Inert Gas Fire Suppression Systems Using IG541 (INERGEN)
Solving the Hydraulic Calculation Problem
by

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Regardless of the specific agent used, fire systems need to deliver a required quantity of agent to the fire zones within a specified time. The science of hydraulics enables us to design piping and node networks to accomplish this task. In this presentation we will examine the application of classical flow theory to inert gas **541,** better known **as** INERGEN. We will also examine one major enhancement to the flow theory which was required to produce accuracy sufficient for calculation of complex pipe networks.

In the paper, after a brief review of flow theory, we will examine the assumptions used in developing the INERGEN flow theory. We will also look at the effect of heat entry into the INERGEN during discharge. Finally we will review some of the transient effects that had to be accounted for in order to produce a flow calculation method which was sufficiently accurate to meet the demands of the fire protection industry.

Although this theory was developed and perfected specifically for inert gas **541,** it likely could be applied to other inert gas mixtures. One must proceed with caution because the applicability of specific items used in this theory to other inert gases must be thoroughly verified by testing.

For fire suppression systems flow calculations consist of determining nozzle pressures, discharge times and the quantity of agent discharged from each nozzle. Each of these items has a special relation to the problem of extinguishing fires. Inert gas systems operate at relatively high storage pressures. For INERGEN our storage pressure at 70" F. is almost 2200 pounds per square inch. The systems use a pressure reducing device at the outlet of the cylinder manifold to drop the high manifold pressure to approximately 1000 pounds per square inch. During an INERGEN discharge there are very rapid pressure changes both in the cylinder and in the pipe line.

Figure 1 is a diagram of a basic inert gas distribution system. The system includes storage cylinders, cylinder manifold, a pressure reducing device, the downstream discharge piping and delivery nozzles.

For the INERGEN systems studies, the pressure reducing device was an orifice plate

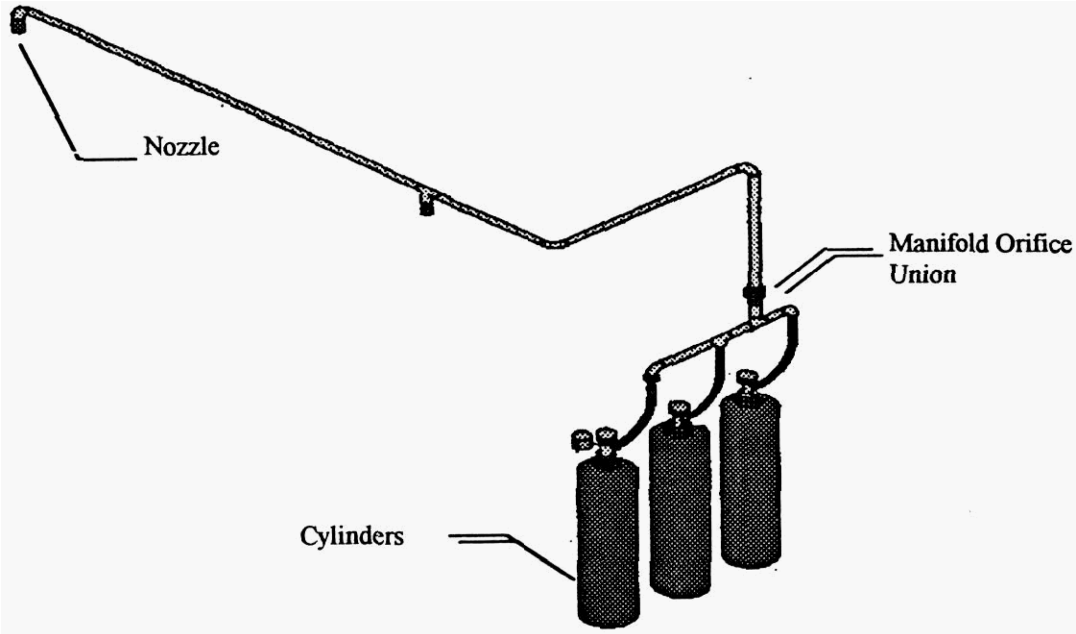


Figure 1 Typical Inert* Gas *Discharge System

To calculate the "flow" conditions in the cylinder during discharge, pressure loss in the piping, nozzle flow characteristics and a number of transient conditions are considered. **INERGEN** flow is a single phase flow of a compressible vapor. To determine the cylinder pressure recession during discharge. Assuming adiabatic expansion of the gas in the cylinder, theoretical calculation of pressure and density is a function of percent discharge from the cylinder was done. The results of that calculation are shown by the red lines in Figures 2 and 3. The data points are actual test data points taken during discharge tests.

These are typical of the verification tests which were run. In both cases, there is a very good correlation between the predicted and the measured cylinder pressure as a function of percent discharged.. Using this theoretical base line for cylinder pressure conditions, pressure loss in the pipe may be calculated.

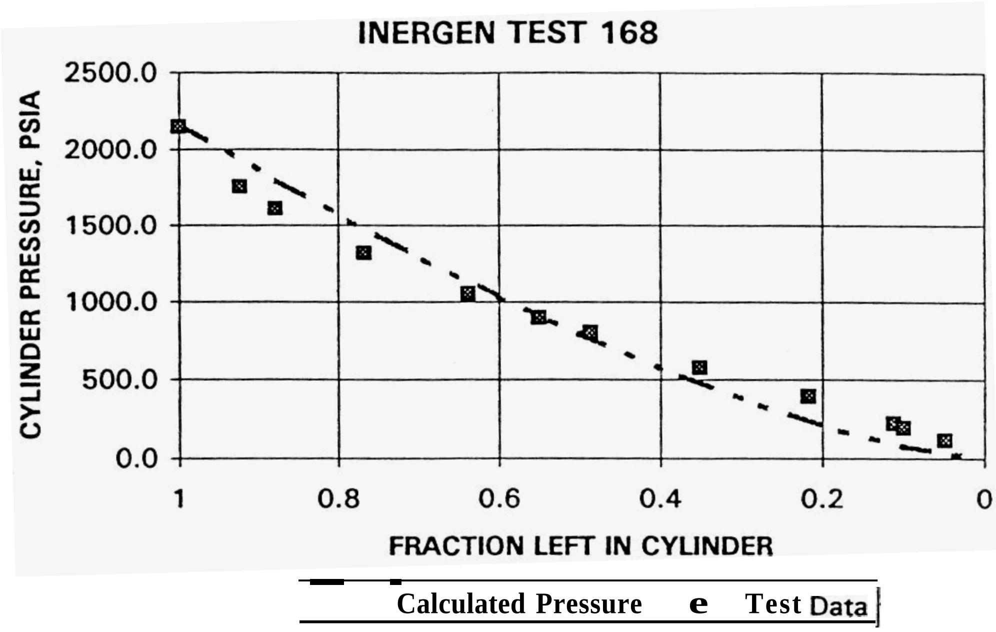


Figure 2

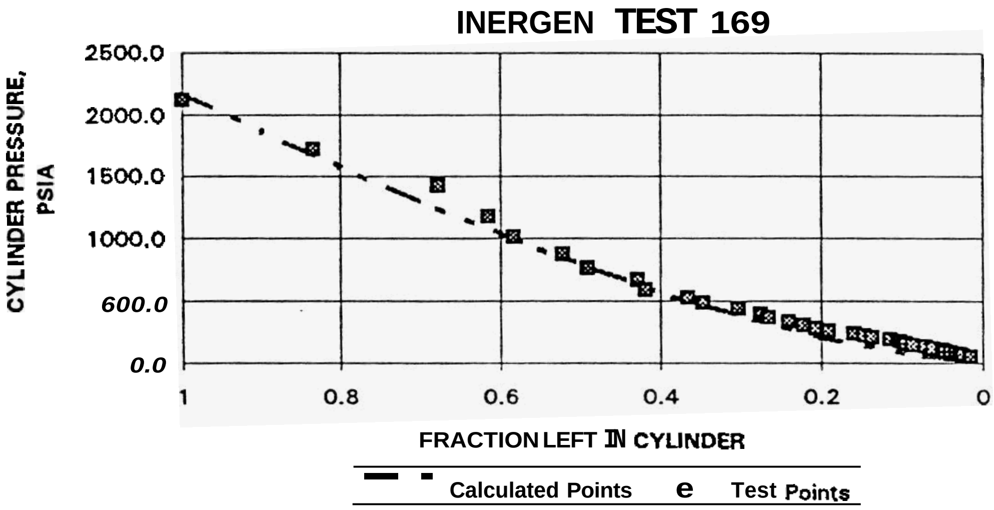


Figure 3

PIPELINE PRESSURE LOSS

$$Z + \frac{144P}{\rho} + \frac{v^2}{2g} = H$$

Bernoulli's equation (show above) is the basic equation of flow dynamics. In the form shown, it **is** specifically applicable to non compressible flow, where fluid density is essentially constant with changes in pressure. Inert gas, however, is a compressible fluid. A change in pressure yields a marked change in the density of the gas.

$$43.5 \cdot Q^2 \cdot f \cdot L - 7.97 \cdot Q^2 \cdot D \cdot \int_{P_o}^{P_f} \rho \, dP + D^5 \cdot \int_{\rho_o}^{\rho_f} \frac{d\rho}{\rho} = 0$$

Bernoulli's equation was modified to be used for compressible fluids (above). **This** equation was derived by James Hesson in the early **1950's.** A form of the equation is published in NFPA standards 12 and 12A. It has been successfully **used** to calculate the flow of many gaseous fire fighting agents, including Halon 1301, Halon **1211,** carbon dioxide, HFC 23, HFC 227EA, and now, inert gas.

In designing a fire extinguishing system, one generally starts knowing the required flow rate and, after developing a piping layout, the equivalent length of pipe from the storage container to each nozzle. Knowing flow rate and equivalent length, the pipe diameter ***can*** be estimated, and the friction factor for that pipe may be determined. Friction factor varies with Reynolds number until we reach a complete and turbulent flow regime. To simplify the calculation, set minimum flow rates for each pipe size sufficiently high to insure flow in the complete turbulent flow regime. Thus a constant friction factor for every given pipe diameter may be used.

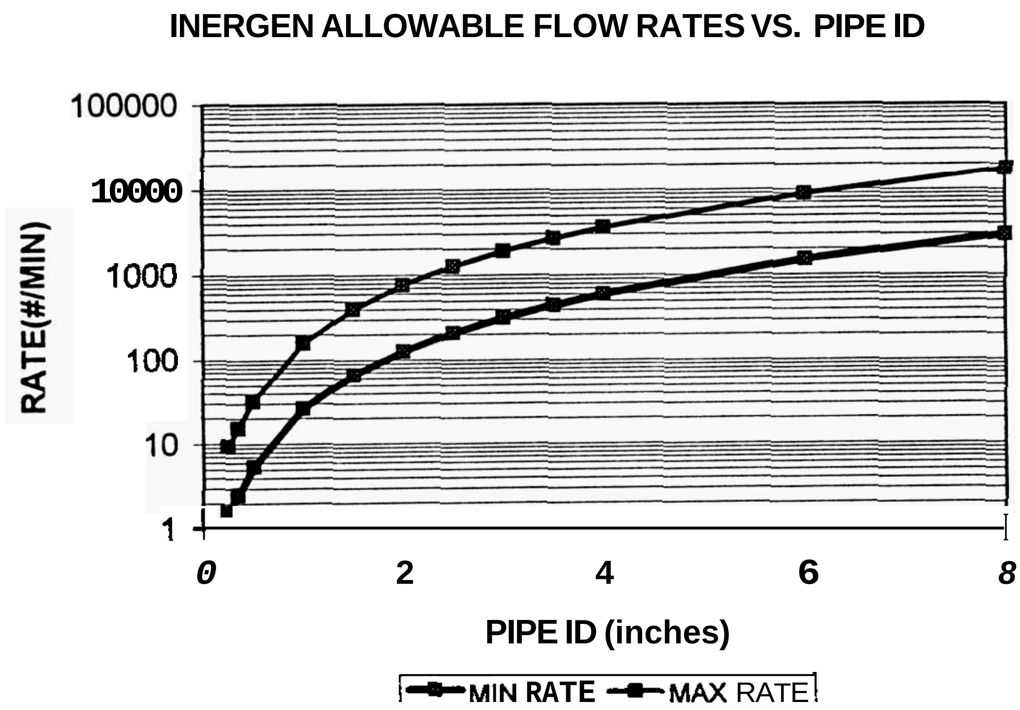


Figure 4

Under normal conditions, the maximum flow in a gas is determined by the sonic velocity for the gas. The maximum flow rate was set to avoid sonic velocity, or choked flow conditions, in the INERGEN system pipe. Figure **4** shows the maximum and minimum permitted flow rates versus pipe ID. The accuracy of system calculations has been verified for systems operating anywhere in the envelope between the two lines.

At this point, all of the quantities in the pressure-flow equation are known except for the density and the pressure, with pressure being the desired solution. The next task is to determine the density of the agent at various pressures in the pipeline.

There are two classical theoretical limits for pipeline flow. One is isothermal flow where the flowing media is at a constant temperature along the entire pipeline. A cross country gas transmission line is an example of a system where isothermal flow is used for calculating the hydraulics. The other extreme is adiabatic flow. In an adiabatic system no heat exchange takes place between the flowing substance and its surroundings. This type of flow is approximated in systems with very short pipelines which are perfectly insulated and also in systems where the discharge times are very rapid. In systems with very fast discharges, the amount of time that the agent is actually in contact with the pipe is so short that very little heat can be transferred between the agent and the piping.

Density in Pipe - Adiabatic Conditions

$$\rho = \frac{\rho_0 P T_0}{T P_0}$$

$$T = T_0 \left(\frac{P}{P_0}\right)^{\frac{\chi - 1}{\chi}}$$

$$\chi = c_p / c_v$$

For the early INERGEN system work, adiabatic expansion was assumed. Using standard equations for adiabatic expansion, density as a function of pressure for Inert Gas 541 in a pipeline was calculated. (Figure 5) Initial test work which involved relatively short piping networks in generally balanced systems indicated the assumption of adiabatic expansion provided acceptable results. However, as systems became more complex and longer pipe runs were tested, the assumption of adiabatic conditions broke down. Actual pressures were less than predicted pressures, actual discharge times became longer than predicted discharge times and the actual quantity of agent discharge from more remote nozzles in the system became less than the predicted quantity. All of these facts indicate that the actual density of the agent in the pipe was probably less than the predicted density.

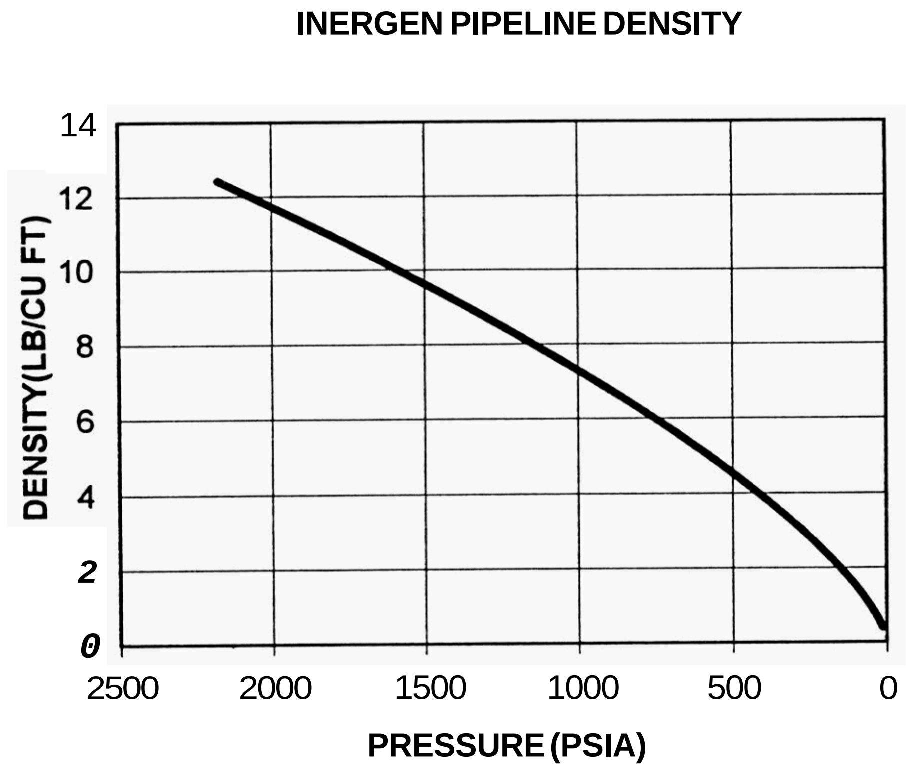


Figure 5

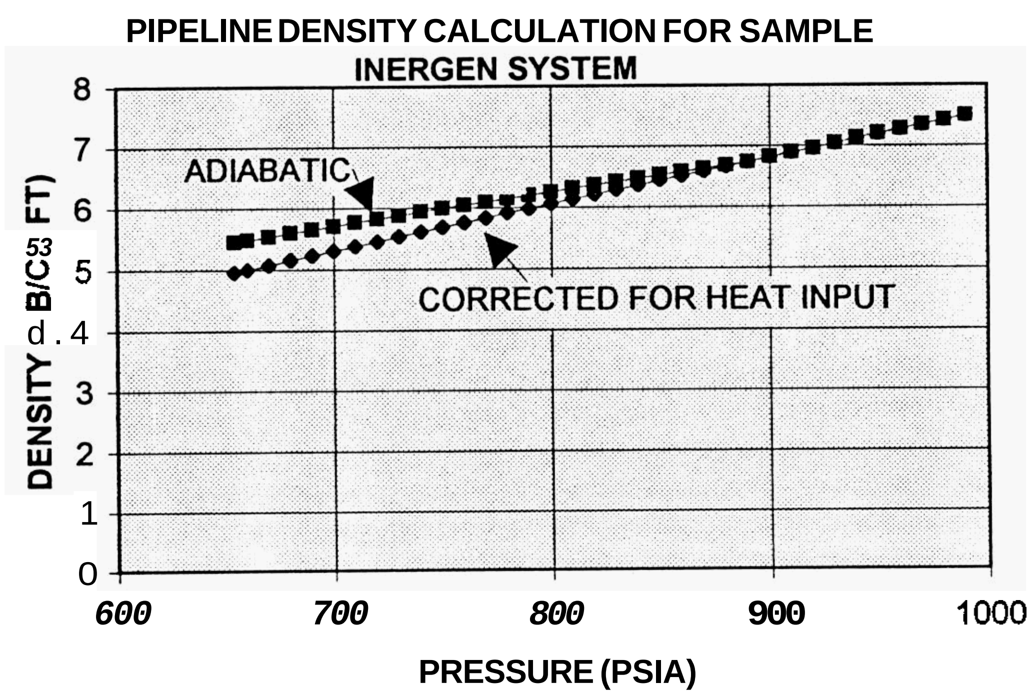


Figure 6

Assuming some heat absorption from the pipe during discharge, a corrected agent temperature was used in the density calculation. The net effect of this was new densities used for the flow calculation tended to be lower than the densities determined from the adiabatic expansion calculation. For very long pipes, the corrected density would approach the density based on isothermal expansion.

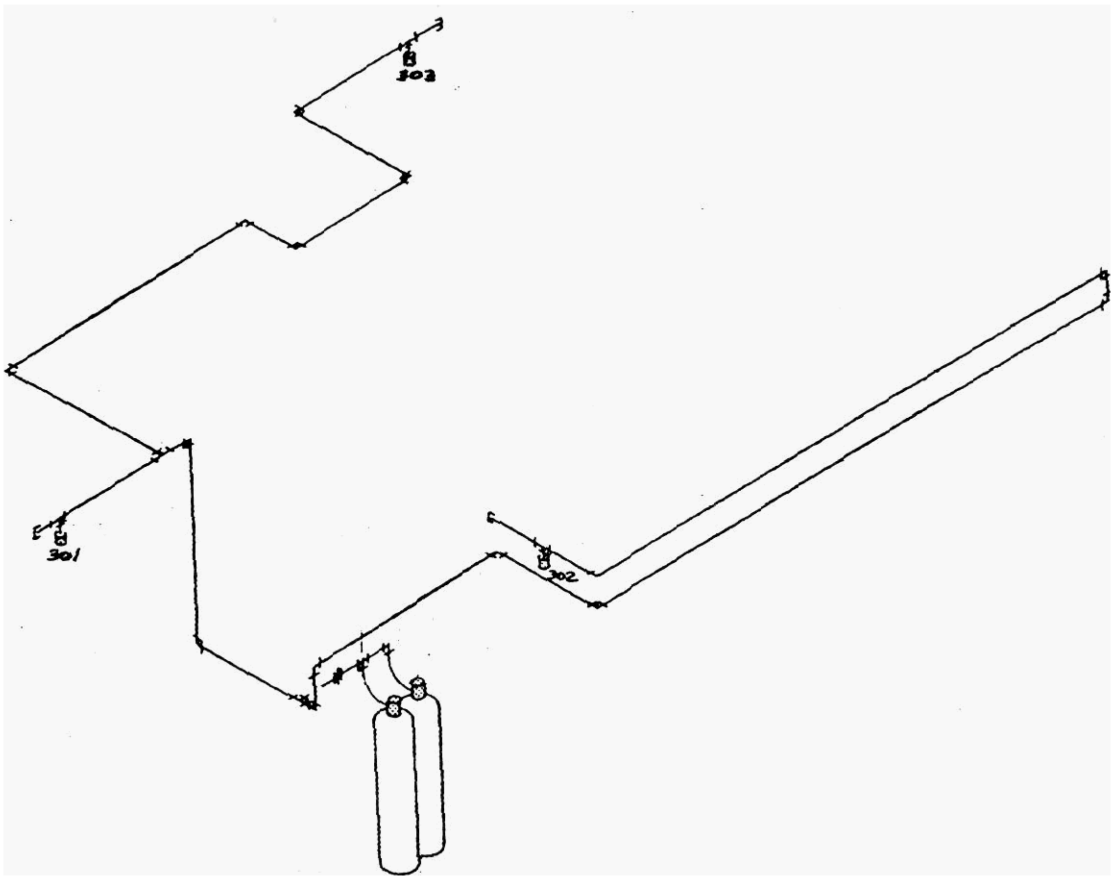


Figure 7 (above) is an isometric diagram of a rather unbalanced piping system which could not be accurately calculated by assuming either adiabatic or isothermal expansion. In fact the initial calculation of quantities discharged from the various nozzles showed a very large error for node 302. Nozzle 302 was fed by a 55 ft. long branch of 1/4" schedule pipe. Even though the discharge was quite rapid, approximately 22 seconds, heat entry into the agent in the 1/4" branch was enough to produce a 17% error in the predicted quantity discharged. When the system was re-calculated using the heat entry correction, our calculations came to within less than 1 ½ % of the actual amounts of the INERGEN discharged from individual nozzles. Similar results were seen in numerous tests

RUN THRUFXD1	CU FT ACT	CU FT CALC	%ACT/CALC	CU FT CALC	%ACT/CALC
101	192.7	189.7	1.6%	*194.5*	*-0.9%*
102	44.3	53.5	-17.2%	*43.7*	*1.4%*
103	183	176.8	3.5%	*181.9*	0.6%

Italic is corrected for heat entry.

NOZZLE FLOW In considering nozzle flow, standard theory for the compressible flow of gases through nozzles is used. The standard equations relating flow through nozzles for a compressed gas are used. "Y" is a standard expansion factor based on ratio of inlet to outlet pressure, k – the ratio of specific heats, and β -- the ratio of orifice diameter to pipe diameter.

$$Q = 0.525 \cdot Y \cdot D^{2} \cdot C\sqrt{P \cdot \rho}$$

"TRANSIENTS" In addition to calculation of basic flow rates and pressure drops, certain transient conditions in these systems must be considered. The major "transient" and miscellaneous effects to be considered are:

The initial pressure wave and the amount of time it took that pressure wave to reach the individual nozzles.

Very rapidly changing pressures and flow rates throughout the course of the discharge.

Discharge time.

The maximum initial flow into the system is restricted first by the manifold orifice and second by the velocity of sound in the inert gas. The amount of time to build full pressure at each nozzle depends on the length of pipe between the manifold orifice and the nozzle and the bulk modulus of air in the pipe leading to the nozzle.

Figure 8 is a recording of Pressure versus time for a three nozzle test system. Nozzle 103 was closest to the cylinder, and on this graph we can see the very rapid rise in pressure at nozzle 103. Nozzle 101 was next along the branch and we see a somewhat slower rise in pressure. Nozzle 102 was quite a distance downstream of the first two nozzles and there is a very slow rise in pressure at that nozzle.

The discharge time for an inert gas system is quite difficult to measure directly. With liquefied compressed gas systems, there is a noticeable demarcation between the predominate liquid phase of the discharge and the point at which the system runs out of liquid and discharges only vapor. Thermodynamically we can calculate the quantity of agent which would leave the cylinder as liquid, and based on that calculate a discharge time. With INERGEN there is only a vapor discharge. By testing, a reference point for discharge time was determined by relating cylinder pressure to the percent of agent discharged. For this test an INERGEN cylinder was placed on a recording scale and tracings of weight versus time were made.

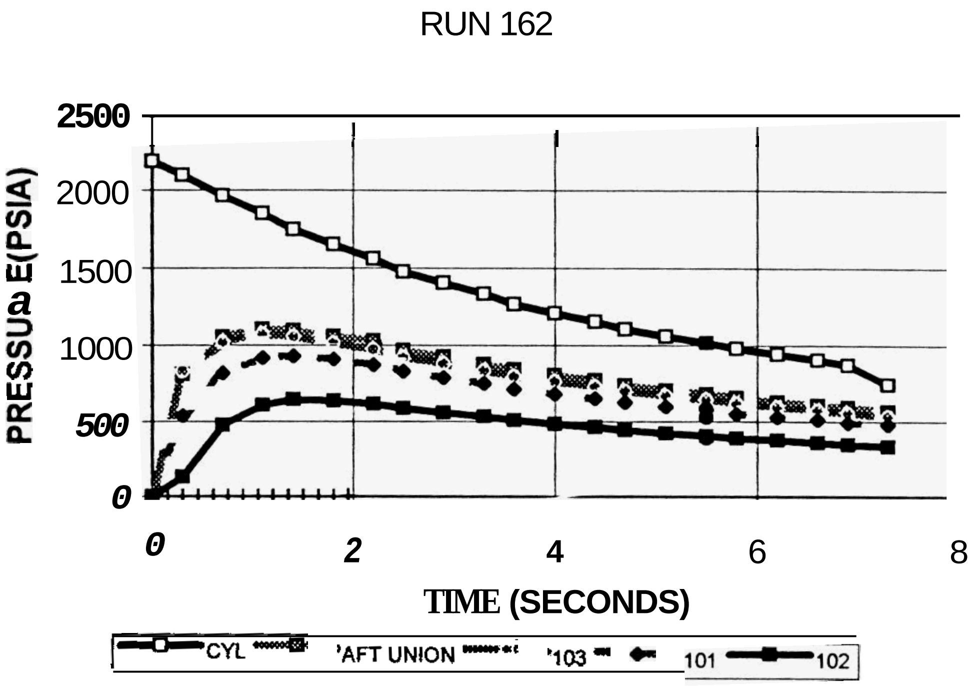


Figure 8

Figure 9 (next page) is an isometric for the system whose pressures are shown in Figure 8.

Because of the very rapid changes in cylinder and nozzle pressure conditions, it is appropriate to calculate pressure and flow rate for each 10% increment discharged from the cylinders.

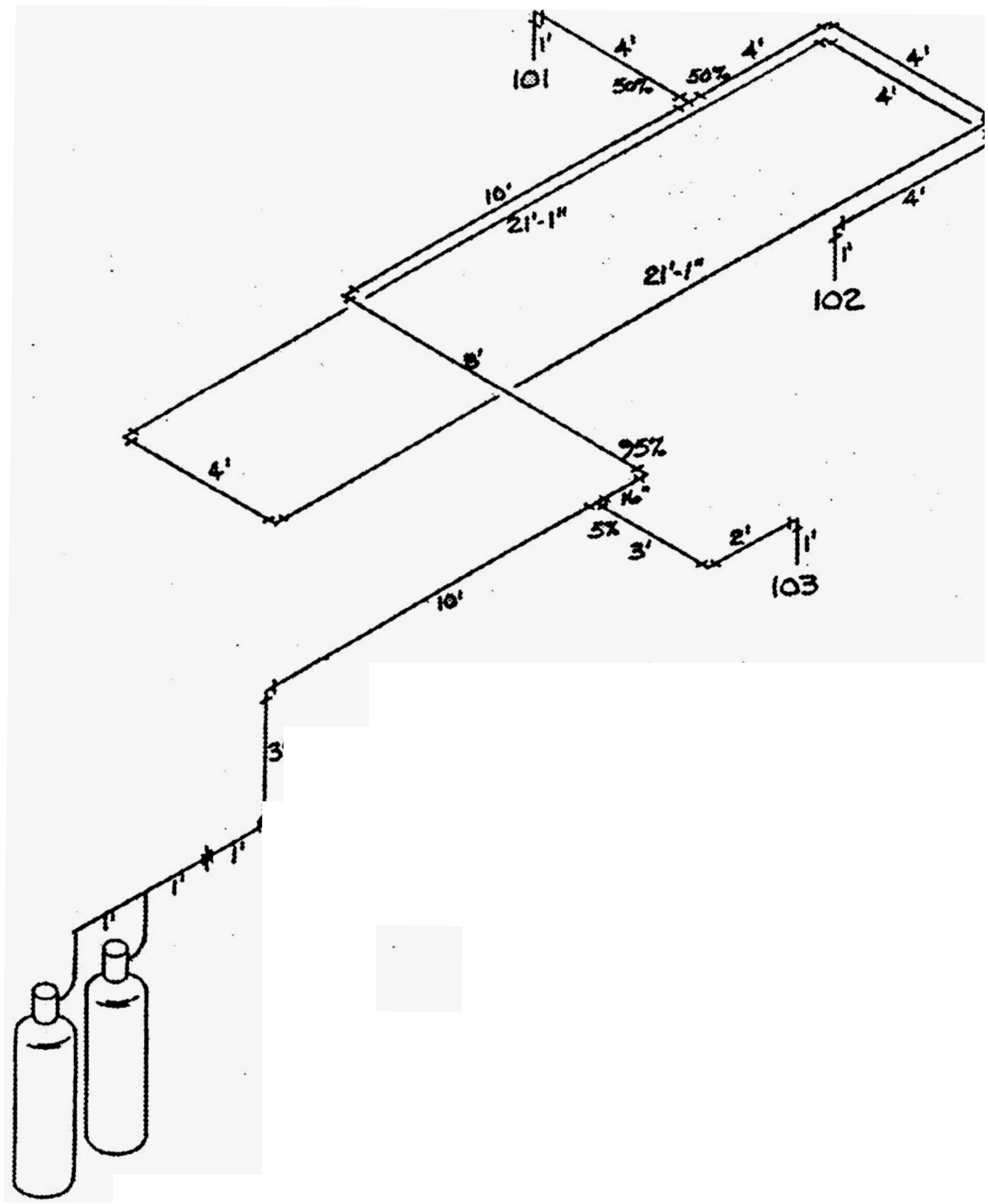
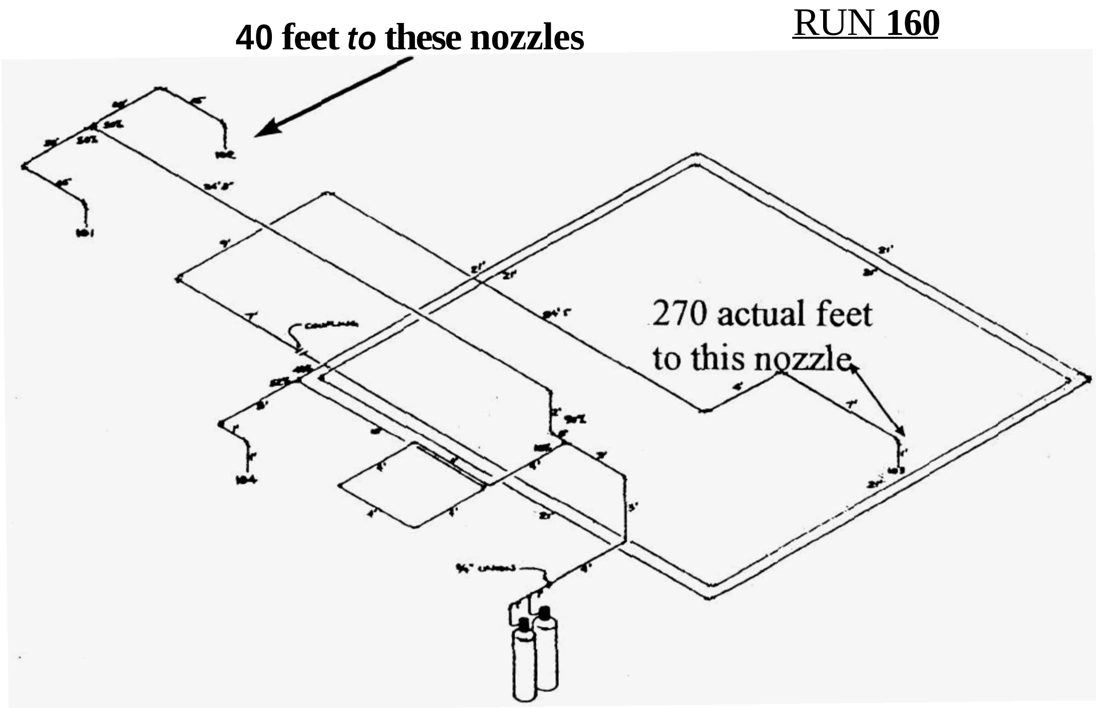


Figure 9 Isometric for RUN 162

CONCLUSION In reviewing the work that was done on the **IG 541** flow calculations, the major accomplishment was the recognition of the fact that **INERGEN** flow did deviate from both the adiabatic and the isothermal extremes enough that neither theoretical assumption could be **used** for complex systems. The introduction of a theoretical heat correction permitted calculation of extremely complex systems such as the one shown in the following isometric diagram. The results of the discharge test for this system are compared with the calculated predictions on this chart. Nozzle 103 which is **fed** by some 270 ft. of pipe has the worst error, with that error less than **3.5%.**



5/5/96 Inert Gas 46

RUN 160	CU FT ACT	CU FT CALC	%ACT/ CALC
101	134.2	135.6	-1.03%
102	138.9	138.4	0.36%
103	43.3	41.87	3.4246
104	103.6	104.2	4.53%

Though we do not recommend designing a system with such extreme piping, the demonstrated ability to calculate flow from such a system is heartening. It gives a great deal of confidence in the accuracy *of* these flow calculations when applied to reasonable systems in the field.