CHAPTER 11

PUMPS IN PIPE SYSTEMS

The designers of liquid conveyance systems are frequently faced with a pump selection problem. While vendors of pumps and pumping appurtenances are generally quite helpful in the selection process, it is better to be well informed on pumps and their operating characteristics, particularly under transient conditions. For this reason Chapter 2 presented some fundamental elements of pump theory and operation; this chapter will address the issue of transients, building on the knowledge of similarity relationships from Chapter 2. We will restrict our coverage to centrifugal, turbine, and axial-flow pumps. Positive displacement pumps are not considered.

11.1 PUMP POWER FAILURE RUNDOWN

The sudden loss of energy to a pump can be caused by an unexpected power failure or simply because an individual has switched off the power. In either case the rotating pump impeller begins to decelerate with the pressure dropping on the discharge side of the pump and rising on the suction side (if it is an inline booster pump configuration). The resultant transient may quickly lead to column separation with ensuing hard-to-predict consequences, including cavity collapse or exceedingly high pressures, perhaps caused by the closure of a check valve. Whatever the cause, it can be very important to be able to simulate this rather common occurrence to determine whether dangerous pressures develop.

As the pump slows down after power failure, its head vs. discharge and torque vs. discharge characteristics change. It is customary to assume as the pump speed changes that the pump characteristics at any speed can be found by using the similarity relations that are presented in Chapter 2 for homologous pumps. While the changes in pump torque are important in the rundown process, we will first concentrate on how the pump head itself varies with discharge.

The pump characteristics at various speeds can be displayed as shown in Fig. 11.1. The solid line labeled N_0 is the pump characteristic curve during steady-state conditions, while the similarly-shaped dashed lines represent that characteristic curve at successively lower speeds. The position of each dashed characteristic line can be calculated from the steady-state line by using the similarity rules from Chapter 2. Noting that the diameter of the slowing pump is a constant, we can incorporate it into the similarity constant to yield a form of the similarity equations which applies directly to pump power failure rundown:

$$\frac{Q}{N} = \text{constant} \tag{11.1}$$

$$\frac{h_p}{N^2} = \text{constant} \tag{11.2}$$

$$\frac{T}{N^2} = \text{constant} \tag{11.3}$$

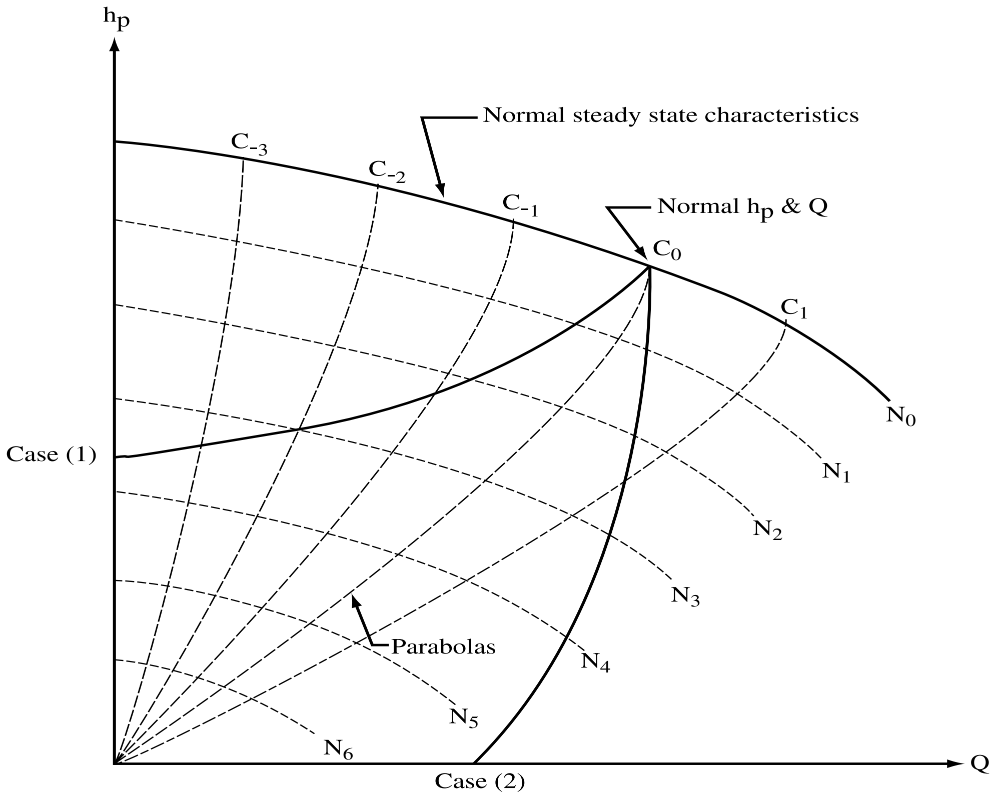


Figure 11.1 Multi-characteristics for a given pump at various speeds.

The curves at successive speeds N_i can be created by selecting a range of values for Q_0 and h_{p_0} at a set of points C_{-3}, C_{-2}, etc. along the original characteristic curve and calculating the corresponding values for each other speed with the following equations:

$$Q_i = Q_0 \frac{N_i}{N_0} \tag{11.4}$$

$$h_{p_i} = h_{p_0} \left(\frac{N_i}{N_0} \right)^2 \tag{11.5}$$

In fact, by using the same Q_0 and h_{p_0}, say at point C_1, and varying N, we can generate a large set of corresponding points, each on a different characteristic curve. These points all lie on a parabola passing through C_1 and the origin. If the same procedure is followed for points C_{-3}, C_{-2}, etc. on the original curve, then a set of pump characteristic lines can be drawn for a set of speeds, and in principle can be drawn for all speeds. Figure 11.1 shows one set of parabolas drawn through the origin.

As the speed of the pump decreases, a path develops on the characteristic diagram of Fig. 11.1 which traces the changes of pump head and discharge as the rundown progresses. However, this trace does not follow a particular parabolic curve; instead the path is determined by the pump and motor rotational inertia and the back pressure exerted by the water in the pipe on the pump impeller. Two typical examples are shown in Fig. 11.1:

Case (1) occurs when the static lift of the pipeline is high, and the line is relatively short. In this case the inertia of the water in the pipeline is relatively small, and gravity helps to decelerate the flow. As a consequence, the discharge through the pump drops to zero rather quickly while a positive head across the pump still exists. Then flow backward through the pump occurs unless a check valve has been installed in the line.

Case (2) occurs when the pipeline is relatively long, and a large portion of the head that has been generated by the pump is needed to overcome the friction loss in the line. When power fails in this case, the large inertia of the moving fluid prevents the flow from decelerating rapidly. The rotation rate of the pump also decreases more slowly, and the head across the pump drops to zero before the discharge does. At this point the flow will either continue to flow forward through the pump, doing work on the pump and causing it to "windmill," or else the flow goes around the pump in a bypass line.

There are actually four possible flow configurations through the pump:

(1) Flow is forward through the pump, and the pump rotates forward.
(2) Flow is in the reverse direction while the pump is still rotating forward (generally of short duration).
(3) Flow is in the reverse direction while the pump also rotates backwards.
(4) Flow is in the forward direction while the pump rotates backwards (also of short duration).

The actual occurrence of any of these situations depends on the inertia of the pump and motor and on the existence of check valves, bypasses and other appurtenances. Unfortunately, the data needed to simulate these conditions are usually only available from manufacturers for the first situation. Even in this situation, the data are not available for Case (2) in Fig. 11.1 when there is a head *loss* through the pump. If the pump is expected to operate in any of the other three modes, then additional information must be sought either through model tests of that pump or by a study of data from tests of similar pumps.

We will analyze a common pump power failure situation, assuming there is a check valve in the pump discharge line. If the pump is a booster pump, we will assume there is a low-loss bypass line around the pump station. This will permit us to complete an analysis while using only information that is commonly available from pump manufacturers.

11.1.1. SETTING UP THE EQUATIONS FOR BOOSTER PUMPS

Along each of the parabolic curves passing through C_{-3}, C_{-2}, etc. in Fig.11.1, the values of Q/N and h_p/N^2 are constant. Hence we can represent all of the pump characteristic behavior in Fig. 11.1 by a single plot of Q/N vs. h_p/N^2. The same reasoning applies to the torque description where a single plot of Q/N vs. T/N^2 will suffice. A typical example of each curve is shown in Fig. 11.2. The curves can be constructed by selecting h_p and Q pairs from the manufacturer's curves, dividing by N and N^2, respectively, and plotting the results. To see how this works, let's look at an example.

Consider a booster pump station in the interior of a line (see Fig. 11.3). There is a check valve on each pump discharge line, and there is a bypass line around the pump station which also has a check valve in it. All pumps are assumed to experience power failure simultaneously. The appropriate equations are the following:

Suction side C^+
$$V_{P_s} = C_1 - C_2 H_{P_s} \tag{11.6}$$

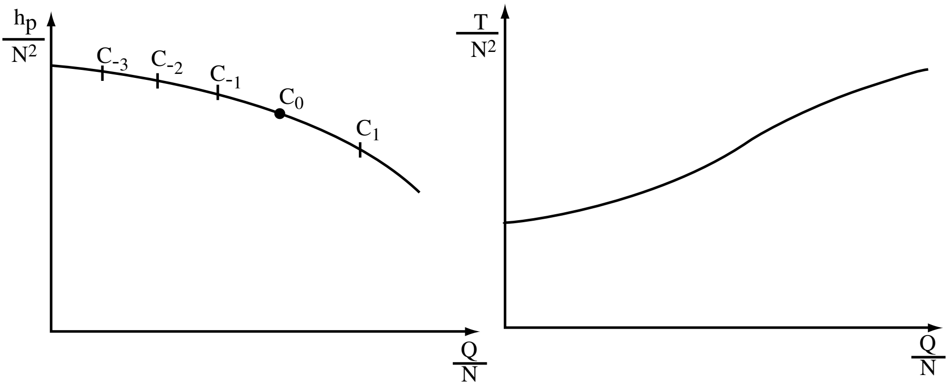


Figure 11.2 Typical h_p/N^2 and T/N^2 curves for a pump.

Discharge side C^- $$V_{P_d} = C_3 + C_4 H_{P_d} \tag{11.7}$$

Conservation of mass $$V_{P_s} A_s = V_{P_d} A_d \tag{11.8}$$

Work-energy $$H_{P_s} + h_p = H_{P_d} \tag{11.9}$$

Pump characteristic $$h_p = f(Q) \text{ or } f_1\left(V_{P_d}\right) \text{ or } f_2\left(V_{P_s}\right) \tag{11.10}$$

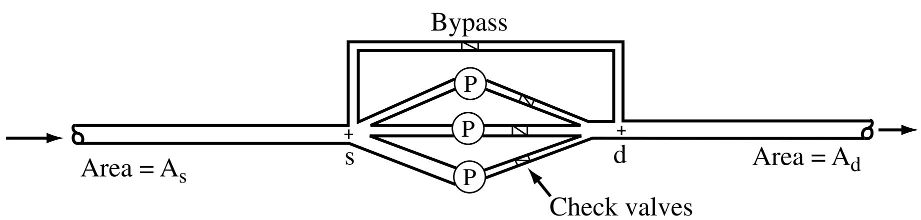


Figure 11.3 A typical parallel pump booster configuration.

This set of equations contains the unknown value of N plus five additional unknowns. If we presume that N can be found before we begin the solution for the other five unknowns, then we can proceed to seek a solution.

In the work-energy equation, Eq. 11.9, we find only the head increase across the pump, as is given by the pump characteristic diagram. If there are significant losses in the pump discharge column, discharge head, check valve or isolation valve, then adjustments to h_p must be made. That is, h_p must be reduced by the amount of the hydraulic losses for a given discharge. Consequently, we must redraw the pump characteristic diagram, corrected for the head losses occurring for each discharge. Because these losses are typically of the form $K_L V^2/2g$, the loss coefficients for the individual local losses can be summed appropriately for the full range of pump discharge and then applied to reduce the pump head. Since h_p is the sum of the head increases across each stage of a multistage pump, the local losses for a given discharge must be divided by the number of stages before computing the head per stage and the pump characteristic diagram is redrawn. This step is necessary because the pump characteristic curve for only one stage is entered into the computer program and is then internally multiplied by the number of stages. One might ask why a head loss term was not included in Eq. 11.9 so one could then proceed in a straightforward manner. The answer can be seen in Section 11.1.3. In that section we linearize the pump characteristic curve to avoid parabolic or higher-order interpolation techniques. To reintroduce a quadratic equation now would defeat this strategy. In short, we adjust the pump characteristic diagram for local losses in order to retain its subsequent linear representation.

It is apparent that we need a representation for Eq. 11.10 that can be combined with the other four equations. In Chapter 9 we modeled the pump curve with a parabolic equation. This approach was restrictive in that it worked well only for characteristic curves which were already nearly parabolic in shape. We will now follow a much more general approach and represent the Q/N vs. h_p/N^2 curve by a series of straight-line segments. At any point the curve is then a straight line valid over a limited range of Q/N. The details of this process will follow after we find the current speed N.

11.1.2. FINDING THE CHANGE IN SPEED

To this point we have assumed that the new speed is known at the time when the new head and velocity values are to be computed. The change in speed is found by calculating the decelerating torque and using rotational dynamics to find ΔN. The rotating portions of the pump — shaft, motor, and impeller — have a rotational moment of inertia I. Normally the motor is by far the largest contributor. However, the rotation of the water-filled impeller and the pump shaft must be included. Values for these elements must be obtained from the manufacturer or estimated by comparison with values for similar pumps whose moments of inertia I are known. In the United States pump manufacturers usually give the value of I as Wr^2 in units of lb-ft^2. This is the value the computer program expects. Should the value of I for a particular motor not be immediately available, then the following formula (Thorley, 1991), adjusted to U.S. units, may be used as an estimate:

$$I = 1818\left(\frac{HP}{N}\right)^{1.48} \tag{11.11}$$

Here I is in lb-ft^2, HP is in horsepower, and N is in rev/min or rpm.

Under steady-state conditions the driving torque of the motor is balanced by the resisting torque exerted by the water on the impeller vanes. When power fails, the driving torque disappears and the resisting torque decelerates the pump. This deceleration is described by

$$T = I\alpha = I\frac{d\omega}{dt} = \frac{2\pi}{60}I\frac{dN}{dt} \tag{11.12}$$

where N is again the speed in rev/min, and I is the total rotational moment of inertia of the rotating parts.

To find the change in speed which occurs over a time increment Δt, we now integrate Eq. 11.12:

$$\int dN = \frac{60}{2\pi I}\int Tdt \tag{11.13}$$

Since the functional relation for T is not known, we choose to keep Δt small and let T be constant over Δt at its known value at the previous instant in time. This approximation is a good one because torque normally does not change much over even the full discharge range. The new rotational speed can then be calculated as

$$N(t+\Delta t) = N(t) - \frac{60}{2\pi I}T(t)\Delta t \tag{11.14}$$

The first change in speed will be calculated from the steady-state values of N_0 and T_0. Subsequent torque values are interpolated from the table of Q/N vs. T/N^2 using the just-computed values of Q. We can now proceed with the solution of Eqs. 11.6 through 11.10 for the new head and velocity.

11.1.3. SOLVING THE EQUATIONS

The first step in solving the equations is to represent Eq. 11.10 as a linear function over a finite range of Q/N. Figure 11.4 shows h_p/N^2 vs. Q/N as a sequence of linear segments. The linear equation over the segment is

$$\frac{h_p}{N^2} = N_{st}\left[\left(\frac{\left(h_p/N^2\right)_A - \left(h_p/N^2\right)_B}{(Q/N)_A - (Q/N)_B}\right)\frac{Q}{N} - \left(\frac{\left(h_p/N^2\right)_A - \left(h_p/N^2\right)_B}{(Q/N)_A - (Q/N)_B}\right)\left(\frac{Q}{N}\right)_B + \left(\frac{h_p}{N^2}\right)_B\right] \tag{11.15}$$

in which the values of h_p/N^2 and Q/N are for one stage of the pump, and N_{st} is the number of pump stages. To simplify the algebra, we rewrite Eq. 11.15 as

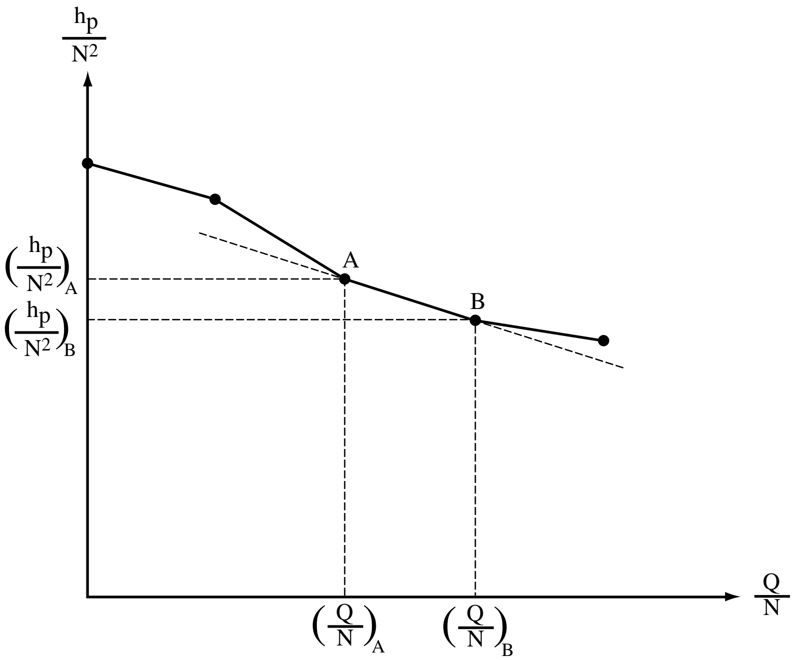


Figure 11.4 Piecewise linear representation of h_p/N^2 vs. Q/N.

$$\frac{h_p}{N^2} = N_{st}\left[C_7\frac{Q}{N} + C_8\right] \tag{11.16}$$

Now the simultaneous solution of the five equations for the pipeline velocity on the discharge side of the pump station produces

$$V_{P_d} = \frac{\dfrac{C_1}{C_2} + N_{st}N^2C_8 + \dfrac{C_3}{C_4}}{\dfrac{1}{C_4} + \dfrac{A_d}{C_2A_s} - \dfrac{N_{st}NA_dC_7}{N_{pu}}} \tag{11.17}$$

in which N_{pu} is the number of pumps in parallel. If $V_{P_d} > 0$, Eqs. 11.6 through 11.10 can be used to find the remaining unknowns. However, if $h_p < 0$, then we must open the bypass line by setting $h_p = 0$ and $H_{P_s} = H_{P_d}$; then we must recompute the velocity from the following two equations:

$$V_{P_d} = \frac{C_1C_4 + C_2C_3}{C_2 + C_4\dfrac{A_d}{A_s}} \tag{11.18}$$

$$V_{P_s} = \frac{A_d}{A_s}V_{P_d} \tag{11.19}$$

Now Eqs. 11.6 and 11.7 can be used to determine H_{P_s} and H_{P_d}.

If the solution of Eq. 11.17 yields a negative velocity, we must set both V_{P_s} and V_{P_d} to zero and use Eqs. 11.6 and 11.7 to compute H_{P_s} and H_{P_d}. Finally, we must compute Q/N and verify that we are indeed within the interval between A and B in Fig. 11.4. If not, we must recompute C_7 and C_8 and repeat the solution process.

Example Problem 11.1

To examine the effect of booster pump power failure, we will place a four-pump station in the interior of a 45,000-ft pipeline. The 30-in diameter pipeline is constructed of welded steel with a friction factor of 0.013 and a wave speed of 3590 ft/s. The line extends between two reservoirs, and the booster station is 15,000 ft downstream from the first reservoir.

The pumps are three-stage Ingersoll-Dresser 15H277 turbine pumps with 11.83-inch impellers having pump characteristics shown in Appendix B. For each pump and motor unit Wr^2 is approximately 475 lb-ft^2. To set up data tables for pump performance, we select six data points along the Q-axis, $Q =$ 0, 1000, 2000, 3000, 4000, and 4500 gal/min. We then read the corresponding h_p and bhp values for each Q and enter them into the input data file.

The program determines the steady discharge, so no preliminary hydraulic computations are needed. One need only select an accuracy standard for the iterative process. In this case an accuracy of 0.50 gal/min was chosen.

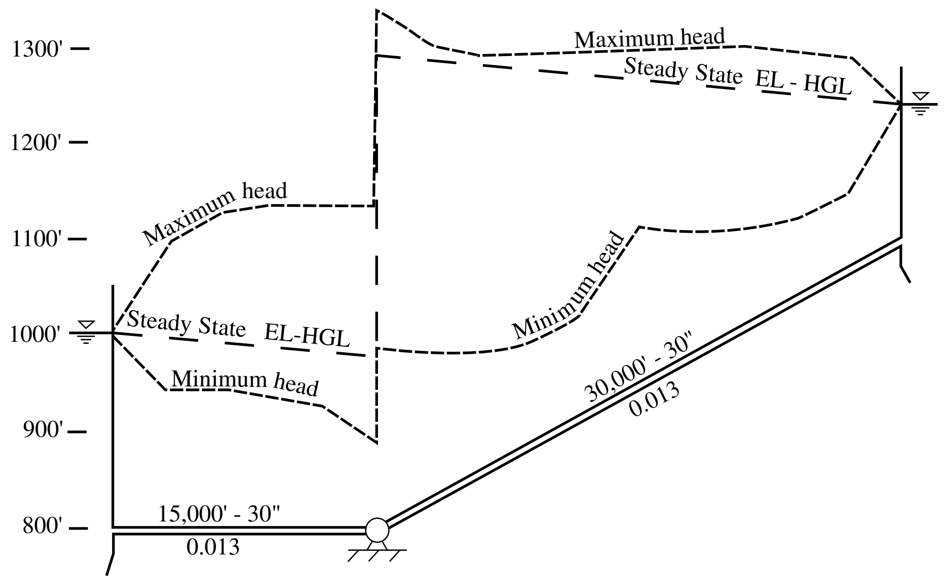
We have also elected to obtain additional output detail at two points. The first is at the suction side of the pump; the second is at the discharge side. This information is read by the PGRAPH subroutine.

The input data file follows:

```
DEMONSTRATION OF PROGRAM NO. 4 - INPUT DATA FILE "EP111.DAT"
BOOSTER PUMP POWER FAILURE, FOUR INGERSOLL-DRESSER 15H277 3-STAGE PUMPS
&SPECS NPIPES=2,NPARTS=5,IOUT=10,HRESUP=1000.,HRESDN=1240.,
       ZEND=1100.,HATM=30.,QTRY=0.,QACC=0.50,TMAX=60.,
       PFILE=T,HVPRNT=T,PPLOT=T,GRAPH=T,RERUN=F/
1  30.  15000.  0.013  3590.  800.
2  30.  30000.  0.013  3590.  800.
&PUMPS NPUMPS=4,NSTAGE=3,IPUMP=1,RPM=1775.,WRSQ=475.,
```

```
        QN=0.,1000.,2000.,3000.,4000.,4500.,
        HNSQ=129.,127.5,121.,103.5,67.5,0.,
        TNSQ=50.,58.,78.,92.,97.,80./
&GRAF NSAVE=2,IOUTSA=1,PIPE=1,2,0,0,NODE=999,1,0,0/
```

The booster pump power failure program PROG4 is used to analyze the problem. The source and executable programs are on the CD. The following plot of extreme pressure values along the pipeline is one of the primary results to come from this analysis. We observe that high heads occur on the suction side of the pump, as well as low heads on the discharge side. No column separation occurs in this case.



* * *

11.1.4. SETTING UP THE EQUATIONS FOR SOURCE PUMPS

We will follow the same general procedure as for booster pumps. There is a check valve on each pump discharge line. There is also a low-friction, essentially frictionless, bypass line with a check valve around the pump station to supply the pipeline if the pump head should drop to zero during the transient. We again assume all pumps fail simul-taneously.

Four equations are needed to model the pump behavior:

Discharge side C^-
$$V_{P_d} = C_3 + C_4 H_{P_d} \tag{11.20}$$

Conservation of mass
$$N_{pu} Q = V_{P_d} A_d \tag{11.21}$$

Work-energy
$$H_{sump} + h_p = H_{P_d} \tag{11.22}$$

Pump characteristic
$$\frac{h_p}{N^2} = N_{st}\left[C_7 \frac{Q}{N} + C_8\right] \tag{11.23}$$

Here N_{pu} is the number of pumps in parallel, and H_{sump} is the pump sump elevation.

When N is found as in the previous case, we have only four unknowns here. With Eqs. 11.20-11.23 we can proceed with a solution. In this case we find

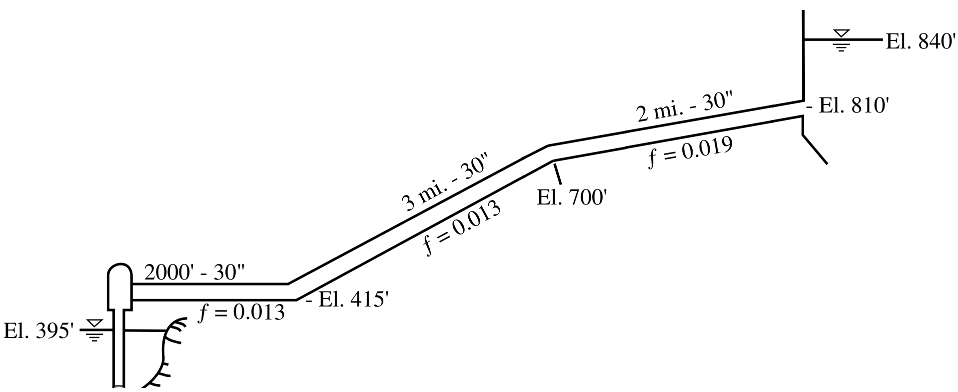
$$H_P = \frac{H_{sump} + \dfrac{N_{st}N}{N_{pu}} C_7 C_3 A_d + N_{st} N^2 C_8}{1 - \dfrac{N_{st}N}{N_{pu}} C_7 C_4 A_d} \tag{11.24}$$

If H_P is less than H_{sump}, then the pump bypass is open, and we must then equate H_P to H_{sump} and use Eq. 11.20 to compute the velocity. If the velocity were found to be negative, then we would set $V_{P_d} = 0$ and compute H_P from Eq. 11.20.

The next example problem applies this analysis to a source pump configuration subjected to a power failure.

Example Problem 11.2

Four pumps in parallel are used to pump approximately 12,000 gal/min from a reservoir at elevation 395 ft to a storage reservoir at elevation 840 ft, as shown below (not to scale). The pump discharge lines have check valves and lead into a manifold which in turn supplies the 30-in welded steel pipeline.



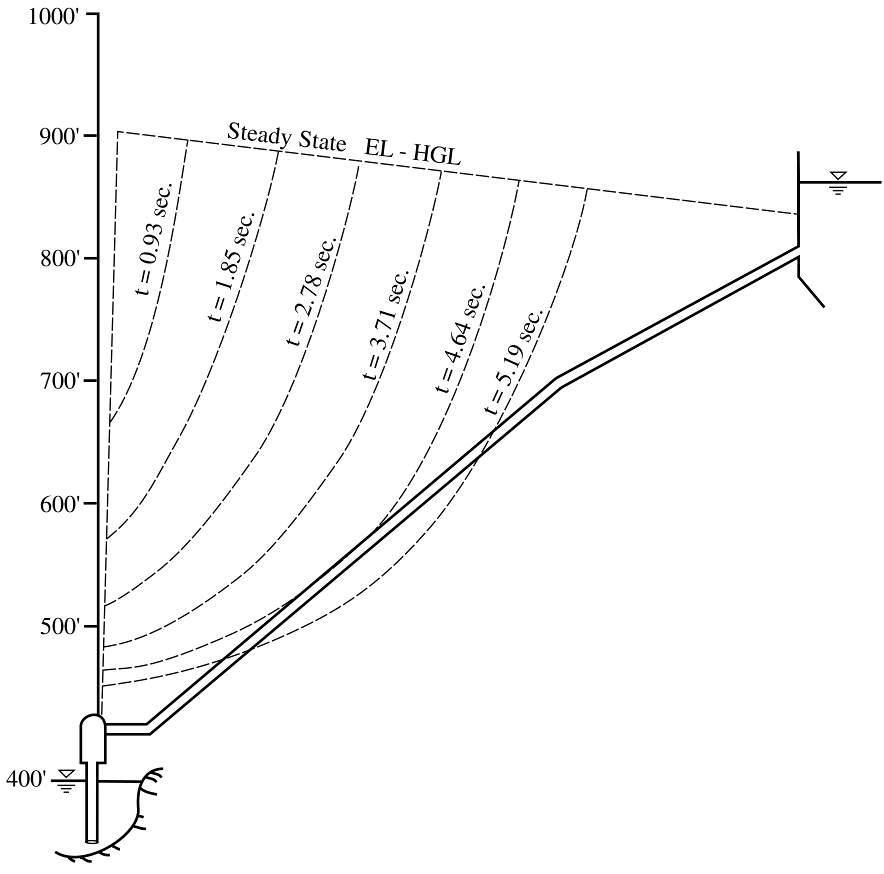
The pipeline extends 2000 ft horizontally from the pump station at an elevation of 415 ft. It then slopes upward for three miles to elevation 700 ft. The remaining two miles of pipe are reinforced concrete and slope gradually upward to enter the storage reservoir at elevation 810 ft. The friction factor and the wave speed for the steel pipe are 0.013 and 3590 ft/s, respectively. For the concrete pipe these values are 0.019 and 3490 ft/s. The pumps are the same Ingersoll-Dresser 15H277 turbine pumps used in Example Problem 11.1, except they now have five stages. Refer to the previous Example Problem for the pump characteristics. The 11.83-in impeller will be used. The total rotary moment of inertia of each pump and motor unit is 475 lb-ft^2.

Find the consequences of pump power failure.

This is a source pump configuration, so PROG3 will be used to determine the effect of pump power failure. The input data file for this program is shown below. This program also uses the subroutine PGRAPH which makes it possible to generate additional tables of output data, printer plots, and data files for external plot programs.

```
DEMONSTRATION OF PROGRAM NO. 3, INPUT DATA FILE "EP112.DAT"
SOURCE PUMP FAILURE, 4 INGERSOLL-DRESSER 15H277 5-STAGE PUMPS
&SPECS NPIPES=3,NPARTS=3,IOUT=5,HRES=840.,HSUMP=395.,
       ZEND=810.,HATM=33.,QACC=0.50,TMAX=10.,DTNEW=0.,
       PFILE=T,HVPRNT=T,PPLOT=T,GRAPH=T,RERUN=F/
1  30.   2000.  0.013  3590.  415.
2  30.  15840.  0.013  3590.  415.
3  30.  10560.  0.019  3490.  700.
&PUMPS NPUMPS=4,NSTAGE=5,RPM=1775.,WRSQ=475.,
       QN=0.,1000.,2000.,3000.,4000.,4500.,
       HNSQ=129.,127.5,121.,103.5,67.5,0.,
       TNSQ=50.,58.,78.,92.,97.,80./
&GRAF NSAVE=3,IOUTSA=1,PIPE=1,2,3,0,NODE=1,1,1,0/
```

The results show that column separation occurs about 5 sec after power failure. At that time the program execution ends because this program is not prepared to analyze vapor cavities. A plot of the EL-HGL for times prior to column separation is presented.



* * *

11.2 PUMP STARTUP

Pressure surges caused by pump startup can be difficult to predict, particularly if air is initially in the system. This air may be in the pump discharge column or located at high spots along the pipeline. Whatever the case, the air-exhaustion process must be simulated in order to approximate the pressures which could occur during startup.

On the other hand, if the pump startup is associated with a recent power failure shutdown, then the restart sequence is important in controlling the pressures. Because the power failure may have caused air to be drawn into the system, we once again must confront the problem of modeling the air removal process from the pipeline.

To simulate this sequence of events, PROG3 has been modified to examine source pump power failure followed by a restart procedure. And PROG8 also can simulate the column separation that may occur during the power failure phase of the sequence and the subsequent air removal process that takes place during the startup phase. Both column separation and air exhaustion are simulated in the simplest manner. The air and vapor cavities are concentrated at the nodes; the pressures at these nodes are set at atmospheric and vapor pressure, respectively, so long as an air bubble or vapor cavity exists. The calculation of pressures that are caused by vapor cavity closure follows the procedure detailed in Section 10.7. Pressures resulting from the elimination of an air bubble through an air valve are treated in the same way. While this simulation procedure is not sophisticated, there are enough uncertainties in the understanding of vapor cavity formation and collapse, the location and movement of vapor cavities and air bubbles, and the extent to which air can be removed from a pipe, that a more thorough analysis is unwarranted.

PROG8 allows air-vacuum valves to be located at pipe junctions along the pipeline which can admit and exhaust air. Vapor cavity formation and collapse are modeled. A bypass line with friction is provided around the pump station to supply water when the pump head drops to zero. The program calculates the steady-flow situation, simulates the power failure rundown and then permits us to explore the behavior of a restart procedure to bring pumps back on line with various ramp times. To simulate power failure without restart, the restart time is simply chosen to be greater than the total execution time for the program. To look at only pump startup, let the program simulate a power failure and then delay the restart until the flow has stabilized. *Always keep in mind that the simulation of vapor cavity behavior and the removal of air from pipelines is a very uncertain process, and the results of such analyses should be viewed very conservatively.*

We will now look further at the pipeline and pumping configuration of Example Problem 11.1 to observe some effects of column separation and pump restart.

Example Problem 11.3

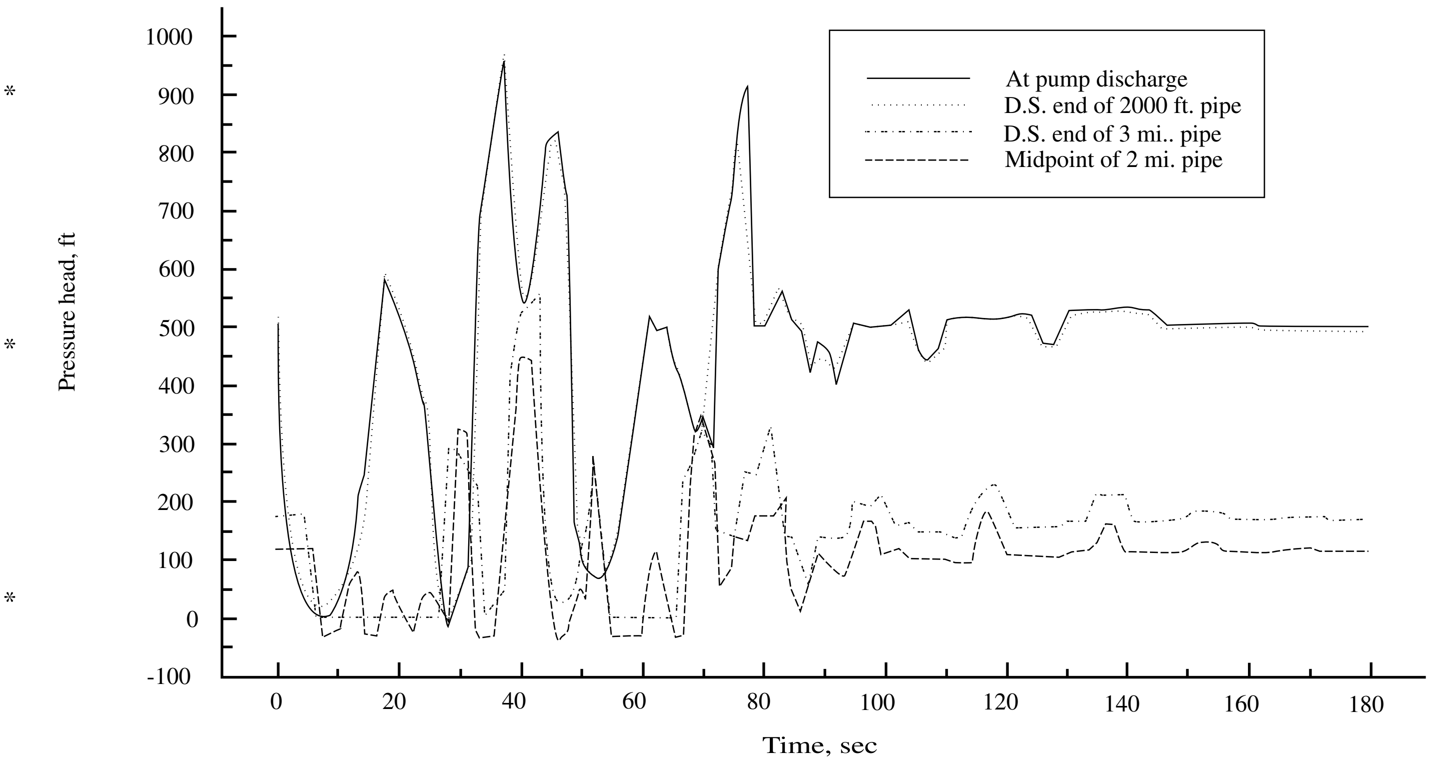
The description in Example Problem 11.2 still applies. In addition, we will restart the pumps at 20-sec intervals, beginning 60 sec after power failure. We will ramp up each pump from 300 rev/min to the full speed of 1775 rev/min in 10 sec. We assume there are air-vacuum valves at the two interior junctions. The loss coefficient K_L for the 24-inch bypass line is 2.5.

The input data file for this analysis follows:

```
DEMONSTRATION OF PROGRAM NO. 8, INPUT DATA FILE "EP113.DAT"
PUMP POWER FAILURE AND RESTART, SAME CONFIGURATION AS EP11.2
&SPECS NPIPES=3,NPARTS=3,HRES=840.,HSUMP=395.,ZEND=810.,
       HATM=33.,QACC=0.50,TMAX=180.,DTNEW=0.,DB=24.,KLB=2.5,
       PFILE=F,HVPRNT=T,PPLOT=F,GRAPH=T,RERUN=F/
1  30.   2000.  0.013  3590.  415.  0
2  30.  15840.  0.013  3590.  415.  1
3  30.  10560.  0.019  3490.  700.  1
&PUMPS NPUMPS=4,NSTAGE=5,RPM=1775.,RPMZ=1775.,WRSQ=475.,
```

```
NSTART=4,QN=0.,1000.,2000.,3000.,4000.,4500.,
       HNSQ=129.,127.5,121.,103.5,67.5,0.,
       TNSQ=50.,58.,78.,92.,97.,80./
&RESTART TSTART(1)=60.,TSTART(2)=80.,TSTART(3)=100.,
       TSTART(4)=120.,TRAMP(1)=10.,TRAMP(2)=10.,TRAMP(3)=10.,
       TRAMP(4)=10.,RPMSTRT(1)=300.,RPMSTRT(2)=300.,
       RPMSTRT(3)=300.,RPMSTRT(4)=300.,RPMEND(1)=1775.,
       RPMEND(2)=1775.,RPMEND(3)=1775.,RPMEND(4)=1775./
&OUTCTRL IOU=1000,    2,    1000,    2,    1000,    1000,
        TIOU= 0.,   60.,    60.5, 179.7,   180.,    400./
&GRAF NSAVE=4,IOUTSA=4,PIPE=1,2,3,3,NODE=1,1,1,7/
```

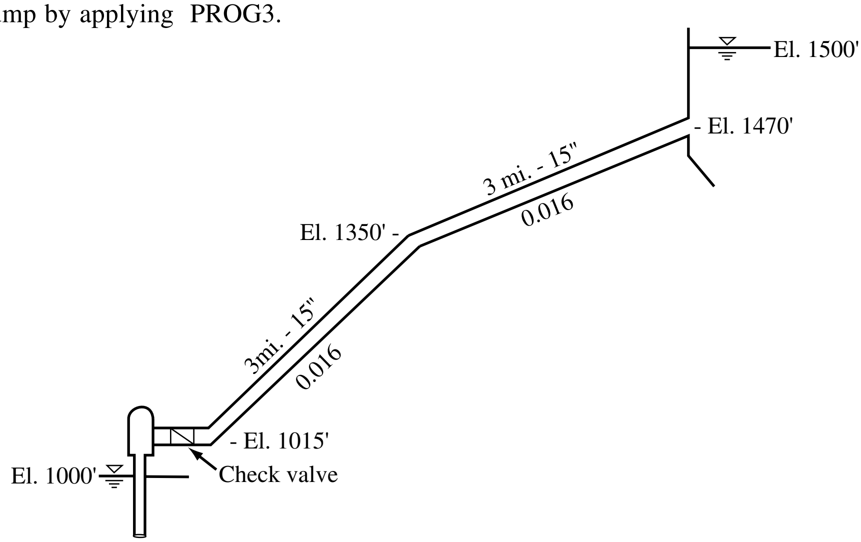
Example Problem 11.3
Pump power failure rundown and restart



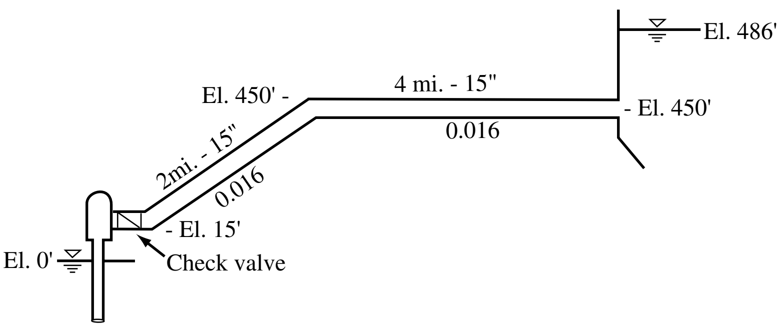
11.3 PROBLEMS

11.1 A seven-stage Ingersoll-Dresser 15H277 pump (see Appendix B) runs at 1775 rev/min with 11.83-in impellers. Water is pumped between two reservoirs with a 500-ft lift. The rotating parts have $Wr^2 = 510$ lb-ft^2. The wave speed is 3500 ft/s.

Investigate the possibility of column separation occurring after power failure to the pump by applying PROG3.



11.2 The pump of Problem 11.1 is used in the system shown below. If Wr^2 has been reduced to 427 lb-ft^2, determine if, when, and where column separation occurs after power failure to the pump. Use PROG3 for your analysis.



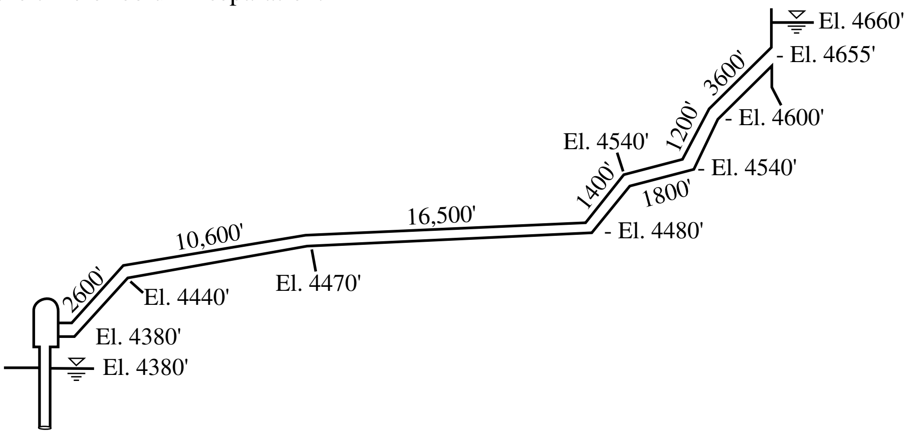
11.3 A 24-in pretensioned reinforced concrete pipe ($f = 0.0135$) extends 37,700 ft between two reservoirs. The wave speeds for the three separate pressure zones in the pipeline are as follows:

First	2,600 ft:	$a = 3550$ ft/s
Middle	10,600 ft:	$a = 3400$ ft/s
Last	24,500 ft:	$a = 3270$ ft/s

The pump station is equipped with four three-stage pumps turning at 1760 rev/min. For each pump $Wr^2 = 138$ lb-ft^2. Data from the pump characteristic diagram are tabulated atop the next page.

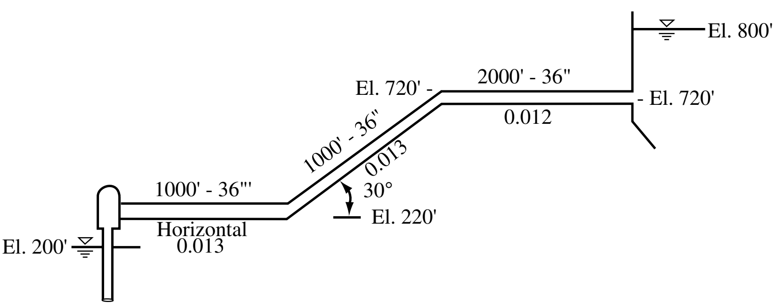
Q (gal/min)	Head/stage (ft)	BHP/stage (hp)
0	213	97
2500	162	130
3000	156	140
3250	151	142
3500	145	145
4000	130	150

Investigate the possibility of column separation following power failure to the pumps by applying PROG3. If it does occur, find the time and location and plot the EL-HGL at the time of column separation.



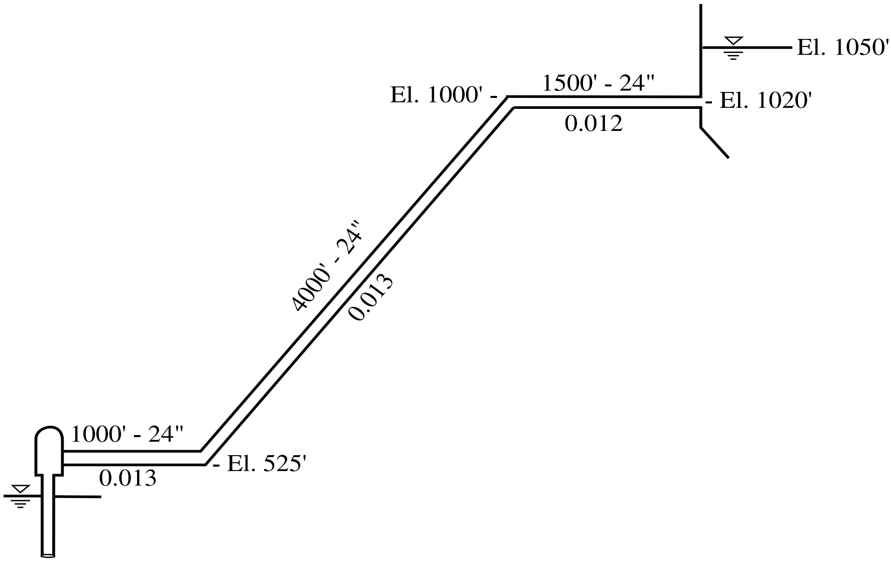
11.4 A proposed project will employ five six-stage 15H277 Ingersoll-Dresser turbine pumps (see Appendix B) to lift approximately 14,000 gal/min of water from the Columbia River gorge to a storage reservoir on a plateau above. The 11.83-in impeller will be used, and the rotating parts have an estimated Wr^2 of 510 lb-ft^2.

The pumps are connected by a manifold to one 36-in line constructed of two materials. The first 2000 ft is welded steel, and the second 2000 ft is asbestos cement. The wave speeds in the pipes are 3190 ft/s and 2860 ft/s, respectively. Use PROG3 to conduct a power failure analysis to determine whether, when, and where column separation occurs.

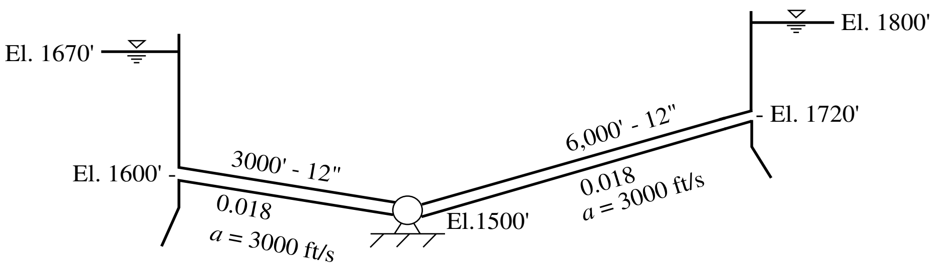


11.5 A project plans to use four five-stage Ingersoll-Dresser 15M185 turbine pumps (see Appendix B) to pump water from the Snake River 6500 ft to a storage reservoir. The approximate pipeline profile is shown below. The four pumps have 11.83-in impellers and are connected via a check-valve and manifold to a 24-in pipeline. For each pump and motor unit it is estimated that Wr^2 = 200 lb-ft^2. The Snake River water level fluctuates between 500 and 520 ft over the pumping season. The deck of the pumping station and the pump manifold are at elevation 525 ft. The portion of the pipeline from the pump station to the top of the incline is welded steel having a wave speed of 3190 ft/s. The remainder of the line is asbestos cement pipe with a wave speed of 2860 ft/s.

Use PROG3 to determine if, when, and where column separation occurs during pump power failure.

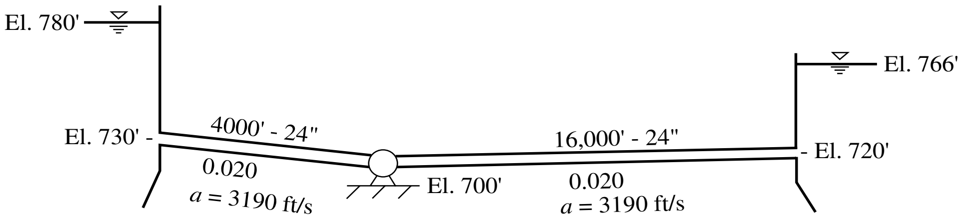


11.6 The pump station in the system below is equipped with three five-stage Ingersoll-Dresser 14JKH pumps operating in parallel (see Appendix B). The pumps use the 10.5-inch impellers and run at 1175 rev/min. Each impeller has $Wr^2 = 2.6$ lb-ft^2; the inertia of the pump shaft can be neglected. For the motor Wr^2 can be estimated with Thorley's formula.



Analyze the system for power failure using PROG4, and determine the maximum and minimum pressures, their location and time of occurrence.

11.7 The booster pump station in the next figure houses two single-stage Ingersoll-Dresser 20KKH turbine pumps. The pumps run at 1180 rev/min and have 15-in impellers. Assume the rotary inertia for the pump and motor unit is 225 lb-ft^2.



Investigate the consequences of power failure for the pipeline using PROG4, and find the extreme pressures occurring and their locations and time of occurrence.

11.8 Use PROG8 to estimate the consequences of column separation in Problem 11.1. There are no air-vacuum valves in the system.

11.9 If an air-vacuum valve exists at the downstream end of the 2-mile pipe in Problem 11.2, use PROG8 to investigate the effects of power failure on the extreme pressures occurring in the system.

11.10 Analysis of Problem 11.3 with PROG3 revealed that column separation occurred. Use PROG8 to estimate the effects of column separation on the maximum and minimum pressures occurring in the pipeline.

11.11 Column separation was found to occur in Problem 11.4 as a result of power failure. Analyze again the system by using PROG8 to estimate the effects of this event. An air-vacuum valve is at the downstream end of the 1000-ft pipe at elevation 720 ft.

11.12 We have established that column separation will occur in Problem 11.5 as a consequence of power failure. Using PROG8, estimate the extreme pressures which will then occur. An air-vacuum valve is at the downstream end of the 4000-ft pipe.