EFFECT OF EXPANSION TANKS AND TEE LOSSES
ON PRESSURE TRANSIENTS -
PRELIMINARY PARAMETER STUDY

by

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ABSTRACT

The effects of expansion tanks and Tee junction losses on pressure transients are investigated numerically using the one-dimensional nonlinear fluid-hammer approach. The losses in a Tee junction are modeled using steady state empirical pressure drop coefficients. Large expansion tanks and gas volumes are represented by a quasi-steady model coupled directly to the dynamic response of the fluid pipes connected to them. A simple constant head pump representation is used in the analysis. The results indicate that pressure losses at a Tee junction are minor. In particular, no significant differences of placing a rupture disc in the run or the branch of a Tee can be established by means of the one-dimensional analysis. Typical thermal expansion tank configurations are found to be ineffective in attenuating pressure pulses. The effective mitigation of pressure transients by such gas volumes requires very large connecting passages from the primary flow path to the tank or the direct in-line placement of surge tanks. It is concluded that the effect of rupture disc placement in the run or the branch of a Tee must be investigated by means of multi-dimensional analysis. Also, more realistic pump models, which take into account the actual response of the device, must be developed to study their pressure pulse transmission characteristics.

I. INTRODUCTION

One of the objectives of the Pressure Transient Analysis Program is to perform parameter studies in support of design issues concerning LMFBR intermediate heat transport systems. Using the physical parameters of the CRBRP intermediate sodium loop as a guide, a preliminary parameter study dealing with the pressure attenuation effectiveness of expansion tanks and the placement of rupture discs was recently completed. The investigation was conducted within the limitations of one-dimensional fluid-hammer modeling of the transient flow phenomena. The NATRANSIENT [1]* computer code developed for this type of analysis was modified to permit the treatment of expansion tanks, cover gas spaces or accumulators. Simple modeling of pump response was also introduced as well as a treatment for pressure losses at a Tee junction.

One of the tasks of the study was to ascertain if expansion tanks placed within the flow system to absorb the thermal expansion of the sodium under operating conditions would also provide some attenuation of transient pressure pulses which may be generated due to a tube rupture in the steam generator units. A second aspect investigated was the effect of rupture disc placement in either the run or branch of a Tee on the pressure relief afforded by the breaking of the disc.

In the following we outline the fluid-hammer modeling of these phenomena and the computer code modifications. A brief summary of the computational results is provided together with a discussion of their limitations and implications. Finally, recommendations for future efforts and model improvements are made.

* Numbers in brackets refer to references listed on page 25.

II. MODIFICATIONS OF PRESSURE TRANSIENT ANALYSIS

In order to carry out the parameter studies on the effect of expansion tanks and rupture disc placement a number of modifications were introduced into the existing NATRANSIENT [1] computer code. Both changes in the analysis method and the programming of the code were required. An outline of these modifications is given in the following sections.

A. Pressure Losses at Tee Junctions

A generally accepted method for the computation of pressure drop between the branches of a Tee for unsteady flow does not seem to exist [2]. Due to the absence of any theoretical analysis or well-tested empirical method to predict the effect of a Tee on the passage of a transient, it was decided that the methods developed for steady flow be applied in order to obtain at least an order of magnitude estimate of the pressure drops at a Tee. It is found that even in the case of steady flow only empirical methods are used and that there are many of them. Among these, a relatively simple method given in the Crane Catalog [3] was chosen mainly because of the ease with which it can be adapted for computation.

The Crane Catalog [3] gives the pressure losses for various fittings in terms of equivalent pipe length expressed in pipe diameters (L/D). For a standard Tee the L/D values for flow through a run and through a branch are 20 and 60 respectively. The resistance coefficient or the loss coefficient is then obtained as the product of the L/D value and f , the pipe friction factor. The pressure loss is then expressed as $(f \cdot L/D) \rho u^2/2$ where $\rho u^2/2$ is the kinetic energy of the steady flow.

Formulas for the pressure and velocities at a Tee for unsteady flow, neglecting pressure losses, were given by Y. W. Shin [1] in a form suitable for incorporation into the NATRANSIENT code. In the following derivation,

these formulas are modified so as to include the effect of losses. Let

Figure 1. Tee Junction

i , $i+1$ and j be the pipe branches joining at a Tee (see Fig. 1). Following the notation of Shin [1], the nodes joined at the Tee will be denoted as i,n ; $i+1,1$ and $j,1$. Velocity in each branch is considered positive when flow is in the direction of increasing node numbers. The formulas for pressures and velocities depend on the pattern of flow at a Tee. The patterns of combining and dividing flows are as shown in Figs. 2 and 3.

Figure 2. Combining Flows

Figure 3. Dividing Flows

Consider, for instance, the flow pattern shown in case (d) of Fig. 3.

The equation of continuity at the Tee gives

$$A_{i,n} u_{i,n}^{t+\Delta t} = A_{i+1,1} u_{i+1,1}^{t+\Delta t} + A_j u_{j,1}^{t+\Delta t} \quad (1)$$

The energy balance equations at the Tee are:

$$p_{i,n}^{t+\Delta t} + \frac{1}{2} \rho (u_{i,n}^{t+\Delta t})^2 = p_{i+1,1}^{t+\Delta t} + \frac{1}{2} \rho (u_{i+1,1}^{t+\Delta t})^2 + \frac{1}{2} K_1^{i+1} \rho (u_{i,n}^{t+\Delta t})^2 \quad (2)$$

$$p_{i,n}^{t+\Delta t} + \frac{1}{2} \rho (u_{i,n}^{t+\Delta t})^2 = p_{j,1}^{t+\Delta t} + \frac{1}{2} \rho (u_{j,1}^{t+\Delta t})^2 + \frac{1}{2} K_1^j \rho (u_{i,n}^{t+\Delta t})^2 \quad (3)$$

In Eqs. (1)-(3), A stands for area of cross section of flow, p the pressure, u the velocity, ρ the density and K_1^j is the loss coefficient for flow from the i th to the j th branch. In addition, three characteristic equations - one for each pipe branch - may be written.

Along C^+ in the i th branch:

$$u_{i,n}^{t+\Delta t} - u_{i,A}^t + \frac{1}{\rho c_i} (p_{i,n}^{t+\Delta t} - p_{i,A}^t) + \frac{f_{i,A}^t u_{i,A}^t |u_{i,A}^t| \Delta t}{2D_i} = 0 \quad (4)$$

Along C^- in the i+1 th branch:

$$u_{i+1,1}^{t+\Delta t} - u_{i+1,B}^t - \frac{1}{\rho c_{i+1}} (p_{i+1,1}^{t+\Delta t} - p_{i+1,B}^t) + \frac{f_{i+1,B}^t u_{i+1,B}^t |u_{i+1,B}^t| \Delta t}{2D_{i+1}} = 0 \quad (5)$$

Along C^- in the j th branch:

$$u_{j,1}^{t+\Delta t} - u_{j,C}^t - \frac{1}{\rho c_j} (p_{j,1}^{t+\Delta t} - p_{j,C}^t) + \frac{f_{j,C}^t u_{j,C}^t |u_{j,C}^t| \Delta t}{2D_j} = 0 \quad (6)$$

In Eqs. (4)-(6), c is the sonic speed, D the diameter, f the friction factor and A, B, C are points where the characteristic lines intersect the $t = \text{const}$ line. The notation is similar to that used in Reference 1.

Equations (1) to (6) are a set of six simultaneous equations for six unknowns. Among these Eqs. (2) and (3) contain non-linear terms of $u_{i,n}^{t+\Delta t}$, $u_{i+1,1}^{t+\Delta t}$ and $u_{j,1}^{t+\Delta t}$. Therefore, an attempt to solve the above system

results in two simultaneous quadratic equations in two unknowns. In order to avoid a tedious iterative procedure for solving these equations, the following approximation is made: $u_{i,n}^{t+\Delta t}$, $u_{i+1,1}^{t+\Delta t}$ and $u_{j,1}^{t+\Delta t}$ are replaced by $u_{i,n}^t$, $u_{i+1,1}^t$ and $u_{j,1}^t$ in Equations (2) and (3) only. This is similar to the approximation made in the friction term of Eqs. (4)-(6). Now the six equations may be solved to give the solutions as:

$$p_{i,n}^{t+\Delta t} = \frac{1}{a_i + a_{i+1} + a_j} \{ (a_i p_{i,A}^t + a_{i+1} p_{i+1,B}^t + a_j p_{j,C}^t) + (A_i u_{i,A}^t - A_{i+1} u_{i+1,B}^t - A_j u_{j,C}^t) + \Delta t (b_{i+1} f_{i+1,B}^t u_{i+1,B}^t |u_{i+1,B}^t| + b_j f_{j,C}^t u_{j,C}^t |u_{j,C}^t| - b_i f_{i,A}^t u_{i,A}^t |u_{i,A}^t|) + \frac{1}{2} (\rho a_{i+1} [(u_{i+1,1}^t)^2 - (1 - K_i^{i+1}) (u_{i,n}^t)^2] + \rho a_j [(u_{j,1}^t)^2 - (1 - K_i^j) (u_{i,n}^t)^2]) \} \quad (7)$$

$$p_{i+1,1}^{t+\Delta t} = p_{i,n}^{t+\Delta t} - \frac{1}{2} \rho \{ (u_{i+1,1}^t)^2 - (1 - K_i^{i+1}) (u_{i,n}^t)^2 \} \quad (8)$$

$$p_{j,1}^{t+\Delta t} = p_{i,n}^{t+\Delta t} - \frac{1}{2} \rho \{ (u_{j,1}^t)^2 - (1 - K_i^j) (u_{i,n}^t)^2 \} \quad (9)$$

$$u_{i,n}^{t+\Delta t} = u_{i,A}^t - \frac{f_{i,A}^t u_{i,A}^t |u_{i,A}^t| \Delta t}{2D_i} + \frac{1}{\rho c_i} (p_{i,A}^t - p_{i,n}^{t+\Delta t}) \quad (10)$$

$$u_{i+1,1}^{t+\Delta t} = u_{i+1,B}^t - \frac{f_{i+1,B}^t u_{i+1,B}^t |u_{i+1,B}^t| \Delta t}{2D_{i+1}} - \frac{1}{\rho c_{i+1}} (p_{i+1,B}^t - p_{i+1,1}^{t+\Delta t}) \quad (11)$$

$$u_{j,1}^{t+\Delta t} = u_{j,C}^t - \frac{f_{j,C}^t u_{j,C}^t |u_{j,C}^t| \Delta t}{2D_j} - \frac{1}{\rho c_j} (p_{j,C}^t - p_{j,1}^{t+\Delta t}) \quad (12)$$

where $a_i = \frac{A_i}{\rho c_i}$ and $b_i = \frac{A_i}{2D_i}$ etc.

The above equations replace equations (31a)-(31c) of Reference 1. The corresponding equations for the remaining 5 cases may be derived following

the same procedure. These results are omitted from here for the sake of brevity.

B. Expansion Tank Modeling

No unique method for modeling the transient response of an expansion tank, a surge tank, or a cover gas volume exists. Models of various complexity have been used in the past and range from simple lumped mass thermodynamic treatments to a complete analysis of fluid oscillation and sloshing. The method most appropriate for a particular application depends on the relative magnitude of the forces applied to the surge tank, the mass of liquid contained in the tank, and the size of the gas volume. Also considered must be the rates of pressure application. For the large gas volumes and fluid masses of interest here, it appears that a simple quasi-static response of the gas volumes coupled to the fully transient inflow, as given by the method of characteristic solution, is quite satisfactory for predicting the response of expansion tanks.

Figure 4. Expansion Tank Model

Assume that the tank in Figure 4 contains a constant mass of gas, the instantaneous volume and pressure of which are respectively V and p . Prescribing a perfect gas behavior the relationship between pressure and volume is simply

$$pV^\gamma = p_0 V_0^\gamma = \text{constant} \quad . \quad (13)$$

Here p_0 , V_0 are some initial values and the exponent γ may range from unity, for isothermal response, to $\gamma = c_p/c_v$ (ratio of specific heats) in the case of adiabatic behavior. Neglecting fluid velocities within the tank and also pressure losses at intake and outlet pipes one finds that the gas pressure is simply equal to the pressure at the connecting node of each pipe, i.e.,

$$p = p_{i,n} = p_{j,1} \quad . \quad (14)$$

Following the sign convention of Reference 1, we consider flow in the direction of increasing node numbers along a pipe as positive. Hence if k pipes are connected with their first node to the surge tank and ℓ pipes are connected with their last node to the tank, then the rate of change of gas volume is given by,

$$\frac{dV}{dt} = \sum_{j=1}^k u_{j,1} A_j - \sum_{i=1}^{\ell} u_{i,n} A_i \quad . \quad (15)$$

Here u_i and A_i are the fluid velocity and pipe cross-sectional area of the i th pipe. Equation (15) can be written in finite difference form for the time step Δt .

$$V^{t+\Delta t} = V^t + \Delta t \left[\sum_{j=1}^k u_{j,1}^t A_j - \sum_{i=1}^{\ell} u_{i,n}^t A_i \right] \quad . \quad (16)$$

A simple forward difference in time is used. A more correct approximation would be to use average velocities over the time step Δt . However, this entails the use of the unknown velocities at time $t + \Delta t$ and because of

equation (13) would lead to a nonlinear relationship for the unknown variables. Such a system could only be solved by iterative procedures. To avoid these difficulties the velocities at the beginning of the time step are used in equation (16). This is consistent with the treatment of the friction terms in the compatibility equations.

The solution of the expansion tank problem at each time step then simply requires the computation of a new gas volume V from equation (16), calculating a new pressure p from (13), and obtaining the pipe node pressures from relationship (14). Finally new pipe velocities are computed using the characteristic compatibility equations and interpolation equations, i.e., for the C^- characteristic

$$u_{j,1}^{t+\Delta t} = u_{j,1}^t + \frac{1}{\rho c_j} (p_{j,1}^{t+\Delta t} - p_{j,B}^t) - \frac{f_{j,B}^t u_{j,B}^t |u_{j,B}^t|}{2D_j} \quad (17)$$

C^+ characteristic

$$u_{i,n}^{t+\Delta t} = u_{i,n}^t - \frac{1}{\rho c_i} (p_{i,n}^{t+\Delta t} - p_{i,A}^t) - \frac{f_{i,A}^t u_{i,A}^t |u_{i,A}^t|}{2D_i} \quad (18)$$

Here f is the friction factor, c the sound velocity, ρ the density, and A and B refer to the intersection points of the characteristics with the line $t = t$. The values at the latter points are obtained by linear interpolation, following the procedures outlined in Reference 1.

$$u_{i,A}^t = \frac{u_{i,n}^t - c_i \theta_i (u_{i,n}^t - u_{i,n-1}^t)}{1 + \theta_i (u_{i,n}^t - u_{i,n-1}^t)} \quad (19)$$

$$p_{i,A}^t = p_{i,n}^t - \theta_i (c_i + u_{i,A}^t) (p_{i,n}^t - p_{i,n-1}^t) \quad (20)$$

$$u_{j,B}^t = \frac{u_{j,1}^t + c_j \theta_j (u_{j,2}^t - u_{j,1}^t)}{1 + \theta_j (u_{j,2}^t - u_{j,1}^t)} \quad (21)$$

$$p_{j,B}^t = p_{j,1}^t + \theta_j (c_j - u_{j,B}^t) (p_{j,2}^t - p_{j,1}^t) \quad (22)$$

Here $\theta_i = \Delta t / \Delta x_i$ and Δx_i is the grid size for the i th pipe. The friction factors $f_{i,A}^t$ and $f_{j,B}^t$ are computed using Darcy's law and are based on their respective velocity and the specified pipe roughnesses. Equations (13) to (22) provide a simple procedure to calculate the response of an expansion tank or gas cover irrespective of the number of pipes connected to it. The model neglects any gas outflow or the interconnection of multiple gas spaces. However, for large gas volumes this procedure is acceptable.

C. Pump Representation

The behavior of centrifugal pumps under transient flow conditions has been modeled with some success [4]. Such modeling requires that the performance curves (e.g., head vs. discharge) be known for the normal operating range as well as under abnormal conditions such as flow reversal. Since the major objective of the parameter study was to evaluate the influence of expansion tanks, and since detailed pump performance information was not available, a very simple pump representation was used here. In essence the pump is treated as a constant head device and is modeled in terms of a multiple pipe junction with a sudden head jump on the discharge side. A similar representation is employed in the TRANSWRAP code [5]. The simple model is justified on the basis that interest is restricted to short time durations, i.e., of the order of one pump revolution. Large inertial effects prevent any substantial speed response during these short durations. To represent the various inflow, outflow and internal flow passages of the pump, any number of pipes can be used. The pump tank and gas cover can be modeled in a manner identical to that of the expansion tank. The pump configuration of interest to the parameter study is shown in Section III (see Fig. 6) where the results are presented.

III. RESULTS AND DISCUSSION

The modified NATRANSIENT computer code was used to carry out parametric calculations on the effect of rupture disc placement and the influence of expansion tanks on pressure transients. As mentioned earlier, the transient flow modeling is strictly one-dimensional; thus no multidimensional geometric effects can be included. No attempt was made to simulate actual pressure transients, e.g., pressure pulses resulting from water leaks in the steam generating units. Instead arbitrary pressure time histories were prescribed at the source location.

A. Placement of Rupture Disc

In order to study the influence of pressure loss at a Tee on rupture disc performance, the two configurations shown in Fig. 5 were considered.

Figure 5. Rupture Disc Configurations (Initial Conditions - $u = 0$, $p = 100$ psi, Pressure Source - 500 psi step increase)

The initial conditions ($u = 0$, $p = 100$ psi) and pipe dimensions are identical for cases (a) and (b). The only difference between them is in the placement of the rupture disc as indicated in the figure. The pressure source was represented as either an instantaneous pressure jump to 500 psia or as a steep ramp rate to the same peak value. Burst pressure for the discs was assumed to be 200 psi. The pressure and the velocity at each node for a time duration of 0.01 sec (about 8 times the time required for the wave to traverse the longest pipe) were computed in each case. The following is a brief summary of the results obtained in the computations.

- 1) The pressure history in branch ① is not significantly affected by the placement of the rupture disc. The maximum pressure and the time at which it occurs in branch ① are identical for both cases (a) and (b). The difference in pressure v as between case (a) and case (b) were on the order of 0.4% the end of 0.01 sec.
- 2) The pressure histories in branches ② and ③ of case (a) are interchanged for case (b) with a very small difference. The maximum pressure in branch ②/③ of case (a) is identical to that in branch ③/② of case (b). The maximum pressure difference at corresponding nodes between case (a) and case (b) increases with time, but even after 0.01 sec, it is only on the order of 2%.

Thus a one-dimensional analysis, which considers only the pressure losses at a Tee, does not indicate any significant effect of rupture disc placement on the pressure either upstream or downstream of the relief point. This is an expected result, since in general the pressure term p in the energy equations is much larger than the loss term $\frac{1}{2} K \rho u^2$ even for very high velocities. While the fact that the loss coefficients are based on steady flow empirical data may alter the results slightly, the major shortcoming of the analysis is thought to be its one-dimensional limitation. The flow through a Tee junction is basically multidimensional and pressure pulse propagation is influenced by the refractions and reflections of the wave when a sudden directional change is encountered. These aspects of the problem are not at all represented in the current analysis. The directional

effects may, however, at times immediately following the rupture disc bursting, significantly influence the pressure relief phenomena.

B. Effect of Expansion Tanks and Gas Covers

The piping and equipment configuration shown in Figure 6 were used to investigate the effects of expansion tanks and gas covers on the propagation of pressure transients. The equipment arrangement and pipe dimensions are approximately those of the equivalent portion in the secondary sodium loop for the CRBRP. As shown, the expansion tank is connected to both the low pressure pump side and the discharge pipe. The pump is modeled as a multiple pipe junction with a sudden head increase between points B and C. Access to the pump tank and gas cover from the pump inlet side was either completely open or restricted to the equivalent of a six inch diameter pipe. Initial pressure upstream of the pump and in the gas spaces was assumed to be 100 psi. A constant pump head of 100 psi was used, resulting in an initial discharge pressure of 200 psi. The initial flow velocity corresponded to the nominal operating flow for the intermediate sodium loop, i.e., 21 fps. A pressure source having the form of an instantaneous step increase to 500 psi on the intake side (see Figure 6) was used in most calculations. A short duration pressure pulse (10 msec) which linearly rose to 500 psi in 2 msec and then linearly decayed to the base pressure of 100 psi was also used in some computations. Finally a calculation was performed with the step pressure source located in the discharge leg of the pump. The calculations were carried out for a duration of 20 to 30 msec, i.e., a duration at least 3 times as long as the time required for a pressure pulse to traverse the entire piping system. A time step of 0.5 msec was used in all computations. Trials using a finer time step (0.2 msec) gave essentially identical results to those obtained with the coarser step.

Figure 6. Schematic of Pump-Expansion Tank System

The results of the computations are summarized in Figures 7 to 13. They represent pressure and velocity histories at the locations indicated by capital letters in Figure 6. It can be observed that for all cases considered the pressure in the gas volumes remains essentially constant. This implies that little gas compression should be expected in large gas volumes at least for short time durations. Figure 7 indicates that, for a configuration which appears to be most representative of the current design of the intermediate sodium loop, essentially the entire pressure pulse arriving at the pump intake is transmitted to the discharge side. For this calculation it was assumed that a restriction equivalent to a 6 inch diameter pipe exists between the pump intake and the pump tank and gas cover. Velocity increases (Figure 8) are moderate, except for the restriction leading from the inlet to the pump tank (point B). Thus in this configuration no pressure attenuation from either the expansion tank or pump gas cover is apparent. The pressure oscillation at point D in the line from the expansion tank to the discharge pipe is due to the fact that compression and expansion waves move continuously between the high pressure in the pipe and the low constant pressure in the tank.

If the pump inlet side is completely open to the large diameter pump tank and its gas cover, then any pressure pulse through the pump is very effectively attenuated as shown in Figure 9. Pressure oscillations now occur at point A in the intake line due to the alternating compression and expansion waves coming from the constant pressure source and the pump gas cover respectively. The effect of increasing the diameter from 4 inches to 24 inches of the pipe from the low pressure pump side to the expansion tank is illustrated in Figure 10. Again the passage to the pump tank and gas cover is restricted. While peak pressures on the discharge side are

Figure 7. Pressure-Time Variations (500 psi Step Pulse Pressure Source at Inlet Side, 6" Diameter Restriction to Pump Tank, 4" Line to Expansion Tank)

Figure 8. Velocity-Time Variations (500 psi Step Pulse Pressure Source at Inlet Side, 6" Diameter Restriction to Pump Tank, 4" Line to Expansion Tank)

Figure 9. Pressure-Time Variations (500 psi Step Pulse Pressure Source at Inlet Side, Open to Pump Tank, 4" Line to Expansion Tank)

Figure 10. Pressure-Time Variations (500 psi Step Pulse Pressure Source at Inlet Side, 6" Diameter Restriction to Pump Tank, 24" Line to Expansion Tank)

Figure 11. Pressure-Time Variations (500 psi Peak Pressure, 10 msec Triangular Pulse at Inlet Side, 6" Diameter Restriction to Pump Tank, 24" Line to Expansion Tank)

Figure 12. Pressure-Time Variations (500 psi Step Pulse Pressure Source at Discharge Side, 6" Diameter Restriction to Pump Tank, 4" Line to Expansion Tank)

Figure 13. Velocity-Time Variations (500 psi Step Pulse Pressure Source at Discharge Side, 6" Diameter Restriction to Pump Tank, 4" Line to Expansion Tank)

lowered by about 100 psi, a very substantial pressure pulse still passes through the pump. In practice this type of configuration would probably require some type of rupture device to open up the larger pipe. Figure 11 represents results for a configuration identical to that used in the preceding example. However, the pressure source generates only a short duration (10 msec) triangular pressure pulse described earlier. Peak pressures transmitted through the pump are slightly lower but again most of the pressure impulse passes through the pump. It is interesting to note that pressures on the intake side drop below the steady state value and that actually cavitation or column separation must occur in the inlet branch as indicated by the negative pressure at point A. This effect is caused by the rapid pressure decay at the source location coupled with the sudden expansion in the pump region to more than twice the initial flow area. Figures 12 and 13 show respectively the pressure and velocity histories for a step increase pressure source located in the discharge leg of the pump. Again passage to the pump tank is restricted and the line from the expansion tank to the pump inlet side is 4 inches in diameter. An undiminished pressure signal is transmitted through the pump. Flow reversal does not occur in the discharge pipe, thus the simple constant head pump model should still be valid.

In summary it is obvious that the expansion tank in the current intermediate sodium loop configuration does not provide any pressure attenuation. A much more effective pressure pulse suppression may be obtained from the pump gas cover. The effectiveness of this gas space depends on the cross-sectional area of the passages from the pump intake side to the pump tank and gas cover. Large openings provide good pressure pulse attenuation. The pressure suppression effectiveness of the expansion tank can be enhanced by a substantial diameter increase and a simultaneous length reduction of the pipe connecting the

pump to the expansion tank. In general, to obtain significant pressure mitigation the primary flow passage must be directly connected to a sudden very large area expansion, e.g., a tank, which in turn is coupled to a gas space of significant size. Such a configuration can be obtained in numerous ways, an example being the open pump tank. A similar effect could be achieved through the employment of an in-line surge tank. For efficient normal operation all of these devices may have to be isolated from the main flow passage by means of rupture discs or other sudden pressure relief hardware.

C. Recommendations

Since the one-dimensional treatment of rupture disc placement did not yield any conclusive results it is recommended that this problem be analyzed in a multidimensional formulation. While the actual geometries are three-dimensional it is believed that the salient features of rupture disc placement in the run or the branch of a Tee can be analyzed in a plane two-dimensional geometry. The formulation for two-dimensional or axisymmetric fluid-hammer problems in terms of a nearcharacteristic approach was recently completed [6] and a computer program for the treatment of simple problems has been coded. It is thus recommended that this program be modified to accommodate the rupture disc placement problem and that parametric calculations be performed.

Concerning the effectiveness of expansion tanks and gas covers as pressure suppression devices, it is recommended that any design changes or new concepts be evaluated using the approach outlined in this study. Improvements in the modeling of tanks and gas spaces may be required to reflect the dynamic response of these devices, particularly for small volumes. The major modification which must be introduced in the analysis is a more realistic modeling of pump response. Such models must reflect the actual pump characteristics both for the normal operating range and for abnormal

conditions. In particular the effect of flow reversal, speed changes and cavitation must be properly accounted for.

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