Hydraulic Analysis of Sudden Flow Changes in a Complex Pumping Circuit

GEORGE E. ALVES

E. I. du Pont de Nemours and Company, Inc., Wilmington, Delaware

A problem arose as to what would happen if there were sudden failures of power to one or more pumps of a large-scale complex pumping circuit composed of several individual pumping systems. This paper describes the application of several published methods of hydraulic transient analysis to the problem. The performance of the system computed under several assumptions is discussed, and a comparison is made with experimentally determined values.

In the design and operation of most process pumping systems, consideration is given only to the normal steady state conditions; that is, based on continuous uninterrupted operation, the pump, motor, and piping are specified for the given process flow rate and pressure. However, there are a few process pumping systems in which sudden flow changes would cause damage to the pumping facilities or adversely affect the process. This was the case on a problem involving a large-scale, complex pumping circuit.

The pumping circuit, Figure 1, consists of several individual pumping systems in parallel between a large suction header and a large discharge header. Each individual pumping system contains a suction line; a large double-suction, double-volute, high-capacity, high-head centrifugal pump with a flywheel, driving motors, and a brake; and a discharge line with a valve and a piece of process equipment. Two electric motors are used to power the pump; one is an a.c. motor for normal operation at design speed, and the other is a d.c. motor with power supplied by a motor-generator set for emergency operation at reduced capacity at one third of design speed. In the event of failure of a driving motor, the flywheel extends the pumping time while the circuit is being shut down.

Use of a tilting-disk check valve in the discharge line of each system was contemplated to prevent reverse flow in the line in case of failure of power to a pump. However, a question arose as to what might happen if the check valve did not operate properly. When power to a pump fails, the check valve could "hang up," allowing an appreciable backflow to build up, and then close suddenly, causing a high-pressure surge which could rupture the line. If there is no check valve, a considerable backflow could follow such a failure, thus producing a further decrease in flow through the discharge header. Also the high torque on the pump could produce reverse rotation of the d.c. motor, thereby causing damage, if d.c. power is on, to the motor-generator set (powered by a Diesel engine).

To answer such a question, an analysis was required. Several published methods were applied to determine possible pressure buildup if check valves were used and to determine in the absence of check valves the flow through the circuit and the torque on the pump-motor unit as a function of time for various abnormal operating conditions. Owing to the lack of assurance of the various methods of analysis and to the importance of the problem, some of the computed values were verified by tests. The results of the analysis led to the decision not to install check valves but to use brakes or nonreversing clutches to protect the systems from the consequences of running the pumps as turbines.

That the methods of analysis have been verified for a large-scale, complex pumping circuit may be of interest to designers faced with the necessity of conducting similar hydraulic-transient analyses. This paper presents a summary of the methods of computation and a comparison of the computed and experimental results.

DISCUSSION

In the hydraulic analysis of pumping systems involving sudden flow changes, consideration must be given to the effect of slowing down the mass of fluid flowing in the line and the effect of the rotating parts, such as pump impeller, flywheel, and motor rotors.

Water Hammer

When a column of flowing fluid is suddenly stopped, a pounding of the line is observed. This pounding of the line is commonly known as *water hammer*. As the flowing fluid is decelerated, pressure waves are set up owing to the inertia of the fluid in the line.

A brief discussion of water-hammer theory will be presented here as background for the discussions of check-valve operation and hydraulic transients. For sudden flow stoppage, the pressure rise due to the deceleration of a truly incompressible fluid in a nonexpandable pipe would be infinite; that is, all the fluid in the line would behave as a "plug" and the pressure rise would be that corresponding to the inertia effects of this plug. Experiments in pipe lines, however, have shown that there is a finite maximum pressure change, because part of the kinetic energy of the moving fluid in the pipe is expended in stretching the pipe walls and compressing the fluid. The equation for the maximum pressure rise produced by a sudden flow change can be derived from Newton's second law, relating force to the rate of change of momentum. The resulting equation is referred to in the literature as the Joukowsky, or water-hammer, equation and is given as

$$h_{wh} = a(\Delta V)/g_c \qquad (1)$$

where

$$a = \left\{1 \Big/ \left[\frac{\rho}{g_c}\left(\frac{1}{k} + \frac{D}{bE}\right)\right]\right\}^{1/2} \qquad (2)$$

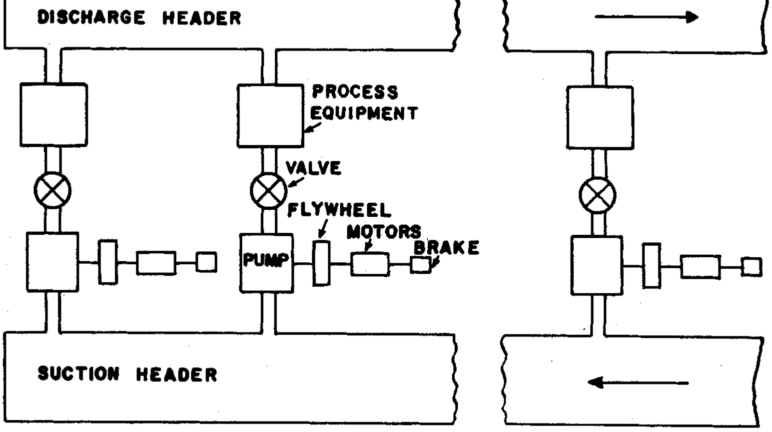

Fig. 1. Pumping circuit.

The maximum pressure rise given by Equation (1) can also be developed if the flow is changed within the time it takes the pressure wave to travel from the point of stoppage to the end of the pipe or to the point of total wave reflection and return; that is, within one pipe period, as given by

$$\tau = \frac{2L}{a} \quad (3)$$

For example, the discharge piping between the pump and the piece of process equipment and between the piece of process equipment and the discharge header may be considered:

$$a \cong 3{,}800 \text{ ft./sec.} \quad (4)$$

$$h_{wh} \cong 120\Delta V, \text{ ft.} \quad (5)$$

$$\tau \cong 0.01 \text{ sec.} \quad (6)$$

If the time of flow stoppage is somewhat longer than one pipe period, the pressure rise will not be so great as that given by Equation (1) as part of the direct pressure waves will be canceled by the reflected pressure waves. The actual pressure rise can be determined by use of the Allievi equations or charts, which are solutions of the wave equations (*1, 2, 4, 10*).

The foregoing analysis also applies to the pressure reduction for the reflected wave or on acceleration of flow. If the pressure reduction results in a static pressure at any point in the line below the vapor pressure of the fluid, the fluid in the line will separate or pull apart as the pressure wave passes that location. The pipe might collapse as the fluid separates; or, barring that, it might burst as the fluid rejoins, provided there are no protective devices such as relief valves to admit air when the fluid separates and to release the air and some fluid when the fluid rejoins.

Additional details on water-hammer theory can be obtained from references *2, 3, 4, 5, 10*, and *12*.

Check Valves

If a check valve in the discharge line from the pump were closed within one pipe period or 0.01 sec., the maximum pressure rise in the line could be calculated from Equation (5):

$$h_{wh} \cong 120 V_0, \text{ ft.} \quad (7)$$

If the check valve were to "hang up" for a time, with a resulting backflow through the system, and then slam shut, the maximum pressure rise in the line would be

$$h_{wh} \cong 120 V_b, \text{ ft.} \quad (8)$$

Information on closing time of check valves (*8*) indicates that the normal closing time of tilting-disk check valves would be about 0.5 sec. With this length of time the backflow could build up to be between 30 and 60% of the normal forward flow, as will be evident later on in this discussion.

Pump Performance

It should be noted that in normal operation, conditions in the system will be unchanging. However, in the event of a power failure, the pump impeller will gradually slow down. Then when the torque supplied by the inertia of the rotating parts is insufficient to pump fluid, the flow in the line will reverse direction and the pump impeller will keep on rotating and slowing down in the normal direction of rotation; that is, the pump acts as a brake or energy dissipator. Finally the pump impeller will reverse direction of rotation and the pump will operate as a turbine with no load. For a constant head on the pump the flow in the reverse direction will increase to a maximum and then decrease somewhat to a constant value at the point of turbine operation with no load. Thus, for the study of hydraulic transients or unsteady state operation in pumping systems, a complete set of pump-performance data, commonly referred to as "complete pump characteristics," is required—complete in the sense that data on head, capacity, speed, and torque are required for the pump in normal or pump operation, as an energy dissipator, and in turbine-type operation.

A typical manufacturer's pump-performance curve for the double-suction, double-volute pump is shown in Figure 2. However, complete pump characteristics were not available from the manufacturer for the actual pumps or even from the literature for any double-suction, double-volute pump. At the time that the computations were made, complete pump characteristics were available for other types of centrifugal pumps namely, single-suction, single-volute (*6, 7*), single-suction, double-volute (*7*), and double-suction, single-volute (*6, 11*). A comparison of these curves indicated that for constant capacity and speed, the variations in head and torque between the different designs were only about 10 to 20%; therefore, as shown in Figure 3, the plot of the data for the double-suction, single-volute pump presented in reference *11* was used in the computations.

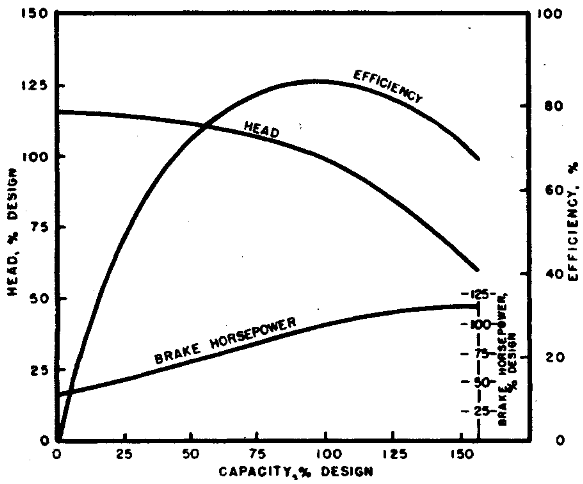

Fig. 2. Typical pump performance curves for a double-suction, double-volute pump.

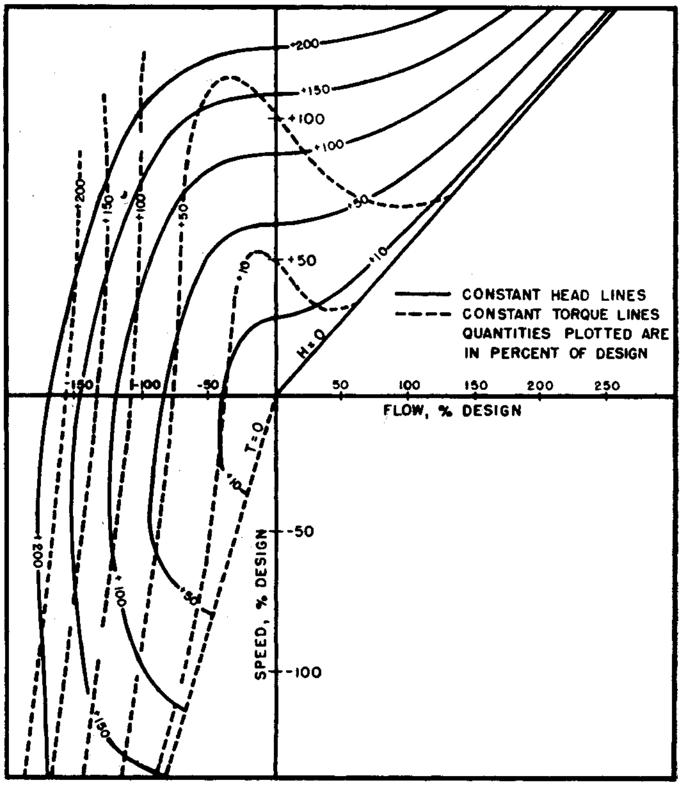

Fig. 3. Complete pump characteristics for a double-suction, single-volute pump.

Hydraulic Transients

Hydraulic transients or unsteady state phenomena considered herein include the flow through the circuit and, for the pumps with sudden power failures, the flow through the pump, the speed of the pump, and the torque on the pump-motor unit. All these transients are considered as a function of time for various types of power failure. The pump undergoing a sudden power failure will be referred to as the *idle* pump.

There are several methods of computing the foregoing information by the application of the various water-hammer equations, complete pump characteristics, and the equation of motion of a rotating system. In addition, there is a method which does not take into account the water-hammer effects. The several methods of computation are described below.

Water-hammer Effects Neglected

A simple method neglecting water-hammer effects is the graphical integration of the equation of motion of a rotating system (*6*).

The equation of motion of a rotating system

$$t = -(I/g_c)(d\omega/d\theta) \qquad (9)$$

is integrated and then the units are changed to those customarily used, to give

$$\theta_{n+1} - \theta_n = -[(\pi I n_0)/(30 g_c t_0)] \int_{N_n}^{N_{n+1}} (dN)/T \qquad (10)$$

Equation (10) can be solved by a graphical integration. First a plot is made of (constant)/T vs. N and then the time is determined from the area under the curve. The constant is the term in brackets in Equation (10). Values of N and T are obtained from the complete-pump-characteristics plot, Figure 3, for an assumed flow and computed head, with the notation that

$$H = h/h_0 \qquad (11)$$

$$Q = q/q_0 \qquad (12)$$

$$N = n/n_0 \qquad (13)$$

$$T = t/t_0 \qquad (14)$$

When two or more pumps are in operation and power to one pump fails, the pressure in the discharge header will drop and the flow from the idle pump will decrease. After a length of time there will be backflow through the idle pump. Therefore, in the foregoing computations the head on the idle pump should be corrected for both discharge-header pressure reduction and flow through the idle pump. Assuming a flow, either in the normal direction or in the reverse direction, depending upon the time, through the idle pump at a given speed, one can compute the head on the idle pump and then check on the complete-pump-characteristics plot; the correct value of the flow is thus found by trial and error. Finally the value of torque T on the idle pump is read from the complete-pump-characteristics plot by use of the foregoing determined values of head, flow, and speed. From these values and the application of Equation (10), the time-vs.-speed curve for the idle pump can be obtained; and from the values of the flow through the idle pump and the flow delivered by the other pumps, a plot of flow through the discharge header vs. time can be made.

This method was applied to two cases of power failure, and the results were verified experimentally in order to assure the reliability of the computational procedure.

Case 1. Sudden Failure of One Pump with A.C. Power Remaining on the Other Pumps

Principal attention was given to the condition in which both a.c. and d.c. power would fail simultaneously. Consideration also had to be given to the possibility of a shaft breakage between the impeller and the flywheel. For the latter condition the reduction in the moment of inertia had to be considered.

On the basis of the method described above and the assumption of a constant discharge-header pressure at the design value, since this would give conservative results, values of $(\pi I n_0)/(30 g_c t_0 T)$ were plotted against N, as shown in Figure 4, for the condition of simultaneous failure of both a.c. and d.c. power. By a graphical integration of the area under the curve the values of time required to retard the impeller of the idle pump to given speeds were determined. Figure 5 shows curves of the flow through the idle pump and the speed of the idle pump as a function of time. Note that the flow through the idle pump decreases rapidly to zero; then the flow reverses direction and gradually increases to a maximum corresponding to the lowest point in Figure 5; and finally the flow decreases to a steady state value at the terminal or steady state reverse speed of the idle pump.

A comparison of the computed values with the experimental results showed good agreement, as indicated in Table 1.

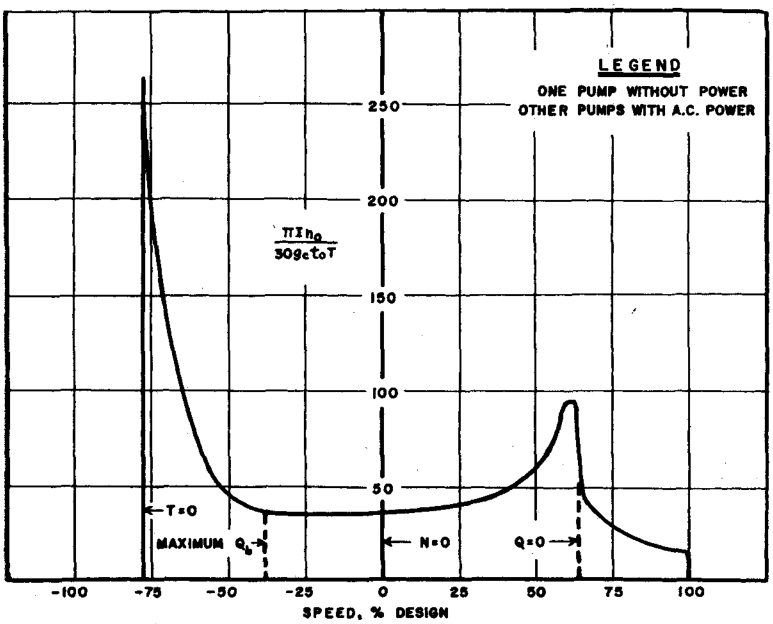

Fig. 4. Graphical integration of the equation of motion of rotating masses.

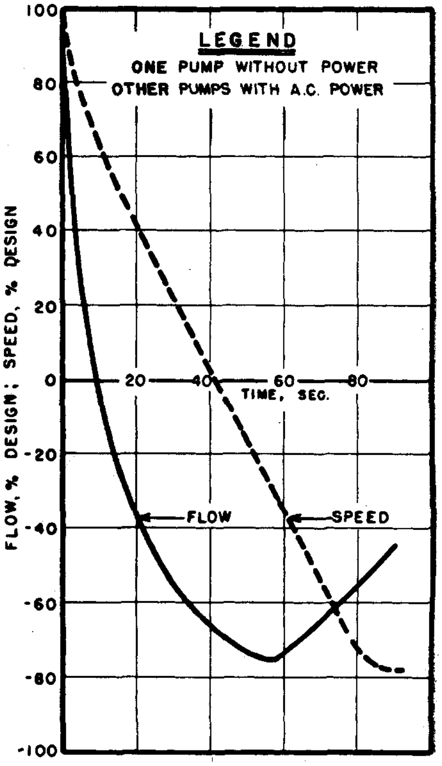

Fig. 5. Flow and speed vs. time for the idle pump.

An interesting note is the surprisingly short interval of time that would be required for the pump impeller to change direction of rotation in case the shaft should break between the impeller and the flywheel. In a practical case the moment of inertia of the impeller might well be about 1% of the combined moment of inertia of the impeller, flywheel, and motor rotors. If this were the case, the time for the impeller to change from design speed in the normal direction to about 80% of design speed in the reverse direction would be less than 1 sec. It can be seen that such a short time interval would preclude the possibility of experimental verification because of the probability of damage to the installation.

Case 2. Sudden A.C. Power Failure on All Pumps with Sudden D.C. Power Failure on the Majority of the Pumps

The computations were performed for sudden d.c. power failure on the majority of the pumps. For the retardation from 100 to 33.5% design speed, the contribution of the operating d.c. motors was neglected. From 33.5% design speed to zero the contribution of flow from the pumps with d.c. power had to be taken into account. The same procedure applied to Case 1 was used to compute the times.

Table 2 presents the results obtained both by the computation described above and by experiment for d.c. power failure on all pumps. Little difference in time would be expected for the retardation from 100 to 33.5% design speed, since comparatively little contribution to the torque would be made by the few operating d.c. motors. For retardation below 33.5% design speed the time for the computed case would be expected to be greater than for the case with d.c. power failure on all pumps.

TABLE 1. RETARDATION TIMES

Sudden Failure of One Pump with A.C. Power on Other Pumps

Flow, % design	Speed, % design	Time, sec. Power failure, a.c. and d.c. Experimental	Power failure, a.c. and d.c. Computed for constant discharge-header pressure	Broken shaft at pump (no flywheel) Computed for constant discharge-header pressure*
Zero flow				
			8.7	0.09
0	64.7			
	70.1	11		
Zero speed				
−67.7	0		41.2	0.41
	0	45		
−66.1	0 (Brake)	Experimental		
−55.1	0 (Brake)	Computed, allowing for decrease in discharge-header pressure		
Zero torque (terminal speed)				
−44.9	−78.2		89.2	0.89
−36.6	−80.2	120		

*Moment of inertia of the pump impeller assumed as 1% of the combined moment of inertia of the rotating masses.

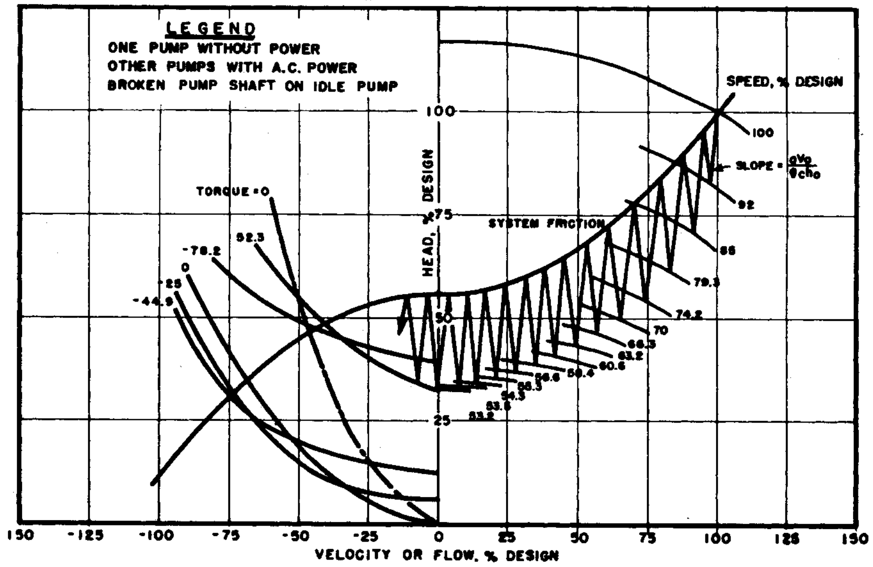

Fig. 6. Graphical procedure for hydraulic transients.

Allowance for Water-hammer Effects

This method takes into account the raising or lowering of the pressure at the pump discharge owing to surges or water hammer.

There are two methods of attacking the problem: (*a*) arithmetic and (*b*) graphical. The arithmetic method is tedious and involves a trial-and-error procedure. This method is adequately described in the literature (*10*) and will not be discussed here. However, the graphical counterpart is only briefly described in the literature (*2, 5, 9, 10, 12*), and an outline of this method will be given.

Graphical Procedure. The graphical procedure is essentially a solution of Equation (10), with Equation (1) being incorporated for the change in head on the idle pump due to surges or water hammer. A coordinate system is prepared of head vs. flow in the discharge line (Figure 6). Added then are the curve of pump performance for 100% design speed and the curve of the pumping-system friction. A *surge-characteristic* line having a slope of (aV_0/g_ch_0) [*see* Equations (1) and (11) and note that the velocity at any time is equal to QV_0] is drawn downward from the intersection of the pump-performance curve and the system-friction curve until it meets the pump-performance curve for the speed existing after a retardation time of one pipe period. This pump-performance curve for a retarded speed is computed from the pump-performance curve at 100% of design speed and the following pump relations (*11*):

$$H = H_a(N/N_a)^2 \qquad (15)$$

$$Q = Q_a(N/N_a) \qquad (16)$$

The speed is determined by application of Equation (10) for a small interval of

TABLE 2. RETARDATION TIMES

A.C. Power Failure on All Pumps and D.C. Power Failure as Indicated

Speed, % design	Time, sec. Power failure, d.c., on all pumps, experimental	Power failure, d.c., on majority of pumps, calculated
100 (initial)	0	0
33.5	33.5	29.2
0	390	453

time:

$$(T_n + T_{n+1})/2 = [(\pi I n_0)/(30 g_c t_0)] \cdot [(N_n - N_{n+1})/(\theta_{n+1} - \theta_n)] \quad (17)$$

where

$$\theta_{n+1} - \theta_n = 2L/a \quad (18)$$

This involves a trial-and-error solution to determine with accuracy the point of intersection. However, where the interval of time is very small and the moment of inertia is large, Equation (17) can be approximated to serve as a practical check on the other methods and to eliminate the trial-and-error solution by replacing $(T_n + T_{n+1})/2$ with T_n.

The *reflected-surge-characteristic* line is then drawn upward with a slope of $-aV_0/g_c h_0$ to the system-friction curve. The process is repeated until the point of zero flow is reached. After this point the procedure is the same except that the plot of complete pump characteristics is used instead of the normal pump-performance curve and the pump relations. From the plot of complete pump characteristics, data are obtained for the pump with reverse flow and positive speeds. Maximum reverse speed (negative speed at zero torque) is approximately the negative speed at which the surge-characteristic line crosses the zero-torque curve when being drawn to the system-friction curve or the pump-performance curve for the next highest negative speed.

After the point of zero flow is reached for a system with a very small pipe period and a large moment of inertia of the rotating elements, there may be a large number of pipe periods of time to be considered before the point of maximum reverse speed (zero torque) is reached. An approximation can be made for the number of intervals of time from zero flow to zero speed and from zero speed to maximum reverse speed by use of an approximation of Equation (10):

$$T_{avg} = [(\pi I n_0)/(30 g_c t_0)] \cdot [(N_n - N_{n+1})/(\theta_{n+1} - \theta_n)] \quad (19)$$

an inspection of the construction shown in Figure 6, and the plot of complete pump characteristics, Figure 3. First an average head can be obtained by inspection of the construction and then, by use of this average head and the points of zero flow and zero speed, or zero speed and estimated maximum reverse speed, an average value of the torque is read from the plot of complete pump characteristics. Thus the difference in time is obtained from Equation (19) and the number of pipe periods obtained by dividing this time by the length of time of 1 pipe period.

Figure 6 is the construction for the case of a possible sudden shaft failure on one pump and a.c. power on the other pumps (*see* case 1). For the idle pump to decelerate from 100% of design speed to a speed corresponding to zero flow, a time of 14 pipe periods would be required. From zero flow to zero speed, the time was approximated, because of the large number of pipe periods required, as follows. Inspection of Figures 3 and 6 shows that $30\% < H < 55\%$, $0\% < N < 55\%$, and $T_{avg} \cong 25\%$; thus from Equation (19) $\Delta\theta = 0.27$ sec., or 27 pipe periods. From zero speed to maximum reverse speed, the same procedure as above was applied: $30\% < H < 45\%$, $-80\% < N < 0\%$, and $T_{avg} \cong 35\%$; thus $\Delta\theta = 0.29$ sec., or 29 pipe periods.

A summary of times is as follows:

	Time, sec.
100% Speed to zero flow	0.14
Zero flow to zero speed	0.27
100% Speed to zero speed	0.41
Zero speed to maximum reverse speed	0.29
100% Speed to maximum reverse speed	0.70

These values compare favorably with those given in Table 1. Thus it is indicated again that the simple graphical integration of the equation of motion of a rotating system, neglecting water-hammer effects, is of sufficient accuracy for these problems on unsteady state flow.

Charts. For the quick estimation of various hydraulic transients in pumping systems, Parmakian (*9*) has presented charts based upon experience gained on large pumping installations of the United States Bureau of Reclamation. However, it is believed that at those installations the system friction was negligible compared with the pumping head, and therefore for systems with considerable friction the charts would indicate times appreciably shorter than those computed by the methods previously discussed.

For the case of a possible shaft failure (*see* case 1), the times obtained from the charts are from 15 to 50% of those given in Table 1.

CONCLUSIONS

1. The graphical integration of the equation of motion of a rotating system, water-hammer effects neglected, is a rapid method of computing hydraulic transients in a pumping system.

2. The periods given by the charts prepared by Parmakian were considerably shorter than those obtained by computation. The charts probably do not apply to systems having high friction losses.

3. The agreement was good between the computed and experimental values of the hydraulic transients.

4. This analysis led to the decision not to install check valves but to employ brakes or nonreversing clutches to protect the systems from the consequences of running the pumps as turbines.

NOTATION

- a = velocity of wave propagation, ft./sec.
- b = pipe-wall thickness, ft.
- D = pipe inside diameter, ft.
- E = modulus of elasticity of pipe wall material, lb. force/sq. ft.
- g_c = conversion factor, 32.2 (lb./lb. force)(ft./sec.^{2})
- H = h/h_0
- h = head, ft.
- I = moment of inertia = WR^2, lb.-ft.^{2}
- k = bulk modulus of elasticity of the fluid, lb. force/sq. ft.
- L = length of pipe from the location of flow stoppage to the location of total wave reflection (used in water-hammer computation), ft.
- N = n/n_0
- n = rotational speed, rev./min.
- Q = q/q_0
- q = rate of flow, cu. ft./sec.
- R = radius of gyration, ft.
- T = t/t_0
- t = torque, lb. force-ft.
- V = velocity, ft./sec.
- ΔV = change in velocity, ft./sec.
- W = weight of rotating element, lb.
- θ = time, sec.
- ρ = fluid density, lb./cu. ft.
- τ = $2L/a$ = pipe period, sec.
- ω = rotational speed, radians/sec.

Subscripts

- 0 = initial steady state or design condition
- a = normal operation at 100% design speed
- avg = average
- b = backflow
- n = any point
- wh = water hammer

LITERATURE CITED

1. Allievi, L., "Theory of Water-Hammer," translated by E. E. Halmos, Riccardo Garroni, Rome (1925).
2. Angus, R. W., *Proc. Inst. Mech. Engrs.* (*London*), **136**, 245 (1937).
3. ———, "Hydraulics for Engineers," Pitman and Sons, Toronto (1943).
4. "A.S.M.E. - A.S.C.E. Symposium on Water Hammer, 1933," published by Am. Soc. Mech. Engrs.
5. Kerr, S. Logan, *Trans. Am. Soc. Mech. Engrs.*, **72**, 667 (1950).
6. Knapp, R. T., *loc. cit.*, **59**, 683 (1937).
7. *Ibid.*, **63**, 251 (1941).
8. Parmakian, John, *Memo.*, U. S. Bur. Reclamation, Denver, Colo. (April 26, 1949).
9. ———, *Trans. Am. Soc. Mech. Engrs.*, **75**, 995 (1953).
10. Rich, G. R., "Hydraulic Transients," McGraw-Hill Book Company, Inc., New York (1951).
11. Stepanoff, A. J., "Centrifugal and Axial Flow Pumps," John Wiley and Sons, New York (1948).
12. ———, *Trans. Am. Soc. Mech. Engrs.*, **71**, 515 (1949).

Presented at Nuclear Engineering and Science Congress, Cleveland