THE UNIVERSITY OF MICHIGAN

INDUSTRY PROGRAM OF THE COLLEGE OF ENGINEERING

WATERHAMMER ANALYSIS OF PIPELINES

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November, 1963

IP-642

ACKNOWLEDGEMENTS

This study was supported by a National Science Foundation Grant (NSF-GP340). Facilities of the computing centers at the University of Colorado and the University of Michigan were utilized for the computational work.

Mr. John Parmakian and Mr. Thomas Logan made many helpful suggestions during preparation of the paper. Mr. Thomas Propson aided in drawing the figures.

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NOMENCLATURE

a	speed of pressure pulse wave in pipe
A	point of previous computation of head and velocity
B	point of previous computation of head and velocity
C	point of previous computation of head and velocity
D	diameter of discharge pipe
DELH	steady state head loss across open discharge valve
DS	diameter of suction pipe
f	Darcy-Weisbach friction factor
g	gravity
h	dimensionless head; head ratio for a pump
h_{pump}	dimensionless head produced by pump
h_{vo}	head loss across the valve of an operating pump/rated head
h_{vf}	head loss across the valve of a failing pump/rated head
H	elevation of hydraulic gradeline
I	moment of inertia of pump, motor and entrained water
K_1	factor involving moment of inertia
L	length of discharge line
LS	length of suction line
M	number of pumps
M_f	number of failing pumps
M_o	number of operating pumps
N	pump speed
N_s	specific speed (gpm units)

P	computation point; upstream end of discharge line
Q	discharge
R	point on characteristic curve
S	point on characteristic curve; downstream section of suction line
t	time
tdh_o	total dynamic head of an operating pump/rated head
tdh_f	total dynamic head of a failing pump/rated head
T	torque applied to pump shaft
v	dimensionless velocity
V	velocity at a point
VA	velocity at upstream end of discharge pipe
x	distance along pipe
α	speed ratio of pump (rpm/rated rpm)
β	torque ratio of pump (torque/rated torque)
θ	$\Delta t/\Delta x$
τ	dimensionless valve opening
ϕ	angle pipeline makes with horizontal
ω	angular velocity of pump

ABSTRACT

Waterhammer transients caused by failure of power to pumping stations are studied by use of the high-speed digital computer. Dimensionless-homologous complete pump characteristics are presented for three specific speeds and are utilized in the solutions because of their convenience for storage in the computer. The method of characteristics equations for waterhammer are employed, including friction loss.

Comparisons are made between gravity loading and friction-loss loading on centrifugal pumps during power failure, for failure of one or more of a set of pumps operating in parallel to supply a pipeline. For gravity loading, high pressure transients must be provided for; for friction loading, low pressures with possible column separation must be avoided. Effects of changes in diameter, moment of inertia of rotating parts, and length of pipeline are studied. Special valve stroking methods are employed to control transients for gravity loading.

I INTRODUCTION

An analysis of the hydraulic transients in pipelines caused by failure of power to pumps is undertaken in this paper, using dimensionless-homologous pump data and the method of characteristics equations for waterhammer. The equations are solved by means of a high-speed digital computer. To illustrate the analysis, a pumping station is located in a pipeline between two reservoirs. In the station are a number of identical pumps, in parallel. Any number of the pumps may suddenly lose their power. A valve may be located at each pump discharge, and may be closed in an arbitrary manner (or as a check valve) when triggered by the loss of power. The effects of large friction heads as compared with large gravity heads are discussed, along with the effects of diameter change and valve closure timing.

When power is lost to a pump lifting water to a reservoir the following events take place (in the absence of a discharge valve): the flow rapidly diminishes to zero and then reverses; negative pressure waves are propagated downstream from the pump and positive pressure waves are propagated upstream through the suction pipe. The pump rapidly loses its forward rotation and reverses shortly after reversal of flow. As the pump increases in speed in the reverse direction it causes greater resistance to flow which produces high pressures in the discharge line.

When the load on the pumping system is primarily due to fluid friction, as in the case of a long discharge line, vapor pressure and column separation may occur in the discharge line. Problems of flow reversal through the pumps are not serious for this case.

The proper operation of a valve at the pump discharge can greatly reduce the high pressures caused by reverse flow; its operation, however, is not effective in alleviating the low pressures in a long pipeline.

Pump characteristics are first discussed, followed by a summary of the computer waterhammer calculations. Pump speed changes are next discussed, and then boundary conditions are developed for failure of one or more pumps. The resulting computer program is then applied to situations where several of the parameters are varied. The paper concludes with a discussion of valving.

II DIMENSIONLESS-HOMOLOGOUS PUMP CHARACTERISTICS

Pump manufactures usually supply data on performance of their units for the zone of normal operation only; data on the zones of energy dissipation and turbine operation must usually be estimated from the meagre data available in the literature.[1,2] The zone of energy dissipation occurs with forward rotation and reversed flow, and the zone of turbine operation refers to the case with reverse rotation and reverse flow. Benjamin Donsky[2] has presented data taken by Professor Aladar Hollander at California Institute of Technology for the three specific speeds (g.p.m. units) 1800, 7600, and 13,500, representative of centrifugal, mixed, and axial flow pumps. He presents the data on a dimensionless head-discharge plot for constant values of speed and torque ratios. These curves have been developed by the principles of homologous units and from these curves the data may be reduced to a few curves of dimensionless-homologous parameters for convenient storage and interpolation for computing.

Dimensionless-homologous ratios of use are:

For head:

$$\frac{h}{\alpha^2} \text{ vs } \frac{v}{\alpha} \quad \text{or} \quad \frac{h}{v^2} \text{ vs } \frac{\alpha}{v}$$

For torque:

$$\frac{\beta}{\alpha^2} \text{ vs } \frac{v}{\alpha} \quad \text{or} \quad \frac{\beta}{v^2} \text{ vs } \frac{\alpha}{v}$$

in which h is the head (total dynamic head) divided by the rated head,

β is the torque divided by the rated torque, α is the pump speed divided by the rated speed, and v is the pump discharge divided by the rated discharge.

Since both α and v pass through zero during the course of a pump reversal, it is necessary to use both h/α^2 vs v/α and h/v^2 vs α/v to avoid having the curves go to infinity. Figure 1 shows the torque ratios for the Hollander data as taken from Donsky's curves. For each of the three specific speeds 6 curve segments, each starting at the ordinate axis, are specified as follows: Three letters and a number are used; first, B or H to designate torque or head ratio, second A or V to designate division by α^2 or v^2 ; third N, D, or T to indicate normal, energy dissipation, or turbine operation zone; and fourth 1, 2, or 3 to designate N_s = 1800, 7600, or 13,500. Figure 2 shows the corresponding head ratios. Hence the six designations of torque ratios are: BAN, BVN, BAD, BVD, BAT, and BVT.

They are stored in the computer by listing the values of points on the curves for equal values of $\Delta(v/\alpha)$ or $\Delta(\alpha/v)$, whichever is appropriate, and the value of $\Delta(v/\alpha)$ or $\Delta(\alpha/v)$. Table I lists the head and torque ratios needed to portray the pump characteristics for failure of power to pumps. When data on specific pumps are given by the manufacturer for the normal zone, they should be used; the data for the energy dissipation and turbine operation zone may be estimated from Table I. Fortunately the exact shape of the curves in the energy dissipation and turbine operation zones are not critical, and become of little significance when optimum valve closure programming is used.

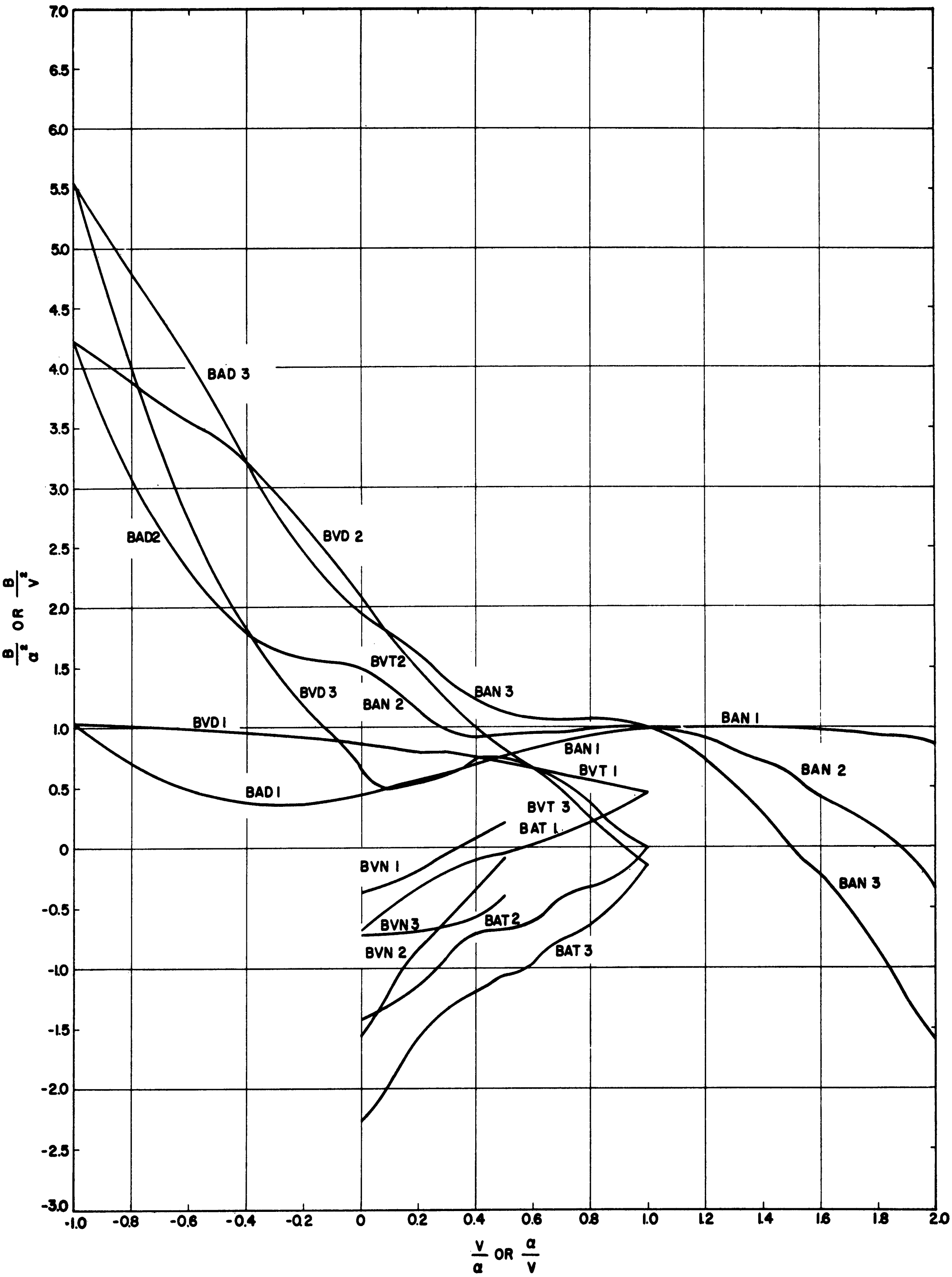


Figure 1. Dimensionless-Homologous Torque Data for the Three Specific Speeds 1800, 7600, and 13,500.

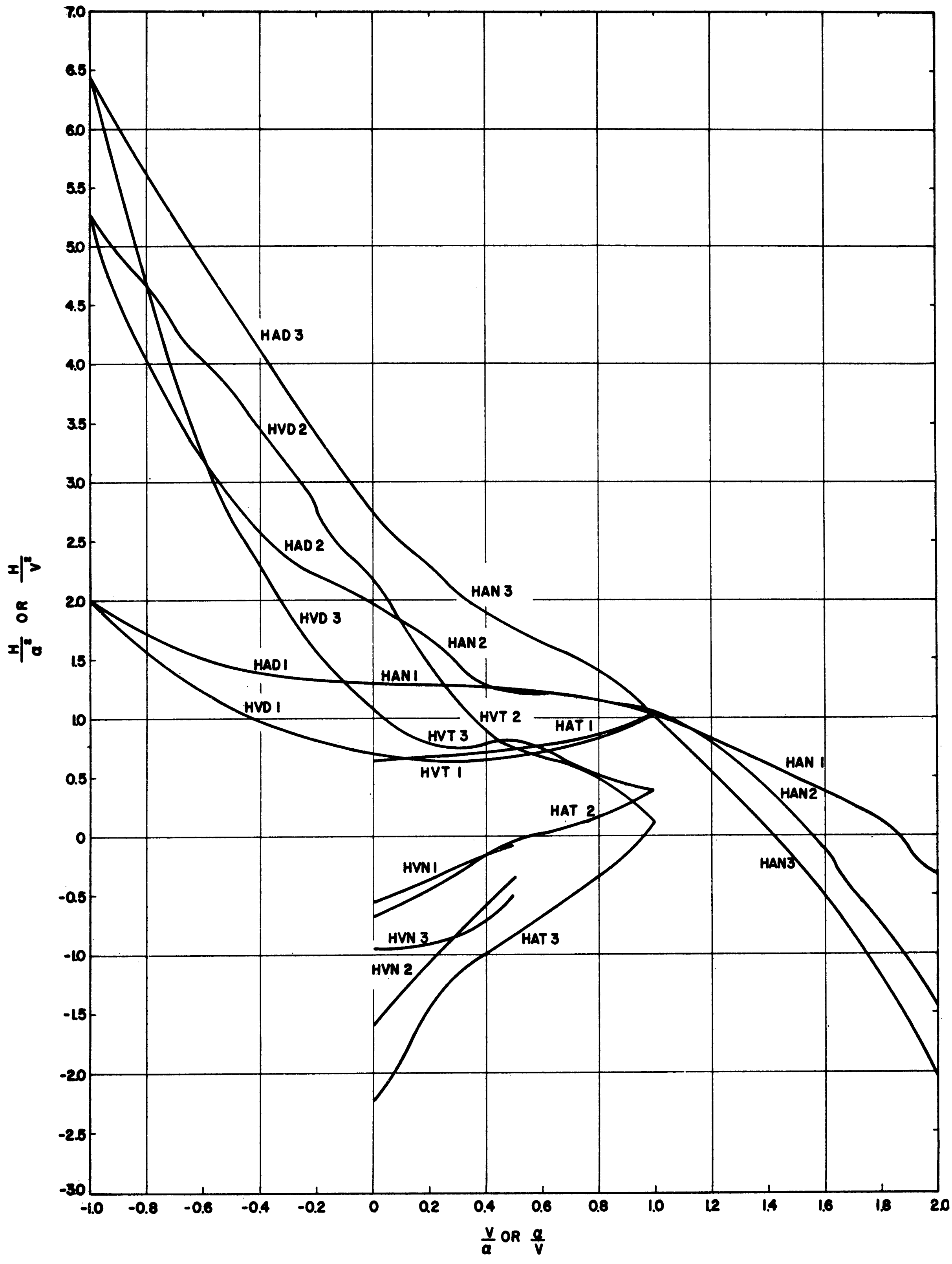


Figure 2. Dimensionless-Homologous Head Data for the Three Specific Speeds 1800, 7600, and 13,500.

TABLE I

HEAD AND TORQUE DATA FOR THREE SPECIFIC SPEEDS

($\Delta \frac{v}{\alpha} = \Delta \frac{\alpha}{v} = .1$ for N and T zones; -.1 for D zone)

NS = 1800 (gpm units)

	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21
HVN1	-.556	-.472	-.376	-.270	-.167	-.080															
HAN1	1.288	1.287	1.284	1.272	1.256	1.236	1.206	1.166	1.118	1.061	1.000	.913	.812	.707	.599	.489	.381	.271	.124	-.133	-.320
HAD1	1.288	1.291	1.312	1.347	1.382	1.431	1.500	1.600	1.715	1.844	1.992										
HVN1	.692	.742	.807	.883	.975	1.086	1.212	1.371	1.542	1.755	1.992										
HVT1	.692	.656	.631	.631	.641	.663	.700	.761	.834	.910	1.011										
HAT1	.634	.652	.668	.684	.705	.732	.764	.806	.861	.927	1.011										
BVN1	-.372	-.292	-.192	-.049	.075	.215															
BAN1	.450	.504	.567	.633	.707	.772	.833	.886	.931	.969	1.000	1.008	1.008	1.006	.996	.989	.981	963	.931	.927	.860
BAD1	.450	.393	.372	.367	.381	.419	.484	.582	.700	.850	1.040										
BVD1	.865	.895	.915	.935	.961	.978	.990	.999	1.015	1.024	1.040										
BVT1	.865	.831	.800	.800	.750	.700	.650	.600	.558	.508	.455										
BAT1	-.684	-.499	-.332	-.196	-.098	-.042	.023	.110	.200	.316	.455										

SPECIFIC SPEED = 7600

	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21
HVN2	-1.600	-1.330	-1.068	-.828	-.584	-.360															
HAN2	1.964	1.821	1.660	1.458	1.280	1.220	1.206	1.176	1.120	1.058	1.000	.891	.735	.530	.324	.104	-.120	-.460	-.760	-1.080	-1.440
HAD2	1.964	2.089	2.206	2.357	2.574	2.859	3.187	3.587	4.018	4.575	5.265										
HVD2	2.165	2.469	2.783	3.133	3.437	3.770	4.025	4.300	4.658	4.944	5.265										
HVT2	2.165	1.821	1.493	1.158	.875	.742	.666	.603	.511	.430	.380										
HAT2	-.678	-.578	-.456	-.322	-.162	-.045	.018	.077	.160	.265	.380										
BVN2	-1.553	-1.180	-.832	-.600	-.338	-.083															
BAN2	1.493	1.332	1.143	.976	.921	.949	.958	.966	.996	1.008	1.000	.982	.928	.811	.716	.622	.424	.293	.150	-.040	-.332
BAD2	1.493	1.544	1.579	1.645	1.787	2.012	2.306	2.663	3.094	3.610	4.220										
BVD2	2.090	2.387	2.677	2.976	3.213	3.412	3.548	3.687	3.888	4.058	4.220										
BVT2	2.090	1.763	1.478	1.223	.997	.821	.665	.545	.376	.136	.000										
BAT2	-1.424	-1.289	-1.144	-.900	-.711	-.664	-.600	-.400	-.329	-.200	.000										

SPECIFIC SPEED = 13,500

	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21
HVN3	-.956	-.948	-.920	-.836	-.720	-.511															
HAN3	2.735	2.490	2.275	2.057	1.900	1.762	1.645	1.535	1.406	1.223	1.000	.770	.534	.287	.047	-.222	-.510	-.833	-1.200	-1.618	-2.044
HAD3	2.735	3.054	3.400	3.750	4.116	4.493	4.850	5.228	5.622	6.010	6.423										
HVD3	1.075	1.310	1.567	1.888	2.288	2.688	3.239	3.873	4.675	5.535	6.423										
HVT3	1.075	.876	.784	.738	.788	.800	.730	.610	.490	.321	.117										
HAT3	-2.233	-1.883	-1.456	-1.163	-1.000	-.838	-.691	-.520	-.354	-.160	.117										
BVN3	-.720	-.712	-.692	-.640	-.560	-.398															
BAN3	1.954	1.786	1.618	1.402	1.237	1.126	1.078	1.065	1.069	1.056	1.000	.894	.735	.517	.280	.000	-.222	-.513	-.852	-1.250	-1.592
BAD3	1.954	2.188	2.475	2.811	3.200	3.660	4.086	4.457	4.814	5.161	5.535										
BVD3	.670	.939	1.179	1.471	1.846	2.225	2.743	3.323	4.022	4.780	5.535										
BVT3	.670	.487	.540	.600	.740	.746	.663	.476	.247	.034	-.145										
BAT3	-2.278	-1.972	-1.574	-1.325	-1.196	-1.052	-.965	-.760	-.636	-.420	-.145										

III COMPUTER WATER HAMMER CALCULATIONS

The methods of computing head and velocity at small time intervals for equally-spaced sections along a pipeline, with inclusion of friction, have been reported elsewhere in the literature.[4,5,6] The important working equations are briefly summarized:

For interior sections along the pipeline

$$V_P = \frac{V_R+V_S}{2} + \frac{g}{2a_C}(H_R - H_S) - \left(\frac{f}{2D}V^2\right)_C \Delta t \qquad (1)$$

$$H_P = \frac{H_R+H_S}{2} + \frac{a_C}{2g}(V_R - V_S) - V_C \sin\varphi \, \Delta t \qquad (2)$$

in which V_P and H_P are velocity and elevation of hydraulic grade line at section P . φ is the angle the pipeline makes with the horizontal measured positive downwards. Referring to Figure 3, it is considered that V and H at A, B, and C have previously been computed. V_R, V_S, H_R, and H_S are obtained by linear interpolation between A, C, and B by the formulas:

$$V_R = V_C + \theta(V + a)_C (V_A - V_C) \qquad (3)$$

$$H_R = H_C + \theta(V + a)_C (H_A - H_C) \qquad (4)$$

$$V_S = V_C + \theta(V - a)_C (V_C - V_B) \qquad (5)$$

$$H_S = H_C + \theta(V - a)_C (H_C - H_B) \qquad (6)$$

in which $\theta = \Delta t/\Delta x$ the preselected mesh ratio for the calculations. C_+ and C_- designate the characteristics. For the method to yield meaningful results it is essential that

$$\theta < \frac{1}{|V| + a} \qquad (7)$$

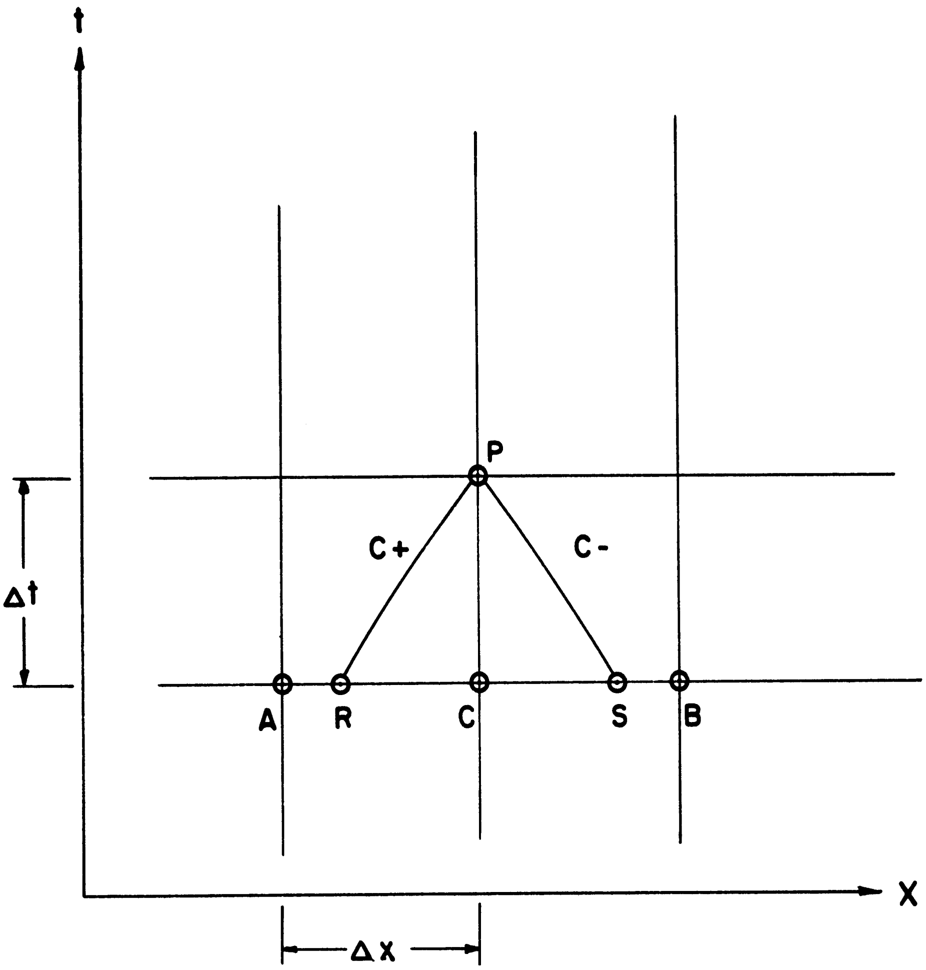


Figure 3. Interior Point Calculation.

in which a is the speed of the pressure pulse wave in the pipe. In Equation (1) f is the Darcy-Weisbach friction factor, D the inside diameter of the pipeline, g the acceleration due to gravity, and the subscript C means evaluation of the quantity at the section under computation for the preceding time.

At each end of the pipeline only one equation is available in terms of the two unknowns V_P and H_P ; these equations are:

The right end:

$$V_P = V_R - \frac{g}{a_C} (H_p - H_R) - (\frac{fV^2}{2D})_C \Delta t - \frac{g}{a_C} V_C \sin \varphi \Delta t \quad (8)$$

The left end:

$$V_P = V_S + \frac{g}{a_C} (H_p - H_S) - (\frac{fV^2}{2D})_C \Delta t + \frac{g}{a_C} V_C \sin \varphi \Delta t \quad (9)$$

External conditions must supply the extra relationship so that V_P and H_P may be computed at each end.

IV CALCULATION OF SPEED CHANGE

When power to a pump is interrupted, an unbalanced torque, T, is applied to the moving parts which depends upon the rotational speed ω and the discharge through the pump. The basic equation for speed change is

$$T = -I \frac{d\omega}{dt} \tag{10}$$

in which I is the moment of inertia of rotating parts, including the liquid within the impeller and $d\omega/dt$ is the angular acceleration. For computation period Δt, the change in rotational speed $\Delta\omega$ (radians per sec) is given by

$$\Delta\omega = -\frac{T \Delta t}{I} \tag{11}$$

The average torque is estimated for the short period Δt, from Figure 1, by extrapolating previous values of α and v for the midpoint of the period, thus

$$\bar{v} = v + dV \tag{12}$$

$$\bar{\alpha} = \alpha + d\alpha \tag{13}$$

Then from the appropriate curve of Figure 1 for $\bar{\alpha}/\bar{v}$ or $\bar{v}/\bar{\alpha}$ and the proper zone (determined by sign of $\bar{v}$ and $\bar{\alpha}$) β/α^2 or β/v^2 is obtained by parabolic interpolation. From this expression, by multiplying by $\bar{\alpha}^2$ or $\bar{v}^2$, β is obtained. Equation (11), in terms of β and α, becomes

$$\Delta\alpha = -\beta \frac{30 T_R \Delta t}{I N_R \pi} \tag{14}$$

which yields the new speed ratio α when added to the previous value of α . T_R and N_R are rated torque and speed, respectively. In the computer program the values of $\overline{\alpha}$, $\overline{v}$, and $\overline{\alpha}/\overline{v}$ or $\overline{v}/\overline{\alpha}$ permit the correct curve to be selected. With the speed known for the new time interval, the head and velocity ratios may be computed at the upstream end of the pipeline by taking into account the waterhammer pulses and valve losses, as outlined in the following section.

V. BOUNDARY CONDITIONS FOR A PARALLEL PUMP SYSTEM

In Figure 4 a pumping station is connected to a suction pipe and a discharge pipe, which in turn are connected to reservoirs. Several identical pumps are connected in parallel within the station, each of which may have a control valve at its discharge. H_o is the rated head and Q_o the rated discharge of each pump. When the difference in elevation of reservoirs is other than for rated conditions, H_s is the steady-state total dynamic head produced by a pump. The elevation of hydraulic gradelines are computed for the transient problem by taking the elevation datum as the elevation of centerline of the pumps. The actual transient head at any instant is the difference in elevation of hydraulic gradeline and pipe at any section. The heads above the centerline of the pumps are made dimensionless by dividing by the rated head H_o.

At the instant of failure of power to one or more pumps, all discharge valves on the failing pumps are assumed to close simultaneously in an arbitrarily determined manner, except when they act as check valves or are omitted completely. In solving the transient problem, the first step is to find the steady state conditions, including elevation of hydraulic gradelines and discharge corresponding to the elevations of reservoirs. At t = 0 the power is assumed to be cut off for M_f of the M pumps, leaving M_o pumps in operation at constant speed $\alpha = 1$. For consecutive, equal small time increments Δt, the following unknown dimensionless quantities

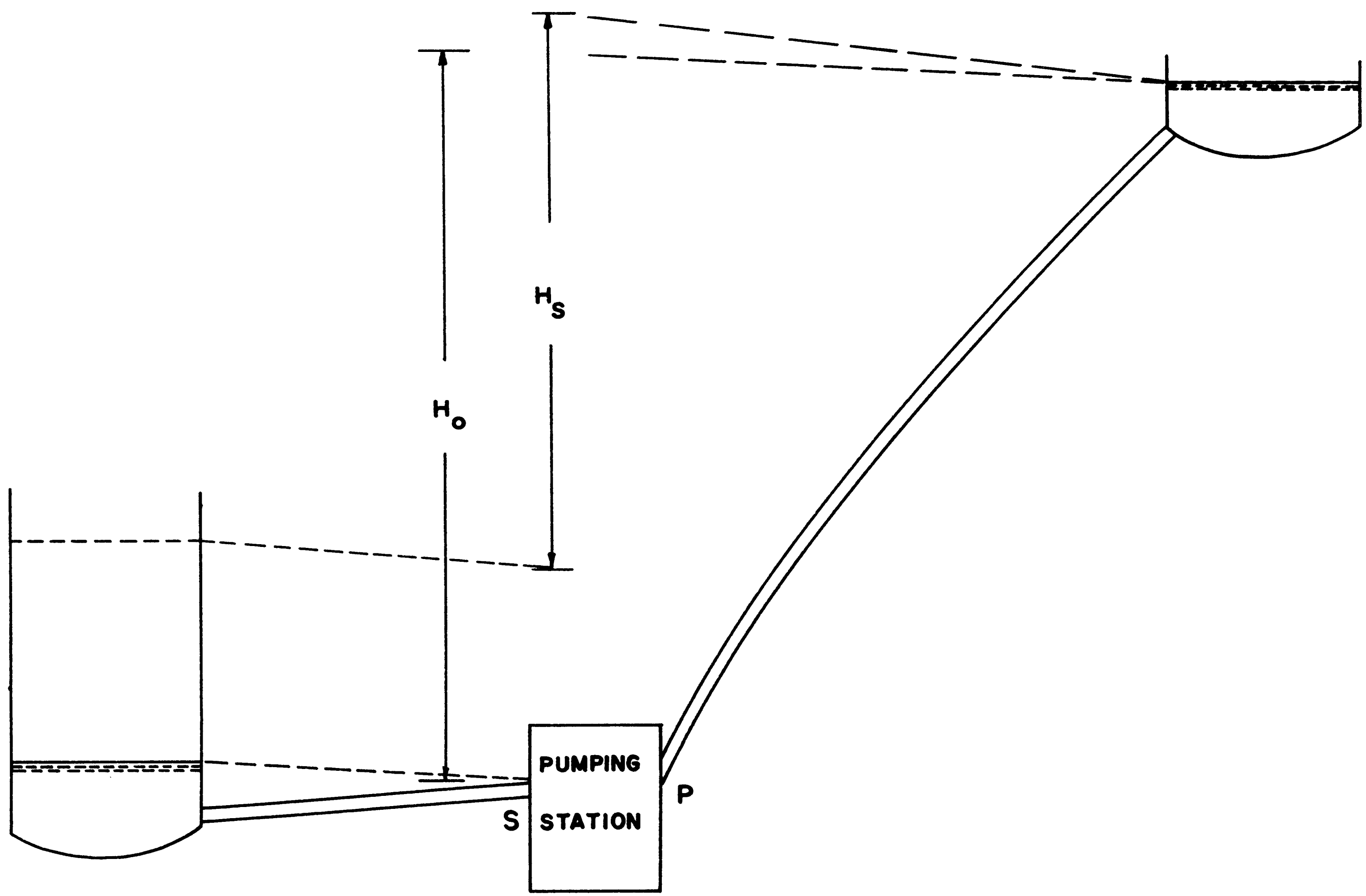


Figure 4. Parallel Pump System with Suction Line.

must be computed to determine the boundary conditions at the pump (the speed for the new time is computed first by the methods of Section IV):

$v_1 = \frac{Q_1}{M \cdot Q_o}$ in which Q_1 is the discharge through one operating pump.

$v_2 = \frac{Q_2}{M \cdot Q_o}$ in which Q_2 is the discharge through one failing pump.

$v_P = \frac{Q}{M \cdot Q_o}$ in which Q is the transient flow through the system at P, Figure 4.

tdh_o = total dynamic head (produced by an operating pump) divided by H_o.

tdh_f = total dynamic head (produced by a failing pump) divided by H_o.

h_P = El. of hydraulic gradeline at P divided by H_o.

h_S = El. of hydraulic gradeline at S (Figure 4) divided by H_o.

h_{vo} = head loss across the valve of an operating pump divided by H_o.

h_{vf} = head loss across the valve of a failing pump divided by H_o.

By defining τ as the dimensionless valve opening such that for steady-state conditions it has the value unity for head loss DELH and steady-state discharge Q_s, and by considering that τ is a known function of time, nine equations are available for solving for the nine unknown. These equations are:

$$tdh_o = a_o + a_1 v_1 + a_2 v_1^2 \qquad (15)$$

in which, for $\alpha = 1$, a_o, a_1, a_2 are determined from Table I to replace the curve HAN by Equation (15). For example, if $\bar{v}/\alpha = .75$ for $N_s = 1800$, then from the table $v/\alpha = .7$, $h/\alpha^2 = 1.166$; $v/\alpha = .8$, $h/\alpha^2 = 1.118$; $v/\alpha = .9$, $h/\alpha^2 = 1.061$. Then a_o, a_1, a_2 are computed so that Equation (15) passes through these points. This is done for each time increment so that the characteristic pump curve is represented by an appropriate parabola for each calculation.

$$tdh_f = b_o + b_1 v_2 + b_2 v_2^2 \tag{16}$$

in which, for known α, b_o, b_1, and b_2 are obtained by passing a parabola through three adjacent points on the appropriate head-discharge curve defined in Table I.

$$v_P = c_1 + c_2\, h_P \tag{17}$$

in which c_1 and c_2 are obtained from the left-end boundary condition for the discharge pipe, Equation (9). Everything is known in the equation except V_p and H_p, and these are made dimensionless in Equation (17).

$$v_P = c_{S_1} + c_{S_2} h_S \tag{18}$$

The right-end boundary conditions, Equation (8), applied to the suction pipe, yields the values of c_{S_1} and c_{S_2}.

$$v_P = M_o v_1 + M_f v_2 \tag{19}$$

which is the continuity equation in dimensionless form

$$h_{vo} = \frac{DELH}{H_o} v_1 |v_1| \tag{20}$$

Since for steady state conditions $M \cdot V_1 = 1$, this equation yields the dimensionless head loss across the valve (which remains open) of an operating pump.

$$h_{vf} = \frac{M^2}{\tau^2} \frac{DELH}{H_o} v_2 |v_2| \tag{21}$$

which yields the dimensionless head loss across the valve of a failing pump. τ is given as a function of time, or as a function of v_2 for a check valve.

$$tdh_o = h_P - h_S + h_{vo} \tag{22}$$

$$tdh_f = h_P - h_S + h_{vf} \tag{23}$$

These two equations state that the head difference between junctions P and S (Figure 4), is equal to the total dynamic head produced by a pump minus the head loss across the valve, for either operating or failing pumps.

The last nine equations, in nine unknowns, yield all needed information for the boundary conditions at the pumping station when the speed of failing pumps is known for the time of calculation.

The general procedure for solving the transient problem is to first set up the steady-state case for the given elevations of reservoirs, and pump and pipeline characteristics, computing H_S, Q_S and the velocity and elevation of hydraulic gradeline at equal reaches along the suction and the discharge pipes. Next the speed change on the failing pumps is computed for time Δt as discussed in Section IV. With this speed known, all the constants in the nine equations (15) - (23) may be determined and the nine unknowns determined. The waterhammer equations for interior points along the suction and discharge lines are next used to compute new heads and velocities, then the boundary conditions at each reservoir, which fixes the elevation of hydraulic gradelines there, are used. The results may then be printed out if desired, and the time incremented to repeat the procedure until the desired maximum time is reached.

VI. DISCUSSION OF THE COMPUTER PROGRAM

Although several papers have dealt with the failure of power to pumps[3,7,8,9] most of the work has been concerned with lines of such a length that the friction head loss is small compared with the total dynamic head. For the situation where there is no suction pipe, no discharge valving, and friction is neglected, curves have been prepared[2,7] showing maximum and minimum surge heads at the pump and midpoint of discharge pipe as a function of $K_1 \; 2L/a$ for various values of aV_o/gH_o, with

$$K_1 = \frac{30T_R}{IN_R} \qquad (24)$$

from Equation (14).

With the additional complications of a long suction pipe, arbitrary valving, and large friction heads, the preparation of dimensionless curves for a wide variation of parameters would not be feasible. A typical water hammer problem resulting from failure of one or more pumps in a pipeline takes about 15 seconds of time on an IBM 7090 computer. For any specific project investigation, solutions may be obtained for appropriate ranges of all parameters including inertia of rotating parts, pipe diameters, inclusion of relief valves or surge tanks, and special valving.

To illustrate the use of the computer program several transient cases have been solved. First, to check on the general validity of the program, data have been obtained and used to check against field tests for the Tracy Pumping Plant.[8] The pump characteristics for the normal zone were furnished by the pump manufacturer. In the energy dissipation and turbine operation zones characteristics were obtained from the California Institute of Technology N_s = 1800 data. The results of field tests and computer results are shown in Figure 5 with the plotted points obtained from the computer program. The agreement is excellent for the normal zone where characteristics are known, and is generally good for the speed calculations, but the computer gives somewhat higher heads for the zones of energy dissipation and turbine operation.

In Figures 6 through 11 results of calculations are shown for a hypothetical pump having the complete pump characteristics given by the N_s = 1800 data in Table I, with speed N_o = 327 rpm, head $H_o = H_s$ = 200 feet, discharge Q_o = 191 cfs for steady-state operation. In each case the suction line has a length of about 23.5 per cent of the discharge line and has the same diameter as the discharge line. The calculations for Figures 6 through 8 were without a discharge valve and Figures 9 through 11 were made with check valves at the pump discharge. The pipelines were taken as horizontal with elevation datum through their centerlines. The elevations of suction and discharge reservoirs were calculated so that rated head and flow is obtained for steady-state operation, and the elevation of hydraulic gradeline at the pump suction is zero.

In Figure 6 the loci of maximum and minimum points along the hydraulic gradlines are shown for two cases, in which all three operating pumps fail.

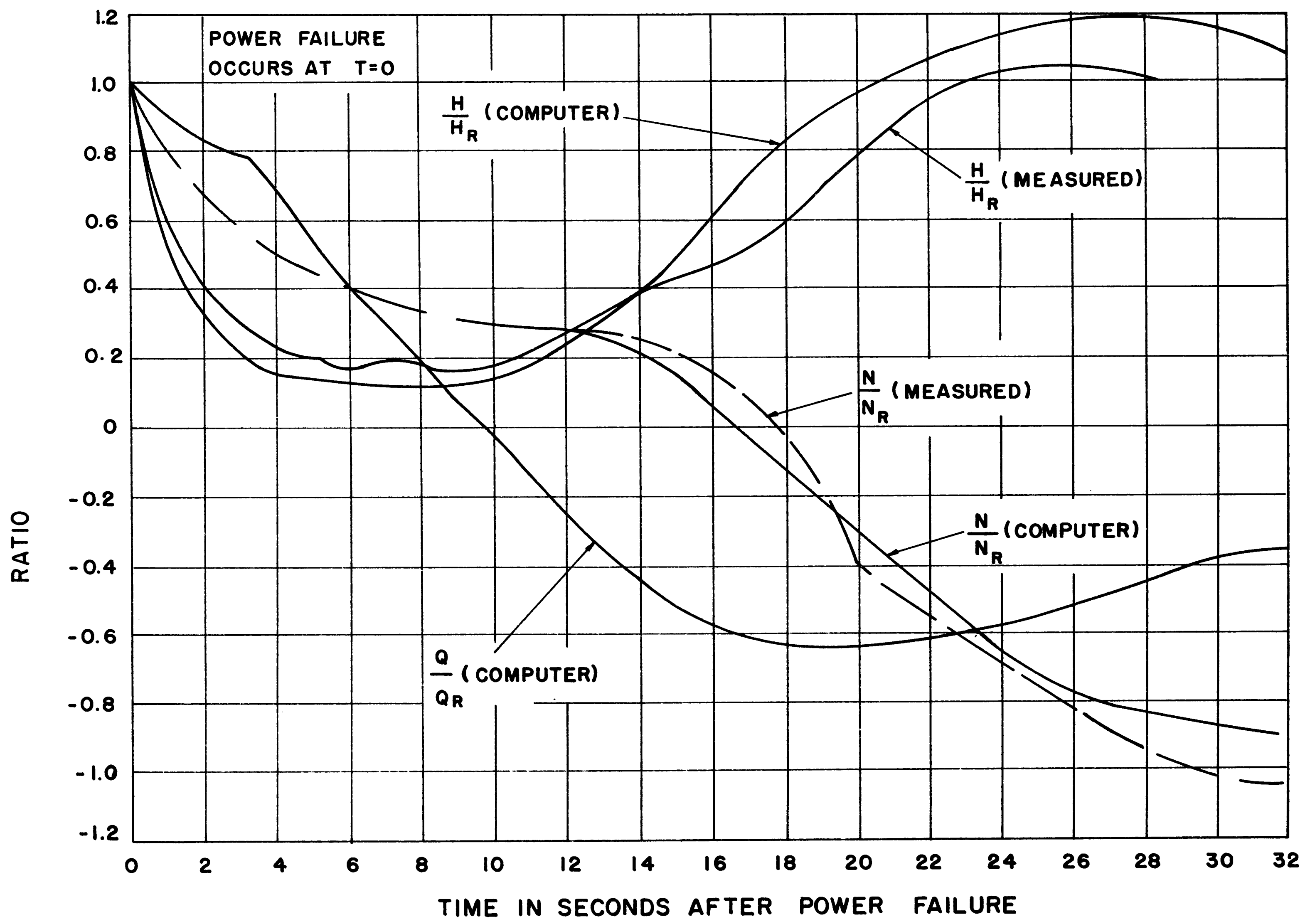


Figure 5. Tracy Pumping Plant Failure Results from Field Tests and from Computer Results. Simultaneous Failure of Two Pumps Discharging to Same Pipe.

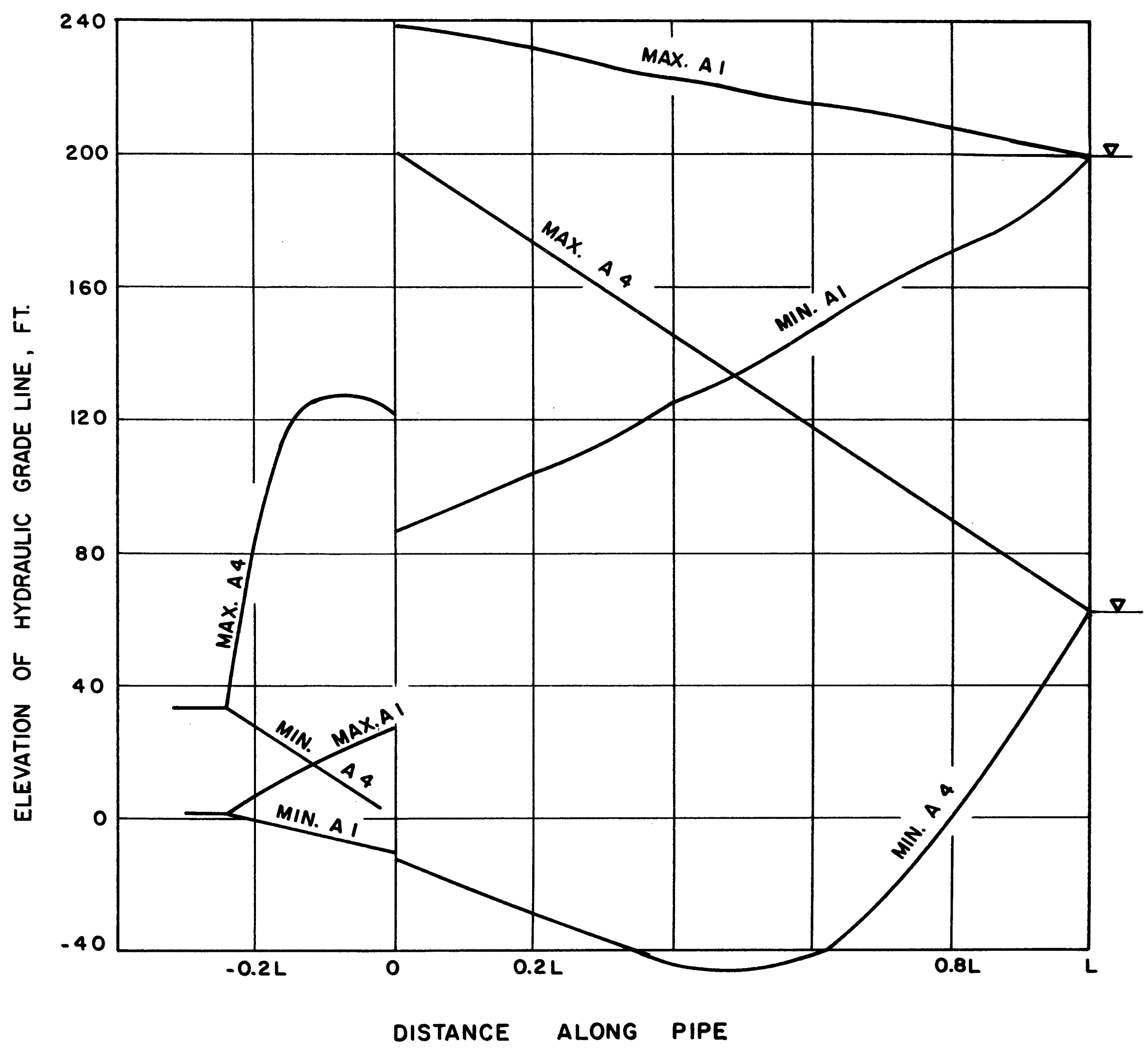


Figure 6. Maximum and Minimum Values of Hydraulic Gradeline for Simultaneous Failure of 3 Pumps without Discharge Valves in a 11-foot Diameter Line.

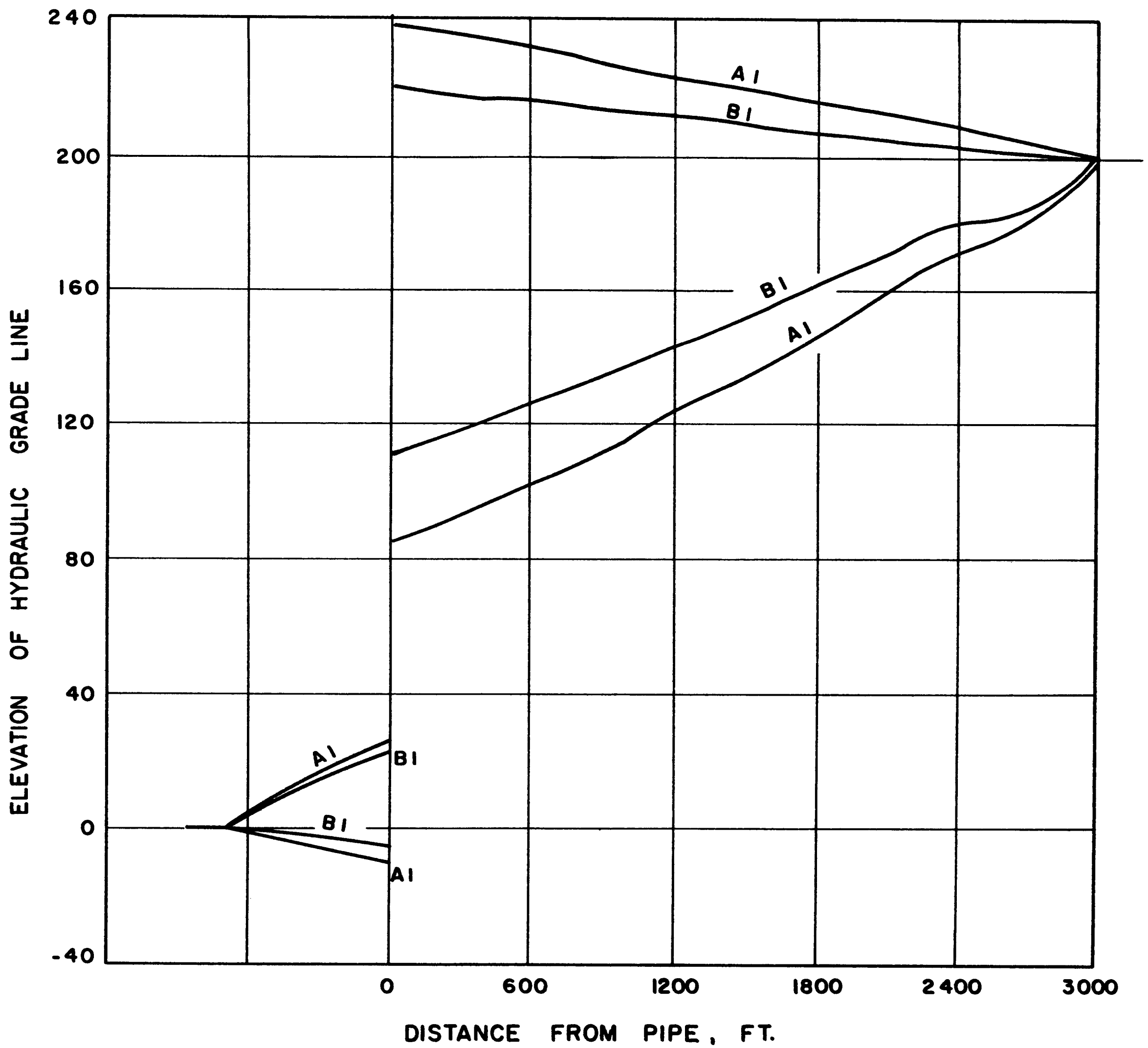


Figure 7. Maximum and Minimum Values of Hydraulic Gradeline for Simultaneous Failure of 3 Pumps without Discharge Valves in a 11-foot Diameter Line.

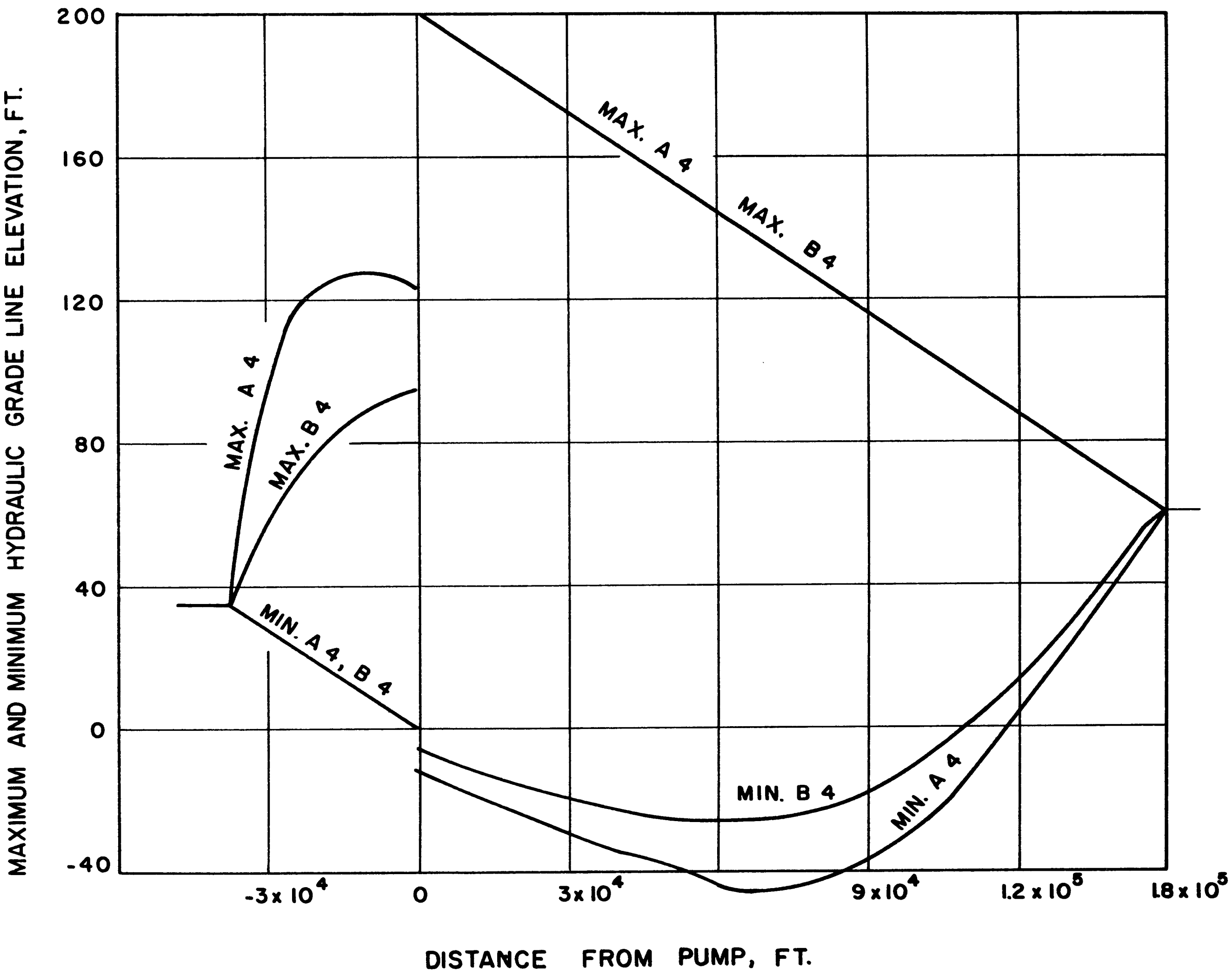


Figure 8. Maximum and Minimum Values of Hydraulic Gradeline for Simultaneous Failure of 3 Pumps without Discharge Valves in a 11-foot Diameter Line.

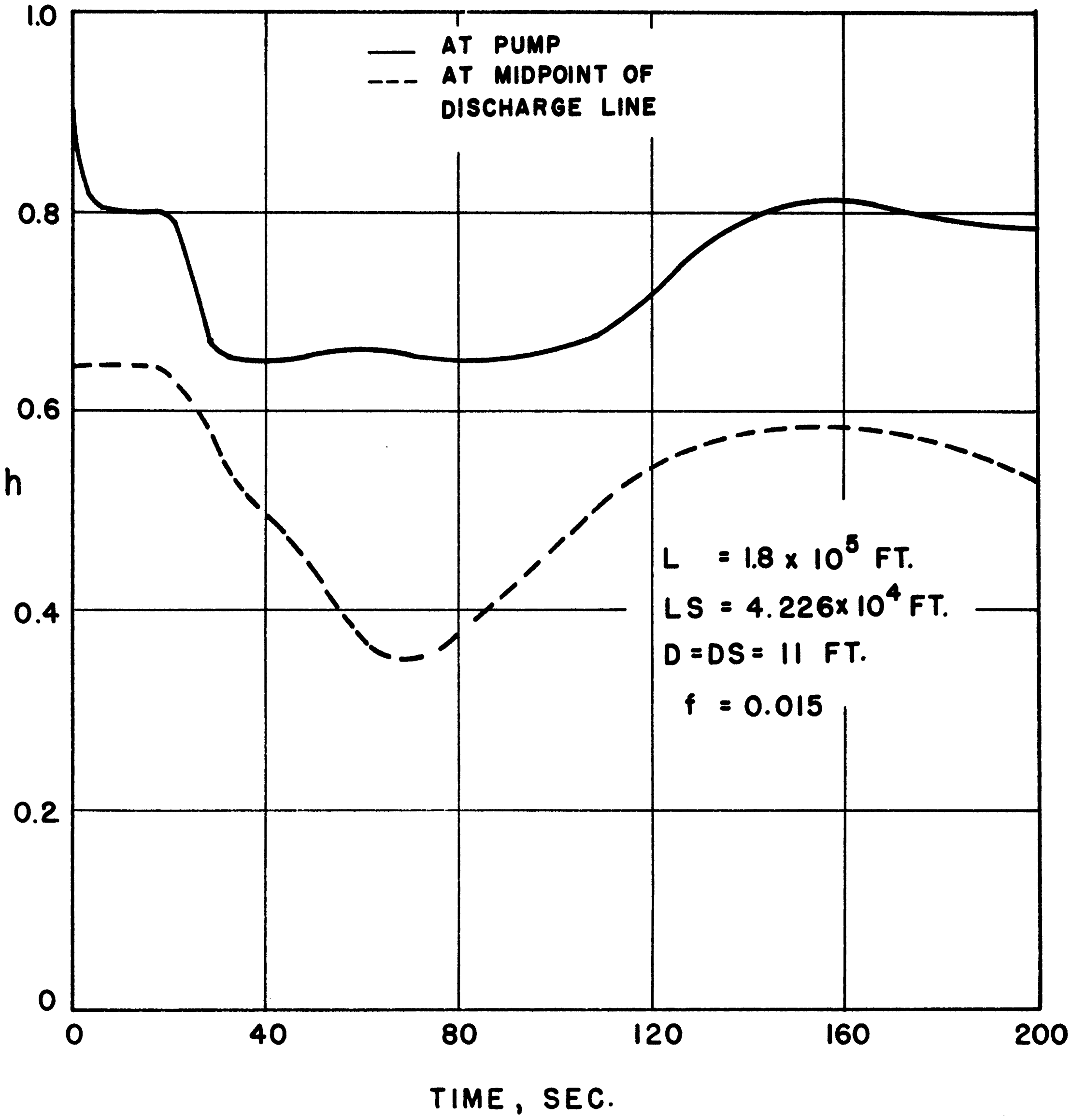


Figure 9. Dimensionless Head at Upstream and Middle of Discharge Line for 3 Pumps When 1 Fails.

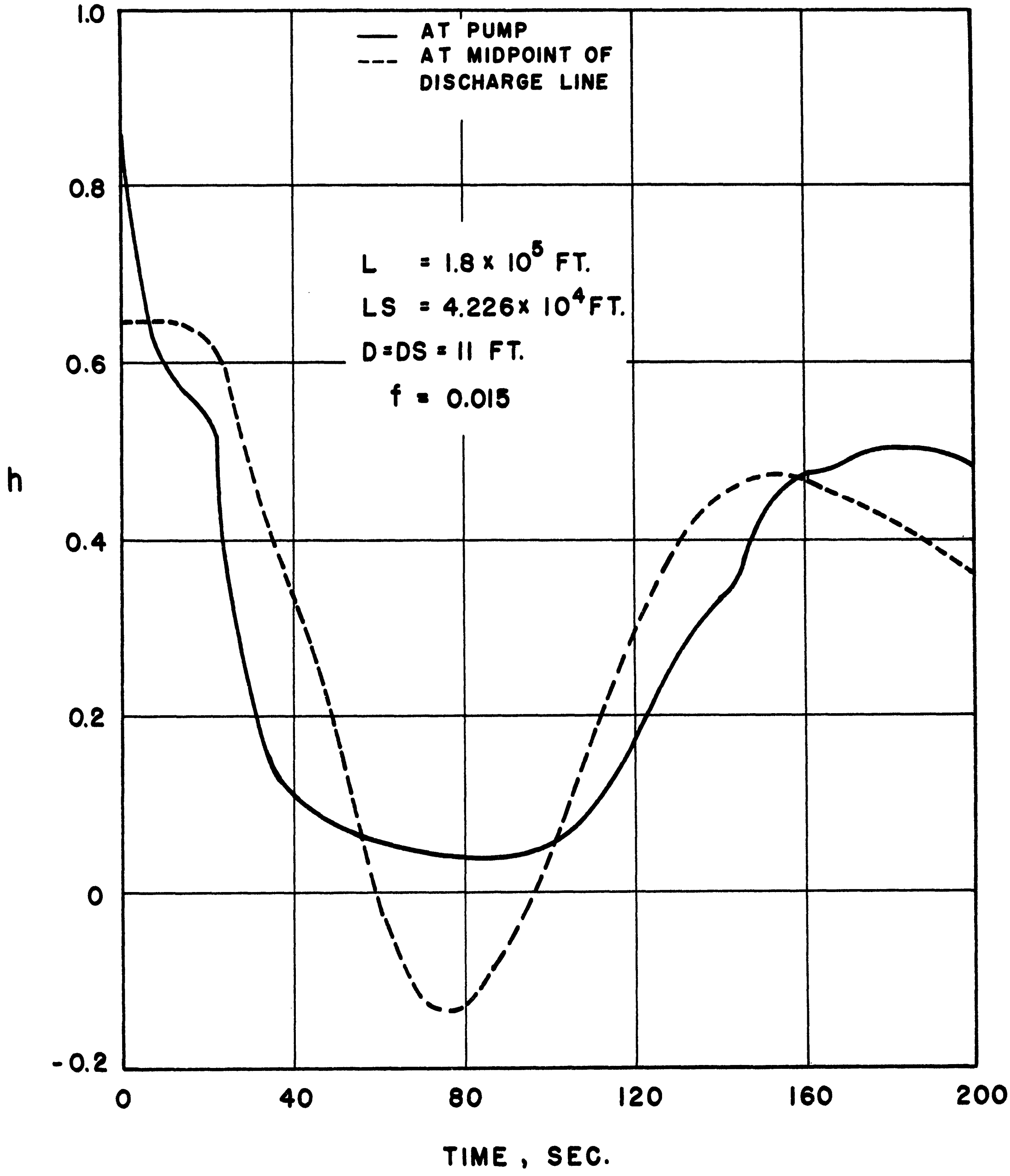


Figure 10. Dimensionless Head at Upstream and Middle of Discharge Line for 3 Pumps When 2 of Them Fail.

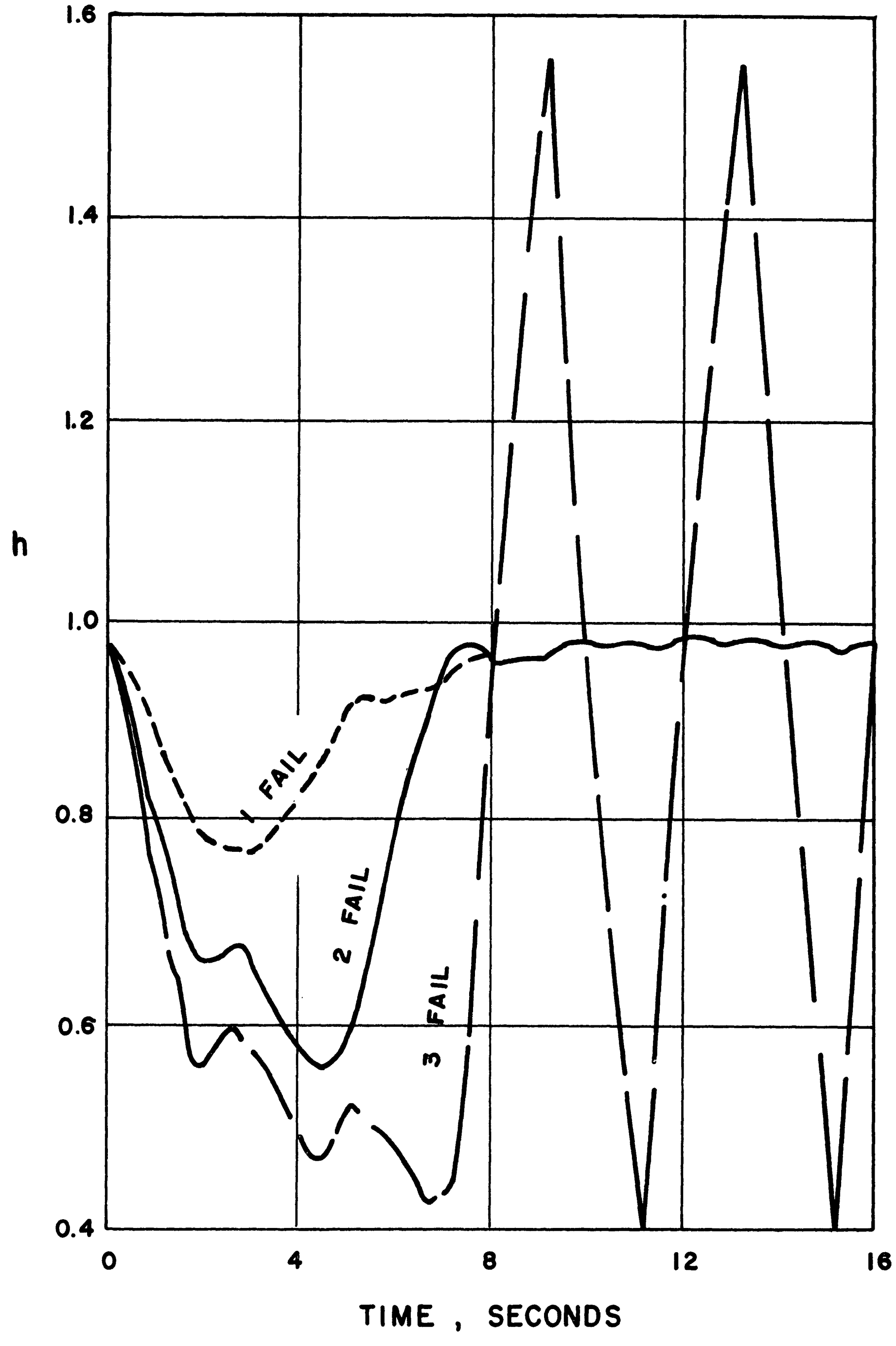


Figure 11. Head at Pump for 3 Pumps Operating When 1, 2, and 3 Fail, with Undamped Check Valves.

A1: $L = 3000$ feet, $L_s = 704$ feet, $D = D_s = 11$ feet, $f = f_s = .015$. The friction drop is very small for this case, hence the loading is due to the gravity lift. The maximum steady state head is 200 feet and the transient head about 238 feet. The minimum head varies from about 88 feet linearly to about 198 feet. The suction pipe transient extremes are also shown.

A4: $L = 180{,}000$ feet, $L_s = 42{,}260$ feet, $D = D_s = 11$ feet, $f = f_s = .015$. The friction drop predominates for this case, with the maximum transient heads equal to the steady-state head in the discharge pipe, and the minimum transient heads in the suction pipe equal to the steady state heads. The minimum hydraulic gradlines cause the main threat, since column separation may occur with subsequent rejoining with high heads resulting. The discharge pipeline would need to be placed at about -20 feet through its midlength to avoid column separation.

Figure 7 shows the effect of doubling the moment of inertia of rotating parts for power failure when the load is primarily due to gravity. The maximum pressure rise along the pipeline is reduced by about one half by the increased moment of inertia. There is considerable improvement in the minimum pressures along the discharge pipe. Whether they are significant depends upon the actual pipeline profile. Figure 8 shows the effect of increasing the moment of inertia by 4 times for the very long pipeline with most of the pump loading due to friction. In neither of these cases do the transient pressures exceed the steady state pressures. Improvement is quite significant with respect to the minimum pressures, depending again upon the actual pipeline profile.

Transients are generally less severe the greater the pipeline diameter, other parameters being the same, since head rise due to velocity change is given by $\Delta H = a\Delta V/g$ and velocities vary inversely as the square of the diameter. To conserve space the computer results are not shown.

Figures 9 through 11 are plots of dimensionless head against time for failures when check valves are located at each pump discharge. Figure 9 is the case of three pumps operating on the long pipeline when one pump fails. The check valve on the failing pump closes about 5 seconds after power failure and the maximum discharge drop is about 16 per cent. The minimum point on the hydraulci gradeline occurs at the midpoint of the discharge line as shown.

When two pumps out of three fail on a long pipeline, Figure 10, the check valves close at about 143 seconds, with a flow reduction of about 47 per cent at 200 seconds.

When all three pumps fail on a long pipeline the minimum head along the pipeline is given by the Min A4 line in Figure 6. The mimimum head occurs at the midpoint of the discharge line at 72 seconds and at the pump at 86 seconds. The check valves would not close for several hundred seconds after power failure.

Figure 11 shows dimensionless head variation at the pump discharge (downstream from the check valves) as a function of time for a short pipeline for the three cases of failure of one, two, or three of three operating pumps. As long as one pump remains in operation the steady-state pressures are not exceeded by the transient pressures. For the gravity loading with all pumps failing with undamped check valves very severe fluctuating transients occur. Upon closure of the check valves the water in the pipeline acts as a liquid spring with the hydraulic gradeline at the valve oscillating about the downstream reservoir. When one pump fails its check valve closes in about 3 seconds, when two pumps fail their check valves close in about 5.2 seconds and when the three pumps fail their values close at 7.2 seconds for the cases shown.

Since the transients are so extreme for complete failure with check valves for gravity loadings, special precautions should be taken. One procedure is discussed in the following section.

VII. SPECIAL VALVE PROGRAMMING

In the preceding section it was shown that undamped check valves are unsatisfactory for simultaneous failure of all pumps when pumping against a gravity head. In the case of Tracy Pumping Plant, for example, the discharge valves are butterfly valves and close first at a rapid rate followed by a second slower rate. This procedure permits reasonable control of maximum pressures, but must be found by a trial method, i.e., assume the closing rates, then calculate the resulting transients to see if they are satisfactory. One objection is that the pumps are permitted to reverse; and, since pump characteristics are in general not accurately known for the turbine zone, considerable error may result.

By use of previously developed valve stroking methods,[5] undamped check valves may be used for all the pumps, with a bypass line and control valve around one of the check valves. Figure 12 shows the computer calculation of transients for application of the valving to the Tracy Pumping Plant (see Figure 5). The transients are substantially the same up to the instant of flow reversal in the discharge line at the valve t = 9.6 seconds. At the instant reversal, from the computer solution, the velocity and hydraulic gradeline elevation are known for 21 equally spaced sections along the discharge line. By proper motion of the control bypass valve for the next 2L/a seconds, the hydraulic gradeline can be made a straight line from the valve to the downstream reservoir with arbitrary maximum head at the pumps. During this period the flow is also made uniform throughout the discharge pipeline. Friction is not important for this period of time as the velocities are quite small. With the uniform flow and straight hydraulic grade line, the procedures of the valve stroking method[5] are used to bring the flow to rest without increase in hydraulic gradeline. In Figure 12 the maximum permitted head was arbitrarily chosen as HM = 1.10. Advantages of the method are nonreversal of the pumps, very small loss of water in the reversed flow, and arbitrary selection of the maximum stress in the discharge pipeline.

Figure 13 shows the equations and graphical visualization of the conversion of an arbitrary hydraulic gradeline ABC to a preselected maximum hydraulic gradeline HM - C. In the sketch of the system the discharge pipe has N+1 equally spaced sections numbered 0 at A to N at C. Subscript 0 represents the time at which the conversion of hydraulic gradeline ABC starts, and subscript 1 refers to time 2L/a later when the hydraulic grade line is HN - C. Intermediate times are indicated by subscripts I/2N or 1-I/2N.

On the Allievi[10] chart (Figure 13) of h versus v the horizontal lines through H(N) (representing the reservoir elevation) are drawn. The required velocity and head, VA and HA at the valve are to be specified for the 2N+1 equal time increments in 2L/a seconds. Let the point A_0 on the plot represent h and v at A at time 0; this condition is reflected at the reservoir at $C_{1/2}$ and returns to head HM at A_1 as shown. This determines the uniform velocity V1 for the pipe at time 1. For a straight-line hydraulic gradeline at time 1 with

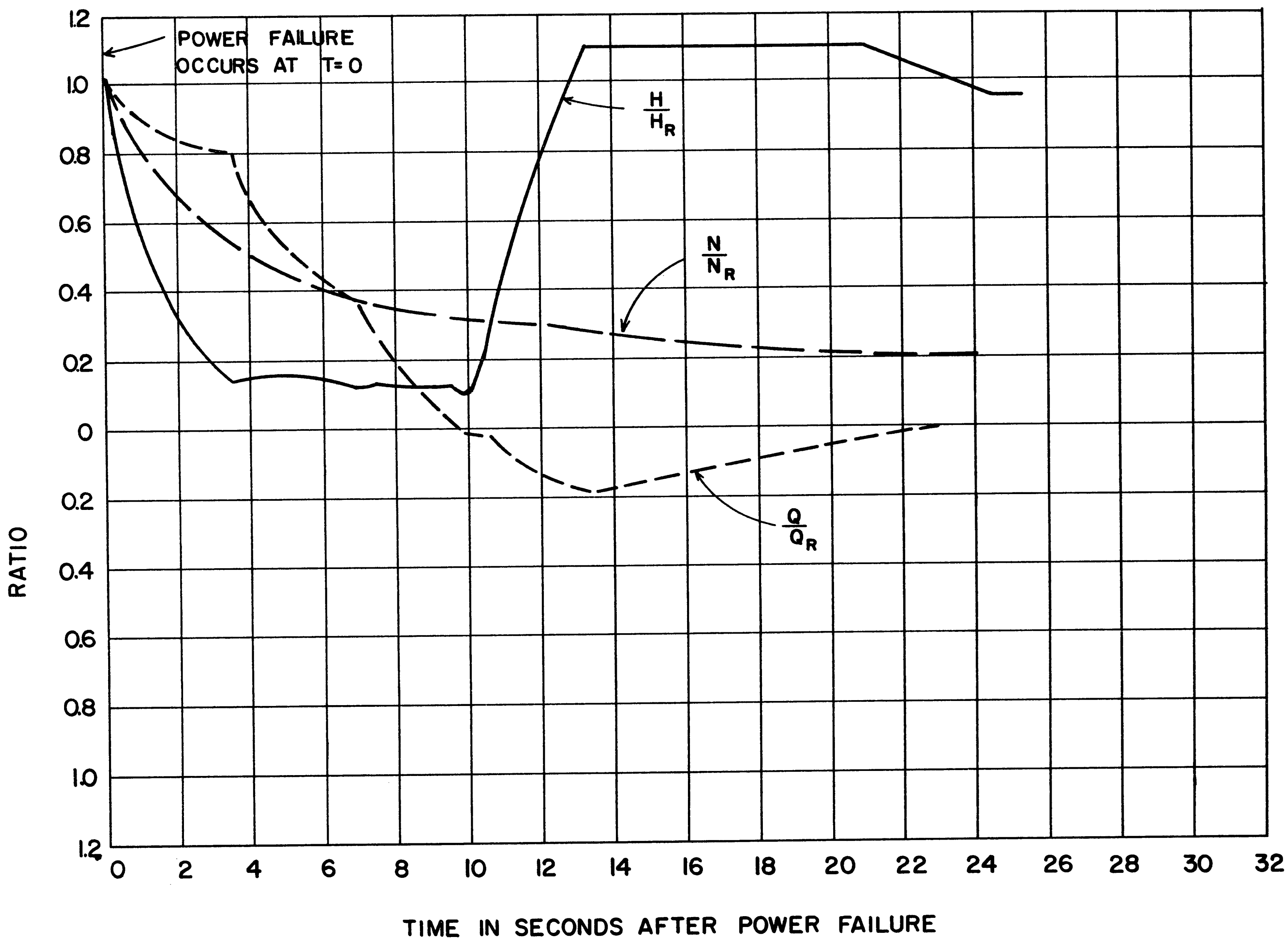
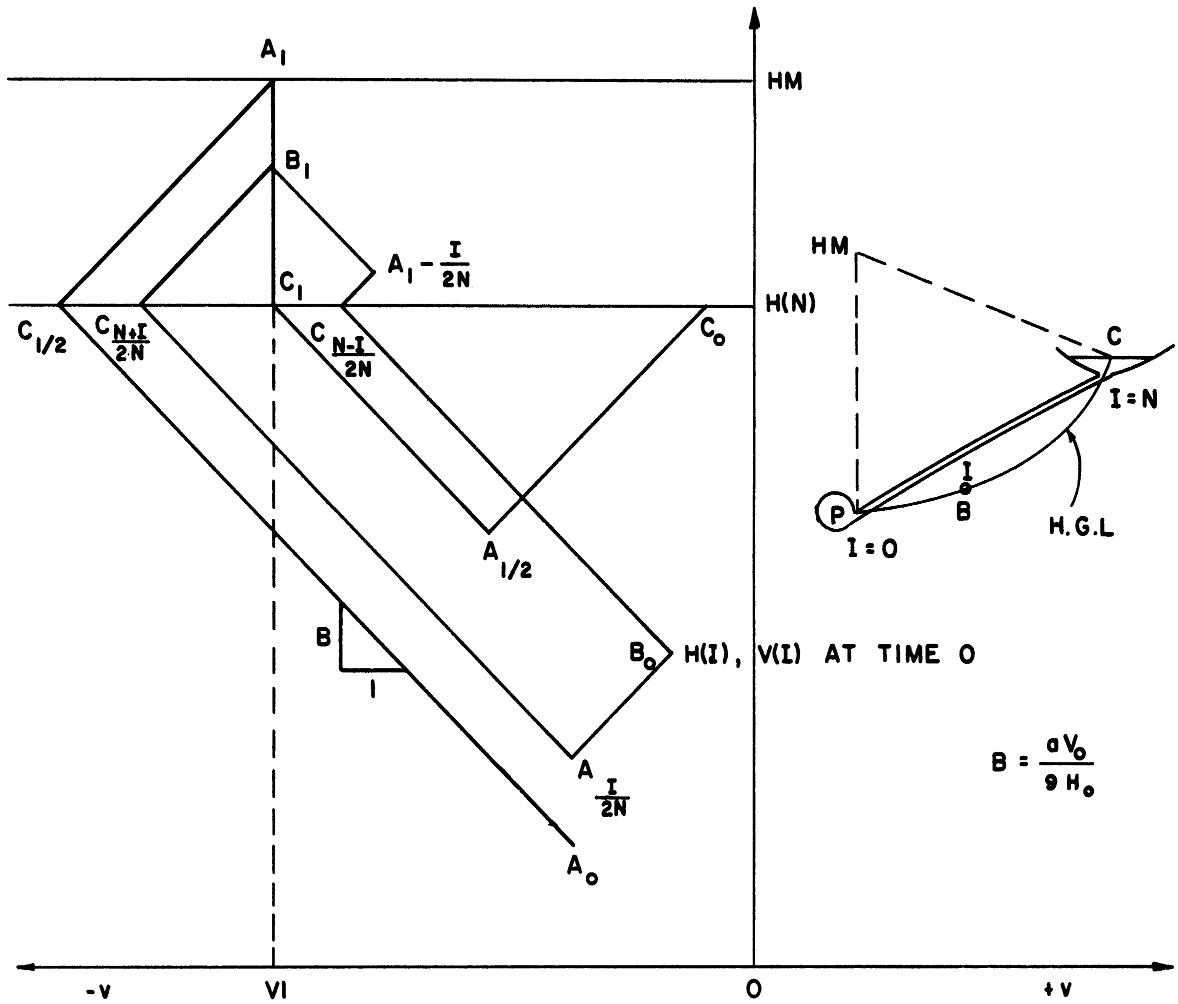


Figure 12. Programmed Valve Closure at Instant of Flow Reversal, Tracy Pumping Plant.



$$VI = V - (2H(N) - H - HM)/B$$

$$HA_{\frac{I}{2N}} = 0.5\left[B(VI - V(I)) + H(I) - HM + 2H(N) + I(HM - H(N))/N\right]$$

$$HA_{1-\frac{I}{2N}} = 0.5\left[B(VI - V(I)) + 2H(N) - H(I) + HM - I(HM - H(N))/N\right]$$

$$VA_{\frac{I}{2N}} = V(I) - (H(I) - HA_{\frac{I}{2N}})/B$$

$$VA_{1-\frac{I}{2N}} = V(I) - (2H(N) - H(I) - HA_{1-\frac{I}{2N}})/B$$

Figure 13. Equations for Changing Hydraulic Gradeline from Its Position at Time of First Flow Reversal to a Straight Line with Uniform Velocity V1, 2L/a Seconds Later.

uniform velocity V1 in the pipeline the N+1 sections at time 1 must be represented by uniformly spaced points between A_1 and C_1 on the chart. C_O represents h and v at the reservoir at time 0; h and v at the valve at time $\frac{1}{2}$ must be as given by $A_{1/2}$ for the wave to be reflected back to C_1. For the I-th section B, represented by B_O at time 0 and the known point B_1 for desired conditions, the points $A_{I/2N}$ and $A_{1-I/2N}$ show necessary avlues of v and h at the valve at the times I/2N and 1-I/2N respectively. General equations are shown for computation of h and v at the valve, Figure 13. Applying these equations for the 2N+1 times, the required h and v for conversion of hydraulic gradeline from ABC to HM - C are determined. By using the computer program to determine the dimensionless head at the pump, h_{pump}, for these velocities and heads, the valve position τ may be determined

$$\tau_{I/2N} = \sqrt{\frac{VA_{I/2N}}{HA_{I/2N} - h_{pump}}} \qquad (25)$$

and similarly for the period 1/2 to 1.

VIII. SUMMARY AND CONCLUSIONS

Dimensionless-homologous complete pump characteristics for three specific speeds have been developed from test data at California Institute of Technology and are presented in tabular form for easy storage in a computer. When used with waterhammer equations developed by the method of characteristics, including friction, and with arbitrary closure of valves, predictions of hydraulic transients in pipelines due to power failure to pumps may be made.

A computer program has been developed for failure of one or more of a set of identical parallel pumps connected to a common suction pipeline and a common discharge line with arbitrary valving at the pump discharge. Comparisons of hydraulic transients as a result of gravity pump loading or friction-loss pump loading have been made, as well as the effects of pipe diameter and of changes in moment of inertia of moving parts. For gravity lifts high gradients are encountered when the pumps run away backwards, and for long pipelines with friction drop causing the pump loading, low pressures occur in the discharge pipe with danger of column separation.

By use of special valve programming techniques the high pressures resulting from gravity pump loading may be held to a preselected value.

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