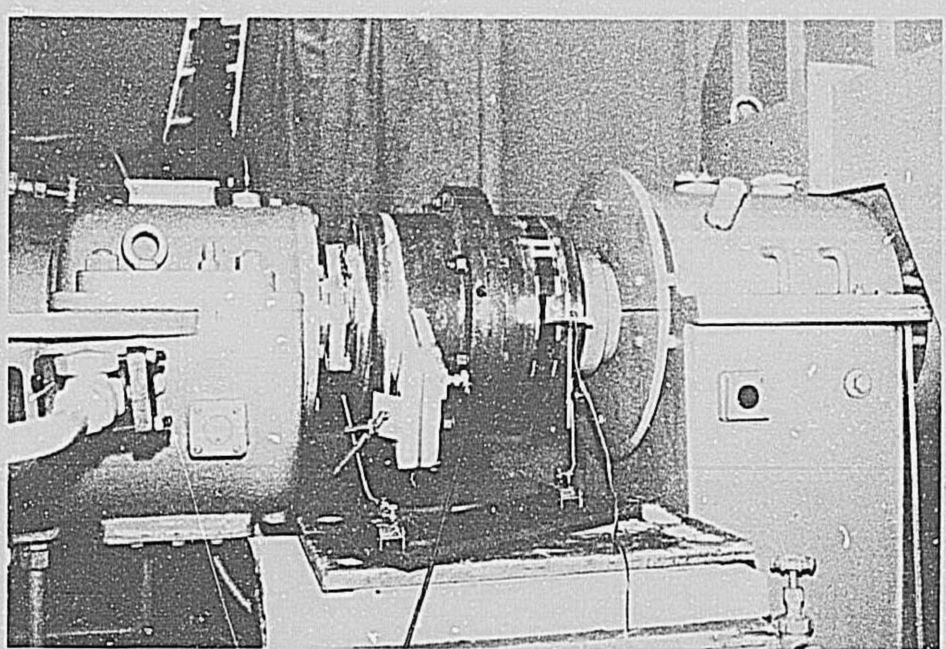


Figure 4. — Speed and torque measuring devices on the pump shaft at Pumping Plant 1A.
Photo C801-D-78995

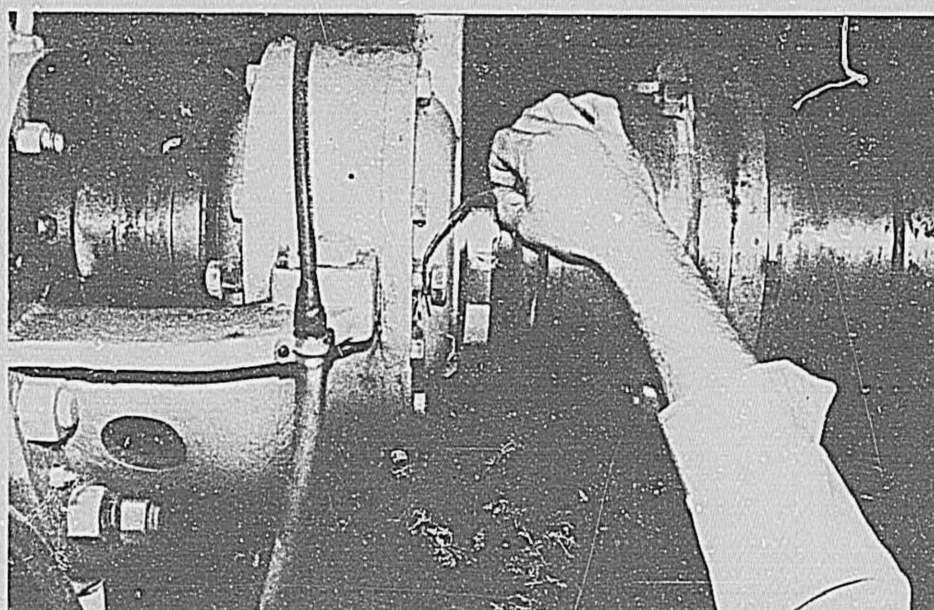


Figure 5. — Strain gages being placed on the pump shaft for obtaining torque measurements.
Photo C801-D-78994

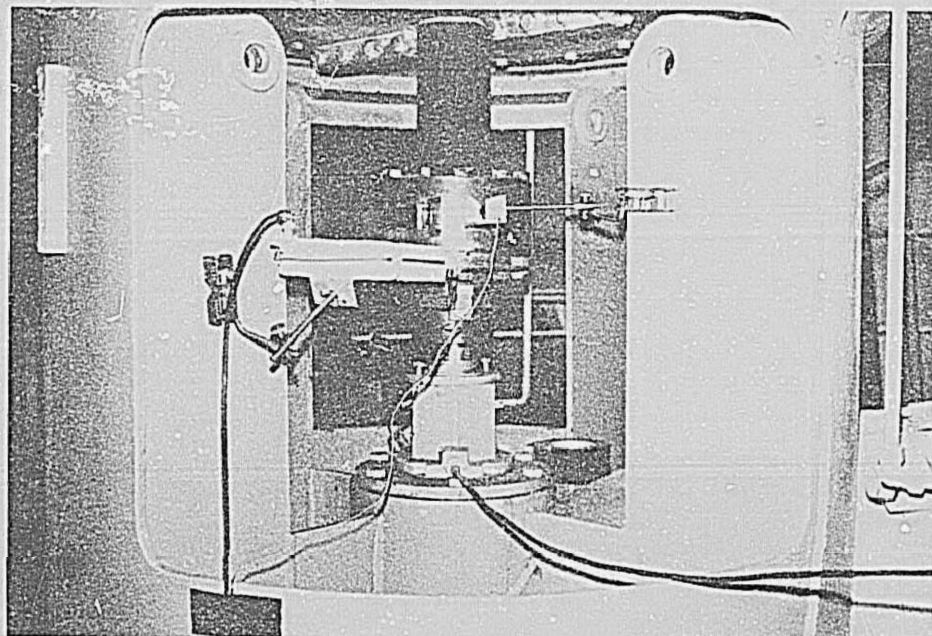


Figure 6. — Speed and torque measurement apparatus for the vertical unit at Pumping Plant 1 Photo C801-D-78997

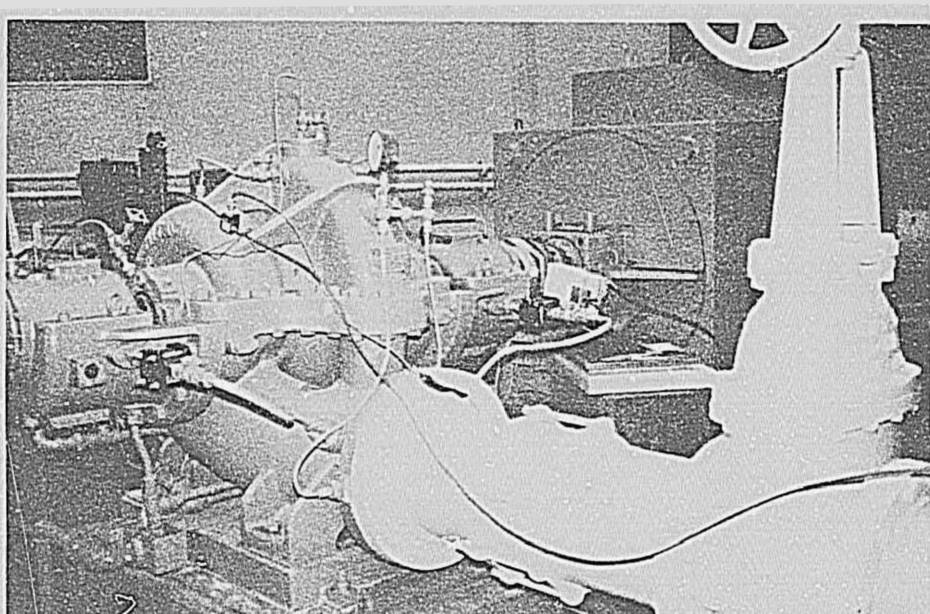


Figure 7. — Pressure transducers attached to each side of the pump for differential pressure head measurement. Photo C801-D-78996

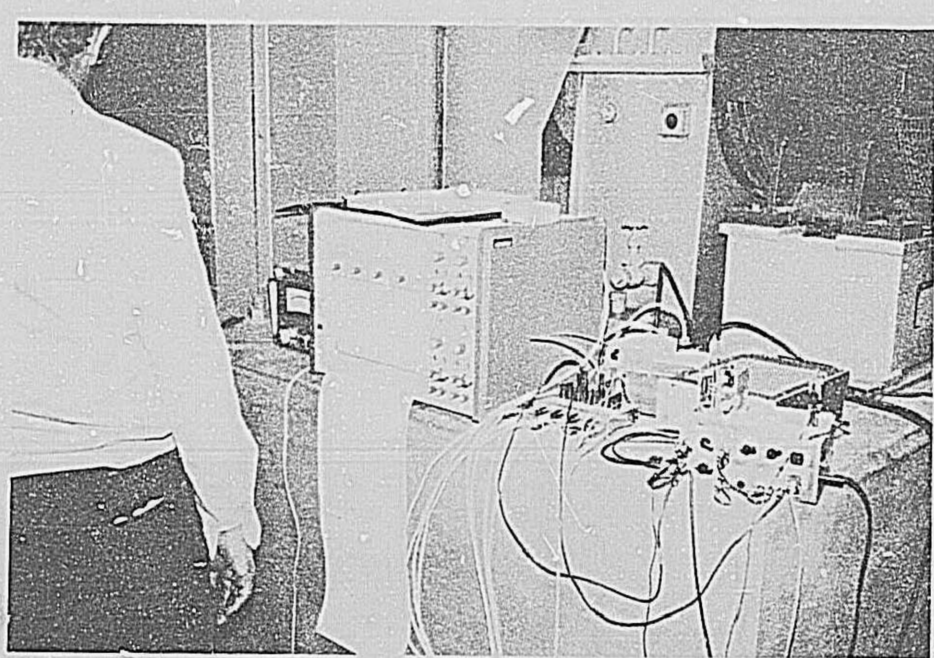


Figure 8. — Four channel oscillograph in operation. Photo C801-D-78998

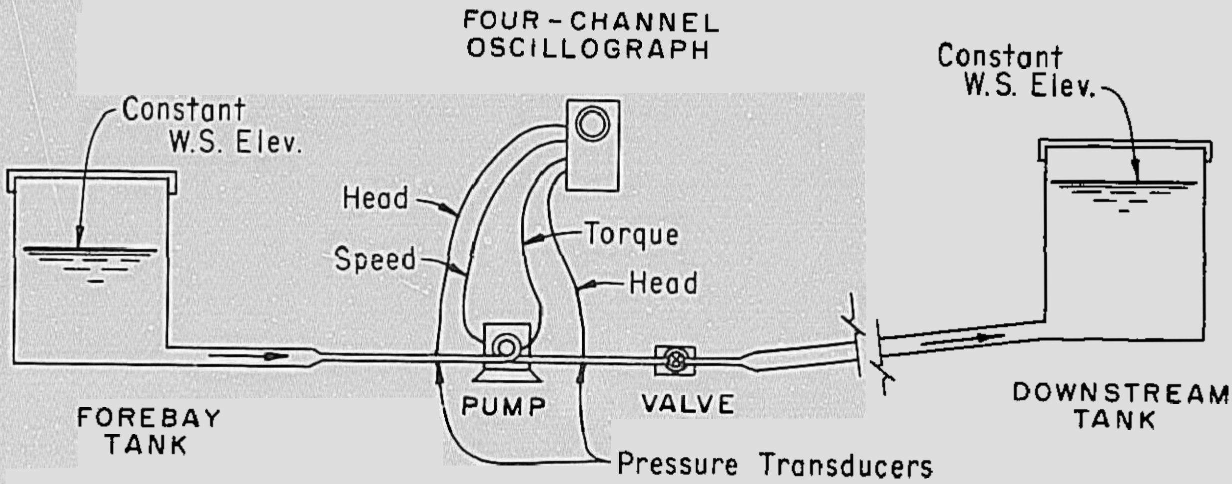


Figure 9. — Typical test setup.

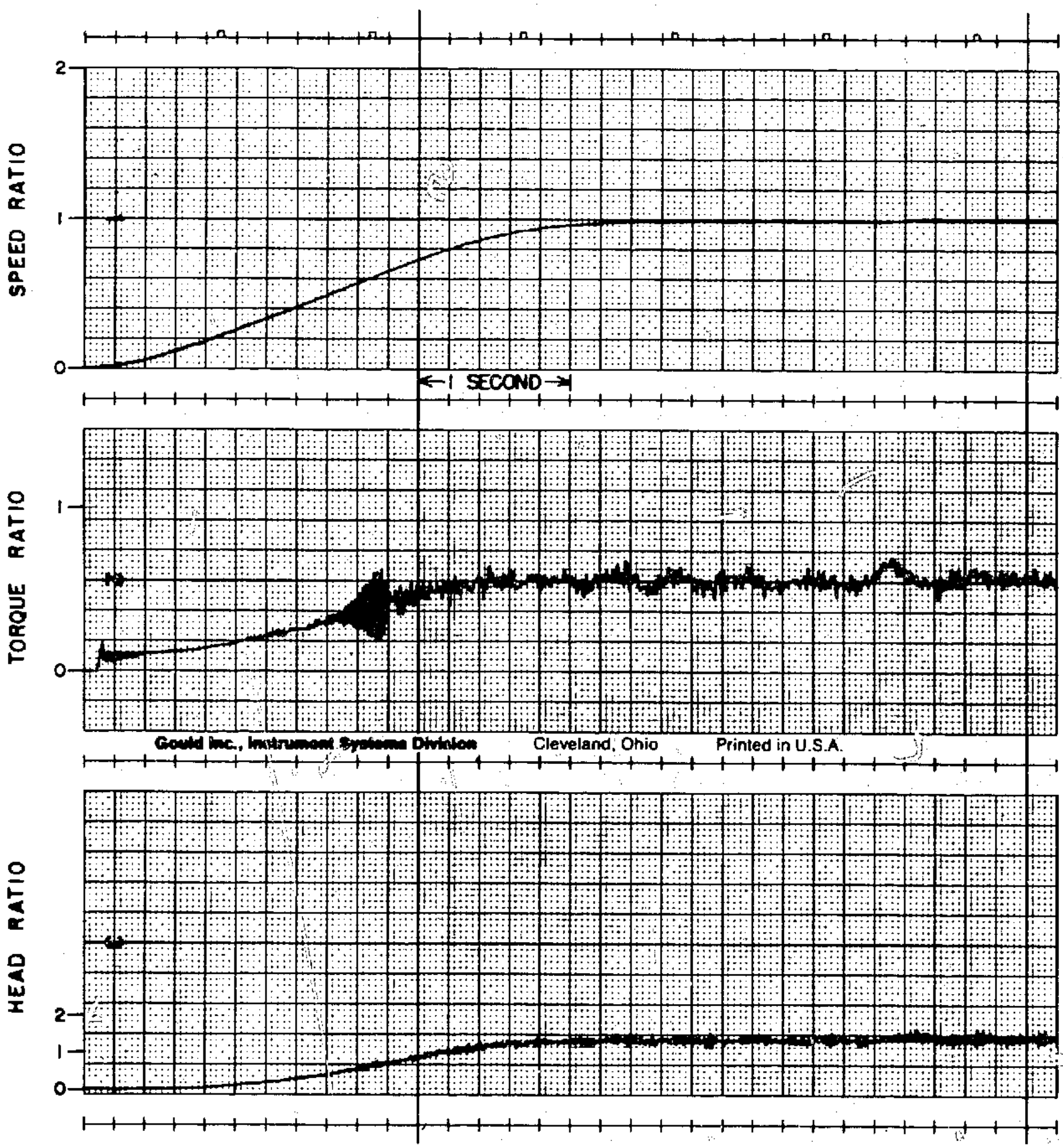


Figure 10. — Oscillograph record of pump startup.

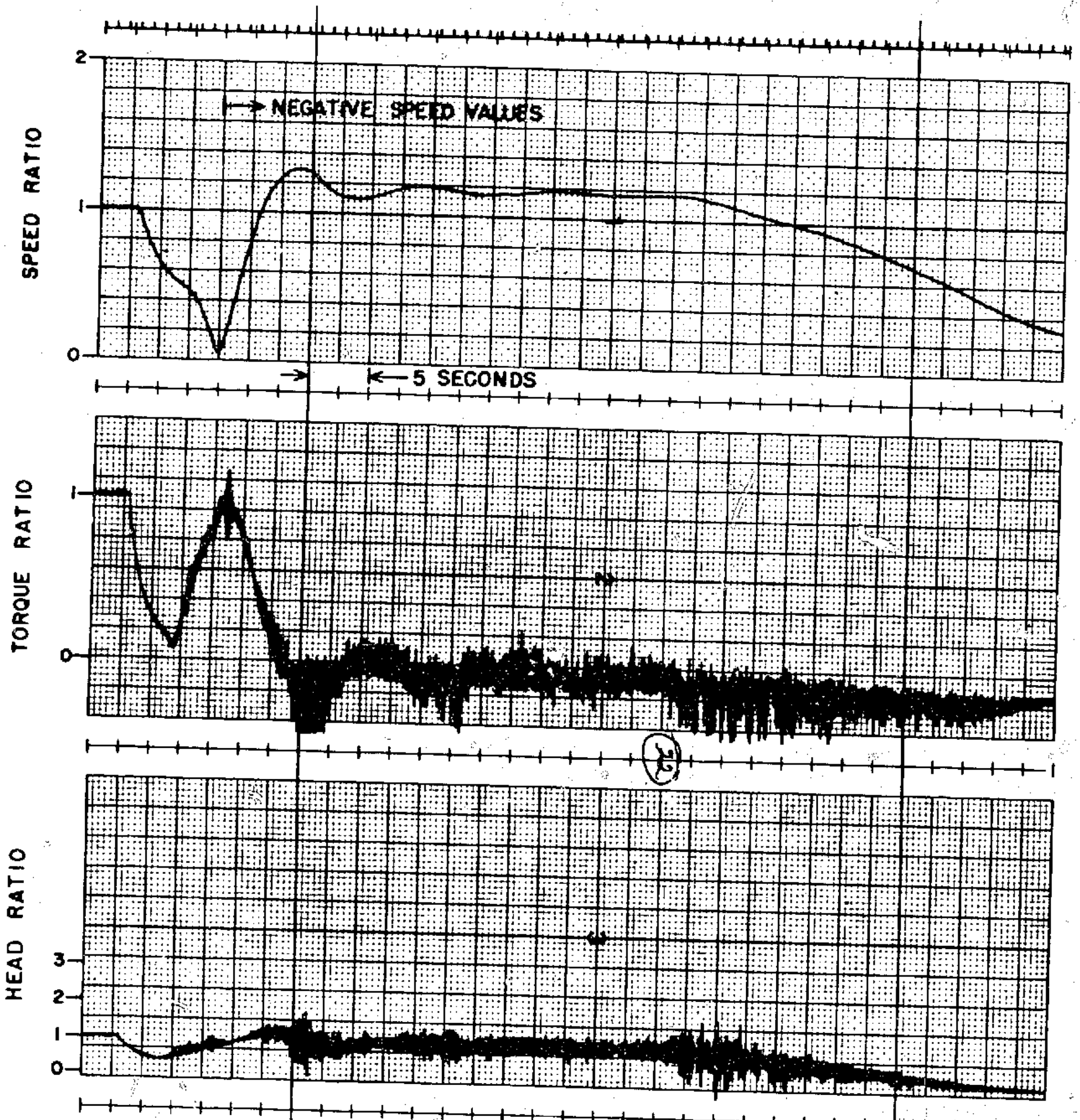


Figure 11. — Oscillograph record of power failure at pump motor.

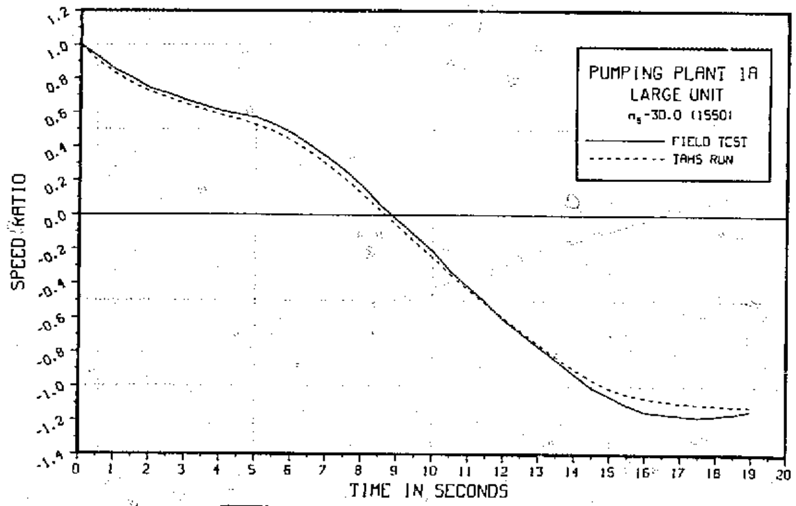


Figure 12. — Pump speed following power failure at Pumping Plant 1A — large unit.

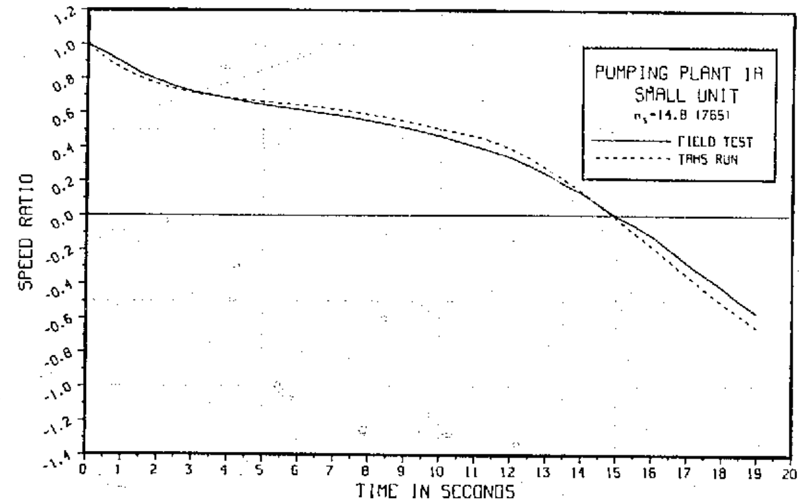


Figure 13. — Pump speed folowing power failure at Pumping Plant 1A — small unit.

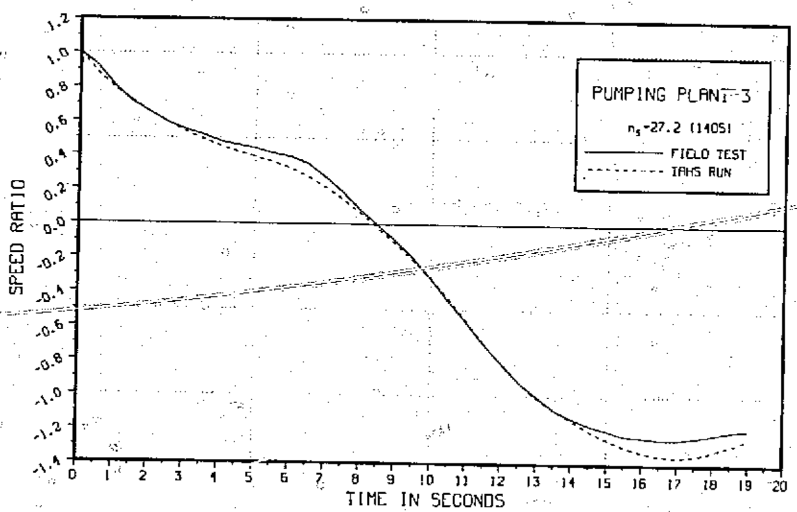


Figure 14. — Pump speed following power failure at Pumping Plant 3.

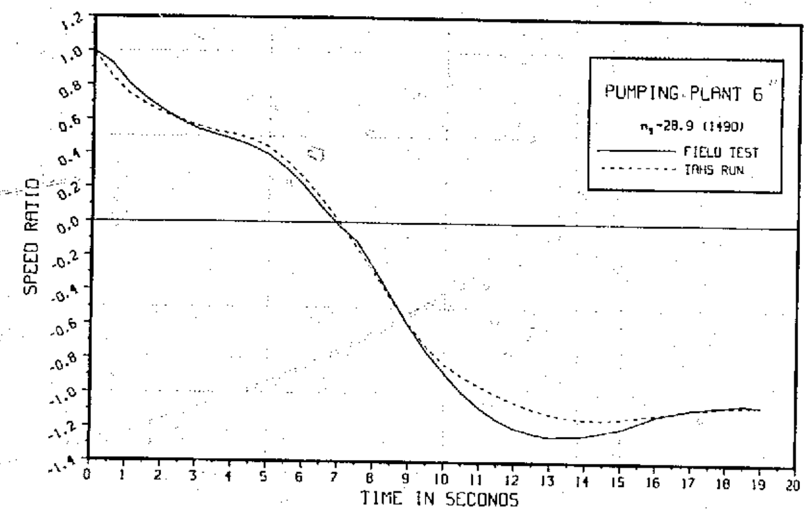
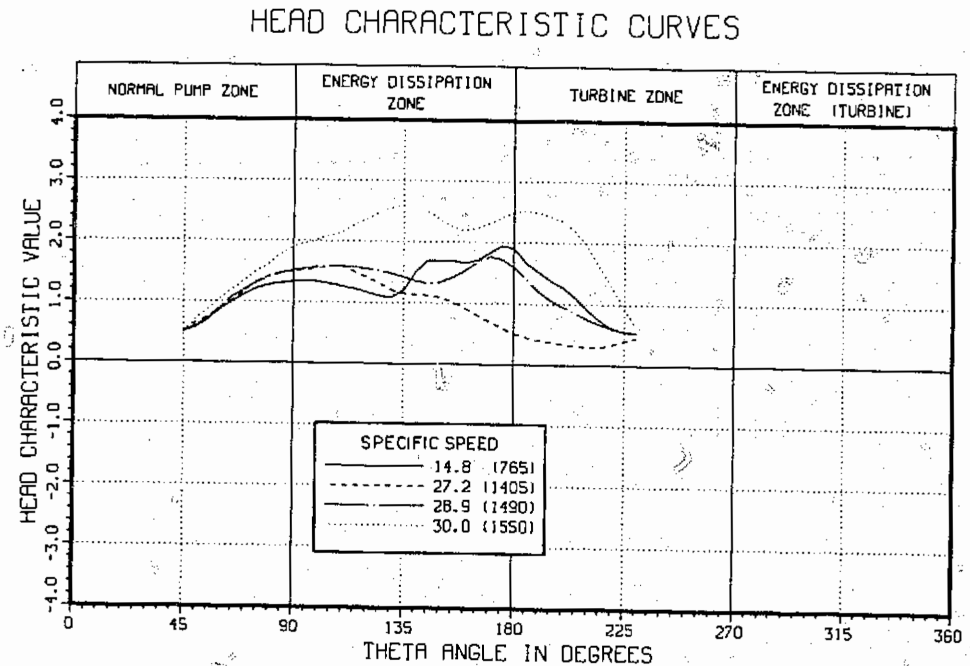


Figure 15. — Pump speed following power failure at Pumping Plant 6.



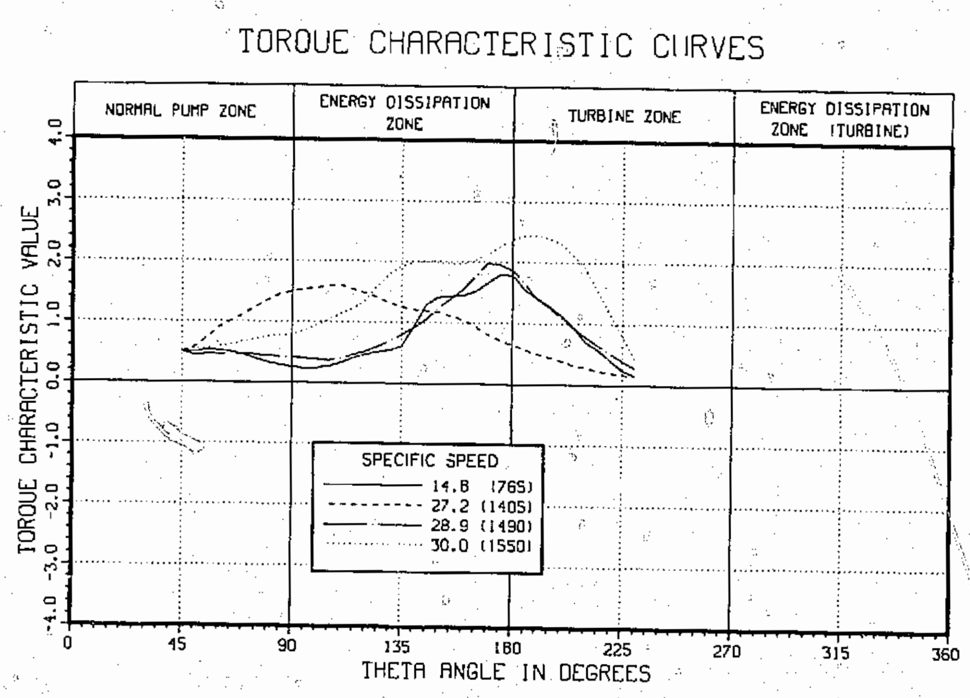
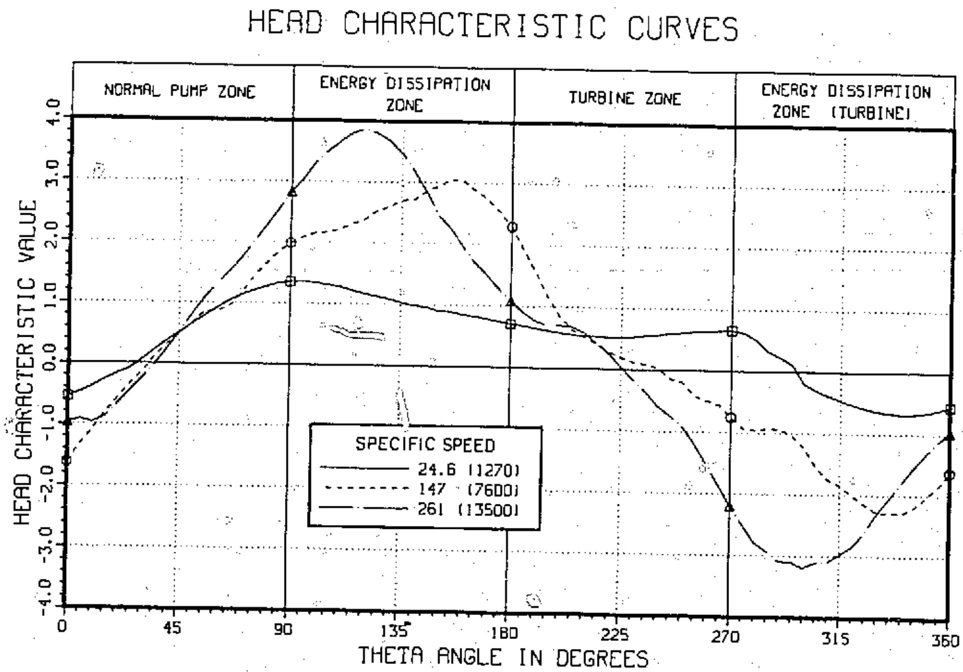


Figure 16. — Southern Nevada pump characteristics. Head characteristic value = $h/(\alpha^2 + q^2)$, torque characteristic value = $\beta/(\alpha^2 + q^2)$.



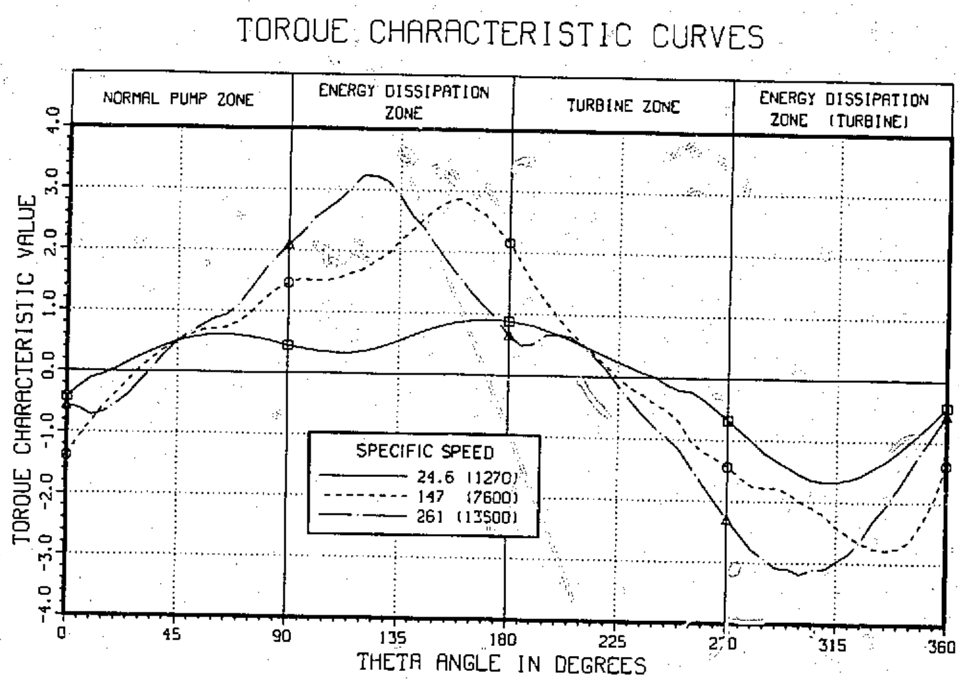
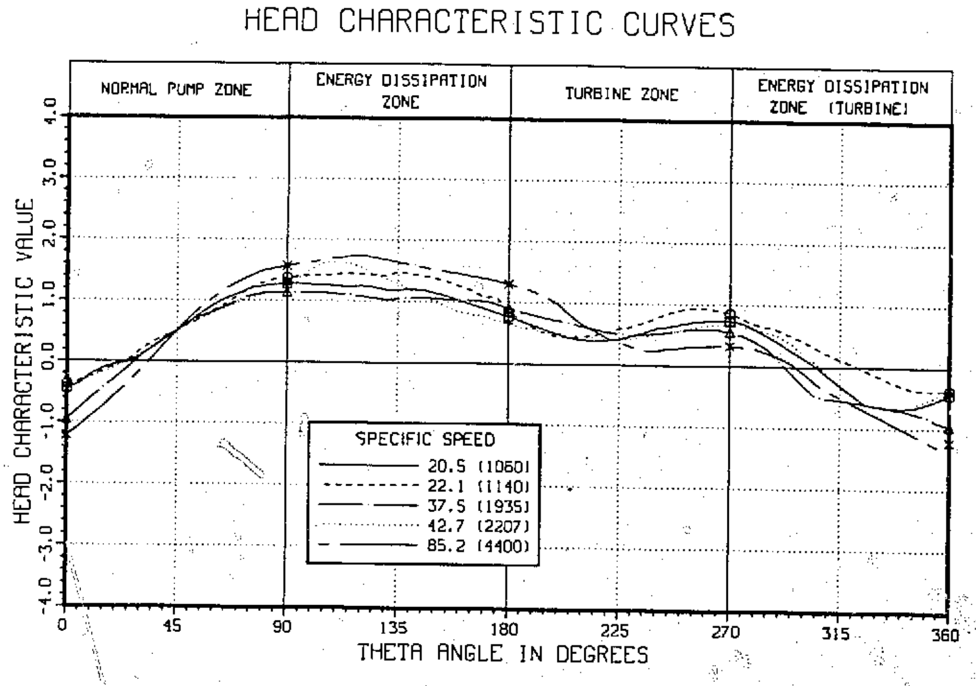


Figure 17. — Three complete sets of pump characteristics.



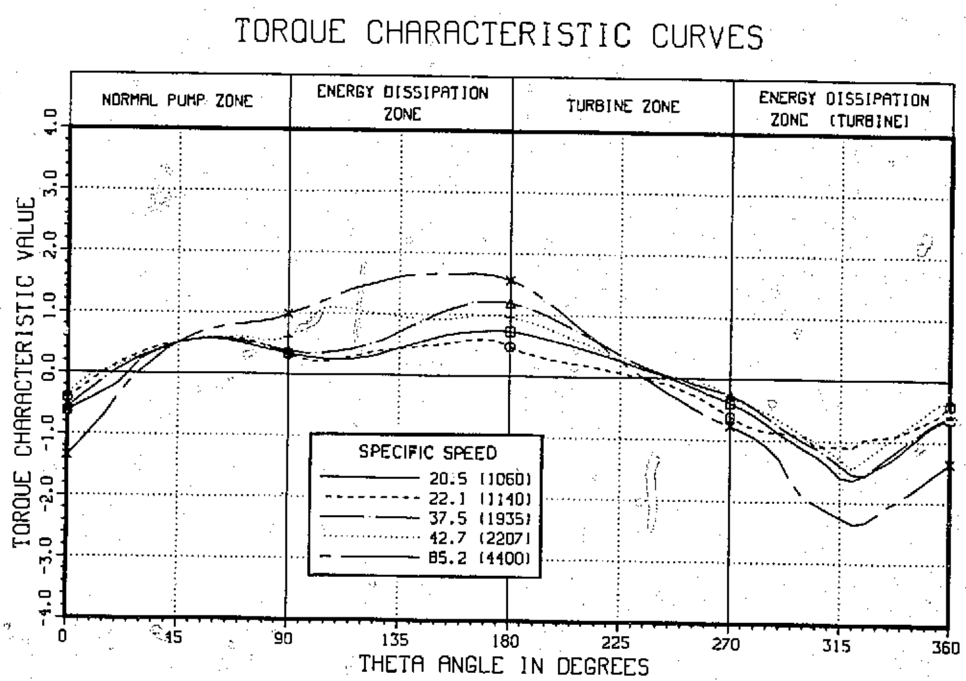


Figure 18. — Five complete sets of pump characteristics.

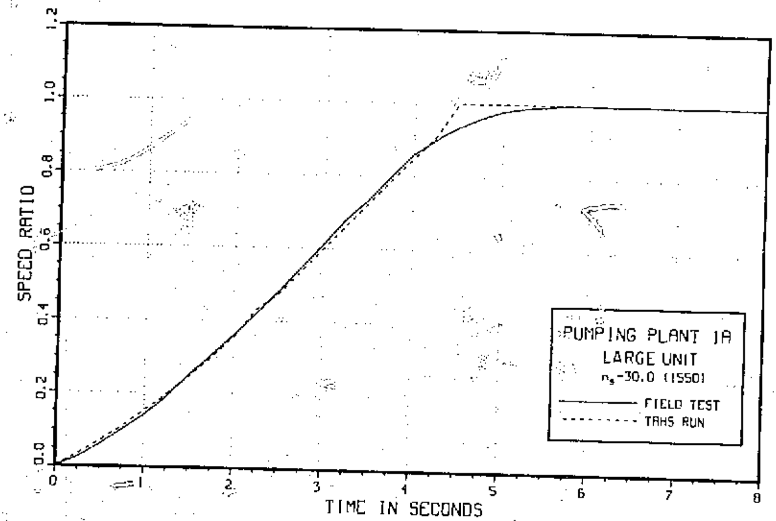


Figure 19. — Pump speed following startup at Pumping Plant 1A — large unit.

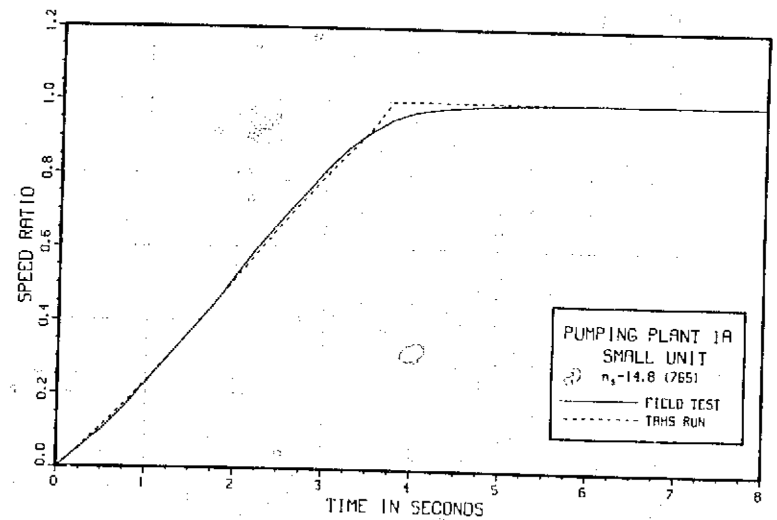


Figure 20. — Pump speed following startup at Pumping Plant 1A — small unit.

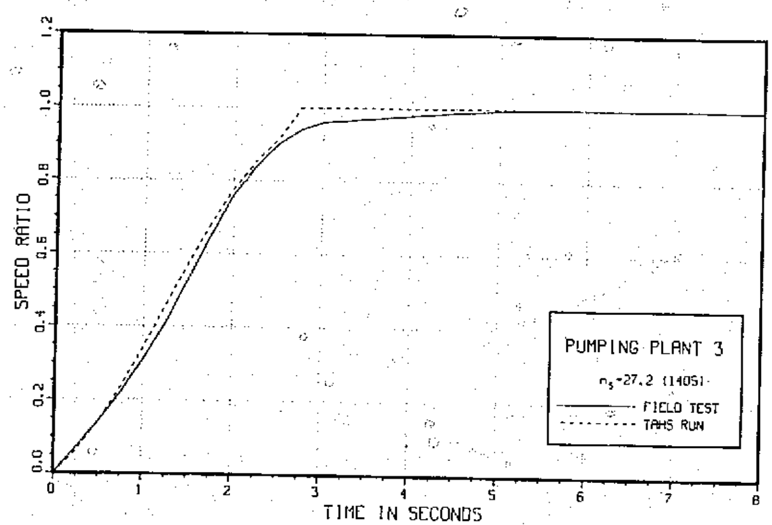


Figure 21. — Pump speed following startup at Pumping Plant 3.

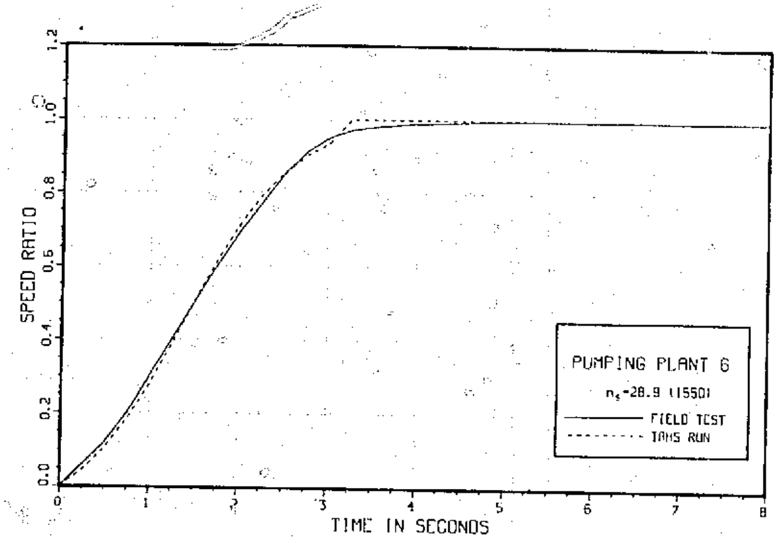


Figure 22. — Pump speed following startup at Pumping Plant 6.

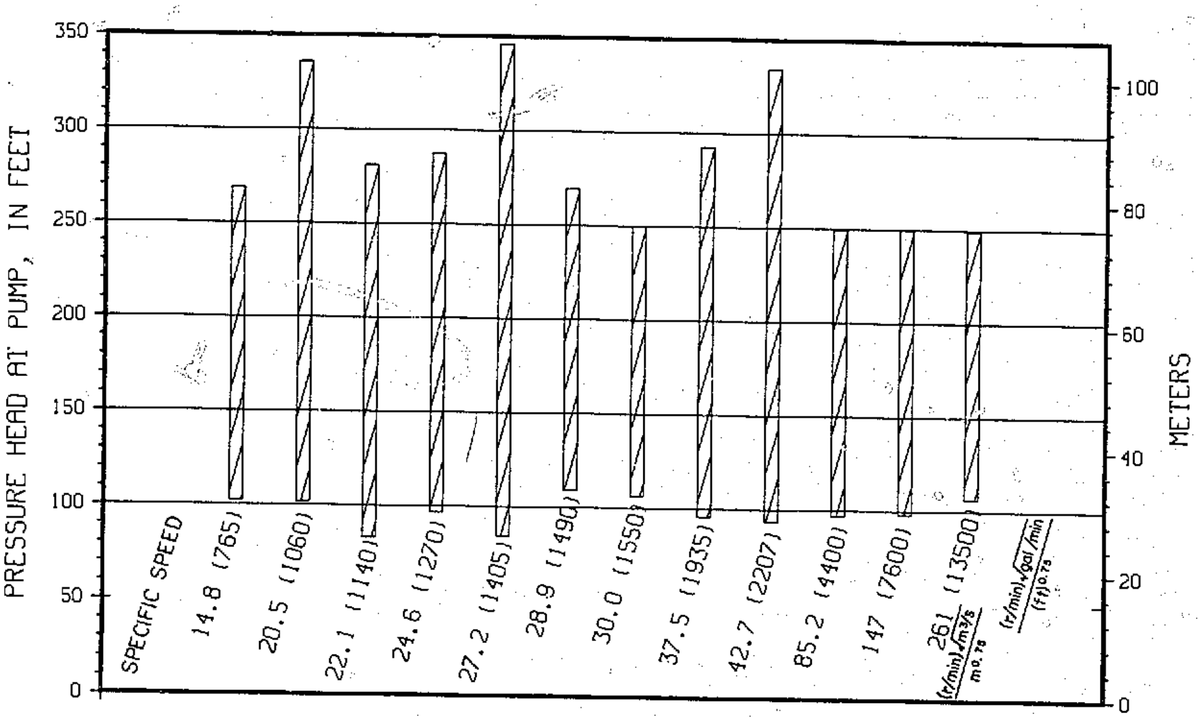


Figure 23. — Comparison of different pump characteristics. Each bar represents the head range values produced using the corresponding set of characteristics.

APPENDIX

Motor Torque Derivation

$$M_m = M_p + I\,\frac{d\omega}{dt}$$

$$M_m = M_p + \frac{WR^2}{g}\,\frac{d\omega}{dt}$$

$$\frac{M_{m_1} + M_{m_2}}{2} = \frac{M_{p_1} + M_{p_2}}{2}\left(\frac{WR^2}{g}\right)\left(\frac{\omega_2 - \omega_1}{\Delta t}\right)$$

$$M_{m_1} + M_{m_2} = M_{p_1} + M_{p_2} + \left(\frac{2WR^2}{g}\right)\left(\frac{2\pi}{60}\right)\left(\frac{n_2 - n_1}{\Delta t}\right)$$

$$\frac{M_{m_1} + M_{m_2}}{M_r} = \frac{M_{p_1} + M_{p_2}}{M_r} + \left(\frac{\pi WR^2}{15gM_r}\right)\left(\frac{n_2 - n_1}{\Delta t}\right)\left(\frac{n_r}{n_r}\right)$$

$$\beta_{m_1} + \beta_{m_2} = \beta_{p_1} + \beta_{p_2} + \left(\frac{\pi WR^2 n_r}{15gM_r}\right)\left(\frac{a_2 - a_1}{\Delta t}\right)$$

let $K_1 = \dfrac{15gM_r}{\pi WR^2 n_r}$

$\beta_{m_1} + \beta_{m_2} = \beta_{p_1} + \beta_{p_2} + \dfrac{a_2 - a_1}{K_1\,\Delta t}$, this is the motor torque as a function of pump shaft torque and speed for obtaining the pump start-up equations.

The subscripts 1 and 2 refer to times t_1 and t_2; i.e., $t_2 - t_1 = \Delta t$.

Given: β_{p_1}, β_{p_2}, a_1, a_2, K_1, Δt, and β_{m_1}, a value can be calculated for β_{m_2}.

At the time zero, β_{m_1} must be estimated.

Parameters

- g = Gravitational acceleration
- I = Mass moment of inertia
- M_m = Motor torque
- M_p = Pump shaft torque
- M_r = Rated torque
- n = Rotational speed
- n_r = Rated speed
- t = Time
- WR^2 = Product of weight of revolving parts and the square of the radius of gyration
- a = Speed ratio
- β = Torque ratio
- π = 3.141 59 . . .
- ω = Angular velocity

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