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8. OPTIMIZATION OF SYSTEM DISCHARGE

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8.1 Overview

Current aircraft fire suppression bottles for dry bay and engine nacelle applications, which are designed to meet Military Specification MIL-C-22284A (proof pressure of 9.62 MPa and minimum burst pressure of 12.37 MPa), are normally filled with liquid halon 1301 (CF_3Br) to about half of the bottle volume, and the bottle is then pressurized with nitrogen to a specified equilibrium pressure (typically 4.1 MPa) at room temperature. The purpose of using the pressurization gas is to expedite the discharge of the agent and to facilitate the dispersion of the agent. Without nitrogen pressurization, the bottle pressure, which is simply the vapor pressure of the agent, can be so low (even sub-atmospheric) at extremely cold ambience that there is virtually no driving force to expel the agent from the bottle in case of a fire, thus hindering a rapid release of the mixture.

From the above description, three important issues have emerged and need be considered when using a halon alternative as an in-flight fire suppressant: (1) the system hardware, (2) the thermophysical properties of the agent/nitrogen mixture, and (3) the agent/nitrogen mixture behavior during a discharge.

In this section, the last two issues which are pertinent to agent storage and its subsequent discharge will be addressed in detail. With respect to the first issue, a survey of the system constraints on existing in-flight halon 1301 fire protection systems will be given. This survey is also used to provide guidelines for identifying important experimental parameters for subsequent agent/nitrogen storage and discharge studies. The information on system constraints consists of bottle size, percent liquid-fill, nitrogen pressurization, range of operating temperatures, and proof and burst pressures of the vessel. In addition, the information on piping systems used to distribute halon 1301 for engine nacelle fire protection applications will be discussed. The thermophysical properties of selected agents and agent/nitrogen mixtures are presented under conditions commensurate with those identified in the system constraint studies. This information is also relevant to the establishment of the initial conditions of the vessel before the discharge of the mixture. Finally, parametric studies on the discharges of agent/nitrogen mixtures without (to simulate dry bay applications) and with (to simulate engine nacelle applications) a piping system will be discussed. Numerical calculations using CONCHAS-SPRAY and KIVA II were also performed in order to explore the feasibility of applying such complex codes to the study of a flashing discharge into an enclosure. This section focuses on the four selected agents (HFC-125, FC-218, CF₃I, and HFC-227ea) for dry bay (HFC-125, FC-218, and CF₃I) and engine nacelle (HFC-125, CF₃I, and HFC-227ea) applications.

The results obtained from this study provide important technical information on bottle design and agent discharge for new generation aircraft that may use the halon alternatives, for the existing aircraft that may undergo retrofitting, or simply for possible "drop-in" replacements.

8.2 System Constraints

This section describes a survey summarizing the current in-flight halon 1301 systems. This work was conducted with the cooperation of several aircraft fire extinguisher manufacturers, aircraft and airframe manufacturers, defense contractors, the three branches of the U.S. Military Service, and the Federal Aviation Administration (FAA). This survey is the result of examining the halon 1301 system specifications for over fifty types of aircraft (both military and commercial). The halon 1301 systems included in this survey are used for fire protection in engine nacelles, dry bays, and auxiliary power units (APU). Table 1 lists the participants in the survey and the acquired information. Since some of the information obtained was considered proprietary, it was not possible to tabulate the system

Participant	Acquired information
Boeing	Piping
Booz-Allen & Hamilton, Inc.	Bottle sizes, agent fill volumes, operating temperaures, vessel growth potential, nitrogen pressurization
Federal Aviation Administration	Bottle sizes and agent fill volumes
Lockheed Martin	Piping, and bottle sizes
McDonnell Douglas	Vessel growth potential
Northrop Grumman	Vessel growth potential
Pacific Scientific	Bottle sizes, agent fill volumes, operating temperatures, and nitrogen pressurization
Systron-Donner	Bottle sizes, agent fill volumes, operating temperatures, and nitrogen pressurization
Walter Kidde	Bottle sizes, agent fill volumes, operating temperatures, and nitrogen pressurization
U.S. Air Force	Bottle sizes, agent fill volumes, operating temperatures, and nitrogen pressurization
U.S. Army	Bottle sizes, agent fill volumes, operating temperatures, and nitrogen pressurization
U.S. Navy	Bottle sizes, agent fill volumes, operating temperatures, and nitrogen pressurization

Table 1.Names of the participants in the survey and the acquired information on current halon1301 system constraints

specifications for each individual aircraft in detail. Only the minimum and maximum operating ranges of the parameters were compiled in this study. Note that no differentiation among the areas of application were made in identifying the parameters. The survey is summarized in Table 2. The percent of liquid fill was calculated based on the total amount and liquid density of *pure* agent at 25 °C. In *most* applications, the bottle specifications closely conform to Military Specification MIL-C-22284A. The vessel growth potential is defined as the allowable percent increase in the original vessel size. However, it is not clear from the literature whether the growth potential refers to the increase in vessel size at the same mounting location of the vessel or different, new location when rerouting of agent distribution is feasible.

Based on the volume factor, it is not possible, in most cases, to use the existing containers for the selected halon alternatives as "drop-in" replacements because the containers may not be large enough to hold the amount of agent required for fire extinguishment or may encounter the so-called liquid-full condition (Grosshandler *et al.*, 1994) sooner even at moderately high ambient temperatures due to an initial small ullage as a result of an increased initial fill volume of the liquid halon alternative. The volume factor is defined as a measure of the liquid agent volume in the storage vessel *normalized* by the liquid volume of halon 1301 required to suppress a flame under identical conditions (Grosshandler *et al.*, 1994). The following calculations, using HFC-125 as an example, illustrate how the system constraints on the bottle size can be assessed based only on thermodynamic considerations. Note that

Table 2.Summary of current halon 1301 system specifications for engine nacelle, dry bay, and
auxiliary power unit applications

Property	Range	
Bottle size (cm ³)	650 to 23000	
Percent liquid fill (%)	25 to 70	
Operating temperature (°C)	- 54 to 160	
Nitrogen pressurization (MPa)	1.4 to 5.7	
Proof pressure (MPa)	6.4 to 24.7	
Burst pressure (MPa)	11.3 to 37.1	
Vessel growth potential (%)	0 to 400	

other constraints (*e.g.*, mechanical properties of the vessel, discharge time) have to be taken into account in the overall system constraint evaluation. The intent here is not to provide a comprehensive methodology for system constraint evaluation, but rather to provide system designers a set of engineering tools to perform design calculations (in this case, on bottle size).

Assuming the original halon 1301 bottle is a military standard CF-6 type vessel with an internal volume of $3.67 \times 10^{-3} \text{ m}^3$ and contains 2.96 kg of halon 1301 pressurized to 4.12 MPa with nitrogen at 21 °C, the percent of liquid fill at this condition is calculated to be 49.7 % using the computer code PROFISSY (to be discussed in detail in Section 8.3). The volume of liquid HFC-125 required is $5.47 \times 10^{-3} \text{ m}^3$, based on an average volume factor of 3 as reported in Grosshandler *et al.* (1994). If there is no vessel growth potential, then the original halon 1301 vessel can not be used for HFC-125 because the liquid volume of the agent is greater than the vessel volume. Table 3 summarizes the system constraint evaluation based on other vessel growth potentials. It is clear that even with a growth potential of 500 %, which results in 24.9 % liquid fill, the pressure at 150 °C is still very close to the burst pressure of a standard CF-6 vessel. In the calculations, a temperature of 150 °C was assumed to be the extreme operating temperature of the vessel. Similar calculations were also performed for the other three agents, and the results are given in Tables 4, 5, and 6.

To the best of our knowledge, no military specifications exist for storage and delivery systems for dry bay discharge applications. Since the events typically occurring in a dry bay are on the order of milliseconds, it is expected that the discharge will be directly from the vessel without going through any piping systems. For engine nacelle applications, piping systems are required to transport halon from the bottles (normally located outside the engine nacelle) to the point of application in the nacelle. The design of the piping system centers around Military Specification MIL-E-22285 (1959) and its amendment (1960). According to these documents, the discharge time, defined as the time measured from the instant the agent (halon 1301) starts to leave the tubing ends to the time the required amount of agent has been discharged, should be 1 second or less. The initial amount of agent in the container(s) should be able to provide an agent concentration of at least 6 % by volume in all parts of the affected zone for at least 0.5 second at normal cruising condition. The pipe lines should be less than 3.1 m (10 ft) long, where practicable, and be straight if possible. The fittings and turns in the piping system should be kept to a minimum. Discharge piping should terminate in open ends without

Vessel growth potential (%)	New vessel volume (m ³)	Percent liquid fill (%)	Vessel pressure @ 150 °C (MPa)	
0	3.67 x 10 ⁻³	Not possible	Not calculated	
50	5.51 x 10 ⁻³	99.4	> 41.23	
100	7.34 x 10 ⁻³	74.6	26.65	
200	11.01 x 10 ⁻³	49.7	16.77	
300	14.68×10^{-3}	37.3	14.01	
400	18.35 x 10 ⁻³	29.8	12.61	
500	22.02×10^{-3}	24.9	11.73	

Table 3.	Sample calculations showing vessel size constraint for HFC-125 based on an average
	volume factor of 3 and various vessel growth potentials

nozzles, and tubing and fittings should have a minimum burst pressure of 10.3 MPa. Tubing sizes are determined based on the desired discharge rates and on system-volume considerations (*i.e.*, minimizing the system volumes). It is likely that any system employing a halon replacement agent would, at least, have to meet similar performance criteria and system constraints.

8.3 Thermophysical Properties of Selected Agents and Agent/Nitrogen Mixtures

8.3.1 Introduction. The total pressure in the bottle is a complex function of ambient temperature because of the temperature dependence of the agent vapor pressure, the partial pressure of nitrogen in the gas ullage, and the solubility of nitrogen in the liquid agent. For halon 1301, the bottle pressure-temperature relationship and the solubility of nitrogen in halon 1301 have been well characterized. By contrast, such a relationship and solubility data are scarce or do not exist for some of the halon alternatives.

The major objectives of this study are very similar to those reported in the previous work (Grosshandler *et al.*, 1994), the only difference being the initial liquid agent fill conditions at room temperature and the number of agents used. The present study expands the temperature-pressure database for the four selected agents (HFC-125, FC-218, CF_3I , and HFC-227ea) using two additional initial fill conditions and an experimental apparatus that can accommodate much higher final pressures at elevated temperatures than the earlier study.

The first subtask of this study was to determine the solubility of nitrogen in the four selected agents at room temperature. Halon 1301 was also included in this study for the purposes of reference and comparison with the alternative agents. Specifically, the objective was to measure the amount of nitrogen that was needed to pressurize the bottle to a specified equilibrium pressure.

In the second subtask, the thermodynamic properties of the four selected agents together with halon 1301 were measured. The objective was to determine the final pressure of the bottle when the bottle, filled with either a pre-determined amount of pure agent or with agent pressurized with nitrogen, was exposed to two extreme temperatures (-60 °C and 150 °C). Experiments on pure

Vessel growth potential (%)	New vessel volume (m ³)	Percent liquid fill (%)	Vessel pressure @ 150 °C (MPa)
0	3.67 x 10 ⁻³	Not possible	Not calculated
50	5.51 x 10 ⁻³	92.8	30.81
100	7.34×10^{-3}	69.6	18.43
200	11.01×10^{-3}	46.4	12.88
300	14.68 x 10 ⁻³	34.8	11.22
400	18.35 x 10 ⁻³	27.9	10.38
500	22.02 x 10 ⁻³	23.2	9.83

Table 4.Sample calculations showing vessel size constraint for FC-218 based on an average
volume factor of 2.8 and various vessel growth potentials

agents were performed in order to obtain pressure-temperature relationships for some of the agents (especially CF_3I) where documented data were not readily available.

Furthermore, in order to assimilate the results obtained from the two aforementioned subtasks into useful engineering correlations that could be easily used by the bottle designers, the third subtask was to develop a computer code that could facilitate such calculations.

For the purpose of reference, selected thermophysical properties of the four selected agents are tabulated in Table 7 where MW is the molecular weight, T_b is the normal boiling point (at 0.101 MPa), T_c is the critical temperature, P_{sat} is the saturation vapor pressure at 25 °C, P_c is the critical pressure, ρ_c is the critical density, ρ_l is the saturated liquid density at 25 °C, C_{pl} is the isobaric liquid heat capacity at T_b , C_{p2} is the isobaric liquid heat capacity at T_b . The references from which these data were taken are also given in the table.

8.3.2 Experimental Apparatus and Procedures. This section describes the experimental techniques used to measure the amount of nitrogen required to pressurize the vessel initially filled with a fixed amount of agent to a specified equilibrium pressure at room temperature and the pressure-temperature relationship for pure agents and agent/nitrogen mixtures. The initial amounts of liquid agent dispensed in the vessel before nitrogen pressurization corresponded to approximately half and two-thirds of the vessel volume at room temperature. For agent/nitrogen mixtures, two initial equilibrium nitrogen charge pressures (2.75 MPa and 4.12 MPa) at room temperature were used in this study. The amount of nitrogen required to pressurize the vessel with agent to 2.75 MPa or 4.12 MPa, were measured only at room temperature. To determine the final vessel pressure as a function of temperature, experiments were performed at -60 °C, 22 °C, and 150 °C.

8.3.2.1 Agent Filling Calculations. This sub-section describes the method used to calculate the total amount of agent needed such that the liquid volume fraction corresponds to the amount that is one-half or two-thirds of the vessel volume at room temperature. The calculation procedure is as follows.

Since two phases (liquid-vapor) exist in equilibrium for all the four selected agents at room temperature, the total mass $(m_{a,t})$ of agent in the vessel with a total volume V_t is equal to the sum of the masses of agent in both vapor and liquid phases.

Vessel growth potential (%)	New vessel volume (m ³)	Percent liquid fill (%)	Vessel pressure @ 150 °C (MPa)	
0	3.67×10^{-3}	39.8	11.53	
50	5.51×10^{-3}	26.5	10.46	
100	7.34×10^{-3}	19.9	9.85	
200	11.01 x 10 ⁻³	13.3	9.04	
300	14.68 x 10 ⁻³	10.0	8.52	
400	18.35 x 10 ⁻³	8.0	8.18	
500	22.02×10^{-3}	6.6	7.93	

Table 5. Sample calculations showing vessel size constraint for CF_3I based on an average volume factor of 0.8 and various vessel growth potentials

$$m_{a,t} = m_{a,y} + m_{a,l} = \rho_{a,y}V_y + \rho_{a,l}V_l$$
(1)

where *m* is the mass, ρ is the saturation density of the agent, *V* is the volume, subscripts *t*, *a*, *v*, and *l* represents total, agent, vapor phase, and liquid phase respectively. For a given agent at room temperature, $\rho_{a,v}$ and $\rho_{a,l}$ are known or can be estimated. If the vessel is half filled with liquid agent,

$$V_l = V_v = \frac{V_l}{2} \tag{2}$$

Similarly, for a two-thirds liquid full vessel,

$$V_l = \frac{2V_l}{3} \tag{3}$$

$$V_{\nu} = \frac{V_t}{3} \tag{4}$$

With V_l and V_v calculated by the method shown above, $m_{a,t}$ can then be determined from Equation (1). In the calculations for HFC-227ea, FC-218, HFC-125, and halon 1301, $\rho_{a,v}$ and $\rho_{a,l}$ were obtained from the database developed by Gallagher *et al.* (1993). For CF₃I, $\rho_{a,l}$ was estimated by the modified Rackett method (Reid *et al.*, 1987), and $\rho_{a,v}$ was calculated by using the vapor pressure at room temperature and assuming an ideal vapor phase.

Vessel growth potential (%)	New vessel volume (m ³)	Percent liquid fill (%)	Vessel pressure @ 150 °C (MPa)
0	3.67×10^{-3}	Not possible	Not calculated
50	5.51×10^{-3}	82.9	26.65
100	7.34 x 10 ⁻³	62.2	15.74
200	11.01 x 10 ⁻³	41.4	11.80
300	14.68×10^{-3}	31.1	10.71
400	18.35 x 10 ⁻³	24.9	10.07
500	22.02×10^{-3}	20.7	9.62

Table 6.	Sample calculations showing vessel size constraint for HFC-227ea based on an average
	volume factor of 2.5 and various vessel growth potentials

8.3.2.2 Agent Preparation. Before the agent was dispensed into the test vessel, it was degassed to remove any dissolved noncondensible gases which might be present in the agent. Figure 1 is a schematic of the degassing apparatus. The entire apparatus was first evacuated to 1.33 Pa for at least half an hour before agent vapor was condensed in the sublimizer which was chilled in a dry ice bath. The sublimizer was a stainless steel vessel with a re-entrant well at the center. After sufficient liquid had been condensed, liquid nitrogen was poured into the well. The liquid agent in the sublimizer was then made to boil by applying vacuum. As the liquid boiled, it would give off its own vapor and the noncondensible gases. Because the wall of the well was very cold due to the presence of liquid nitrogen, the agent vapor was condensed on the wall, and the condensate then dripped back to the boiling liquid agent. In this way, only the noncondensible gases and a very small amount of agent vapor were removed from the sublimizer by the vacuum pump. Depending on the initial amount of liquid agent in the sublimizer, this degassing process normally went on for at least half an hour.

Because the initial batch of CF_3I was found to contain trace amounts of HI, CO_2 , H_2O , and I_2 , a purification system was constructed in order to remove impurities which might be present in other batches. The purification system is shown schematically in Figure 2. After the entire system was evacuated, the agent vapor passed through a column packed with potassium hydroxide whose main function was to remove HI, then through a molecular sieve filter to remove water vapor and carbon dioxide, and finally through a stainless steel coil which was immersed in an ice bath to remove I_2 . The purified vapor was then retrieved by condensation in a stainless steel container. The collected agent was again subjected to boiling using a vacuum at dry-ice temperature for about half an hour to further remove noncondensible gases.

8.3.2.3 Pure Agent Measurements. For low temperature $(-60 \, ^{\circ}\text{C})$ measurements of the pure agents, the experimental set-up, which is similar to the one used in the previous agent screening project (Grosshandler *et al.*, 1994), is shown in Figure 3. It consists of a stainless steel (SS 304)

Agent	MW	T _b	T _c	P _c	P _{sat}	ρ _c	Ρι	C _{pl}	C _{p2}	h _{fg}
	(kg/mol)	(°	C)	(M	Pa)	(kg/	/m ³)	(kJ/k	(g K)	(kJ/kg)
HFC-227ea	0.170	-16.4 ^a	101.9ª	2.95ª	0.46 ^a	621 ^b	1389 ^a	1.074 ^a	1.177 ^a	126 ^a
CF ₃ I	0.196	-22.0 ^c	122.0 ^d	4.04 ^d	0.49 ^e	871 ^d	2106 ^f	0.542 ^g	0.592 ^g	106 ^h
FC-218	0.188	-36.8ª	72.0 ^a	2.68 ^a	0.87ª	629 ⁱ	1251 ^a	0.977 ^a	1.190 ^a	101 ^a
HFC-125	0.120	-48.6ª	66.2 ^a	3.63ª	1.38 ^a	571 ^j	1190 ^a	1.126 ^a	1.426 ^a	164 ^a
CF ₃ Br	0.149	-57.8 ^a	67.0 ^a	4.02 ^a	1.61 ^a	745 ^k	1551 ^a	0.670 ^a	0.881 ^a	111 ^a

Tał	ole 7.	Selected	thermoph	vsical i	propertie	es of	agents

^aFrom Gallagher et al. (1993)

^bFrom Great Lakes Chemical Corporation (1993)

^cFrom Kudchadker et al. (1979)

^dFrom Sladkov and Bogacheva (1992)

^eEstimated by the method of Lee and Kesler (Reid et al., 1987)

^tEstimated by the modified Rackett method (Reid et al., 1987)

^gEstimated by the method of Rowlinson (Reid et al., 1987)

^hEstimated by Pitzer acentric factor correlation (Reid et al., 1987)

ⁱFrom Braker and Mossman (1980)

^jFrom Allied Signal Inc. (1990)

^kFrom ASHRAE Inc. (1969)

vessel (tested hydrostatically to 20.6 MPa), a needle valve (Whitey SS-ORS2)¹ for dispensing the agent and nitrogen, a K-type thermocouple (Omega TJ36-CASS-116U-12), and a pressure transducer (Druck Model PDCR 330) with an uncertainty of 6.9 kPa. The volume of the whole system was determined by filling the system with nitrogen to various specified pressures at room temperature and weighing the pressurized system on an electronic scale with an uncertainty of 0.1 g. With the mass of nitrogen, temperature, and pressure known, the volume was calculated by using a generalized correlation for the compressibility factor (Smith and Van Ness, 1975). The total volume of the apparatus was determined to have a mean of 52.2 cm^3 with a standard deviation of 0.3 cm³.

The experimental procedure was as follows. The apparatus was evacuated to 1.33 Pa for at least ten minutes. The set-up was then connected to the agent supply bottle (not shown in Figure 3). The vessel was placed in a Dewar flask and chilled by using dry ice before gaseous agent was dispensed through the needle valve into the vessel. Liquid agent was obtained by condensing the agent vapor, and the total amount was metered by placing the vessel and the Dewar flask on an electronic balance. When the amount of agent reached the target mass, the needle valve was closed. The vessel was then removed from the Dewar flask, warmed back to room temperature, and weighed on an electronic balance with an uncertainty of 0.1 g to determine the actual mass in the system.

The filled vessel was then immersed in a heat transfer fluid (Dow Corning Syltherm XLT) which was chilled by two immersion coolers (NESLAB Models IBC-4A and CC100-II) to -60 °C. The

¹Certain commercial products are identified in this report in order to specify adequately the equipment used. Such identification does not imply recommendation by the National Institute of Standards and Technology, nor does it imply that this equipment is the best available for the purpose.



Figure 1. Schematic of the degassing apparatus.



Figure 2. Schematic of CF₃I purification system.



Figure 3. Experimental set-up for cold temperature measurements.

final vessel pressure was recorded from the pressure transducer read-out at least an hour later, after the internal temperature had reached thermal equilibrium with the bath temperature.

For high temperature (150 °C) measurements, a different set-up was constructed because the vessel used in the cold temperature experiments could not withstand the pressure that resulted from the high temperature condition. The apparatus is shown schematically in Figure 4. It consisted of a stainless steel (SS 316) high pressure vessel with a working pressure rated for 68.7 MPa at 204 °C (Jerguson Series 51 liquid level sight gage), a magnetic pump/mixer, a pressure transducer (Druck Model PDCR 330), and taper seal high pressure valves (HiP Company). Heating was achieved by placing the test apparatus in a temperature controlled oven. The internal volume of the entire system with the valves to the vacuum pump, vent, nitrogen bottle, and sublimizer closed was determined by evacuating the system followed by sucking distilled water into the system from a beaker. The amount of water required to fill the system was then determined from the loss of weight of the beaker. Knowing the mass of water and its density at room temperature, the internal system volume was calculated to have an average of 42.6 cm³ with a standard deviation of 0.2 cm³. Each window on the pressure vessel was also calibrated in terms of the volume percent of liquid filled to facilitate the filling process. The calibration was conducted by measuring the amount of water required to fill up to each window. In order to minimize photolysis of CF₃I, agent filling was performed under a darkroom safelight. In addition, when CF₂I was used in a test, extreme caution was taken to prevent any stray room light from getting into the oven.

The experimental procedure was as follows. The entire system was evacuated for at least half an hour before liquid agent was dispensed from the sublimizer. The exact amount of agent in the system was determined by weighing the sublimizer. The oven was then turned on, and the test apparatus was heated to the desired temperature. Because of space limitation, it was not possible to insert a thermocouple into the vessel to monitor the internal temperature. To ensure that thermal equilibrium had been reached, two thermocouples were placed at two different locations on the vessel wall, and the final pressure was recorded only when the two thermocouple readings were within 1.0 °C of the set temperature of the oven and the final pressure maintained a steady reading for at least half an hour.

8.3.2.4 Nitrogen Solubility in Agents. To measure the total amount of nitrogen required to pressurize the vessel to a specified equilibrium pressure at room temperature, the set-up for the cold temperature experiments was used. The agent filling procedure was the same as in the cold temperature studies. After a pure liquid agent was dispensed in the vessel, the nitrogen gas supply line (not shown in Figure 3) was connected to the vessel. The supply line was evacuated for at least five minutes before initiating a slow flow of nitrogen. The needle valve was then opened to allow nitrogen to bubble slowly through the liquid agent until an equilibrium pressure of approximately 2.8 MPa was reached. Shaking the vessel intermittently and vigorously and repetitive bubbling of nitrogen was required before the final equilibrium pressure was attained. The amount of nitrogen needed to pressurize the vessel was obtained by weighing the apparatus. This amount corresponded to the sum of the mass of nitrogen in the gas ullage of the vessel and that dissolved in the liquid agent. The charged vessel was then ready for the cold temperature experiments, as described in Section 8.3.2.5. When the cold experiments were completed, the charged vessel was warmed back to room temperature and was pressurized again with nitrogen to approximately 4.1 MPa. The amount of nitrogen was then determined, and the vessel was ready for another cold temperature experiment.

8.3.2.5 Agent/Nitrogen Mixture Measurements. The same two apparatus (see Figures 3 and 4) used for pure agent studies were also utilized to perform agent/nitrogen mixture experiments. In fact, since the experimental procedure for the solubility studies was equivalent to the preparation of the vessel at room temperature for subsequent pressure-temperature measurements, cold temperature

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Figure 4. Apparatus for high temperature experiments.

experiments were therefore performed immediately after the solubility measurements were made by simply immersing the same vessel in the chilled heat transfer fluid.

For high temperature measurements, the experimental procedure was as follows. The same agent filling procedure in the pure agent studies was used. After the agent was dispensed, the pressurization of the vessel with nitrogen to a specified equilibrium pressure (approximately 2.8 MPa) at room temperature was initiated with the magnetic pump running to facilitate the mixing of nitrogen with the agent. After pressurization, the apparatus was heated to the final temperature (150 °C), and the final pressure of the vessel was recorded the same way as in the pure agent experiments. The apparatus was then cooled back to room temperature, pressurized with nitrogen to approximately 4.1 MPa, and heated to the final temperature (150 °C), where the equilibrium pressure was recorded.

8.3.3 Modeling Thermodynamic Properties of Agent/Nitrogen Mixtures

8.3.3.1 Introduction. The thermodynamic properties of agent/nitrogen mixtures are modeled using a computer program PROFISSY (acronym for Properties Of Fire Suppression Systems). The program incorporates a model known as "extended corresponding states" (ECS). This model has been quite successful in modeling the thermodynamic and transport properties of fluids. An early model of this type, known as the TRAPP model (Ely and Hanley, 1981a, 1981b, and 1983), has been successful in the prediction of hydrocarbon transport properties, including LNG (McCarty, 1982). Much of the development of TRAPP occurred in the early 1980's at the National Institute of Standards and Technology. Since that time, the model has been revised, improved, and used successfully to predict the properties of hydrocarbon mixtures (Ely, 1982; Huber and Ely, 1990; Friend, 1992), air (Ely, 1990), and CO₂-rich hydrocarbon mixtures (Sherman et al., 1989), and forms the basis of two current Standard Reference Data Products (Huber and Ely, 1990; Friend, 1992). ECS has been used successfully to predict equilibrium properties such as density, heat capacity, speed of sound, entropy, enthalpy, Joule-Thomson coefficient, viscosity, and thermal conductivity. It is a powerful method, applicable to the entire range of fluid states, from dense liquid to dilute gas, as well as to the supercritical fluid regime. It may be used with only minimal information on a fluid: the critical point, the normal boiling point, and the molecular weight. Additional information on a fluid, such as vapor pressures, saturated liquid densities, and liquid viscosities can be used to refine the model predictions.

Recently ECS models have been used to model thermophysical properties of alternative refrigerants and their mixtures (Huber and Ely, 1992; Huber *et al.*, 1992; Gallagher *et al.*, 1995). In this work, we extended the model to calculate the thermodynamic properties of agent/nitrogen mixtures.

8.3.3.2 Extended Corresponding States Model. The central idea of extended corresponding states is that all points on the *PVT* surface of any fluid may be represented by scaling the *PVT* surface of a reference substance. These "scale factors" involve the critical properties of the fluid of interest and the reference fluid and may also be functions of temperature and density.

The basic equation in the extended corresponding states theory relates the dimensionless residual Helmholtz free energy of a pure fluid, denoted by subscript j, to that of a reference fluid, denoted by subscript o, whose thermodynamic properties are known, in principle, with great accuracy:

$$a_i^r(\rho_i, T_i) = a_o^r(\rho_o, T_o)$$
⁽³⁾

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where the residual Helmholtz free energy is defined as

$$a^{r} = \frac{A(\rho,T) - A^{*}(\rho,T)}{RT}$$
(6)

In the above two equations, A is the Helmholtz free energy, the asterisk indicates an ideal gas state value, ρ is the molar density, T is the absolute temperature, and R is the universal gas constant. The temperature and molar density of the fluid j are scaled with the corresponding values for the reference fluid o according to the following two equations:

$$T_o = \frac{T_j}{f_j} \tag{7}$$

$$\rho_o = \rho_j h_j \tag{8}$$

where the scale factors f_j and h_j are called the equivalent substance reducing ratios. In extended corresponding states between two pure fluids j and o, they are related to the critical parameters of the two fluids, namely

$$f_j = \frac{T_{c,j}}{T_{c,o}} \, \theta_j(\rho_j, T_j) \tag{9}$$

$$h_j = \frac{\rho_{c,o}}{\rho_{c,i}} \phi_j(\rho_j, T_j)$$
(10)

where the functions θ_j and ϕ_j are called shape factors, and subscript *c* denotes a critical property. In addition, the condition that the compressibility factors of the fluids be equal, $Z_j^r = Z_o^r$, is imposed. Other dimensionless residual thermodynamic properties of a fluid can be derived from differentiation of Equation (5) and can be found in Huber and Ely (1994).

To calculate properties of mixtures using extended corresponding states, we use what is known as a "one-fluid" mixture model. The properties of a mixture are found by first assuming the mixture to be a single hypothetical pure fluid (subscript x) through the use of some mixing and combining rules. In this way, the properties of the hypothetical pure fluid are then found by scaling with a single reference fluid o.

$$a_{mix}^{r}(\rho, T, [x_{j}]) = a_{x}^{r}(\rho_{x}, T_{x}) = a_{o}^{r}(\rho_{o}, T_{o})$$
(11)

In order to obtain the equivalent substance reducing ratios, f_x and h_x , for the hypothetical pure fluid, we use the van der Waals one-fluid mixing rules,

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$$h_{x} = \sum_{i=1}^{n} \sum_{j=1}^{n} x_{i} x_{j} h_{ij}$$
(12)

$$f_{x}h_{x} = \sum_{i=1}^{n} \sum_{j=1}^{n} x_{i}x_{j}f_{ij}h_{ij}$$
(13)

and the combining rules,

$$f_{ij} = \sqrt{f_i f_j} (1 - k_{ij}) \tag{14}$$

$$h_{ij} = \frac{(h_i^{1/3} + h_j^{1/3})^3 (1 - l_{ij})}{8}$$
(15)

where x is the mole fraction, n is the number of components, subscripts i and j refer to component i and j respectively, k_{ij} and l_{ij} are the binary interaction parameters that may be non-zero when $i \neq j$ and k_{ii} and l_{ii} are defined to be zero. The parameters are generally found empirically by fitting experimental data. However, in order to test the robustness of the ECS model without resorting to the experimental data which have to be obtained *a priori*, the binary interaction parameters are set equal to zero in the model.

Several factors contribute to the ability of the ECS model to accurately predict thermophysical properties. Two important factors are the choice of the reference fluid and the determination of the shape factors. It is important to have a very well characterized empirical reference fluid equation of state. The reference fluid used in this work is HFC-134a (1,1,1,2-tetrafluoroethane) which is represented by the Jacobsen-Stewart (1973) modified 32 term Benedict-Webb-Rubin equation of state (MBWR-32) with coefficients as determined by Huber and McLinden (1992). This equation has been used extensively to represent the properties of hydrocarbons, common inorganics, and most recently refrigerants. Its functional form is essentially a polynomial in density and temperature and is given by

$$p = \sum_{n=1}^{9} a_n(T) \rho^n + e^{-(\rho/\rho_c)^2} \sum_{n=10}^{15} a_n(T) \rho^{2n-17}$$
(16)

The temperature dependence of the $a_n(T)$ is summarized in Table 8. The 32 coefficients $(b_1 \text{ to } b_{32})$ for the reference fluid HFC-134a are tabulated in Huber and Ely (1994). Equation (16), when applied to HFC-134a, typically reproduces the pressures within 0.4 % and the density within 0.2 %. Exceptions are large density deviations in the near critical region, and large pressure errors in the low-temperature compressed liquid region near the saturation boundary.

Table 8.	. Т	emperature	dependence	of the	MBWR-32	coefficients

$a_I = RT$	$a_9 = b_{19} / T^2$
$a_2 = b_1 T + b_2 T^{1/2} + b_3 + b_4 / T + b_5 / T^2$	$a_{10} = b_{20} / T^2 + b_{21} / T^3$
$a_3 = b_6 T + b_7 + b_8 / T + b_9 / T^2$	$a_{11} = b_{22} / T^2 + b_{23} / T^4$
$a_4 = b_{10}T + b_{11} + b_{12} / T$	$a_{12} = b_{24} / T^2 + b_{25} / T^3$
$a_5 = b_{13}$	$a_{13} = b_{26} / T^2 + b_{27} / T^4$
$a_6 = b_{14} / T + b_{15} / T^2$	$a_{14} = b_{28} / T^2 + b_{29} / T^3$
$a_7 = b_{16} / T$	$a_{15} = b_{30} / T^2 + b_{31} / T^3 + b_{32} / T^4$
$a_8 = b_{17} / T + b_{18} / T^2$	

The accuracy one obtains from ECS models also depends on how well the shape factors are determined. The shape factors are functions of both density and temperature. In principle, if one has accurate, analytical equations of state for the two fluids, the exact shape factors relating them can be found, as discussed by Ely (1990). However, often only one fluid (the reference) has an accurate, analytical equation of state available. In this situation, one usually assumes an empirical form for the shape factors. Leach *et al.* (1968) expressed the shape factors as weak functions of temperature and density using the acentric factor as a third correlating parameter. Since the shape factors are much stronger functions of temperature than density and most of the data which are available for refrigerants are for the saturation boundary, we have chosen a different approach utilizing information along the saturation boundary of the fluids (Huber and Ely, 1994) which results in density independent shape factors. It can be shown (Huber and Ely, 1994) that the vapor pressures and the saturated liquid densities of a fluid can be related to the reference fluid properties through the following equations

$$p_{sat, j}(T_j) = p_{sat, o}\left(\frac{T_j}{f_j}\right) \frac{f_j}{h_j}$$
(17)

$$\rho_{sat, j}(T_j) = \rho_{sat, o}(\frac{T_j}{f_j}) \frac{1}{h_j}$$

(18)

where subscript sat refers to a property of a fluid along the saturation boundary. Equations (17) and (18) are simultaneously solved to obtain f_j and h_j along the saturation boundary. Vapor pressures and saturated liquid densities are provided from empirical correlations of data, if available, or from estimations (Reid *et al.*, 1987). Additional information on the procedure for obtaining shape factors can be found in Huber and Ely (1994).

We do not yet have extensive results on the performance of the ECS model for predicting the density of agent/nitrogen mixtures; however, our experience with hydrocarbon mixtures and with refrigerant mixtures gives us confidence that generally the densities of the liquid phase are accurate to within 3 %. This number can be improved with the use of the binary interaction parameters.

8.3.3.3 Calculation Procedure. The computer code PROFISSY was developed with the primary purpose of helping fire suppression bottle designers or users to obtain temperature-pressure characteristics of the bottle. In other words, given a vessel (V_t) charged with agent and nitrogen at room temperature, one would like to know what the final vessel pressure will be when the vessel is exposed to a different temperature. In this aspect, only the information which is pertinent to the problem is provided in the output although the computer program can perform other thermodynamic property calculations.

The first step in the calculation is to determine the initial conditions of the vessel, that is to determine the compositions of the liquid and vapor phases, the fraction of the mixture (agent/nitrogen) that will be in the vapor phase, and the amount of nitrogen required to pressurize the vessel or the initial total pressure of the vessel given an initial amount of nitrogen. To perform such calculations, it is required that the amount of agent, the fill temperature, the vessel size, and the total pressure of the vessel or the amount of nitrogen in the vessel be known. The calculation is essentially a flash calculation at fixed T and P. The flash calculation can be generally stated as follows:

Given a mixture, in this case agent/nitrogen, whose bulk compositions (z_j) are known at T and P, determine x_j , y_j , and fraction vaporized (α), where x is the liquid mole fraction and y is the vapor mole fraction. The solution of the problem requires mass balance and vapor-liquid equilibrium calculations. A detailed computational procedure can be found in Prausnitz *et al.* (1980) and Walas (1985). In our case, the calculation is slightly different depending on whether (1) the amount of nitrogen $(n_{g,t})$ or (2) the total initial charge pressure (P_i) is given.

In Case (1), with V_{t} , T_{i} , $n_{a,t}$, and $n_{g,t}$ given, we calculate P_{i} , x_{g} , y_{g} , and α , where *n* is the number of moles and subscripts, *t*, *g*, and *a* represent total, nitrogen, and agent respectively. Note that for a binary system x_{a} and y_{a} are known once x_{g} and y_{g} are determined. In Case (2), knowing V_{t} , T_{i} , $n_{t,a}$, and P_{i} , then $n_{g,t}$, x_{g} , y_{g} , and α are calculated. Irrespective of Case (1) or (2), bubble point and dew point calculations (see *e.g.*, Prausnitz *et al.*, 1980) need be carried out to ensure that the flash calculations are performed in the two-phase region. Flash calculations are feasible only when the vessel conditions are above the bubble point and below the dew point. Below the bubble point, the mixture is a subcooled liquid; in this case, $x_{i} = z_{i}$. Above the dew point, the mixture is a superheated vapor; in this case $y_{i} = z_{i}$.

Once the initial conditions of the vessel are established, calculations of temperature-pressure characteristics can proceed. In this case, given V_t , $n_{a,t}$, and $n_{g,t}$, we want to calculate P_f , x_g , y_g , and α at T_f if the resulting condition is in a two-phase region. The computational procedure is similar to Case (1) above. If the resulting condition in the vessel is in a single-phase region, then straightforward PVT calculations will provide P_f given V_t , T_f , and z_i (= x_i or y_i).

Only four pieces of information are required to run the program: (1) agent mass, (2) vessel volume, (3) fill temperature, and (4) either nitrogen mass needed to pressurize the vessel, or the fill pressure of the vessel. Appendix A provides a typical session showing how to run the computer program. The current version of the program only supports the agents used in this study although other agents can be incorporated into the program by modifying the property data base.

8.3.4 Results and Discussion

8.3.4.1 Pure Agents. The results of the temperature-pressure measurements for pure agents are summarized in Table 9. In all cases, the combined standard uncertainty of the measurement was estimated to be less than 0.2 MPa based on the law of propagation of uncertainty with the sensitivity coefficients calculated using the ideal gas law. The total agent mass in the second column corresponds to the initial liquid agents whose volumes in the two cases measured approximate one-half or two-

Agent	Agent, $m_{a,t}$ (g)	P_f (measured) (MPa)	P_f (predicted) (MPa)	error ^a (%)
HFC-227ea	28.3	6.83	6.47	5.3
	30.3	7.10	6.74	5.1
	40.7	10.82	10.33	4.5
· · · · · · · · · · · · · · · · · · ·	38.2	10.09	9.32	7.6
CF ₃ I	43.0	6.51	6.62	1.7
	43.0	6.77	6.77	0.0
	57.3	8.84	9.23	4.4
	58.6	9.52	9.85	3.5
FC-218	28.9	8.33	8.01	3.8
	29.1	8.32	8.01	3.7
	37.4	11.90	11.72	1.5
	37.3	11.85	11.74	0.9
HFC-125	26.1	11.57	11.29	2.4
	34.6	16.61	16.01	3.6
	32.3	15.21	14.83	2.5
CF ₃ Br	34.9	11.98	11.50	4.0
	33.8	11.68	11.21	4.0
	44.4	16.28	15.55	4.5
	45.6	16.97	16.26	4.2

Table 9. Comparison of measured pressures of pure agents at 150 °C ± 2 °C with predicted values

^aerror (%) = $|P_f(\text{measured}) - P_f(\text{predicted})| \times 100 / P_f(\text{measured})|$

thirds of the system volume at room temperature. The measurements were obtained at ~ 150 °C. Only the high temperature results are reported here because at low temperature (- 60 °C) the pressures for all the agents were found to be below 0.1 MPa, which was the lowest value that could be measured from the pressure transducer.

For all the tests with liquid agent initially half filling the vessel, the final pressures at 150 $^{\circ}$ C were found to be less than 12.5 MPa. For an initially two-thirds filled vessel, the final pressures at 150 $^{\circ}$ C were measured to be less than 17.5 MPa.

The predicted final pressures obtained using PROFISSY are also tabulated in Table 9. Note that PROFISSY was developed specifically for agent/nitrogen mixtures with no intention to duplicate the function of the REFPROP computer program (Gallagher *et al.*, 1993) (which is designed to estimate thermophysical properties of pure and mixed refrigerants). An attempt to use REFPROP for the pure agents proved to be unfruitful because the final conditions attained in the experiments exceeded the calculational range of REFPROP. To use PROFISSY for the estimation, 0.00001 g of nitrogen was

artificially input into the program to simulate the calculations for a pure agent. This procedure was found to be justifiable because the calculated initial charge pressures at the initial fill temperatures were very close to the measured agent vapor pressures at the same temperatures. In most cases except CF_3I , PROFISSY tended to underestimate the final pressures. Overall, the predictions were found to be within 7.6 % or less of the measured values.

Since very few previous studies related to CF_3I vapor pressure measurements have been reported in the literature (Smith and Srivastava, 1986), temperature-pressure relationships for $CF_{2}I$ were also measured over a temperature range from -20 °C to 150 °C. From these measurements, the vapor pressure as a function of temperature was determined. Figure 5 shows three sets of PT measurements using initial liquid agent volumes corresponding to one-third, half, and two-thirds of the vessel volume at room temperature, respectively. The trend in Figure 5 can be explained based on a comparison between the fill density, defined as the ratio of the total agent mass to vessel volume, and the critical density of $CF_{2}I$ (Grosshandler *et al.*, 1994). Irrespective of the fill conditions at room temperature, if the vessel is chilled to a lower temperature, the measured pressure will correspond to the vapor pressure at that temperature. On the contrary, if the vessel is heated to a higher temperature, the pressure that one measures depends on whether the liquid and vapor phases still coexist at that temperature. To determine whether the two phases are still present at the temperature of interest, one has to find out whether the fill density lies between the saturated vapor and liquid densities at that temperature (Smith and Van Ness, 1975). Using the same reasoning, vapor pressures that were extracted from Figure 5 in the temperature range between – 20 °C and 70 °C are shown in Figure 6. The data were fitted in the form of log $P_{sat} = A - (B/T)$ where A and B have values of 3.186 and 1037 respectively, P_{sat} is in MPa and T is in K. Note that the above equation should only be used within the stated temperature range.

8.3.4.2 Solubility of Nitrogen in Agents. Table 10 summarizes the experimental results obtained under an initial total pressure (P_i) of approximately 2.8 MPa or 4.1 MPa at room temperature $(T_i = 23 \text{ °C} \pm 1 \text{ °C})$ and with an initial amount of liquid agent corresponding to one-half of the vessel volume. For all the agents evaluated, the amounts of nitrogen $(m_{g,i})$ required to pressurize the vessel to ~ 2.8 MPa and ~ 4.1 MPa were found experimentally to be less than 4 % and 6 % of the total mass of the agent $(m_{a,i})$, respectively. The predicted nitrogen mass in the fifth column and the predicted solubilities of nitrogen $(x_{g,i})$ in the agent under T_i and P_i in the last column of Table 9 were obtained using PROFISSY. The calculations were performed with $m_{a,i}$, P_i , V_t (= 52.2 cm³), and T_i as input and the two binary interaction parameters $(k_{ij}$ and $l_{ij})$ set equal to zero. The solubility is defined as the mole fraction of dissolved nitrogen in the liquid agent/nitrogen mixture at pressure P_i . The predicted nitrogen mass found to be the highest compared to the other four agents, whereas nitrogen exhibited the lowest solubility in CF₃Br when P_i was ~ 2.8 MPa and in CF₃I when P_i was ~ 4.1 MPa.

Table 11 tabulates the experimental results as well as the predictions obtained under similar experimental conditions as in Table 10, except that the initial amount of liquid agent corresponded to two-thirds of the vessel volume. The predicted nitrogen mass also compares favorably with the measured value. For all the agents evaluated, the amount of nitrogen required to pressurize the vessel was found experimentally to be less than 3 % to 4 % of the total mass of the agent. Note that the calculated solubilities in Table 11 are very close to those tabulated in Table 10. This is not surprising because the solubility depends only on P_i and T_i and does not depend on the amount of liquid agent. Again, FC-218 was found to have the highest nitrogen solubility compared to the other four agents, irrespective of P_i . At low P_i (~ 2.8 MPa), the solubility of nitrogen in CF₃Br was calculated to be the



Figure 5. Pressure-temperature measurements of pure CF_3I .



Figure 6. Vapor pressure of CF_3I as a function of temperature.

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Agent	Agent, $m_{a,t}$ (g)	P _i (MPa)	Nitrogen, m _{g,t} (g)	Predicted $m_{g,t}$ (g)	Solubility, $x_{g,l}$ (predicted)
HFC-227ea	36.5	2.90	1.2	1.24	0.083
	36.6	2.93	1.2	1.26	0.084
	36.5	4.29	1.9	1.95	0.129
	36.6	4.28	1.9	1.94	0.128
CF ₃ I	54.9	2.87	1.0	1.15	0.056
	54.8	2.87	1.1	1.15	0.056
	54.9	4.21	1.6	1.79	0.087
	54.8	4.21	1.7	1.78	0.087
FC-218	33.2	2.89	1.0	1.07	0.090
	33.1	2.89	1.1	1.07	0.090
	33.2	4.27	. 1.8	1.77	0.152
-	33.1	4.31	1.8	1.80	0.154
HFC-125	31.3	2.78	0.9	0.81	0.053
	31.1	2.79	0.8	0.81	0.053
	31.3	4.24	1.6	1.62	0.106
	31.1	4.23	1.6	1.61	0.106
CF ₃ Br	40.9	2.89	0.7	0.75	0.046
	40.7	2.87	0.6	0.75	0.046
	40.9	4.29	1.4	1.53	0.095
	40.7	4.29	1.4	1.53	0.095

Table 10. Amount of nitrogen $(m_{g,l})$ required to pressurize mixture to P_i at $T_i = 23$ °C ± 1 °C with an initial amount of liquid agent corresponding to half of the vessel volume

lowest, whereas the solubility of nitrogen in CF_3I was found to be the lowest at high P_i (~ 4.1 MPa). This is quite explainable by the great difference in the vapor pressures of these two agents, as noted in Table 7.

8.3.4.3 Agent/Nitrogen Mixtures. The experimental results for agent/nitrogen mixtures at -60 °C with an initial volume of liquid agent corresponding to half of the vessel volume are shown in Table 12. In all cases, the combined standard uncertainty of the measurement was found to be less than 0.1 MPa. The sensitivity coefficients in the uncertainty estimation were calculated based on PROFISSY. Comparing to the vapor pressure of a pure agent at -60 °C, the measurements showed that there were still significant residual pressures in the vessel. These residual pressures are deemed necessary to expel the cold liquid agents if the vessels are discharged under such extremely low temperature. The fifth column in the table tabulates the predicted final pressures using PROFISSY.

Agent	Agent, $m_{a,t}$ (g)	P _i (MPa)	Nitrogen, m _{g,t} (g)	Predicted m _{g,t} (g)	Solubility, $x_{g,l}$ (predicted)
HFC-227ea	48.7	2.87	1.1	1.16	0.082
	48.7	2.98	1.2	1.21	0.086
	48.7	4.25	1.8	1.81	0.127
	48.7	4.25	1.8	1.82	0.127
CF ₃ I	72.8	2.77	0.9	1.02	0.054
	72.7	2.79	0.9	1.02	0.054
	72.8	4.14	1.4	1.62	0.086
	72.7	4.16	1.4	1.63	0.086
FC-218	44.1	2.94	1.0	1.07	0.096
	44.2	2.93	1.0	1.06	0.096
	44.1	4.36	1.6	1.81	0.161
	44.2	4.29	1.6	1.75	0.157
HFC-125	41.9	2.70	0.8	0.79	0.051
	41.8	2.85	0.8	0.83	0.055
	41.9	4.18	1.5	1.56	0.104
	41.8	4.18	1.5	1.51	0.102
CF ₃ Br	54.9	2.92	0.7	0.76	0.048
	54.8	2.87	0.7	0.74	0.046
	54.9	4.25	1.4	1.48	0.094
	54.8	4.25	1.4	1.49	0.094

Table 11.	Amount of nitrogen $(m_{g,i})$ required to pressurize mixture to P_i at T_i (= 23 °C ± 1 °C) with
	an initial amount of liquid agent corresponding to two-thirds of the vessel volume

The calculations were performed with k_{ij} and l_{ij} set equal to zero, $V_t = 52.2 \text{ cm}^3$, and the initial conditions given in the table. In general, the predicted values were all found to be within 6 % of the measured values.

Table 13 summarizes the results obtained from experiments at -60 °C using an initial liquid agent volume equivalent to two-thirds of the vessel volume. Considerable final pressures in the vessel were also noted at -60 °C. Predictions using PROFISSY with k_{ij} and l_{ij} set equal to zero and $V_i = 52.2$ cm³ were also carried out, and the comparisons between predicted and measured final pressures indicate that overall the errors were within 8 % for all the agents.

Agent	$m_{a,t}$ (g)	P _i (MPa)	P _{f,c} (MPa) (measured)	P _{f,c} (MPa) (predicted)	error ^a (%)
HFC-227ea	36.5	2.90	1.69	1.70	0.6
	36.6	2.93	1.72	1.71	0.6
	36.5	4.29	2.59	2.60	0.4
an a	36.6	4.28	2.58	2.61	1.2
CF ₃ I	54.9	2.87	1.71	1.61	5.9
	54.8	2.87	1.71	1.62	5.3
	54.9	4.21	2.62	2.49	5.0
1999 - L	54.8	4.21	2.61	2.47	5.4
FC-218	33.2	2.89	1.44	1.39	3.5
	33.1	2.89	1.43	1.38	3.5
	33.2	4.27	2.34	2.23	4.7
	33.1	4.31	2.38	2.27	4.6
HFC-125	31.3	2.78	1.11	1.07	3.6
	31.1	2.79	1.11	1.08	2.7
	31.3	4.24	2.10	2.04	2.9
	31.1	4.23	2.09	2.03	2.9
CF ₃ Br	40.9	2.89	1.04	1.02	1.9
	40.7	2.87	1.00	1.02	2.0
	40.9	4.29	1.95	1.94	0.5
e Antonio de la composición de la composi Antonio de la composición	40.7	4.29	1.99	1.97	1.0

Table 12. Initial conditions at $T_i = 23 \text{ °C} \pm 1 \text{ °C}$ and final pressures $(P_{f,c})$ at $-60 \text{ °C} \pm 1 \text{ °C}$ for agent/nitrogen mixtures (half filled volume)

^aerror (%) = $|P_{f,c}(\text{measured}) - P_{f,c}(\text{predicted}) | x 100 / P_{f,c}(\text{measured})$

The results for high temperature (150 °C) measurements are shown in Tables 14 and 15 for initial conditions corresponding to half and two-thirds filled volumes, respectively. The combined standard uncertainty, in all cases, was estimated to be less than 0.2 MPa, and the sensitivity coefficients were calculated based on an ideal gas mixture model. The predictions were obtained by using PROFISSY. In the calculations, k_{ij} and l_{ij} were again assumed to be zero and $V_t = 42.6$ cm³ was used. For these two experimental conditions, the errors between the predicted and measured final pressures were found to be within 10 %.

Agent	$m_{a,t}$ (g)	P _i (MPa)	P _{f,c} (MPa) (measured)	P _{f,c} (MPa) (predicted)	error ^a (%)
HFC-227ea	48.7	2.87	1.66	1.67	0.6
	48.7	2.98	1.70	1.73	1.8
	48.7	4.25	2.55	2.57	0.8
	48.7	4.25	2.51	2.59	3.2
CF ₃ I	72.8	2.77	1.60 .	1.52	5.0
	72.7	2.79	1.64	1.53	6.7
	72.8	4.14	2.49	2.43	2.4
	72.7	4.16	2.62	2.41	8.0
FC-218	44.1	2.94	1.41	1.41	0.0
	44.2	2.93	1.39	1.38	0.7
	44.1	4.36	2.32	2.32	0.0
	44.2	4.29	2.30	2.27	1.3
HFC-125	41.9	2.70	1.08	1.06	1.9
	41.8	2.85	1.10	1.11	0.9
	41.9	4.18	2.06	2.05	0.5
	41.8	4.18	2.02	1.99	1.5
CF ₃ Br	54.9	2.92	1.05	1.04	1.0
·	54.8	2.87	1.02	1.02	0.0
	54.9	4.25	1.94	1.94	0.0
	54.8	4.25	1.90	1.96	3.2

Table 13.	Initial conditions at $T_i = 23 \text{ °C} \pm 1 \text{ °C}$ and final pressures $(P_{f,c})$ at $-60 \text{ °C} \pm 1 \text{ °C}$	C for
	agent/nitrogen mixtures (two-thirds filled volume)	

^aerror (%) = $| P_{f,c}(\text{measured}) - P_{f,c}(\text{predicted}) | x 100 / P_{f,c}(\text{measured})$

Agent	$m_{a,t}$ (g)	P _i (MPa)	P _{f,h} (MPa) (measured)	P _{f,h} (MPa) (predicted)	error ^a (%)
HFC-227ea	28.3	2.76	10.70	10.39	2.9
	30.3	2.81	11.25	10.95	2.7
	28.3	4.29	13.58	13.25	2.4
	30.3	4.19	14.43	14.00	3.0
CF ₃ I	43.2	2.75	10.38	10.55	1.6
	43.2	2.74	10.22	10.35	1.3
·	43.2	4.13	12.71	13.03	2.5
	43.2	4.16	12.92	13.05	1.0
FC-218	28.9	2.85	12.13	11.90	1.9
	29.1	2.76	11.84	11.76	0.7
	28.9	4.16	14.64	14.44	1.4
· · ·	29.1	4.25	14.73	14.59	1.0
HFC-125	26.4	2.79	14.45	14.43	0.1
· ·	26.1	2.78	14.35	14.41	0.4
4	26.4	4.18	17.32	17.56	1.4
	26.1	4.21	17.20	17.36	0.9
CF ₃ Br	34.9	2.80	14.32	14.04	2.0
	34.9	4.22	17.11	16.90	1.2

Table 14. Initial conditions at $T_i = 23 \text{ °C} \pm 1 \text{ °C}$ and final pressures $(P_{f,h})$ at 150 °C $\pm 2 \text{ °C}$ for agent/nitrogen mixtures (half filled volume)

^aerror (%) = $|P_{f,h}(\text{measured}) - P_{f,h}(\text{predicted}) | \times 100 / P_{f,h}(\text{measured})$

Agent	m _{a,t} (g)	P _i (MPa)	P _{f,h} (MPa) (measured)	P _{f,h} (MPa) (predicted)	error ^a (%)
HFC-227ea	38.4	2.80	15.56	14.57	6.4
	39.7	2.78	16.77	15.34	8.5
	38.4	4.14	19.19	17.71	7.7
· · ·	39.7	4.29	20.54	18.87	8.1
CF ₃ I	57.3	2.76	13.79	14.34	4.0
	58.6	2.79	14.42	15.08	4.6
	57.3	4.23	17.62	18.38	4.3
	58.6	4.10	18.25	18.41	0.9
FC-218	37.4	2.88	16.39	16.24	0.9
	37.3	2,78	16.51	16.68	1.0
	37.4	4.16	20.03	20.01	0.1
	37.3	4.23	19.99	19.94	0.3
HFC-125	34.6	2.76	20.13	20.37	1.2
	32.3	2.86	18.87	18.94	0.4
	34.6	4.16	24.25	24.71	1.9
	32.3	4.18	22.39	22.87	2.1
CF ₃ Br	44.4	2.76	18.92	18.68	1.3
	45.6	2.76	20.19	19.69	2.5
	44.4	4.18	22.91	22.82	0.4
	45.6	4.21	23.22	23.31	0.4

Table 15.	Initial conditions at $T_i = 23 \text{ °C} \pm 1 \text{ °C}$ and final pressures $(P_{f,h})$ at 150 °C $\pm 1 \text{ °C}$ for
	agent/nitrogen mixtures (two-thirds filled volume)

^aerror (%) = $| P_{f,h}(\text{measured}) - P_{f,h}(\text{predicted}) | \times 100 / P_{f,h}(\text{measured})$

8.3.5 Summary and Conclusions. For all the pure agents studied under two fill conditions (1/2 and 2/3 filled), the final pressure that the vessel experienced when it was heated to 150 °C was found to be less than 17.5 MPa. Irrespective of the initial fill conditions, the final pressures were all below 0.1 MPa when the vessel was chilled to -60 °C and pure agents were used.

For an initial fill condition corresponding to approximately half of the vessel volume, the amount of nitrogen required to pressurize the vessel up to 4.4 MPa was measured to be less than 6 % of the total agent mass in the vessel in all cases. Under an initial condition of two-thirds liquid fill, the required nitrogen mass was found to be less than 4 % of the total agent mass. Independent of the initial amount of liquid agent in the vessel, the mole fractions of nitrogen in the liquid agents (solubilities) at room temperature were calculated to be less than 0.10 and 0.17, respectively, at 2.8 MPa and 4.1 MPa.

For agent/nitrogen mixtures with an initial liquid agent fill condition of one half or two-thirds and initial total pressures of approximately 2.8 MPa and 4.1 MPa at room temperature, the final pressures measured at -60 °C were found to be higher than 0.9 MPa and 1.8 MPa respectively in all cases. Compared to the vapor pressures of the pure agents, significant residual pressures were still present in the vessel at -60 °C. These residual pressures, as the result of the initial nitrogen pressurization, are vital to agent discharges at cold ambience.

For agent/mixtures with an initially half full condition and initial charge pressures of approximately 2.8 MPa and 4.1 MPa, the resulting pressures at 150 °C were measured to be less than 15 MPa and 18 MPa respectively. For the two-thirds condition under similar initial charge pressures, the final pressures were found to be less than 21 MPa and 25 MPa respectively. Comparisons of the measurements among the four selected agents showed that HFC-125 exhibited the highest final pressures at 150 °C under all the experimental conditions used in this study.

The thermodynamic properties of agent/nitrogen mixtures were calculated using the computer code PROFISSY which is based on an extended corresponding states model. The code was developed to help fire suppression bottle designers to facilitate the calculations of bottle temperature-pressure relationships. Comparing to the experimental measurements, the predictions from PROFISSY were generally found to be within 10 % or less of the measured values.

Based on the experimental results from this study, the following conclusions are made:

- 1. Under the current pressure vessel design criteria in military specification MIL-C-22284A (proof pressure of 9.62 MPa and minimum burst pressure of 12.37 MPa), the structural integrity of the vessel will be severely compromised by the resulting high pressure if the vessel, when used with the four selected agents, is exposed to ambient temperatures up to 150 °C.
- 2. When using the computer code PROFISSY, extreme care should be taken because PROFISSY tends to *underestimate* the final pressures at elevated temperatures by about 10 % in some cases.

8.4 Discharge of Agent/Nitrogen Mixtures in a Simulated Dry Bay

8.4.1 Introduction. Currently, CF_3Br (halon 1301) is used for in-flight dry bay fire protection applications. It is normally stored under pressure above its vapor pressure at room temperature by using nitrogen. Due to nitrogen pressurization, nitrogen is dissolved in the liquid agent, and the presence of dissolved nitrogen complicates the discharge process in that the fluid leaving the vessel is

not pure halon but a mixture of halon and nitrogen and that degassing of the dissolved nitrogen (*i.e.*, the dissolved nitrogen coming out of the liquid in the form of bubbles) occurs inside the vessel during discharge (depressurization).

The search for halon alternatives has resulted in three potential agents, HFC-125, FC-218, and $CF_{3}I$, for dry bay fire protection applications from a list of more than ten candidates (Grosshandler *et al.*, 1994). This task is an extension of the study of agent discharge fluid dynamics previously reported as a part of the agent screening work in Grosshandler *et al.* (1994). In the previous study, the agent was only pressurized with nitrogen a few minutes before the initiation of a test. Given such a short contact time of nitrogen with agent, it was not possible to have an appreciable amount of nitrogen dissolved in the agent. The experimental protocol was dictated by an *idealized* discharge mechanism (a rupture disc). Rupture discs, which offered simple flow geometry and quick action and also facilitated the agent screening process, were chosen as the result of a compromise between a simulated and a real discharge event. Furthermore, it was believed that the effect of dissolved nitrogen on the discharge process was similar among all the agents evaluated because the nitrogen solubilities in all the agents were measured to be very similar (Grosshandler *et al.*, 1994) and that any observable differences in the discharge dynamics among the agents studied were attributable to the agents themselves rather than the dissolved nitrogen.

In this study, the role of dissolved nitrogen in the agent discharge dynamics was examined in a systematic way. Actual discharge mechanisms (solenoid valves and squibs) were used to simulate discharge processes. The main objective of this task was to study the discharge characteristics of the three selected agents (CF_3Br was also included as a reference) in an unconfined space to simulate a dry bay (without airflow and obstacles) under various experimental conditions in greater detail than the previous screening project. In particular, the parameters of interest include size of valve opening, fill density, nitrogen charge pressure, initial agent temperature, discharge orientation, and the degree of nitrogen saturation in liquid agent.

8.4.2 Experimental Methods. This section describes the experimental hardware and procedures used to performed the dry bay discharge tests. The simulated dry bay was essentially a room with dimensions of 3 m (width) x 3 m (length) x 3.4 m (height). No clutter or obstacles, except a pressure transducer located about 1.4 m downstream from the vessel outlet, were placed in the flow path of the flashing spray. Depending on the release mechanism used and discharge orientation, the vessel was centrally located either on one side of the room, near the ceiling, or near the floor. There were two major components in the experimental set-up: (1) the pressure vessel and (2) the release mechanism.

Three pressure vessels, a cylindrical, a spherical, and a cylindrical with sight gages, were used in this study. The choice of the vessel was dictated by the release mechanism. The cylindrical vessel, which was also used in the previous study (Grosshandler *et al.*, 1994), was made from a stainless steel SS 304 tube with an internal diameter of 50.8 mm and a wall thickness of 6.4 mm. One end of the tube was welded with an end cap in which four holes were drilled and tapped to provide access for mounting a piezoelectric dynamic pressure transducer (Kistler Model 603B1), a pressure gage, two K-type thermocouples (Omega CASS-18E-12 and KMQSS-125U-6), and a needle valve (Whitey SS-1RS4) for agent filling and subsequent nitrogen pressurization. Occasionally, a fast response static pressure transducer (Druck Model PDCR 330) was also mounted to the vessel in order to validate the readings from the dynamic pressure transducer. A flange for mounting the release mechanism was welded to the other end of the tube. The internal volume of the vessel without the release mechanism in place was found to be $4.97 \times 10^{-4} \text{ m}^3$, determined by the volume of water required to fill the vessel. Although the vessel was hydrostatically tested to 17.2 MPa, it was never operated above 11 MPa for safety reasons. A schematic of the vessel is shown in Figure 7.





Figure 7. Schematic of the cylindrical vessel.

The spherical vessel was custom-made for the National Institute of Standards and Technology by Pacific Scientific HTL Kin-Tech Division. The vessel was made of metal alloy with an internal volume of 5.38×10^{-4} m³. There was a three-outlet manifold welded to the top of the vessel. The three access holes were for mounting a K-type thermocouple to monitor the liquid temperature, a dynamic or static pressure transducer, and a needle valve. The vessel was also equipped with a one-way fill valve (not used in this study). At 21 °C, the vessel had a proof pressure of 12.4 MPa and a burst pressure of 18.6 MPa. Figure 8 shows the schematic of the spherical vessel.

The cylindrical vessel with sight gages was an industrial liquid level gage (Jerguson 14TL10). It was made of stainless steel SS 316 and had two opposing slots on the side wall of the vessel. The slots were machined to provide plane seats for the gaskets and the glass assemblies. The gage glasses were specially processed, heat treated, and tempered. The glass assemblies were secured in place with steel covers and specially designed bolts. There were a 3/4 NPT top end connection for mounting instrumentation and a 600# RF slip-on flanged bottom connection for mounting the release mechanism. A cross was attached to the top end connection to provide three access holes for a dynamic pressure transducer, a pressure gage, and a needle valve. The vessel was hydrostatically tested by the manufacturer to 10.2 MPa at 38 °C. The internal volume of the vessel without the release mechanism was estimated to be $6.10 \times 10^{-4} \text{ m}^3$. The schematic of the vessel is shown in Figure 9.

Four types of release mechanisms were used in this study: two solenoid valves (from Marotta Scientific Model MV121KJ-2 and HTL Pacific Scientific PN 36400036), a pyrotechnic piston-actuated valve (Kidde-Graviner, Jordan Crew Bay System), and a squib (HTL Pacific Scientific). There are some slight differences in the operating procedures for the two solenoid valves. Upon activation, the Marotta valve with an outlet of 4.45 cm remains open and requires re-arming if it is to be used again, whereas the HTL valve, which has an outlet size similar to the Marotta valve, will close automatically and no re-arming of the valve is needed. Note that both valves have a smaller passage upstream of the outlet. The activation of the Kidde-Graviner valve with a straight-through opening of 4 cm is achieved by using a fast acting beam assembly held initially in position by a spindle. A pyrotechnic piston actuator, when it is energized, causes the spindle to rotate and free the beam assembly. Rearming of the valve is also required. The squib assembly consists of an explosive cartridge (HTL PN 30903827), a rupture disc, and a discharge outlet port (Military Standard MS33649-08) with a diameter of 2 cm. Opening of the rupture disc is initiated by the explosive cartridge. The squib was used solely with the spherical vessel. A DC power source was used to activate the release mechanism. Approximately 25 V and 10 A were required for the Marotta and HTL solenoid valves, approximately 10 V and 1 A for the piston-actuated Kidde-Graviner valve, and approximately 20 V and 3.5 A for the HTL squib. The response times of all the release mechanisms were on the order of 10 ms.

The experimental preparation involved the following steps. The vessel was filled approximately two-thirds full (by volume); in some experiments a half-filled condition was used. The total mass of the agent dispensed to the vessel was obtained by weighing on an electronic scale with an uncertainty of 1 g. The vessel was then pressurized with nitrogen to a specified *equilibrium* pressure at room temperature. The attainment of the final equilibrium pressure could be facilitated by shaking the vessel intermittently and vigorously and by repetitive nitrogen pressurization. The vessel was then mounted and was ready to be discharged at room temperature.

Figure 10 shows the experimental set-up for the vessel with a Marotta or HTL solenoid valve. Because of the configuration of the valve outlet, the normal discharge orientation is horizontal in this case. For the Kidde-Graviner valve and the HTL squib, the normal discharge direction is vertically downward. The five lasers and photodiode detectors shown in Figure 10 were used to measure the average dispersion velocities of the leading edge of the evaporating spray propagating downstream from the vessel exit. The measurement technique was based on the principle of laser attenuation, details of which can be found in the previous report by Grosshandler *et al.* (1994). A piezoelectric



Figure 8. Schematic of the spherical vessel.


Figure 9. Schematic of the cylindrical vessel with sight gages.





dynamic pressure transducer (Kistler Model 603B1) oriented normal to the flow direction was placed along the centerline of the valve outlet at a downstream position of approximately 1.4 m and was used to measure the impact pressure and to obtain some qualitative two-phase behavior of the flashing spray. The mounting of the pressure transducer was also discussed in detail in Grosshandler *et al.* (1994).

All the piezoelectric dynamic pressure transducers were regulated by dual mode charge amplifiers (Kistler Model 5004) with 180 kHz frequency filters. The amplifiers were operated in the "long" time constant mode in order to track the temporal variation of pressure over the relatively long discharge time interval (on the order of 100 ms).

The outputs from the amplifiers, the static pressure transducer (when used), and the photodiode detectors were recorded using a 16 channel, 12 bit, high-speed data acquisition board (Strawberry Tree FLASH-12TM Model 1) at a rate of 25 kHz per channel for 0.6 s. The total number of samples collected per channel was 15000. Because of the speed and memory size, a 1 Mbyte daughter board was required for direct memory access. The data acquisition system was controlled by menu-driven software (WorkBench 4.0), and the data obtained were stored in a personal computer (PC-486) for subsequent data analysis.

A high-speed movie camera (Photec IV) operating at 2000 frames per second was used to document the flashing behavior of agent/nitrogen mixtures at the exit of the vessel and the events occurring inside the vessel during discharge when the cylindrical vessel with sight gages was used. Kodak Ektachrome high-speed daylight films (ASA 400) together with two flood lights for front lighting and an intense flood light for backlighting were used.

The experimental sequence was controlled by a timing circuit. At the initiation of the timing sequence, the high-speed camera was first triggered to ensure that the full framing rate had been attained before photographing the discharge process, and the camera was on for 2 s. At 0.8 s, the data acquisition was initiated for a duration of 0.6 s. At 1 s, the timing circuit sent a control signal to turn on the DC power supply for 0.2 s to activate the release mechanism.

In this section, unless otherwise stated, all the discharge experiments were conducted at room temperature with a nitrogen charge pressure of ~ 4.12 MPa. The vessel was initially filled with an amount of pure liquid agent equivalent to approximately two-thirds of the vessel volume. The discharge orientation was the normal configuration of the release mechanism, as discussed above. This condition is termed *standard discharge* in the discussion.

The effects of several experimental parameters on the discharge characteristics were also examined. These parameters included: (1) discharge orifice size, (2) initial amount of agent, (3) nitrogen charge pressure, (4) initial temperature of agent, (5) discharge orientation, and (6) degree of nitrogen saturation in the liquid agent. Table 16 shows the experimental matrix.

The effect of the orifice size on the standard discharge process was evaluated by mounting a stainless steel orifice plate at the Marotta or HTL solenoid valve outlet. The orifice diameter (2.38 cm) used in this work was about half of the valve outlet diameter.

The initial amount of agent in the vessel was also varied from two-thirds fill (used in all the standard discharge tests) to half fill. Only two agents (CF_3I and HFC-125) were used in this part of the parametric studies.

The effect of initial nitrogen charge pressure on the discharge was examined by performing a series of experiments at room temperature with ~ 2.75 MPa, 4.12 MPa, and 5.50 MPa nitrogen pressures. Because the operating pressures of the Marotta and HTL valves were rated to be less than 5.50 MPa, only two charge pressures (2.75 MPa and 4.12 MPa) were used, whereas all three pressures were evaluated using the spherical vessel equipped with a squib.

Some experiments were performed to determine the effect of initial agent temperature on the discharge characteristics. In the low temperature experiments, the vessel was initially filled with agent,

	CF ₃ I	FC-218	HFC-125	CF ₃ Br			
HTL valve ^a	✓ ^b	1	✓ ‡°	1			
HTL valve ^d	*	1	1				
HTL valve ^{a,e}	·	1	1	1			
Marotta valve ^a	✓ ‡	J	✓ ‡	j.			
Marotta valve ^d		J		1			
Marotta valve ^{a,e}	e 🖌 a 🗤	1		1			
Marotta valve ^{a,f}		1					
Marotta valve ^{d,f}		5					
KG valve ^a		1		J A			
KG valve ^g	1		1	J			
Squib ^a		J		J			
Squib ^d	1	J		J			
Squib ^h			1	. · · · · · · · · · · · · · · · · · · ·			
Squib ⁱ		s		5			
Squib ^a			+b.j				

Table 16. Test matrix for discharge experiments

^ainitial nitrogen charge pressure ~ 4.12 MPa

^btwo-third fill condition

^chalf fill condition

^dinitial nitrogen charge pressure ~ 2.75 MPa

^ewith an orifice plate at the valve outlet

^fcylindrical vessel with sight gages

^ghigh temperature experiments

^hinitial nitrogen charge pressure ~ 5.50 MPa

cold temperature experiments

^jeffect of discharge orientation

pressurized with nitrogen to ~ 4.12 MPa at room temperature, and then cooled with dry ice to below - 50 °C before a discharge was initiated. Due to the large thermal inertia of the cylindrical vessel and the limitations for the operating temperatures of the Marotta and HTL solenoid valves, only the spherical vessel with a squib was used in this set of experiments. In the high temperature experiments, the vessel was heated with a heating tape wrapped around it after it was filled with agent and pressurized with nitrogen to ~ 4.12 MPa at room temperature. Depending on which agent was used in the test, the vessel was heated to a temperature (90 °C for CF₃I) at which the corresponding pressure inside the vessel did not go beyond 9.62 MPa; this was considered to be the maximum operating pressure of the closed cylindrical vessel. Because of the limitation of the operating pressure, the cylindrical vessel with sight gages and the spherical vessel were not used in this study. In addition, only the Kidde-Graviner valve was used due to the fact that the final attainable pressure at the elevated temperature was above the operating pressure of the Marotta and HTL solenoid valves. Note that, since no shaking or agitating of the vessel was performed during the cooling or heating process, the

liquid and gas phases might not be in equilibrium before discharge. This condition is similar to that of an actual fire extinguishing vessel in which no mixing mechanism to maintain equilibrium between the liquid and vapor phases exists at cold or hot ambience.

The study of the effect of orientation on the discharge dynamics was conducted at room temperature with a nitrogen charge pressure of ~ 4.12 MPa by using the spherical vessel and varying the discharge direction from vertically downward (standard discharge) to vertically upward, horizontal, and downward with an angle of 30° or 45° with the vertical. The spherical vessel was chosen because it was lightweight and easy to mount. Only HFC-125 was used in this parametric study.

In order to shed some light on the effect of the degree of nitrogen saturation on the discharge process, a limited set of experiments was performed by partially saturated the liquid agent with nitrogen. Partial saturation was achieved by simply bubbling nitrogen through the liquid agent to a specified pressure without attaining the final equilibrium pressure and without agitating the vessel. In other words, if one were to agitate the vessel, an equilibrium pressure lower than the specified pressure would be obtained because more nitrogen would be dissolved in the liquid agent.

8.4.3 Results and Discussion

8.4.3.1 General Visual Observations. The events occurring *inside* the vessel during a discharge of CF₃Br were conceptually described by Elliot et al. (1984) without any confirmation from visual observations. Based on the experimental pressure decay curves during discharge and a simple theoretical flow model, he suggested that there were several sequential events occurring inside the vessel. When the release mechanism was activated, there was a short duration during which the pressure continuously decreased, and the liquid agent/nitrogen mixture was expelled as a superheated liquid through the valve exit. Nitrogen remained dissolved in the liquid agent as a non-equilibrium supersaturated solution, and the liquid/nitrogen mixture remained a clear solution. As the pressure inside the vessel dropped further, degassing of nitrogen in the solution occurred, and nitrogen bubbles started to form. Once the bubbles nucleated, the dissolved nitrogen quickly came out of the solution into the bubbles causing the bubbles to grow. Due to rapid bubble expansion, the liquid level swelled, and the ullage was compressed to a higher pressure which was noted as a short transient pressure recovery in the pressure decay history. The pressure then began to decrease again until an inflection point in the pressure decay curve was reached. A low quality (defined as the fraction of the mass flow rate which is gas) bubbly two-phase CF₃Br/nitrogen mixture was assumed to be discharged from the vessel during this time interval. The inflection point signified the depletion of the low quality twophase mixture and the initiation of the discharge of the ullage gas mixture (CF₃Br vapor and nitrogen) from the vessel. The rate of pressure decay increased because the gas mixture was less dense than the low quality bubbly two-phase mixture and experienced a higher volumetric discharge rate.

The visual observations made in this study by using the cylindrical vessel with sight gages and high-speed photography revealed the following phenomena which were slightly different from the description given by Elliot *et al.* (1984). For CF_3Br , HFC-125, and FC-218, the liquid agent/nitrogen mixture remained clear for a short duration (~ 5 ms) after the initiation of the discharge. This duration was found to be agent dependent, with CF_3Br the shortest and FC-218 the longest. During this period, the receding liquid/vapor interface was clearly visible. The interior of the vessel as seen through the sight gages then became completely foggy, and the liquid/vapor interface was no longer visible. It was not possible to determine from the movies whether the fogginess prevented the observation of the liquid mixture throughout the vessel, thus causing the liquid/vapor interface to disappear. The fogginess was sustained for a period of time during which the liquid mixture was assumed to be completely

discharged. The sight gages on the vessel then became clear once again. This fogginess is *conjectured* to be due both to vapor condensation as a result of cooling by the adiabatic expansion of the ullage vapor, and the evolution of the dissolved nitrogen as bubbles in the liquid mixture. The existence of the nitrogen bubbles could not be visually confirmed from the high speed movies due to the limited spatial resolution of the pictures. For CF_3I , the receding liquid/vapor interface and the liquid mixture remained clear for a relatively long time (at least half of the liquid had been depleted) after the initiation of the discharge. The fogginess then appeared throughout the vessel, and the liquid/vapor interface vanished. The interior of the vessel became clear again after a period of fogginess. It was postulated that nitrogen degassing did not occur in the CF_3I /nitrogen mixture (see discussion in Section 8.4.3.2). Selected photographic sequences of events for CF_3Br , HFC-125, FC-218, and CF_3I are given in Figures 11, 12, 13, and 14 respectively.

The flashing behavior of CF_3Br external to the vessel was also noted by Elliot *et al.* (1984). They reported that to the eye the discharge was a puff of white vapor. Their high speed movies showed an opaque white plume that steadily diminished in size but did not change in density. The appearance of the plume did not seem to change, with either front lighting or back lighting, even when the vapor mixture discharge began because a white cloud continued to form due to vapor condensation. The observations obtained in the current study were very similar to those made by Elliot et al. (1984) although the visualization of the process using the naked eye was almost impossible because the discharge time was much shorter in this work. Due to the difficulty in discerning the transition from the liquid discharge to the remaining vapor discharge from the appearance of the spray at the valve exit, the time for the depletion of the liquid from the vessel could not be determined with great certainty from the high-speed movies. In addition, even when the vessel with sight gages was used, the fact that the liquid/vapor interface inside the vessel was not visible during the course of the discharge also hindered the determination of the liquid depletion time. Figures 15, 16, 17, and 18, respectively, show individual photographic shots taken at the same instant (5 ms after the first appearance of the spray at the valve exit) during discharges of CF₃Br, HFC-125, FC-218, and CF₃I from the cylindrical vessel equipped with a Marotta solenoid valve. When other release mechanisms were used, the general appearances of the flashing sprays (other than the spray angles) were also observed to be very similar to Figures 15 to 18.

8.4.3.2 Marotta Solenoid Valve. This subsection discusses the experimental results obtained using a Marotta solenoid valve. Figure 19 through Figure 22 show respectively the pressure decay curves obtained inside the vessel during *standard* discharges of CF₃Br, HFC-125, FC-218, and CF₃I. Pressures reported in the figures are gauge pressures. The pressure is nondimensionalized by the initial pressure (P_i) in the vessel before the initiation of a discharge. The time t = 0 s is defined as the time when the solenoid valve is first opened partially. This time, with an estimated uncertainty of 0.08 ms, is inferred from the instant when the initial vessel pressure starts to decrease by 34 kPa and continues to decrease. Note that the baseline fluctuation of the initial vessel pressure is less than 34 kPa peak-to-peak.

For the CF₃Br (Figure 19) standard discharges, the pressure decay curves show four regions as suggested by Elliot *et al.* (1984). The first corresponds to the discharge of metastable liquid agent/dissolved nitrogen mixture, followed by a sudden pressure recovery which presumably implies the degassing of nitrogen from the liquid mixture. The third region corresponds to the discharge of a low quality two-phase bubbly mixture, and the last region signifies the discharge of the remaining vapor mixture from the vessel. In Figure 19, the transition from the third to the fourth region in one case ($P_i = 4.05$ MPa) is not very distinct.

For the HFC-125 and FC-218 tests, the aforementioned four regions are clearly shown in Figures 20 and 21 respectively. However, it is unclear, at present, why the rate of pressure decay in





t = 2.5 ms



t = 1.5 ms

























t = 3.5 ms



t = 5.0 ms



Figure 12. Photographic sequence of events as seen through the sight gages of the cylindrical vessel during a discharge of HFC-125.





t = 0 ms





t = 2.5 ms















 $t = 0 \,\mathrm{ms}$



t = 1.5 ms



t = 2.5 ms



t = 3.5 ms



t = 5.0 ms







Figure 15. A snapshot of CF_3Br discharge (taken 5 ms after the first appearance of the spray at the value exit).



Figure 16. A snapshot of HFC-125 discharge (taken 5 ms after the first appearance of the spray at the valve exit).



Figure 17. A snapshot of FC-218 discharge (taken 5 ms after the first appearance of the spray at the valve exit).



Figure 18. A snapshot of CF_3I discharge (taken 5 ms after the first appearance of the spray at the valve exit).







Figure 20. Pressure decay curves during standard discharges of HFC-125 using a Marotta valve.









the fourth region is slower than that in the third region, which is opposite to the trend observed in the CF_3Br tests.

For the CF_3I experiments, the pressure recovery before the third region is not noted in Figure 22. In addition, there is a pressure recovery in the fourth region. This pressure recovery was subsequently found to be caused by some artifact from the dynamic pressure transducer. This is substantiated by Figure 23 which compares the two pressure decay curves obtained using two different pressure transducers, one static and one dynamic, in the same test. Note that this pressure recovery artifact was only observed in the CF_3I tests.

If the pressure recovery is assumed to be related to the degassing of the dissolved nitrogen in the liquid agent and if the degassing process can presumably be explained in terms of homogeneous nucleation theory and can be modeled by an isothermal decompression process, then the homogeneous nucleation pressure, according to the work by Forest and Ward (1977), should depend on the initial pressure, the surface tension of the liquid mixture, and the vapor pressure of the agent and the solubility of nitrogen in the liquid agent at the temperature of interest. The homogeneous nucleation pressure, which is the pressure at which homogeneous nucleation occurs, can be estimated by the following set of equations whose derivations were given in detail by Forest and Ward (1977). To simplify the analysis, it is further assumed that the effect of pressure and concentration of the dissolved nitrogen on the vapor pressure of the liquid is negligible and that the gas mixture inside the bubble is ideal. It is expected that under the experimental conditions used in this study, a very good estimate of the homogeneous nucleation pressure can be obtained by invoking the above assumptions and without resorting to the detailed calculations. The homogeneous nucleation pressure (P_h) is given by

$$P_{h} = P_{sat} + \frac{CP_{h}}{C_{s}} - \left[\frac{16\pi\sigma^{3}}{3kT\ln(Z/J)}\right]^{1/2}$$
(19)

with

$$\ln\left[\frac{f_n^o(1+C_s)}{C_s}\right] = \ln H + \frac{\nu_n P_h}{RT}$$
(20)

and

$$f_n^o = P_h \exp\left(\frac{P_h B_{nn}}{RT}\right)$$
(21)



Figure 23. Comparison of the pressure decay curve obtained by a dynamic pressure transducer with that obtained by a static pressure transducer.

where P_{sat} is the vapor pressure of the agent at temperature *T*, *C* is the ratio of moles of nitrogen dissolved to moles of liquid agent, C_s is the concentration of nitrogen which would be present in the liquid agent if the pressure of the gas above the liquid were at the total pressure P_h , σ is the surface tension, $Z (\text{cm}^{-3} \text{ s}^{-1})$ is the rate constant, $J (\text{nuclei s}^{-1} \text{ cm}^{-3})$ is the nucleation rate per volume, k is the Boltzmann constant (= 1.38048 x 10⁻¹⁶ dyne cm K⁻¹), f_n^o is the fugacity of the pure gas, H is the Henry's constant, v_n is the partial molar volume of the dissolved nitrogen, B_{nn} is the second virial coefficient, and R is the universal gas constant.

The mole ratio C can be determined from the solubility (mole fraction) of nitrogen in the liquid agent at T and P_i by the following equation:

$$C = \frac{x_n}{1 - x_n} \tag{22}$$

where x_n is the solubility of nitrogen and can be calculated using the vapor-liquid equilibrium computer code (PROFISSY).

Following the procedure outlined in Forest and Ward (1977), Henry's constant H can be estimated using Equation (19) without the last term and Equations (20) and (21), with P_h replaced by P_i in all three equations.

The surface tension for pure agent was estimated by the method of Brock and Bird (Reid *et al.*, 1987); the effect of the dissolved nitrogen on the surface tension was assumed to be negligible. The vapor pressure was estimated by the method proposed by Gomez-Nieto and Thodos (Reid *et al.*, 1987). The partial molar volume of the dissolved nitrogen was calculated using the method in Chueh and Prausnitz (1967). The second virial coefficient B_{nn} was obtained from the tabulation in Dymond and Smith (1969). For all the agents used in this study, the rate constant Z was estimated by the method described in Volmer (1945) and found to be on the order of 10^{32} cm⁻³ s⁻¹. For the nucleation rate J, a value of 10^4 nuclei s⁻¹ cm⁻³ was used. The value of the pressure determined was found to be very insensitive to the value of J chosen because J is in the logarithmic term in Equation (19). The Newton-Raphson numerical scheme (Carnahan *et al.*, 1969) was used to obtain P_h . The initial pressure P_i was used as an initial guess for P_h in the iteration. Table 17 shows the calculated results based on the experimental conditions used to obtain Figures 19 to 22. The calculated homogeneous nucleation pressures are possible according to Reid (1976) although at present no experimental data on CF₃I/nitrogen mixtures are available to substantiate the predictions.

The pressure ratio (P/P_i) at which the degassing of nitrogen initiates or not can now be qualitatively explained in terms of the aforementioned, simplified homogeneous nucleation theory. Based on the calculated homogeneous nucleation pressures in Table 17, the degassing of nitrogen from CF_3I /nitrogen liquid mixtures, which was not observed experimentally as shown in Figure 22, was predicted not to occur under the same experimental conditions. No attempt was made to compare quantitatively the calculated homogeneous nucleation pressures to the experimental degassing pressures because of the uncertainties associated with the thermophysical property predictions used in the calculations. It should be noted that although homogeneous nucleation theory appears to provide some qualitative explanations with regard to the degassing process, heterogeneous bubble nucleation (a more complicated process) from the nucleation sites on the vessel wall could also play a role. However, it can be argued that the ease with which the agents wet the inner surface of the container due to their small contact angles (low surface tensions) during vessel filling favors homogeneous nucleation

Agent	Mass (g)	Т (К)	P _i (MPa)	P _h (MPa)	P_h/P_i
CF ₃ Br	591	295	4.10	1.90	0.46
CF ₃ Br	564	295	4.05	1.85	0.46
HFC-125	438	294	4.03	1.63	0.41
HFC-125	428	294	4.16	1.76	0.42
FC-218	450	295	4.04	2.12	0.53
FC-218	435	294	4.10	2.11	0.52
CF ₃ I	723	294	4.12		
CF ₃ I	719	294	4.21		
CF ₃ I	755	294	4.17		

Table 17. Calculated homogeneous nucleation pressure (P_h) at T

because less effective vapor trapping in the irregular cavities on the imperfect container wall surface will be expected (Carey, 1992).

A comparison of the pressure decay curves for all the agents is made in Figure 24. It is clear that different agents exhibit different discharge rate and degassing characteristics. Based on the pressure decay curves, CF_3I discharges the fastest and CF_3Br the slowest under similar initial conditions. These observations were also confirmed qualitatively by the high-speed movies.

The average dispersion velocities of the spray obtained by the laser attenuation technique are summarized in Figure 25. The dispersion of CF_3I has the slowest average downstream velocities. All the other agents have similar but higher spray dispersion velocities than CF_3I .

The measurements from the downstream dynamic pressure transducer can be used to provide a qualitative description of the two-phase behavior of the spray downstream. According to Whalley (1990), if the two-phase flow is assumed to be homogeneous, then the momentum flux M, which in this case is the measured dynamic pressure, is given by

$$M = \frac{G^2}{\rho_h} = \rho_h u^2$$

(23)

with

$$\rho_h = \alpha \rho_{\sigma} + (1 - \alpha) \rho_l \tag{24}$$

where G is the mass flux, u is the average velocity, α is the void fraction, ρ_h is the homogeneous mixture density, and ρ_g and ρ_l are the saturated vapor and liquid densities. The void faction is defined



Figure 24. Comparison of pressure decay curves during standard discharges using a Marotta valve.



Figure 25. Average downstream velocities during standard discharges using a Marotta valve.

as the average fraction of the volume which is occupied by the gas phase. Using Equations (23) and (24), one can solve for α ,

$$\alpha = \frac{\left(\rho_l - \frac{M}{u^2}\right)}{\rho_l - \rho_g}$$
(25)

Since *M* and *u* were measured in the experiments, an estimate of α could be made. Table 18 summarizes the calculated results. The saturated vapor and liquid densities were assumed to be those of the pure agents at their normal boiling points. For CF₃Br, HFC-125, and FC-218, the saturated densities were obtained from the REFPROP database (Gallagher *et al.*, 1993). For CF₃I, the saturated liquid density was estimated using the modified Rackett method described in Reid *et al.* (1987), and the vapor density was calculated by assuming the vapor was ideal. The average velocities *u* were obtained by taking the ratio of the distance between the transducer and the valve exit to the time when the spray hit the transducer. The dynamic pressures (*M*) used in the calculations corresponded to the maximum pressures recorded in the experiments. Among all the agents, CF₃I has the lowest estimated void fraction, which implies that a fair amount of CF₃I has not been evaporated at this downstream location. The low Jakob number (see Grosshandler *et al.*, 1994) of CF₃I and its high normal boiling point also provide some qualitative support of the calculated low void fraction. Since the other three agents have higher Jakob numbers and lower boiling points than CF₃I, it is not surprising to find higher calculated void fractions for all three agents.

The maximum angles sustained by the sprays at the valve exit during discharges were also measured and are tabulated in Table 18. Note that the spray angle is a function of time during discharge. Since it was very difficult to define clearly the boundaries of the sprays because of their flashing characteristics, an uncertainty of 4° was typical. Compared to the other agents, the maximum spray angles in the selected CF_3I tests were found to be the largest.

8.4.3.3 HTL Solenoid Valve. For comparison purposes, selected pressure decay curves obtained during standard discharges of CF₃Br, HFC-125, FC-218, and CF₃I using the HTL solenoid valve are shown in Figure 26. The initial, relatively flat plateaus in the figure suggest that the opening times of the solenoid valve are slow and differ, depending on which agent is used. However, no general trends were observed when all of the experimental data were examined although Figure 26 shows the opening time of the valve for CF₃I is the slowest. The initial, slow, and *continuous* decrease in pressure could not be attributed to the drift of the baseline initial pressure P_i before the opening of the valve because the baseline value for P_i remained relatively constant except for some small fluctuations. Note that such initial slow pressure decays were not observed in all the tests using a Marotta valve.

Similar to the CF₃I discharges using a Marotta valve, the pressure recovery artifact in the CF₃I pressure trace caused by the dynamic transducer is also noted in Figure 26, and no degassing of CF₃I is suggested. If the opening time of the valve were to be ignored, CF₃I would discharge the fastest, and CF₃Br the slowest under similar initial conditions.

Figure 27 summarizes the average downstream dispersion velocities of the sprays for all the agents. Similar to the observations made using a Marotta valve, the dispersion of CF_3I exhibits the slowest downstream velocities. All the other agents have similar but higher spray dispersion velocities than CF_3I .

Agent	Mass (g)	u (m/s)	M _{max} (MPa)	α	Maximum spray angle
CF ₃ Br	591	103	0.30	0.99	100°
CF ₃ Br	564	*			
HFC-125	438	114	0.84	0.96	110°
HFC-125	428	104	0.41	0.98	
FC-218	450	102	0.43	0.98	108°
FC-218	435	94	0.25	0.99	
CF ₃ I	723	64	1.58	0.84	142°
CF ₃ I	719	64	1.65	0.83	141°
CF ₃ I	755	61	1.68	0.81	

Table 18. Calculated void fractions and measured maximum spray angles (standard discharges using a Marotta valve)

*not recorded

Table 19 tabulates the calculated void fractions at the location of the downstream dynamic pressure transducer. Similar to the results for the Marotta valve, the void fraction for CF_3I is estimated to be the smallest, whereas all the other agents show comparable and higher void fraction values. The measured maximum spray angles are also summarized in Table 19. The CF_3I spray angles still appear to be the largest.

8.4.3.4 Kidde-Graviner Valve The results of the standard discharges of all the agents for this valve are shown in Figure 28. The pressure recovery artifact occurring close to the end of the discharge in the CF_3I measurements is again noted in the figure. According to the pressure decay curves, the discharge rate of CF_3I is the fastest and CF_3Br the slowest. The pressure trace also suggests that the degassing of dissolved nitrogen does not occur in the CF_3I test, consistent with the prediction using the homogeneous nucleation theory (see Table 17). For the other three agents, the pressure recoveries due to the degassing of dissolved nitrogen are not very pronounced, that is the pressures do not rebound significantly. Since this valve has a bigger opening, one possible explanation for these flat pressure plateaus is that the rate of the ullage compression due to the degassing is offset by the rate of ullage expansion resulting from the faster liquid discharge.

The average downstream dispersion velocities are summarized in Figure 29. Once again, the dispersion velocities of CF_3I are the slowest among the other three agents. The high temperature results in the figure will be discussed later in Section 8.4.3.9.

The calculated void fractions at the location of the downstream pressure transducer, together with the measured maximum spray angles, are tabulated in Table 20. Care should be exercised in interpreting the calculated void fraction for HFC-125 because the unusual high M, which results in a low void fraction, could have been caused by the accidental impact of the flying debris (from the actuation of the pyrotechnic piston) on the pressure transducer; the debris was frequently observed coming from the valve exit in the high-speed movies. Excluding the result of HFC-125, CF₃I again







Figure 27. Average downstream velocities during standard discharges using a HTL valve.

Agent	Mass (g)	u (m/s)	M _{max} (MPa)	α	Maximum spray angle
CF ₃ Br	586	91	0.83	0.95	103°
HFC-125	431	98	0.58	0.97	105°
HFC-125	426	98	0.65	0.96	105°
FC-218	444	83	0.55	0.95	108°
CF ₃ I	707	46	1.65	0.67	115°
CF ₃ I	713	51	1.67	0.73	120°

Table 19.	Calculated void	fractions	and	measured	maximum	spray	angles	(standard	discharges	using
	a HTL valve)									

shows the lowest calculated void fraction at a fixed downstream location. In addition, CF_3I spray has the largest maximum spray angle at the valve exit.

8.4.3.5 HTL Squibs. Figure 30 summarizes the results of the standard discharges using HTL squibs. Three salient points are noted in the figure. First, no degassing of the dissolved nitrogen in CF₃I is suggested. Second, the inflection points in the pressure decay curves of CF₃Br, HFC-125, and FC-218 are barely distinguishable. Third, the initial CF₃I discharge rate is the fastest, and CF₃Br has the slowest discharge rate. The dynamic pressure transducer still caused a pressure recovery artifact close to the end of the discharge in some of the CF₃I tests; however, the pressure rebound was not as pronounced as in the tests using Marotta, HTL, and Kidde-Graviner valves. All of the pressure curves have much more fluctuation during the initial discharge period; this could be the result of the violent vibration caused by the activation of the explosive cartridge.

The measurements of average spray dispersion velocities could not be carried out because the brightness of the flash caused by the explosion of the cartridge completely saturated the laser detectors. However, estimates of the average velocities could still be made using the time at which the downstream pressure transducer detected the arrival of the spray. Table 21 summarizes the average velocities, the calculated void fractions, and the measured maximum spray angles. The average dispersion velocity u of the CF₃I spray is still the slowest. In Table 21, the unusual high values of M could again be due to the flying debris (resulting from the action of the explosive cartridge) hitting the pressure transducer. The measured maximum spray angles of all the agents are very similar within the measurement uncertainty. Based on the discussion so far, the spray angle appears to be dependent not only on the agent considered but also on the release mechanism used.

8.4.3.6 Effect of an Orifice Plate. The pressure decay curves obtained using an orifice plate and a Marotta valve during standard discharges are respectively shown in Figures 31, 32, 33, and 34 for CF₃Br, HFC-125, FC-218, and CF₃I. The low P_i and half-fill measurements in the figures will be discussed in later sections. There are three important features in these figures.



Figure 28. Comparison of pressure decay curves during standard discharges using a Kidde-Graviner (KG) valve.





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Agent	Agent Mass (g)		ent Mass (g) u (m/s)		M _{max} (MPa)	α	Maximum spray angle	
CF ₃ Br	625	. 89	1.28	0.92	132°			
HFC-125	497	92	2.20	0.83	145°			
FC-218	492	75	1.18	0.87	134°			
CF ₃ I	800	54	1.21	0.83	152°			

 Table 20.
 Calculated void fractions and measured maximum spray angles (standard discharges using a Kidde-Graviner valve)

1. The pressures at which the degassing occurs (see Figures 31, 32, and 33) do not appear to depend on the presence or absence of the orifice plate under similar initial conditions of the vessel. This is consistent with the calculations of the homogeneous nucleation pressures which depend only on the initial conditions of the vessel (see Section 8.4.3.2).

- 2. No degassing is noticeable in the CF_3I pressure decay curves. In Figure 34, the pressure rebound at the end of the pressure decay curves is due to the artifact of the pressure transducer.
- 3. Compared to the standard discharges, the discharge rates are much slower when a restricted orifice plate is present.

Figure 35 compares the results of all the agents under similar initial conditions. The tendency to degas also conforms qualitatively with the predictions using homogeneous nucleation theory. The initial discharge rate of CF_3I is the fastest, whereas CF_3Br again shows the slowest discharge rate. Figure 36 summarizes the results obtained from the HTL solenoid valve with the same orifice plate. Note that CF_3I was not used in this parametric study. The results are very similar to those obtained using the Marotta valve; this is not surprising because both valves are geometrically similar. In Figure 36, it is unclear why the pressure traces of all the CF_3Br runs show jagged appearances when an orifice plate is used.

Figures 37, 38, 39, and 40 show the average spray dispersion velocities of individual agents using the Marotta valve under various experimental conditions. Only the standard discharges with and without an orifice plate will be discussed here. The other parametric effects will be considered in subsequent sections. In the presence of the orifice plate, the measured average velocities closest to the exit (*i.e.*, at 20 cm downstream) are comparable to or lower than those observed in the standard discharges. Similar trends are noted in Figures 41, 42, and 43 when using the HTL solenoid valve. Figure 44 compares the average velocities obtained in the standard Marotta discharges of all the agents with the orifice plate. The average dispersion velocities of CF_3I again are the slowest among all the other agents.

Table 22 tabulates the calculated void fractions and the measured maximum spray angles. There are no significant differences in the spray angles for the different cases. However, CF_3I still has the lowest calculated void fraction at the specified downstream location.





Agent	Mass (g)	u (m/s)	M _{max} (MPa)	α	Maximum spray angle
CF ₃ Br	581	93	2.03	0.89	140°
HFC-125	454	89	2.37	0.81	131°
HFC-125	453	. 95	2.61	0.81	146°
FC-218	476	83	1.72	0.84	144 ^o
CF ₃ I	773	62	1.52	0.84	136°

Table 21.	Calculated void fraction	s and measured	maximum spray	angles (standard	discharges using
	HTL squibs)				

8.4.3.7 Effect of Fill Density. Figures 32 and 34 also compare respectively the discharge characteristics of HFC-125 and CF₃I under two fill conditions, the standard (2/3) versus one-half, using the Marotta valve. Notice the pressure rebound, an artifact, at the end of the CF₃I discharge, and the pressure decay curves of CF₃I do not suggest any degassing of the dissolved nitrogen, an observation consistent with homogeneous nucleation theory. Figure 45 shows the results for HFC-125 from the HTL valve tests. The pressures at which the degassing occurs under two different fill conditions with similar initial nitrogen charge pressures at the same temperature appear to be not significantly different from each other although the results in Figure 45 show some scatter. This observation can also be explained qualitatively by homogeneous nucleation theory. According to the theory, the bubble nucleation pressure only depends on the initial total pressure and temperature. Since the amount of agent required is less under one-half fill condition, the overall discharge rates, based on the inflection points in the curves, are faster than the discharges under two-thirds fill condition.

The calculated void fractions and the measured spray angles are summarized in Table 23. The calculated void fractions of CF_3I are still the lowest irrespective of the fill conditions. The average spray dispersion velocities under one-half fill conditions are also shown in Figures 38, 40, and 42. Comparing Figures 38 and 40 for the one-half fill condition, CF_3I still has lower average velocities than HFC-125. However, no general conclusions can be made when the average velocities under the two fill conditions are compared at various downstream locations because of the scatter in the results.

8.4.3.8 Effect of Initial Nitrogen Charge Pressure. Visual observations of the behavior of the agent/nitrogen mixtures inside a vessel were also made using the cylindrical vessel with sight gages and the low initial charge pressure (~ 2.75 MPa). For the same agent, the high-speed movies revealed similar phenomena as observed in the standard discharges, except that the liquid/vapor interface remained visible and clear for a much longer duration.

Figures 31, 32, 33, and 34 compare the discharge characteristics of individual agents using the Marotta valve under two initial nitrogen charge pressures. At the low initial nitrogen charge pressure, there is no indication of degassing from the pressure decay curves using any of the agents. The homogeneous nucleation theory, discussed above, also predicts that the degassing does not occur. Table 24 shows the calculated homogeneous nucleation pressures of all the agents based on the experimental conditions used in the Marotta valve tests. The nucleation pressure of CF_3I is not shown in the table because it is calculated to be negative (*i.e.*, in tension). It is clear from the pressure decay







Figure 32. Pressure decay curves during discharges of HFC-125 using a Marotta valve under various conditions.


Figure 33. Pressure decay curves during discharges of FC-218 using a Marotta valve under various conditions.



Figure 34. Pressure decay curves during discharges of CF_3I using a Marotta valve under various conditions.







Figure 36. Comparison of pressure decay curves during standard discharges using a HTL valve with an orifice plate.



Figure 37. Average downstream velocities during discharges of CF_3Br using a Marotta valve under various conditions.



Average downstream distance (cm)

Figure 38. Average downstream velocities during discharges of HFC-125 using a Marotta valve under various conditions.



Figure 39. Average downstream velocities during discharges of FC-218 using a Marotta valve under various conditions.



Figure 40. Average downstream velocities during discharges of CF₃I using a Marotta valve under various conditions.



Figure 41. Average downstream velocities during discharges of CF₃Br using a HTL valve under various conditions.



Figure 42. Average downstream velocities during discharges of HFC-125 using a HTL valve under various conditions.



Figure 43. Average downstream velocities during discharges of FC-218 using a HTL valve under various conditions.





Agent	Mass (g)	u (m/s)	M _{max} (MPa)	α	Maximum spray angle
CF ₃ Br*	577	66	0.19	0.98	143°
HFC-125*	452	76	0.41	0.96	142°
FC-218 [*]	442	68	0.21	0.98	145°
CF ₃ I*	695	54	1.32	0.81	147 ⁰
CF ₃ Br**	577	59	0.35	0.95	
HFC-125**	425	61			145°
FC-218**	447	58	0.34	0.94	145°

Table 22.	alculated void fractions and measured maximum spray angles (discharges with orif	ice
	ate)	

Marotta valve

**HTL valve

curves that by the time the P_h/P_i ratio is reached, all the liquid has been discharged from the vessel; therefore, no homogeneous nucleation or degassing is possible.

The discharge rates, based on the inflection points, are slower for a lower initial nitrogen charge pressure. The results obtained by using the HTL valve were very similar. Figures 46 and 47 compare the discharge characteristics among the agents. Note that CF_3I was not used in the HTL valve tests. In Figure 46, the discharge rate of CF_3I is the fastest, and CF_3Br the slowest.

The comparisons of the average dispersion velocities of the leading edges of the sprays between the low initial charge pressure tests and the standard discharges are also shown in Figures 37 to 43 for individual agents. In general, the velocities near the valve exit are higher in the standard discharges than in the low pressure tests. The comparisons of the velocities among the four agents in the low pressure tests are given in Figure 48. Note again that CF_3I has the lowest dispersion velocities.

Table 25 tabulates the calculated void fractions and maximum spray angles from the tests using low initial charge pressure. For CF_3I , the calculated void fraction is the lowest.

A set of experiments using HFC-125 and the Marotta valve was performed in order to examine the effect of a range of initial charge pressures. The results are summarized in Figure 49. In the figure, "pure" refers to no nitrogen pressurization. Curve e in the figure was obtained using the static pressure transducer which, in general, exhibited more noise than the dynamic transducer in the transient pressure measurements. From the figure, degassing appears to occur only for the two highest initial charge pressures. These two runs have the abrupt pressure recovery associated with the degassing of dissolved nitrogen. This observation can also be explained qualitatively in terms of the homogeneous nucleation theory. Table 26 shows the calculated homogeneous nucleation pressures for various initial charge pressures. At pressures below 2.74 MPa, the calculated homogeneous nucleation pressures are negative (*i.e.*, in tension). In other words, homogeneous nucleation is predicted not to occur under the experimental conditions used in this study. The prediction agrees with the observations in Figure 49. Although homogeneous nucleation is predicted for $P_i = 2.74$ MPa, the nucleation pressure is so low by the time this pressure is reached that all the liquid in the vessel has been depleted; this prediction is in agreement with the observation for $P_i = 2.74$ MPa in Figure 49 that degassing is not indicated. The dependence of discharge rate on initial charge pressure can also be



Figure 45. Pressure decay curves during discharges of HFC-125 using a HTL valve under two fill conditions.

Agent	Mass (g)	u (m/s)	M _{max} (MPa)	α	Maximum spray angle
HFC-125*	428	104	0.41	0.98	
HFC-125*	354**	114	0.33	0.99	114°
CF ₃ I*	719	64	1.65	0.83	. 141°
CF ₃ I ^{*.}	547**	57	1.20	0.85	133°
HFC-125 [§]	431	98	0.58	0.97	105°
HFC-125 [§]	342**	94	1.07	0.92	108°

Table 23.	Calculated void fractions and measured	l maximum spray	angles (discharges	with different
	fill densities)			

^{*}Marotta valve

**one-half fill condition

[§]HTL valve

seen in Figure 49. Based on the inflection points of the curves for similar fill densities, higher initial charge pressures result in higher discharge rates.

Figure 50 summarizes the average spray dispersion velocities of HFC-125 for various initial charge pressures. At the location closest to the valve exit (*i.e.*, 20 cm in the figure), it appears that the velocity is slower for lower initial charge pressures. The estimated void fractions and the measured maximum spray angles are given in Table 27. The maximum spray angle initially increases with increasing initial charge pressure, reaches a maximum, and then decreases with increasing initial charge pressures. Despite the scatter in the measured average velocities and momentum fluxes, the calculated void fractions appear to be higher at high initial charge pressures than at low charge pressures. This implies that the high initial charge pressure facilitates the dispersion and evaporation of the agent *downstream*.

The effect of initial charge pressure was also studied using HTL squibs. Figures 51, 52, 53, and 54 show the discharges of CF₃Br, HFC-125, FC-218, and CF₃I respectively. The low temperature condition in the figures will be discussed later. In all the figures, it is clear that there is no indication that degassing occurs at low initial charge pressures. At the highest initial pressure ($P_i \sim 5.48$ MPa), the initial fluctuation in the pressure traces in Figures 52 and 53 due to the initiation of the explosive cartridge could have masked the pressure recovery during degassing; there is no apparent reason why degassing should not occur in HFC-125/nitrogen and FC-218/nitrogen liquid mixtures at higher initial charge pressures. Once again for CF₃I, no degassing is indicated in Figure 54 even at the highest initial charge pressures. The overall discharge rate of CF₃I is the fastest and that of CF₃Br the slowest. Figure 56 compares the results of HFC-125 and CF₃I at high initial charge pressure. Although the two pressure decay curves cross over at $P/P_i < 0.3$, the overall discharge rate of CF₃I is still faster than that of HFC-125 because for $P/P_i < 0.3$, most of the liquid should have been expelled.

Table 28 compares the calculated void fractions and measured maximum spray angles under various initial charge pressures for all the agents. Note that the scatter in the velocities and momentum flux measurements could have been caused by the flying debris from the explosive cartridge. It appears that the initial charge pressure has little or no effect on the maximum spray angle and that high initial charge pressures result in high average velocities and *vice versa*.

Agent	Mass (g)	Т (К)	P _i (MPa)	P _h (MPa)	P _H /P _i
CF ₃ Br	586	294	2.75	0.51	0.19
HFC-125	448	294	2.74	0.39	0.14
FC-218	442	295	2.67	0.69	0.26
CF ₃ I	725	294	2.78		

Table 24. Calculated homogeneous nucleation pressure (P_h) at T

8.4.3.9 Effect of Initial Vessel Temperature. Except the spray angles, visual observations from high-speed movies reveal a similarity in the appearance of the spray between the standard discharges and the discharges where an agent is initially chilled before discharging into the simulated dry bay which is at room temperature. However, after most of the agent/nitrogen mixture had been discharged, large liquid drops were observed dripping out of the vessel outlet.

Figures 51, 52, 53, and 54 also show the pressure decay curves of individual agents obtained at cold temperatures. Figure 57 summarizes the results of all the cold agent discharges. Irrespective of agent, degassing is not noted in the discharges of cold agents. The discharge rates are slower in cold discharges than in standard discharges because the initial pressures are lower.

In Table 29, the calculated void fractions and measured maximum spray angles and average velocities are compared to those obtained from standard discharges. Despite the potentially inaccurate measurements of M_{max} and u caused by the flying debris from the explosive cartridges, the very low calculated void fractions based on the release of a cold agent, when compared to the standard discharges, indicate that the liquid phase still constitutes a significant volume fraction at a location ~ 1.4 m downstream from the vessel exit. This appears to be consistent with the fact, based on droplet evaporation theory (Kanury, 1977), that the liquid agent droplets, formed as a result of flashing, vaporize much slower when the initial droplet temperature is much closer to or lower than its normal boiling point than the droplets whose initial temperature is much higher than its normal boiling point, as in the case of a standard discharges, and there is also a significant, in some cases a factor of two, reduction in the maximum spray angle for cold agent discharges. The observations are very similar to the critical discharges of a pure liquid under different degrees of subcooling (Celata *et al.*, 1982, 1983) in that the spray angle decreases with increasing liquid subcooling.

From the high-speed movies, the spray appearance, other than the spray angle, of a hot-agent discharge is similar to that of a standard discharge spray. Figures 58, 59, 60, and 61 compare the internal pressure traces for standard discharges of CF_3Br , HFC-125, FC-218, and CF_3I to high temperature discharges using a Kidde-Graviner valve. Note that the temperatures in these figures are the measurements from the thermocouple initially located in the liquid phase. In Figures 58 and 59, the high temperature curves suggest the occurrence of degassing, thus implying that the critical points of the agent/nitrogen mixtures may not have been reached; otherwise smooth pressure decay curves of supercritical fluid discharges should have been observed, as in Figure 60 for FC-218. Note that the calculated mixture critical points from the PROFISSY code are very close to (slightly lower than) the two high temperatures used in Figures 58 and 59. However, since there is no mixing to ensure thermal equilibration between the liquid and vapor phases during heating, a temperature gradient in the vessel exists. Therefore, it is not possible to know precisely the temperatures of the mixtures inside



Figure 46. Comparison of the pressure decay curves during discharges using a Marotta valve under low initial charge pressures.



Figure 47. Comparison of pressure decay curves during discharges using a HTL valve under low initial charge pressures.



Figure 48. Average downstream velocities during discharges using a Marotta valve under low initial charge pressures.

Agent	Mass (g)	u (m/s)	M _{max} (MPa)	α	Maximum spray angle
CF ₃ Br [*]	586	75	1.19	0.90	135°
HFC-125*	448	73	1.37	0.83	140°
FC-218 [*]	442	85	0.22	0.99	136°
CF ₃ I [*]	725	47	1.80	0.66	127°
CF ₃ Br**	586	77	1.14	0.91	107°
HFC-125**	433	88	2.41	. 0.80	140°
FC-218**	438	78	0.58	0.94	123°
FC-218 ^{**}	450	77	0.69	0.93	115°

Table 25. Calculated void fractions and measured maximum spray angles (discharges with $P_i \sim 2.75$ MPa)

Marotta valve

**HTL valve

the vessel. On the contrary, in the case of FC-218, the achievable temperature, dictated by the working pressure of the vessel, in the experiment is much higher (at least 20 °C) than the calculated mixture critical point. Despite the existence of a temperature gradient, it is reasonable to expect that the condition inside the vessel in this case may still reach supercritical.

In Figures 58 and 59, the degassing process can be partly explained using homogeneous nucleation theory. The differences in the nucleation pressures (at which the plateaus occur) observed in the standard and high temperature cases can be attributed to differences in the thermodynamic pathways that lead to the metastable states (at which homogeneous nucleation occurs) of the mixtures from their initial states. In the standard discharges, the pathways are likely to be isothermal depressurization, whereas in the high temperature discharges, due to the temperature gradients inside the vessel the pathways are very complicated. For the purpose of illustration, if one assumes the pathways in the high temperature cases to be isentropic (not an unrealistic assumption), then the homogeneous nucleation pressures can be lower (possibly in this case) or higher than those observed during isothermal decompression, depending on the initial entropies of the mixture (Kim-E and Reid, 1983). The occurrence of a plateau instead of an observable pressure recovery in the figures could be due to the resultant effect of the rate of degassing and the rapid discharge rate.

Contrary to the standard discharges, the high temperature result of CF_3I in Figure 61 indicates some degree of degassing, evident from the occurrence of a plateau in the pressure trace. Note that the temperature used in the test is still below the calculated mixture critical point. If one recalls that the homogeneous nucleation pressure under standard discharge conditions was calculated to be negative (*i.e.*, in tension), no degassing was observed in the standard discharges of CF_3I . The occurrence of degassing in the high temperature discharge can be explained based on the different thermodynamic pathways leading to the metastable states as discussed above in the cases of CF_3Br and HFC-125. That is, in this case, the homogeneous nucleation pressure of a high temperature discharge is higher (positive) than that (negative) of a standard discharge. Note also that the *disappearance* of the pressure rebound, an artifact, at the end of the high temperature discharge in Figure 61, implying that temperature has an important role in the performance of the dynamic pressure transducer.





Mass (g)	Т (К)	P _i (MPa)	P_h (MPa)	P _h /P _i
428	294	4.16	1.76	0.42
456	294	3.39	1.02	0.30
448	294	2.74	0.39	0.14
454	294	2.16		
454	294	1.50	<u></u>	
453	294	1.17		·

Table 26.	Calculated homogeneous nucleation pressure (P_h) at T under various initial charge	е
	pressures for HFC-125	

A comparison of the average spray dispersion velocities for all the agents studied in the standard and high temperature discharges is given in Figure 29, and CF_3I again exhibits the slowest velocities. For the same agent, the measured dispersion velocities *nearest* to the valve exit are higher due to higher initial vessel pressure as a result of higher temperature than those measured in the standard discharges.

The measured maximum spray angles of standard and high temperature discharges using a Kidde-Graviner valve are compared in Table 30. The void fractions are not calculated and are not listed in Table 30 because an estimate of a representative downstream temperature cannot be made in the high temperature discharges. For the same agent, the maximum spray angle is smaller in the standard discharges than in the high temperature discharges. The very low values of the maximum downstream dynamic pressures in the case of high temperature discharges of FC-218 also provide some evidence that FC-218 was discharged as a supercritical fluid (gas) at the temperature of interest.

8.4.3.10 Effect of Vessel Orientation. Based on visual observations from the high-speed movies, the direction of vessel discharge does not generally seem to affect the spray appearance at the squib exit although the sprays resulted from vertically upward discharges are narrower downstream. Figures 62, 63, and 64 show snapshots taken at the same instant (5 ms after the activation of the squib) of vertically downward and upward, and horizontal discharges of HFC-125 using HTL squibs. Figure 65 compares the pressure decay curves during discharges of HFC-125 using HTL squibs under various discharge orientations. Two important features are noted in the figure. First, irrespective of the discharge orientation and as long as the initial conditions (*i.e.*, total pressure, initial temperature, amount of agent) remain the same in all cases, the degassing occurs approximately at the same P/P_i although the time to reach P/P_i differs. This is not surprising because bubble nucleation should be independent of any direction relative to the gravitation vector. Second, the discharge rates do not seem to depend strongly on the orientation of the vessel, a contrast to the observations made in Grosshandler et al. (1994). In that study, nitrogen was in contact with the agent for a very short time before the agent was discharged; however, in this work nitrogen was fully saturated with the agent before the experiments began. The dissolved nitrogen clearly plays an important role in the situation where the discharge orientation is a limiting factor in the application. However, extreme care should be exercised when one tries to extrapolate this observation to other fill conditions. For example, in the situation where the initial liquid level is very low (a case which may not be encountered in practical applications) and the discharge orientation is upwards, the maximum swelling of the liquid level may



Figure 50. Average downstream velocities during discharges of HFC-125 using a Marotta valve under various initial charge pressures.

Mass (g)	P _i (MPa)	u (m/s)	M _{max} (MPa)	α	Maximum spray angle
438	4.03	114	0.84	0.96	110°
428	4.16	104	0.41	0.98	
456	3.39	121	2.41	0.90	1 24°
448	2.74	73	1.50	0.83	140 ^o
454	2.16	60	2.34	0.57	133°
454	1.50	67	2.30	0.67	115°
453	1.17	65	2.30	0.64	102°

Table 27.	Calculated void fractions and measured maximum spray angles of HFC-12	5 discharges
	under various initial charge pressures (using a Marotta valve)	

not reach the top of the vessel during depressurization. In this case, a very *high* quality two-phase mixture is expected to be discharged from the vessel. Therefore, it is recommended that irrespective of the fill conditions, the discharge orientation should be downwards so that a very low quality two-phase mixture can be obtained.

Table 31 summarizes the measured maximum spray angles and the calculated void fractions. Because of the flying debris resulted from the cartridge explosion, it is very difficult to make any reasonably meaningful conclusions in the calculated void fractions. Within the uncertainty, the measured maximum spray angles appear to be similar in all cases; however, the overall appearance of the vertically upward discharge spray is different (less dispersed) from that of the standard discharge spray (*cf.* Figures 62 and 63).

8.4.3.11 Effect of Partial Nitrogen Saturation. Figures 66 and 67 compare respectively the discharge characteristics of complete nitrogen saturation (standard discharge) to partial nitrogen saturation for HFC-125 and FC-218 at $P_i \sim 4.12$ MPa and room temperature using a Marotta valve. In the two figures, the degassing of nitrogen is not noted when it is only partially saturated with the agent. Another interesting observation is that the discharge rate is *slower* when the agent is completely saturated with nitrogen.

Figures 68, 69, 70, and 71 compare the two cases under a lower initial total pressure ($P_i \sim 2.75$ MPa) for CF₃Br, HFC-125, FC-218, and CF₃I respectively. No degassing is noted in either case, and the discharge rate is again slower in the case of complete nitrogen saturation. Note that in Figure 71, the pressure rebound at the end of the discharge is an artifact from the pressure transducer, as discussed above. The reason for not recording such pressure rebound in one of the runs is not known. When a HTL valve was used for similar tests (complete *versus* partial), similar trends were observed with regard to the degassing and the discharge rates.

Figures 37, 38, 39, 40, 41, 42, and 43 also display the average downstream spray velocities for the various agents studied when nitrogen is partially saturated with the agent. Compared to the case of complete nitrogen saturation, the average spray velocities *nearest* to the valve exit are consistently slower.







Figure 52. Comparison of pressure decay curves during discharges of HFC-125 using HTL squibs under various initial conditions.



Figure 53. Comparison of pressure decay curves during discharges of FC-218 using HTL squibs under various initial conditions.







Figure 55. Comparison of pressure decay curves during discharges using HTL squibs at low initial charge pressure.



Figure 56. Comparison of pressure decay curves during discharges using HTL squibs at high initial charge pressure.

Agent	Mass (g)	P _i (MPa)	u (m/s)	M _{max} (MPa)	α	Maximum spray angle
CF ₃ Br	581	4.23	93 .	2.03	0.89	140°
CF ₃ Br	586	2.95	71	2.63	0.74	136° .
HFC-125	458	5.49	93	2.22	0.83	145°
HFC-125	454	4.12	89	2.37	0.81	131°
HFC-125	453	4.14	95	2.61	0.81	146°
HFC-125	454	2.63	69	2.24	0.69	140°
FC-218	476	4.21	83	1.72	0.84	144 ^o
FC-218	458	2.79	69	1.68	0.77	147 ⁰
CF ₃ I	773	5.48	79	1.49	0.90	
CF ₃ I	773	4.17	62	1.52	0.84	1 36°
CF ₃ I	753	2.81	47	1.39	0.74	130°

 Table 28.
 Calculated void fractions and measured maximum spray angles under various initial charge pressures (using HTL squibs)

8.4.3.12 Effect of Vessel Geometry. In all the tests conducted, no qualitative differences in the discharge behaviors were observed when the cylindrical vessel equipped with a Marotta, HTL, or Kidde-Graviner valve or a spherical vessel equipped with a HTL squib was used. Although no experiments have been performed using cylindrical and spherical bottles with *identical* internal volumes due to logistics in the experimental hardware design, it is expected that the discharge behaviors should also be strikingly similar, based on the following arguments. First, nitrogen degassing should be independent of the vessel geometry assuming that the process can be explained according to homogeneous nucleation theories. As discussed above, the homogeneous nucleation theories seem to provide a reasonably good, qualitative prediction of the occurrence of the degassing process. Second, the discharge characteristics of all the agents studied, using a cylindrical or spherical vessel with a slightly different internal volume and different release mechanism, are very similar. Third, the discharge process depends mainly on the ullage pressure, and pressure is an intensive variable and independent of the shape of the vessel; therefore, the effect of vessel geometry will likely have an insignificant effect on the discharge rate. It should be pointed out that the *final* drainage of the remaining liquid through a valve attached to the bottom of one of the two flat heads of a cylindrical vessel and from a spherical vessel will be different. However, in practical vessel design, a cylindrical pressure vessel does not normally have two flat heads, but rather has two hemispherical heads. Consequently, the final drainage problem is expected to be dynamically similar to that of a spherical vessel.

8.4.3.13 Model Predictions. Based on the above discussion and experimental observations, a model that predicts the discharge times of an agent under various experimental conditions is proposed. The intent here is to determine whether the proposed model can provide similar qualitative trends when compared to the experimental observations. The major difference between the proposed model





Agent	Mass (g)	T (°C)	P _i (MPa)	и (m/s)	M _{max} (MPa)	α	Maximum spray angle
CF ₃ Br	581	21	4.23	93	2.03	0.89	140°
CF ₃ Br	584	-52	2.03	43	2.25	0.39	82°
HFC-125	454	21	4.12	89	2.37	0.81	131°
HFC-125	453	22	4.14	95	2.61	0.81	146 ^o
HFC-125	457	-52	2.13	47	2.45	0.27	73°
HFC-125	455	-55	2.05				85°
FC-218	476	22	4.21	83	1.72	0.84	144°
FC-218	475	-51	2.21	47	2.00	0.41	70°
CF ₃ I	773	21	4.17	62	1.52	0.84	136°
CF ₃ I	771	-58	2.54	48	1.74	0.68	51 [°]

 Table 29.
 Calculated void fractions and measured maximum spray angles of standard and cold agent discharges using HTL squibs

and the model developed by Elliot *et al.* (1984) is how the process of degassing of dissolved nitrogen is treated. In Elliot's model (for CF_3Br discharge), there is no provision to determine whether degassing will or will not occur. The degassing of dissolved nitrogen is always assumed to occur, and the critical bubble size is fixed at 15 nm. Degassing may not occur when CF_3I is used or when the initial vessel pressure is below 4.12 MPa, as discussed above. Note that in Elliot's experiments, the total initial vessel pressures all exceeded 4.1 MPa. The proposed model, under certain assumptions, can predict the occurrence or the non-occurrence of degassing. In the following, the assumptions and the equations used in the model are presented.

The discharge configuration of the vessel is assumed to be vertically downward, *i.e.*, the liquid mixture will be discharged first, followed by the ullage vapor mixture. During discharge, no heat and mass transfer between the liquid and the vapor phases is assumed to occur. As discussed in Grosshandler *et al.* (1994), the liquid is assumed to undergo an isothermal depressurization during discharge, and the degassing of dissolved nitrogen is modeled by homogeneous nucleation theory, as described in Forest and Ward (1977). The occurrence of degassing is determined by comparing the ullage pressure to the calculated homogeneous bubble nucleation pressure during the liquid discharge period. If the ullage pressure at some instant in this time interval is equal to or less than the homogeneous nucleation pressure, degassing is assumed to occur. When no degassing is predicted by the homogeneous nucleation pressure, the discharge period the ullage pressure never goes below the homogeneous nucleation pressure, the discharge is modeled as a two-step process, a superheated liquid discharge followed by an emptying of ullage vapor. When degassing occurs, the model is formulated as a four-step process, a superheated liquid discharge followed by swelling of the liquid level due to degassing, a two-phase mixture discharge, and finally a vapor discharge.

The control volume under consideration is the ullage above the liquid. Since no heat and mass transfer is assumed to occur between the liquid and the vapor phases, the control volume is an adiabatic closed system. Furthermore, due to the rapidity of the discharge, the ullage expansion or



Figure 58. Comparison of pressure decay curves during discharges of CF_3Br using a Kidde-Graviner valve under two initial agent temperatures.



Figure 59. Comparison of pressure decay curves during discharges of HFC-125 using a Kidde-Graviner valve under two initial agent temperatures.



Figure 60. Comparison of pressure decay curves during discharges of FC-218 using a Kidde-Graviner valve under two initial agent temperatures.


Figure 61. Comparison of pressure decay curves during discharges of CF₃I using a Kidde-Graviner valve under two initial agent temperatures.

Agent	Mass (g)	Т (°С)	P _i (MPa)	u (m/s)	M _{max} (MPa)	Maximum spray angle
CF ₃ Br	625	21	4.14	89	1.28	132°
CF ₃ Br	620	73	9.24	106	1.84	167°
HFC-125	477	21	4.13	92	2.20	145°
HFC-125	481	70	9.10	114	1.05	180°
HFC-125	484	65	9.49	117	1.66	176°
FC-218	492	21	4.14	75	1.18	134°
FC-218	505	75	8.58	111	0.08	170°
FC-218	508	77	8.81	115	0.12	168°
CF ₃ I	800	21	4.15	54	1.21	152°
CF ₃ I	817	90	8.64	68	1.56	161°

 Table 30.
 Comparison of measured maximum spray angles under standard and high temperature conditions using a Kidde-Graviner valve

momentary compression (caused by degassing) is assumed to be reversible. Therefore, the processes occurring in the ullage can be assumed to be *isentropic*. From the first law of thermodynamics for an isentropic process and assuming an ideal vapor phase,

$$PV^{\gamma} = constant \qquad (26)$$

where γ is the ratio of the constant pressure and volume heat capacities of the vapor mixture, P is the ullage pressure, and V is the ullage volume. Since nitrogen is the dominant species in the vapor phase, the γ of nitrogen with a value of 1.4 is used in the model calculations. If a time derivative of Equation (26) is performed, then

$$\frac{dP}{dt} + \frac{P\gamma}{V} \frac{dV}{dt} = 0.$$
 (27)

Before degassing, the initial saturated liquid agent/dissolved nitrogen mixture discharging through a sharp-edged orifice behaves in a completely metastable manner (Henry, 1979), and the discharge rate can be predicted using incompressible flow formulation. Assuming that the flow through the discharge



Figure 62. A snapshot of vertically downward discharge of HFC-125 (taken 5 ms after the activation of the squib).



Figure 63. A snapshot of vertically upward discharge of HFC-125 (taken 5 ms after the activation of the squib).



Figure 64. A snapshot of horizontal discharge of HFC-125 (taken 5 ms after the activation of the squib).



Figure 65. Comparison of pressure decay curves during discharges of HFC-125 using HTL squibs under various discharge orientations.

Orientation	Mass (g)	и (m/s)	M _{max} (MPa)	α	Maximum spray angle
Vertically downward	454	89	2.37	0.81	131°
n	453	95	2.61	0.81	146°
Vertically upward	453	83	2.49	0.77	137°
Downward 45°	449		,		146°
Downward 60°	456				
Horizontal	450	99	0.70	0.96	1 45°

Table 31. Calculated void fractions and measured maximum spray angles of discharges of HFC-125 using HTL squibs at room temperature, $P_i \sim 4.12$ MPa, and different discharge orientations

valve can be approximated by an orifice flow of a superheated liquid, the rate of change of the ullage volume can be expressed in terms of the volumetric flow rate of the liquid as

$$\frac{dV}{dt} = C_d A \sqrt{\frac{2(P - P_a)}{\rho_{l,m}}}$$
(28)

where C_d is the discharge coefficient for the liquid, with the measured value found to be 0.61 in the literature (Henry, 1979), A is the opening of the orifice, P_a is the ambient pressure, and ρ_{Lm} is the liquid mixture density. In writing Equation (28), quasi-steadiness is implicitly assumed, that is the discharge is treated as a steady-state process at any instant in time. In the calculations, the value of 0.61 was used for C_d , and the liquid mixture density was obtained from the computer program PROFISSY. Equations (27) and (28) were solved numerically using a fourth-order Runge-Kutta method (Carnahan et al., 1969) to obtain the ullage pressure and volume as a function of time. At each time step, the calculated pressure was compared to the calculated homogeneous nucleation pressure in order to determine whether degassing occurred or not. In the case of degassing, the swelling of the liquid level is treated with the following simple model. At the onset of bubble nucleation, all the dissolved nitrogen is assumed to form bubbles with a critical radius, and any subsequent bubble growth and rise to the liquid surface is not considered in the formulation. That is all the bubbles, once formed, stay in the liquid. This approach was also taken by Elliot et al. (1984). According to Forest and Ward (1977) who assumed negligible effects of pressure and concentration of the dissolved nitrogen on the vapor pressure of the liquid and an ideal gas mixture inside the bubble, the critical bubble radius can be calculated as



Figure 66. Comparison of pressure decay curves during discharges of HFC-125 using a Marotta valve at $P_i \sim 4.12$ MPa under complete and partial nitrogen saturation.







Figure 68. Comparison of pressure decay curves during discharges of CF_3Br using a Marotta valve at $P_i \sim 2.75$ MPa under complete and partial nitrogen saturation.







Figure 70. Comparison of pressure decay curves during discharges of FC-218 using a Marotta valve at $P_i \sim 2.75$ MPa under complete and partial nitrogen saturation.

1





$$R_c = 2\sigma \left(P_{sat} + \frac{CP_h}{C_s} - P_h \right)$$
⁽²⁹⁾

where R_c is the critical bubble radius. The volume of each bubble (V_{bubble}) can then be calculated as

$$V_{bubble} = \frac{4}{3} \pi R_c^3 .$$
 (30)

The pressure inside the bubble can be obtained from the Young-Laplace equation (Carey, 1992):

$$P_{bubble} - P_h = \frac{2\sigma}{R_c} . \tag{31}$$

The amount of nitrogen mass in each bubble can be calculated assuming an ideal gas. The temperature used in the calculations is the initial temperature of the liquid because the thermodynamic path is isothermal depressurization. The total number of nitrogen bubbles formed (N_{bubble}) can be obtained from the ratio of the remaining mass of dissolved nitrogen before degassing, which can be calculated from Equations (27) and (28), to the amount of nitrogen vapor in a bubble. The swelling of the liquid level, which is assumed to be primarily due to the volume displacement of the nitrogen bubbles, can now be estimated by

$$V_{l,swell} = N_{bubble} V_{bubble}$$
(32)

where $V_{l,swell}$ is the increase in liquid volume due to swelling. The expansion of the liquid level causes compression of and reduction in the ullage volume. If the process is assumed to be isentropic, the pressure $(P_{recoverv})$ in the compressed ullage volume $(V_{compress})$ can be calculated using

$$\boldsymbol{P}_{recovery} = \boldsymbol{P}_{bd} \left(\frac{\boldsymbol{V}_{bd}}{\boldsymbol{V}_{compress}} \right)^{\boldsymbol{Y}}$$
(33)

where P_{bd} and V_{bd} are the immediate pressure and ullage volume before degassing respectively, which can be obtained in solving Equations (27) and (28). The above approach used to estimate the pressure recovery in the pressure decay curves will result in an abrupt rise in pressure rather than a smooth recovery as noted in the experimental results. The smooth pressure rebound is probably due to the combined effects of bubble inception and growth, bubble coalescence, bubble rise to the liquid surface, disengagement of rising bubbles from the liquid surface, and the discharge of the agent during the degassing period. Because of their complexity, all these effects have not been taken into account in the present model.

To simulate the two-phase (vapor/liquid) discharge after degassing, a homogeneous frozen flow model (Whalley, 1990) is assumed. That is, the two-phase mixture is assumed to be homogeneous, and the quality, defined as the mass fraction of gaseous nitrogen in the mixture, is assumed to be constant during the flow through the orifice. The liquid is assumed not to vaporize. Furthermore, no heat transfer between the gas and the liquid is assumed to occur, and the two-phase mixture is treated as incompressible, a reasonable assumption for a low quality two-phase flow. Under these assumptions, Equation (28) can be used as an approximation to describe the homogeneous two-phase flow with $\rho_{l,m}$ being the homogeneous two-phase mixture density in this case. This density can be estimated using the volume and the mass of the two-phase mixture immediately after degassing. A discharge coefficient with a value of 0.61 was assumed in the calculations. The initial pressure during the two-phase discharge period is $P_{recovery}$.

After complete depletion of liquid (in the case of no degassing) or two-phase mixture (in the case of degassing), the remaining ullage vapor discharge can be modeled using the following equation (Kim-E and Reid 1983):

$$\frac{P}{P_{el}} = \left[1 - \frac{C_a A}{V_t} \left(\frac{R T_{el} \gamma^3 K}{m}\right)^{1/2} \left(\frac{1-\gamma}{2\gamma}\right) t\right]^{\frac{2\gamma}{1-\gamma}}, \qquad (34)$$

with

$$K = \left(\frac{2}{\gamma + 1}\right)^{\left(\frac{\gamma + 1}{\gamma - 1}\right)}$$
(35)

where C_a is the discharge coefficient, *m* is the molecular weight of the vapor (assuming to be nitrogen), P_{el} and T_{el} are the pressure and temperature in the vessel at the instant of complete depletion of liquid or two-phase mixture, respectively, and V_t is the vessel volume. A value of 0.61 was also used for C_a in the calculations. T_{el} can be calculated by

$$T_{el} = T \left(\frac{V_{ullage}}{V_t}\right)^{\gamma - 1}$$
(36)

where V_{ullage} and T are the initial vapor ullage volume and temperature, respectively.

If the initial vessel temperature is above the critical temperature of the agent/nitrogen mixture, the discharge of the supercritical fluid may simply be modeled using Equations (34) and (35) with T_{el} and P_{el} being the initial vessel temperature and pressure.

Table 32 summarizes the predicted and observed complete discharge times for all the agents under the standard discharge conditions. Although the time corresponding to the inflection point in the pressure trace could be used to indicate the end of *liquid* discharge, this approach was not used due to

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Agent	Mass (g)	Observed discharge time (ms) ± 5 ms	Predicted discharge time (ms)
CF ₃ Br [*]	591	89	55
CF ₃ Br [*]	564	۰ <u></u> ۰ ۲	54
CF₃Br ^{\$}	625	72	21
CF₃Br [¶]	581	265	79
HFC-125*	438	77	51
HFC-125*	428	 	50
HFC-125 [§]	477	64	20
HFC-125 [¶]	454	257	75
FC-218 [*]	450	73	50
FC-218 [*]	435		49
FC-218 [§]	492	60	19
FC-218 [¶]	476	250	74
CF_3I^*	723	75	56
CF ₃ I*	719	75	56
CF ₃ I*	755		59
CF ₃ I [§]	800	78	22
CF ₃ I [¶]	773	295	86

Table 32. Comparison between predicted and observed discharge times for standard discharges

*Marotta valve

[§]Kidde-Graviner valve

[¶]HTL squibs

the uncertainties in discerning the inflection points in most of the pressure traces. Instead, the *complete* discharge times were used in the comparison.

In Table 32, the observed *complete* discharge times were measured from the high-speed movies. These times are defined as the interval between the instant when the flashing spray first appears at the vessel exit and the time when the spray can no longer be observed. Using this definition, the discharge times for CF_3I are the longest, which seems to contradict the observations from the measured pressure decay curves. However, the high-speed movies show that the denseness of the flashing CF_3I spray diminishes much sooner than those of the other agents, implying that the bulk of CF_3I discharges faster. The calculated discharge times do not show such a distinct trend among agents. This is probably due to the assumptions used in the model. When *all* the dissolved nitrogen is assumed to degas, it will cause an artificially high liquid swelling level, resulting in an artificially high pressure recovery in the ullage and an overall faster discharge rate. Therefore, the agent/nitrogen mixtures with a tendency to degas will have shorter discharge times. In addition, the calculated homogeneous nucleation pressures in the model agree only qualitatively with the observed degassing pressures. Although one can use the experimentally determined degassing pressures as input

parameters in the model and good predictions can be obtained if the *measured* degassing pressures and the pressure recoveries are used as input in the model, it is better to predict these pressures using homogeneous nucleation theories so that no *a priori* information is required and the versatility of the model can be demonstrated. Despite these simplified descriptions of the liquid swelling process and nitrogen degassing, the discharge times can still be estimated within a factor of four (usually better when a solenoid valve was used) by the model. It is conjectured that the action of a squib or pyrotechnic device could render some of the assumptions used in the model invalid.

8.4.4 Summary and Conclusions. Experiments have been conducted in order to delineate the effects of various parameters on agent discharge into an enclosure. The major experimental observations are summarized in the following.

- 1. Irrespective of the release mechanisms (two solenoid valves, one piston-actuated valve, and a squib), CF_3I exhibits the fastest volumetric discharge rate from the pressure vessel at room temperature; however, the average dispersion velocity of the spray is the slowest once the agent leaves the vessel. All the other agents have comparable volumetric discharge rates and spray dispersion velocities.
- 2. For a given initial vessel pressure, reducing the orifice opening prolongs the discharge time. Based on the experimental results, the discharge time is approximately inversely proportional to the area of the orifice opening. Reducing the liquid fill volume decreases the discharge time because less agent mass is available in the vessel. For a given fill condition, increasing initial nitrogen charge pressure decreases the discharge time and *vice versa*. However, due to the dependence of the degassing process on the initial charge pressure, no simple, empirical correlation between the discharge time and the initial vessel pressure was apparent from the experimental data.
- 3. The average dispersion velocities of cold agent discharges are much slower than those of discharges at room temperature. In addition, the discharge rates of cold agents are slower, and the spray angles are narrower.
- 4. Based on the results of nitrogen saturated HFC-125 discharges (2/3 liquid fill), the vessel orientation does not affect the discharge rate significantly. The dissolved nitrogen in the liquid agent plays an important role.
- 5. When the agent is partially saturated with nitrogen, instead of complete nitrogen saturation, the volumetric discharge rate is much faster.
- 6. A discharge model incorporating homogeneous nucleation theories to account for nitrogen degassing predicts all qualitative aspects of the discharge process and the discharge times within a factor of four.

8.5 Computer Simulation of Agent Dispersion

8.5.1 Introduction. The objective of this study is to explore a computational tool which will provide detailed analysis on the dispersion process of a flashing spray in a simulated dry bay. Since the

discharge process under this scenario is extremely rapid (events typically lasting less than 100 ms), detailed spatial and temporal measurements of temperatures, velocities and concentrations of the flashing spray are very difficult. Therefore, numerical simulation may prove valuable for studying such a highly transient process. This work was conducted in connection with the experimental studies on the agent discharge and dispersion processes described in Section 8.4.

8.5.2 Modeling of Agent Dispersion Process. The computer codes KIVA-II and CONCHAS-SPRAY developed at the Los Alamos National Laboratory (LANL) (Amsden *et al.*, 1989; Cloutman *et al.*, 1982) were used to simulate the agent dispersion of a flashing spray which was obtained from a liquid agent being discharged from a nitrogen-pressurized vessel through a simple, nozzle/orifice-like exit. The system under consideration involves a vessel volume of the order of 10^{-3} m³, with a discharge time of the order of 10^{-2} s. To achieve such a rapid discharge, the prepressurization level is typically on the order of several megapascals.

The dispersion of a flashing spray into a confined space is a complicated process which involves many physical phenomena occurring simultaneously and interacting with one another. As the agent exits from the vessel, thermodynamic and fluid-dynamic instabilities lead to flashing and break-up of the agent into a two-phase gaseous/droplet jet mixture. This occurs in a relatively short transition region which starts at the vessel exit and ends at a section of the two-phase jet where thermodynamic and fluid-dynamic instabilities have ceased.

Downstream of this section, called the "initial section," the flow begins to develop as a mixed, two-phase agent/air jet. Beyond the initial section, thermodynamic equilibrium is maintained and droplet collision and agglomeration do not play an important role in either the ensuing jet dynamics or the dispersion of the agent throughout the protected space. Within the initial section, the spray consists of a mixture of liquid-agent fragments of various shapes and sizes and gaseous agent. The spray begins to entrain initially quiescent air from the surrounding environment as it moves away from the bottle exit. Because the temperature of the discharged liquid agent is far above its normal boiling point, it very quickly evaporates and creates a cloud of agent vapor surrounding the moving spray. The entrainment, mixing and evaporation continue as the spray is dispersed throughout the space. The above processes are included in the present analysis.

The dispersion process is described as a two-phase, three-component, turbulent, compressible, dissipative flow. It is assumed that the gas phase consists of the following three components: agent vapor, nitrogen, and oxygen. The liquid phase only consists of pure liquid agent. The vapor pressure equation is a highly non-linear function of temperature. Heat conduction and convection, diffusive and convective mass transfer, momentum transport, and turbulent flow are taken into account in the gas-phase formulations. In addition, the interphase transport processes, such as momentum exchange in the form of Stokes forces and aerodynamic drag between the droplets and the gas phase, heat exchange in the form of forced convection, and mass exchange in the form of evaporation, are considered. It is assumed that the spray droplets can be represented by a Sauter mean diameter. The droplet evaporation process is assumed to be quasi-steady. The effect of turbulence on droplet motion within the spray is also included, and the turbulence in the gas phase is described by relating the velocity gradients to the turbulent viscosity.

The transport coefficients (heat conductivity, mass diffusivity, and viscosity) and the specific heats are temperature dependent. The effect of the gravitational field is also included. No chemical reactions and their associated thermal effects (*e.g.*, heat release) are assumed in the formulation.

The governing equations are the conservation equations of mass, momentum and energy for the medium under investigation. This set of equations constitutes an initial/boundary value problem within a region bounded by the initial section and the far-field boundaries of the protected space. The set of

time-dependent equations is expressed in a three-dimensional form although the discharges were assumed to be axisymmetric.

The total mass density of the fluid, excluding the mass of the spray droplets, is defined in terms of a partial mass density of the component species. The mass balance equates the change in species concentration with time to the sum of the advection, diffusion and source terms due to evaporation of that particular component. The total fluid density satisfies the total mass balance equation, recognizing that the net diffusion must sum to zero.

The axial and radial momentum balance equations for the fluid mixture equate the time rate of change of momentum to the sum of the advection, pressure, viscous stress, droplet drag, and body force terms. The energy equation for the gas phase accounts for the local change in specific internal energy from the sum of the net advection of sensible heat, heat conduction, enthalpy transport accompanying the diffusion of individual species, flow work, viscous heating, and energy transfer between the two phases. The state relations are assumed to be those of an ideal gas mixture. The transport coefficients include the laminar and turbulent contributions to the momentum, heat, and mass transfer. The turbulent viscosity is broken into a constant uniform background term and a variable turbulent viscosity. The thermal and mass turbulent diffusivity are related to the viscosity through the local Prandtl and Schmidt numbers of the gas mixture.

The motion of the droplets in the spray is governed by ordinary differential equations in Lagrangian form. It is assumed that the spray is sufficiently thin, and, as a consequence, the volume displaced by the droplets may be neglected. Velocity components in the axial, radial and swirl directions are tracked, and the transport between phases is dependent upon the Reynolds number computed from the slip velocity and the film temperature and composition. Internal droplet motion is ignored, leading to uniform droplet temperatures.

The model developed by Dukowicz (1979) was used to calculate droplet mass loss due to evaporation. The particle velocity is computed from the drag associated with the slip velocity. The latent heat cools the droplet during evaporation, and the heat transfer between the drop and the environment is determined using the Ranz-Marshall correlation (1952), modified to account for mass leaving the surface. The fluid-particle interactions are all two-way coupled to ensure overall conservation of mass, momentum and energy.

8.5.3 Solution Method. The gas-phase solution procedure is based on a finite volume method called the ALE (Arbitrary Lagrangian-Eulerian) method. Spatial differences are formed on a finite-difference mesh that subdivides the computational region into a number of small cells that are hexahedrons. The corners of the cells are called vertices, and the positions of the vertices may be arbitrarily specified functions of time, thereby allowing a Lagrangian, Eulerian, or mixed description. The arbitrary mesh can conform to curved boundaries and can move to follow allowable changes in geometry. An advantage of the method is that the mesh need not be orthogonal. The transient solution is marched out in a sequence of finite time increments called cycles or time steps. In each cycle the values of the dependent variables are calculated from those in the previous cycle.

8.5.4 Numerical Results and Discussion. Computations using the two-dimensional axisymmetric approach were performed using the LANL CONCHAS-SPRAY and KIVA-II codes. CONVEX and CRAY supercomputers with the UNIX operating system at NIST were used for the calculations.

The results for a generic agent simulant with thermophysical properties very similar to HCFC-22 are presented graphically as contours of constant velocity, density, pressure, temperature, species concentration, vorticity, kinematic viscosity, mass flux, and liquid phase. The only difference in thermophysical properties between the simulant and HCFC-22 is the vapor pressure curves. As will be discussed below, the actual vapor pressure curve of HCFC-22 could not be used in the calculations

because of convergence problems. The shape of the curve was modified slightly so that calculations could be performed. The rationale for selecting this generic simulant is that its thermophysical properties lie within those of the selected agents.

The performance of the two codes, CONCHAS-SPRAY and KIVA-II, were compared. It was found that in one test case, CONCHAS-SPRAY and KIVA-II gave essentially the same results. Since CONCHAS-SPRAY requires less computing time, the computational results reported herein were obtained using CONCHAS-SPRAY. It should be emphasized that the original codes were designed for the use with the LANL computer facility. In addition, the codes employed graphics libraries installed in that system. To apply the codes using the NIST computing facility, it was necessary to configure them to the NIST operating system. The codes have also been modified to write their output into organized ASCII files. The output files were transferred onto a PC computer where they were further graphically processed with a graphics program written in the C language. The program converted the numerical output into postscript files.

The configuration investigated is the discharge of a droplet spray, with properties similar to one of the halon alternatives, into a totally enclosed space. The transitional region between the nozzle exit and a fully-developed spray was not considered but has been discussed by Cooper (1994). Therefore, it is assumed that the spray has been already established at the discharge orifice exit and is injected into the area at a prescribed rate.

Computational parcels representing the droplets are introduced at the injector with an axial velocity of 61.4 m/s, directed at an angle of 0° relative to the symmetry axis. A zero azimuthal component of velocity is assumed for the droplets at the injector. All the gas and liquid parameters are assumed to be uniform across the orifice exit. All the parameters within the compartment are also assumed to be uniform initially.

The surrounding walls of the compartment are treated as solid boundaries maintained at a constant temperature of 21 °C. Free-slip conditions are assumed on the walls for the velocities. Thus the subsequent penetration and shape of the spray result solely from the interactions of the liquid droplets with the ambient air. The numerical solution of the balance equations associated with the initial and boundary conditions and the material properties provides the time evolution of the spray in the compartment.

The nozzle is located on the centerline of the 0.42 m diameter cylindrically shaped volume. The length of the compartment is either 1.0 m or 2.5 m. Agent inflow boundary conditions, including temperature, density, and velocity are estimated from the results of the discharge vessel experiments and specified in the calculation. The ambient temperature is either 21 °C or -30 °C. The ambient pressure is 1 x 10⁵ kPa. The liquid spray mean diameter is either 200 µm or 600 µm. Table 33 lists the set of the input data used as initial parameters in the computations and the figure numbers corresponding to the figures showing the computational results.

Figures 72 to 76 χ illustrate the computational mesh used in the calculations and the temporal variations of liquid spray outline, agent vapor density, temperature, pressure and gas density contour under the following conditions: initial droplet diameter of 200 μ m, enclosure length of 1 m, and initial ambient temperature equal to the 21 °C.

The effect of droplet diameter is demonstrated in Figures 77 to 80, where the initial droplet size varies from 200 μ m to 600 μ m; all other parameters remained constant.

Figures 81 to 84 illustrate the evolution of the spray with an initial Sauter mean diameter of 200 μ m and an initial ambient temperature of - 30 °C. Figures 85 to 88 show the results for the larger initial mean diameter (600 μ m) at - 30 °C.

Figures 89 to 92 show the calculated results using a Sauter mean diameter of 200 μ m and a different compartment length (2.5 m) at 21 °C. Figures 93 to 96 display the results for a Sauter mean diameter of 600 μ m under the same initial ambient temperature.

Figure number	Spray mean diameter (µm)	Ambient temperature (°C)	Compartment height (m)	
72-76	200	21	1.0	
77-80	600	21	1.0	
81-84	200	-30	1.0	
85-88	600	-30	1.0	
89-92	200	21	2.5	
93-96	` 600	21	2.5	

Table 33. Input data used in the computations

Based on the computational results, the penetration distance of the spray (around 0.9 m after 14 ms, which roughly corresponds to the values measured in the totally open discharge apparatus) is about the same for the 200 μ m and 600 μ m drops; however, the spray structure and thickness differ significantly. For the small droplet diameter, the spray is clearly thicker, especially in the head of the spray, the distribution of the parameters within the spray is more uniform, and the higher density area of the gas phase (containing agent vapor) is much larger.

An additional aerodynamic effect is the tendency to form a vortex in the vicinity of the head of the spray with the small droplet mean diameter. This phenomenon enhances the area filled with the agent vapor. The spray with the large droplet mean diameter is significantly narrower and without the fat head.

An unexpected effect is the generation of local temperatures which exceed the initial ambient temperature. This is caused by the strong shock waves created during the discharge process which compress the gas. The possibility of creating such waves is indicated by Thompson *et al.* (1986) and Kurschat *et al.* (1992). Although the vaporization of the agent cools the chamber, it is possible that some local temperature rise due to compression of the strong shock waves can offset the temperature decrease due to evaporation. The maximum gas phase temperature is significantly higher for the spray with the small droplet mean diameter (there is an increase of 40 K relative to the initial ambient temperature), whereas for the large droplet mean diameter, the temperature only increases 10 K. This influences the interphase heat transfer, which depends on the surface area and temperature difference. Similar observations, with one exception, were made at a lower initial ambient temperature (-30 °C). For the small droplet diameter, the maximum gas-phase temperature is just slightly higher than the ambient value, but for the large droplet diameter, it is lower than the ambient value and nearly equal to the liquid phase temperature (boiling point).

Similar observations were also made when a longer compartment was used in the simulation. Again, there is one exception in the gas-phase temperature behavior. For the large droplet mean diameter, the maximum gas-phase temperature is slightly higher than the ambient value. For the small droplet diameter, the temperature is lower by 51° relative to the ambient value. The conclusion is that the dynamics of the interphase heat exchange phenomena depends strongly on the spray Sauter mean diameter, ambient temperature, and geometry.

The effect of compartment geometry on the dispersion process can be assessed by comparing the results obtained using two different compartment heights. In the case of a 1.0 m high compartment, a pressure wave is generated by the fast moving agent just after 1 ms of the process. The pressure wave reflects alternately from the bottom and top walls of the compartment, thereby influencing the spray



Figure 72. Agent discharge. Computational mesh and liquid phase outlines at: 8.3 ms, 10.7 ms, and 14.3 ms (upward). Initial spray angle 0°. Spray mean diameter 200 μ m. Ambient T = 294 K and P = 101 kPa. Compartment 1 m x 0.42 m.









Figure 73. Vapor density maps at: (3.5, 8.3, 10.7, 14.3) ms (upward); the outline value 0.032 kg/m^3 . Initial spray angle 0°. Spray mean diameter 200 µm. Ambient T = 294 K and P = 101 kPa. Compartment 1 m x 0.42 m.







Figure 75. Gas phase pressure maps at: (3.5, 8.3, 10.7, 14.3) ms (upward); the outline value 105 kPa. Initial spray angle 0°. Spray mean diameter 200 µm. Ambient T = 294 K and P = 101 kPa. Compartment 1 m x 0.42 m.







Figure 76. Gas phase density maps at: (3.5, 8.3, 10.7, 14.3) ms (upward); the outline value 1.27 kg/m^3 . Initial spray angle 0°. Spray mean diameter 200 µm. Ambient T = 294 K and P = 101 kPa. Compartment 1 m x 0.42 m.



Figure 77. Vapor density maps at: (3.5, 8.3, 10.7, 14.3) ms (upward); the outline value 0.026 kg/m^3 . Initial spray angle 0°. Spray mean diameter 600 µm. Ambient T = 294 K and P = 101 kPa. Compartment 1 m x 0.42 m.









Figure 79. Gas phase pressure maps at: (3.5, 8.3, 10.7, 14.3) ms (upward); the outline value 113 kPa. Initial spray angle 0°. Spray mean diameter 600 μ m. Ambient T = 294 K and P = 101 kPa. Compartment 1 m x 0.42 m.



Figure 80. Gas phase density maps at: (3.5, 8.3, 10.7, 14.3) ms (upward); the outline value 1.37 kg/m^3 . Initial spray angle 0°. Spray mean diameter 600 μ m. Ambient T = 294 K and P = 101 kPa. Compartment 1 m x 0.42 m.









Figure 81. Vapor density maps at: (3.5, 8.3, 10.7, 14.3) ms (upward); the outline value 0.036 kg/m³. Initial spray angle 0°. Spray mean diameter 200 μ m. Ambient T = 243 K and P = 101 kPa. Compartment 1 m x 0.42 m.





Figure 82. Gas phase temperature maps at: (3.5, 8.3, 10.7, 14.3) ms (upward); the maximum value 249 K. Initial spray angle 0°. Spray mean diameter 200 μ m. Ambient T = 243 K and P = 101 kPa. Compartment 1 m x 0.42 m.



Figure 83. Gas phase pressure maps at: (3.5, 8.3, 10.7, 14.3) ms (upward); the outline value 89 kPa. Initial spray angle 0°. Spray mean diameter 200 μ m. Ambient T = 243 K and P = 101 kPa. Compartment 1 m x 0.42 m.



Figure 84. Gas phase density maps at: (3.5, 8.3, 10.7, 14.3) ms (upward); the outline value 1.29 kg/m^3 . Initial spray angle 0°. Spray mean diameter 200 μ m. Ambient T = 243 K and P = 101 kPa. Compartment 1 m x 0.42 m.



Figure 85. Vapor density maps at: (3.5, 8.3, 10.7, 14.3) ms (upward); the outline value 0.029 kg/m³. Initial spray angle 0°. Spray mean diameter 600 µm. Ambient T = 243 K and P = 101 kPa. Compartment 1 m x 0.42 m.



Figure 86. Gas phase temperature maps at: (3.5, 8.3, 10.7, 14.3) ms (upward); the maximum value 233 K. Initial spray angle 0°. Spray mean diameter 600 μ m. Ambient T = 243 K and P = 101 kPa. Compartment 1 m x 0.42 m.

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Figure 87. Gas phase pressure maps at: (3.5, 8.3, 10.7, 14.3) ms (upward); the outline value 0.93 kPa. Initial spray angle 0°. Spray mean diameter 600 μ m. Ambient T = 243 K and P = 101 kPa. Compartment 1 m x 0.42 m.



Figure 88. Gas phase density maps at: (3.5, 8.3, 10.7, 14.3) ms (upward); the outline value 1.35 kg/m^3 . Initial spray angle 0°. Spray mean diameter 600 µm. Ambient T = 243 K and P = 101 kPa. Compartment 1 m x 0.42 m.

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Figure 89. Vapor density maps at: (3.5, 8.3, 9.4, 9.5) ms (upward); the outline value 0.024 kg/m^3 . Initial spray angle 0°. Spray mean diameter 200 µm. Ambient T = 294 K and P = 101 kPa. Compartment 2.25 m x 0.42 m.











Figure 91. Gas phase pressure maps at: (3.5, 8.3, 9.4, 9.5) ms (upward); the maximum value 121 kPa. Initial spray angle 0°. Spray mean diameter 200 µm. Ambient T = 294 K and P = 101 kPa. Compartment 2.5 m x 0.42 m.









Figure 92. Gas phase density maps at: (3.5, 8.3, 9.4, 9.5) ms (upward); the outline value 1.39 kg/m³. Initial spray angle 0°. Spray mean diameter 200 μ m. Ambient T = 294 K and P = 101 kPa. Compartment 2.25 m x 0.42 m.



Figure 93. Vapor density maps at: (3.5, 8.3, 10.7, 13.8) ms (upward); the outline value 0.024 kg/m^3 . Initial spray angle 0°. Spray mean diameter 600 µm. Ambient T = 294 K and P = 101 kPa. Compartment 2.5 m x 0.42 m.













Figure 95. Gas phase pressure maps at: (3.5, 8.3, 10.7, 13.8) ms (upward); the outline value 102 kPa. Initial spray angle 0°. Spray mean diameter 600 μ m. Ambient T = 294 K and P = 101 kPa. Compartment 2.5 m x 0.42 m.







Figure 96. Gas phase density maps at: (3.5, 8.3, 10.7, 13.8) ms (upward); the outline value 1.19 kg/m^3 . Initial spray angle 0°. Spray mean diameter 600 µm. Ambient T = 294 K and P = 101 kPa. Compartment 2.5 m x 0.42 m.

structure and dynamics. The spray becomes thicker and slower. In the case of a 2.5 m long compartment, the spray penetration is greater, but at the same time the spray and its head are thinner. After 1 ms, the wave generated by the spray still travels downward, approaching the bottom wall. It eventually reaches the wall at a later moment and becomes weaker due to dissipation and collision with the head of the spray. The speed of the pressure wave for the two different compartment heights is about the same. Because of the difference in the compartment height, the pressure wave influences the spray structure 2.5 times more frequently in the short compartment. Moreover, the interactions are shifted in time, thereby affecting the spray dynamics and structure. A similar effect was observed when a different initial droplet mean diameter was used.

By comparing the results for the two different ambient temperatures (with the spray mean diameter fixed at 200 μ m), the penetration of the spray is about the same; however, the spray structures differ significantly. This is evident especially in the temperature, pressure, and density fields. At an ambient temperature of 21 °C, the density outlines of the gaseous phase are significantly broader. The maximum temperature of the gas phase is higher than the initial ambient temperature by about 40 °C, thereby enhancing the interphase heat exchange process. At – 30 °C, the maximum temperature is about the same as the ambient one. This phenomenon may be associated with the much faster evaporation process, which depends on the interphase heat exchange rate and the compressibility effect in the gas phase. To further substantiate these observations, the behavior of the reflected pressure waves from the walls is examined. In the case of the higher ambient temperature, the pressure wave travels faster. After 10.7 ms in the 1.0 m long compartment, the pressure wave interacts with the spray, while in the other case the pressure wave reaches the spray only after 14.3 ms. Thus, the evaporation process affects the phenomena associated with the compressibility effects.

The effect of initial ambient temperature depends, to some extent, on the spray mean diameter. For the 600 μ m diameter, all the phenomena described above occur, with the exception of the maximum gas-phase temperature. In this case, the peak temperature is only slightly higher than the ambient value. This indicates that the process of cooling via evaporation dominates the compression heating.

In all cases, the maximum values of the velocities in the gas phase are between 80 m/s and 290 m/s, the lower values being comparable to those measured in the discharge experiments (refer to Section 8.4). In the computations, the initial velocity of the agent droplets was assumed to be 61.4 m/s, and the ambient air was initially stagnant. In the near-field, momentum is transferred to the air, increasing the air velocity at the expense of the droplets. In the far-field, the velocities of both phases would be close to each other if vaporization were not present. However, the velocity of the gas phase increases dramatically. Under certain conditions the agent evaporation process becomes so fast that the vapor leaving the droplet surface contributes to the resultant gas velocity. The phenomenon occurs in the vicinity of the injector orifice and also downstream, well beyond the injector area. The maximum values occur first in the vicinity of the injector orifice during the first 3 ms of the process, and then in a later stage, between 10 ms (smaller diameter) and 14 ms (bigger diameter) downstream. Such high velocities lead to the creation of pressure waves, reflecting further between the walls. When the local temperature of the gas phase approaches the agent boiling point, the Mach number may exceed the sonic condition because the speed of sound decreases with the square root of absolute temperature. This means that a weak shock wave is created. The wave initially appears close to the injector exit and then travels downstream toward the end wall. It is also stronger in the area downstream, where higher gas velocities occur. A typical maximum increase of pressure at the wave is about 20 %.

It is necessary to remember that the dynamics of the dispersion process depends on the initial conditions in the transitional region where we have no experimental data to use as input for the

calculations. In the above simulations, the droplets were assumed to have a specific size and the temperature was set equal to the agent boiling point, -41 °C. In reality, the droplet size distribution is unknown, and the theories leading to droplet formation under such dynamic conditions have not yet been established. Likewise, the temperature may be higher in the transition zone. Because the evaporation model approximates the agent vapor pressure as an exponential function of the surrounding temperature, small changes in temperature have a large impact on the spray dynamics.

It should be emphasized that realistic results from such models as CONCHAS-SPRAY/KIVA-II will be dependent on the validity of the transition-region concept and on the ability to provide reasonable estimates of the state of the spray at the initial section (*i.e.*, the boundary conditions). In spite of the fact that the initial droplet size distribution may play an important role in the agent dispersion process, a method for estimating such a distribution does not seem to be forthcoming. In the present calculations, the droplet size distribution could only be hypothesized.

A numerical problem was encountered when solving the set of equations describing the droplet evaporation process. The problem arises when the agent vapor pressure equation is described by an exponential function of temperature and when the temperature difference between the liquid and gas phases is over 10 °C; this problem becomes worse with increasing temperature difference between the two phases. Under such conditions, the required time step to obtain stable computation is on the order of 10^{-12} s when the original vapor pressure equation is used. If the vapor pressure equation is analytically modified in such a way to mitigate the sharp pressure increase, the time step is increased by six orders of magnitude to 10^{-6} s. However, this modification is associated with the change of the original thermophysical properties of the compound. Therefore, it is impossible at the present time to compare the computational results with the dispersion characteristics determined experimentally. Note that the LANL codes are originally designed for applications to internal combustion engines, where liquid hydrocarbons are used. It is possible that a separate subroutine could be written to solve the vapor pressure equation with a numerical method appropriate for this highly non-linear behavior. One possible solution to this problem is to adopt the numerical techniques that are used in solving exponential chemical kinetics equations.

A further comparison between KIVA-II and CONCHAS-SPRAY for the same set of conditions is needed to determine the impact of the turbulence and droplet breakup model on the results. If the outcome is independent of the code, additional analyses with CONCHAS-SPRAY could be conducted. If the turbulence and droplet models influence the results, KIVA-II or preferably KIVA-III (Amsden *et al.*, 1993) should be used for future calculations.

The KIVA codes have the additional advantage that fully three-dimensional spaces can be handled. Extended resources on a CRAY computer will be required for these investigations. The spectrum of parameters influencing the spray dynamics should be broadened to include additional compartment geometries, discharge orifice geometry, thermodynamic and gas dynamic parameters of the agent and ambient air, agent injection and atomization parameters, as well as the conditions of the transition region of spray.

8.5.5 Conclusions. Numerical simulation was carried out using the LANL CONCHAS-SPRAY and KIVA-II computer codes in an attempt to better understand agent dispersion processes. The following conclusions can be drawn based on the computational results obtained so far:

1. CONCHAS-SPRAY is at least an order of magnitude more efficient, in terms of the computational time, than KIVA-II when applied to the dispersion process of a simulated halon alternative. Both codes give the same qualitative and quantitative results using the same input data.

- 2. It was found from the calculations that the geometry of a compartment into which the agent was released is an important parameter governing the spray dynamics. The propagation of pressure waves generated by the fast moving agent affects the shape and penetration of the spray and eventually affects the agent vapor mass fraction field due to pressure wave reflections. The smaller the compartment, the thicker the spray, the lesser the spray penetrates, and the broader the agent density field.
- 3. The liquid agent atomization process, expressed in terms of Sauter mean diameter, influences the spray structure significantly. The smaller the spray mean diameter, the better the agent disperses.
- 4. For a higher ambient temperature, the spray is broader, and the penetration of the spray is farther.
- 5. The occurrence of higher gas velocities corresponds to the presence of pressure waves. The phenomenon may be associated with the enhanced evaporation process which contributes to the total gas velocity.

8.6 Pipe Flow Characteristics of Alternative Agents

8.6.1 Introduction. Three potential near-term halon alternatives for engine nacelle fire protection applications have been identified by the Technology Transition Team (Grosshandler *et al.*, 1994): HFC-227ea, CF_3I , and HFC-125. In engine nacelle applications, the fire suppression agent is located remotely from the nacelle, and a piping system delivers the agent to the nacelle injection location(s). It is crucial to deliver the proper amount of agent in a timely manner to achieve the desired agent concentration in the nacelle for the required time interval. The flow regime in the piping is characteristically a two-phase, two-component gas/liquid system. Since no universally accepted flow prediction method is available for two-phase flows, an experimental study was undertaken to investigate the effects of selected parameters on the flow of alternative agents. The work here focused on the performance of alternative agents based on flow time through piping systems. Rapid transport to, and effective mixing in the nacelle are important for an efficient design utilizing an alternative agent.

The initial selection criteria of the alternative agents included more than just environmental concerns and suppression efficiency. NIST Special Publication 861 (Grosshandler *et al.*, 1994) details the measurements and analyses performed by NIST that assisted in the selection of a reduced list of alternatives. Detailed studies of pipe flow characteristics were not included, primarily because it represents a portion of the overall system performance which does not, to some extent, depend only on the chemical properties of the agent. However, design parameters such as pipe diameter, length, and storage conditions play a significant role in the system's discharge performance. In this study, the effects of the following design parameters were examined: bottle fill condition and temperature, pipe diameter, length, elbows, tees, contractions, and expansions. Steady pipe flow experiments were performed by adding make-up nitrogen to the storage bottle as the agent was discharged. From these experiments, mass flow rate and pipe pressure drop as functions of fixed bottle pressure were obtained. Optical access to the flowing agent in the pipe verified the nature of the flow.

8.6.2 Agent Discharge in Piping. The fact that there is two-phase flow of halon or of alternative agents stems from the "optimized design" of a generic suppression system, expressed as the desire to

get most of the agent to the nacelle very quickly, while minimizing the storage space. First, the bulk of the agent is in the liquid phase in the storage bottle which drastically cuts down on the storage volume. The liquid phase is subject to flashing during depressurization. The portion of the bottle volume that contains vapor is referred to as the ullage. Second, nitrogen gas is added to the storage bottle, raising the ullage pressure above the saturation vapor pressure of the agent; the ullage pressure is the main driving force for the expulsion of the agent from the bottle. Nitrogen dissolves in each of the agents to some extent and can come out of solution during depressurization, initiating two-phase flow.

At this point, a qualitative description of halon or alternative agent pipe flow sets the stage for the more detailed discussion to follow. For now consider that the bottle is very large relative to the piping so that conditions in the bottle do not change during the ensuing pipe flow. Thus, upstream conditions are given by stagnation properties and the pipe flow is steady. Traversing down the long horizontal pipe (with no significant entrance effects, valves, etc.) the pressure is dropping due to wall friction losses and acceleration of the expanding fluid. If the agent is originally saturated with nitrogen, a pressure decrease results in a supersaturated liquid and bubbles can form. Ignoring the effects of bubble inception (nucleation) and growth, a differential amount of nitrogen will come out of solution after a differential drop in pressure from the stagnation conditions (assuming an isothermal liquid phase, so that Henry's law constant does not change). These bubbles will contain both nitrogen and agent vapor even though the total pressure may still be above the saturation vapor pressure of the agent. The quality of the flow (defined as the mass fraction in the gas phase) increases as the pressure drops along the pipe. The two-phase flow is compressible; thus it is possible to exhibit choking. In gaseous flows, choking is the realization of a thermodynamic limit on the mass flux as the flow achieves its sonic velocity; critical flow and choked flow have the same meaning. In two-phase gas/liquid flows choking may be realized at velocities below that of the gas-phase sonic velocity. If the flow is choked, the mass flow rate is independent of the discharge pressure (here, the pressure of the ambience).

An actual bottle discharge exhibits highly transient flow. At first, as the pipe is filling up with agent, the pipe pressure at any point is rising rapidly while the bottle pressure is falling rapidly. The pipe pressure achieves a maximum when the pipe just fills, then starts to fall along with the bottle pressure. As the bottle pressure drops, the dissolved nitrogen in the liquid phase may not come out of solution immediately, resulting in a metastable, supersaturated solution. After some time delay, degassing of the dissolved nitrogen can produce a net pressure rise inside the bottle due to the expansion of a bubbly two-phase fluid. The bottle pressure may reach a local maximum then start to drop again as this two-phase fluid continues to empty. At the point where the last of the two-phase mixture is being expelled from the bottle, a pressure rise is observed in the piping for a short period of time. The fluid in the piping is changing from a low quality (mostly liquid) fluid to a high quality (mostly gaseous) fluid, which exhibits a different pressure drop in the piping. At that point, most of the agent has been discharged from the bottle and piping. The pressure in the piping starts dropping again as the ullage contents discharge.

8.6.3 Single-Phase Flow. Single-phase steady pipe flows are well understood, and accurate flow calculations can be readily performed. Single-phase flows are treated as either incompressible (*i.e.*, liquids) or compressible (*i.e.*, gases). As a rule of thumb, if the change in fluid density is greater than 10 % over the flow section of interest, the flow is treated as compressible. Incompressible flow is the simplest case and is described below to illustrate the significant parameters affecting the flow characteristics.

As is characteristic of all fluid flows, the driving force is the pressure gradient. The pressure force is needed to overcome the resistance to flow arising from fluid drag at the pipe walls. For both laminar and turbulent flow, the following equation holds for flow in a straight horizontal pipe.

$$\frac{\Delta P}{L} = \frac{2f\rho u^2}{D} \tag{37}$$

 ΔP is the pressure drop over the pipe length of interest. L is the pipe length, D is the diameter, ρ is the fluid density, u is the mean velocity and f is the Fanning friction factor. The friction factor is a parameter that describes the fluid-wall interaction; it is possible to calculate the friction factor for laminar flow from pipe geometry, flow conditions and fluid properties. For turbulent flow, the friction factor is treated empirically and is correlated with pipe geometry, flow conditions and fluid properties. Correlations for the friction factor are cast in terms of the Reynolds number (a dimensionless group formed by the ratio of the inertia force to the viscous force) and the relative roughness of the pipe, ε/D where ε is the surface roughness and has been measured for many typical piping materials. The Reynolds number also is indicative of the flow regime with laminar flow existing below Re ~ 2300 and turbulent flow above Re ~ 4000 with a transition region between 2300 and 4000 for pipe flows. The concept of equivalent length is very useful in determining flow rates or pressure drops in realistic piping systems with fittings, bends, valves, etc. Simply put, the concept is to equate the observed pressure drop in a pipe fitting with an equivalent length of pipe (L_{eq}) with the same diameter and friction factor as the piping of interest. A complicated piping system is reduced to an equivalent (fictitious) total pipe length. L_{ed}/D can be obtained from head loss coefficients tabulated for many different pipe fittings (SFPE Handbook, 1988). A loss coefficient divided by (2f) yields L_{ed}/D .

If the fluid is compressible, the density is a function of pressure, and the equation above is not appropriate. The flow depends on the thermodynamic path the fluid follows. Usually, to simplify analysis, a compressible flow is assumed to be either isothermal, adiabatic isentropic or adiabatic isenthalpic, which fixes the thermodynamic path. The same friction factor correlations hold for compressible flow as those for incompressible flow, though the friction factor may change locally as the flow progresses.

8.6.4 Two-Phase Flow. Two-phase gas/liquid flows are common in the chemical process industries and in electric power plants (steam/water system). Design calculations for many steady two-phase flow conditions are readily available, although the accuracy of two-phase flow correlations can be poor (Olujić, 1985); all two-phase flow correlations rely on some empiricism. The lack of a generalized physical model stems in part from the number of flow regimes possible in two-phase gas/liquid flows. These flow regimes are descriptive of the phase separation and include bubbly, annular, wavy, slug and mist flows. A correlation which is reasonable for wavy flow may be extremely poor for bubbly flow, etc. A flow map, which is a graphical depiction of the conditions giving rise to the various flow regimes, can be used to assist in selecting the best correlation equations, though it is best to have direct knowledge of the flow regime if the conditions are uncertain. Depending on the flow regime, a particular model may be preferred to describe the flow. The homogeneous flow model assumes that the two phases are well mixed and traveling with the same velocity which is reasonable for bubbly flows. There are equilibrium and non-equilibrium homogeneous flow approximations. Separated flow models, where the liquid and gas phases are not well mixed and may be travelling at different speeds, characterize the rest of the cases. A good discussion of two-phase gas/liquid flow is presented by Whalley (1990).

The literature on transient two-phase flows mainly involves the high-pressure water/steam system. This is due to the importance of nuclear reactor safety assurance during a loss-of-coolant-accident (LOCA). Most of the literature on halon 1301 systems deals with design of systems for total flooding applications with agent discharge times which are an order of magnitude longer than that required for engine nacelle fire protection application. A review of halon discharge testing and system design for total flooding applications is presented by DiNenno and Budnick (1988). They also include an extensive bibliography on two-phase flow in general and halon systems in particular. The most comprehensive study detailing transient discharge of halon 1301 through piping systems reported in the literature is by Elliot *et al.* (1984).

Design calculation schemes for halon systems are for the most part proprietary. System designers only have to provide a design that works for a given case, meaning design concentrations of agent are maintained for the appropriate times. This requires an experimental test to verify a specific design. Three models of halon 1301 pipe flow described in the literature are mentioned below. The National Fire Protection Association's Standard 12A "Halon 1301 Fire Extinguishing Systems" (National Fire Codes, 1995), gives a simplified design method for systems that should meet the performance criteria for extended discharge into enclosed spaces. The pipe flow equation is an application of the steadystate mechanical energy and mass balances with a specified friction factor and data specific to halon 1301 inserted; homogeneous, equilibrium two-phase flow is assumed. Pipe flow calculations are based on a fixed average bottle pressure condition and adiabatic, isenthalpic flow. DiNenno and Budnick (1988) point out potential accuracy and uncertainty problems in this method. This method was designed for extended discharges (discharge times of about 10 s). Elliot et al. (1984) use a homogeneous, equilibrium model of two-phase flow to solve for transient pipe and nozzle flow in halon systems. They also examined a Los Alamos National Laboratory code "Sola-Loop" developed for the steam/water system which uses the "drift-flux model," a separated flow model, to predict the transient discharge of halon. Both of their calculations compare favorably to some limited halon 1301 tests.

These documented flow calculations do well for certain situations, but rely on data explicit to halon 1301, and make various assumptions about the flow. The observations and data from this study verify some of the assumptions made, and their validity for the alternative agents. Here, a model that simulates the transient discharge of an agent through piping has been developed. It is similar to the NFPA 12A methodology and Elliot's model in that it is based on a homogeneous, equilibrium two-phase flow description. Thermodynamic properties and fill conditions are obtained from a separate vapor-liquid equilibrium computer program called "PROFISSY" (see Section 8.3).

8.6.5 Experimental Apparatus. A range of test conditions was examined to better understand the nature of the alternative agent flows under conditions that may be encountered in suppression system piping. Both transient and quasi-steady flow cases were investigated. The quasi-steady flow experiments were performed to obtain the mass flow rate and steady pipe pressure drop for a fixed discharge pressure.

The experimental apparatus was designed so that different configurations and conditions could be studied. A schematic diagram of the experimental apparatus is shown in Figure 97 and each of its components is described below. System constraints, specific to engine nacelle applications, limited the range of conditions investigated. Pipe diameters and lengths were selected based on a review of military aircraft specifications. Also the fill condition and nitrogen pressurization were selected based on knowledge of typical halon 1301 engine nacelle systems.

8.6.5.1 Discharge Vessel and Piping Design. The discharge vessel was constructed from a cylindrical tube of stainless steel with an internal diameter of 100 mm and a wall thickness of



Figure 97. Schematic of the apparatus for pipe flow experiments.

12.7 mm. Flanges were welded to the two ends of the tube so that top and bottom plates, 19.1 mm thick, could be attached with bolts (Figure 98). The seal between a flange and plate was made with an O-ring. The vessel volume with the flanges attached is $4.06 \times 10^{-3} \text{ m}^3$. This volume was verified by water displacement. The combined standard uncertainty in the volume measurement is estimated to be less than $\pm 10 \text{ cm}^3$. The top plate had five threaded access holes for agent filling, two thermocouples, a pressure transducer, and make-up nitrogen. The bottom plate had one threaded access hole for the release mechanism. The vessel was designed to operate at pressures up to 14 MPa. There was a pressure relief valve connected to the filling port to protect the vessel from unexpected high pressure conditions.

The agent release mechanism was a quick-opening solenoid valve (Marotta Scientific Controls Inc., Model MV121KJ-2) with an inlet diameter of 31.8 mm and an exit diameter of 44.5 mm. The valve exit is at a 90° angle to the inlet. This valve can be easily re-armed for quick turn-around of tests. The valve was activated by a DC power source of approximately 25 V and 10 A; the opening time was on the order of 10 ms. Since the valve exit was much larger than the internal pipe diameters used in this study, a smooth tapered connector was fabricated to make the internal diameter transition from 44.5 mm to 15.9 mm (Figure 99).

The nitrogen make-up tanks were two standard high-pressure gas cylinders with a combined internal volume of 88 x 10^{-3} m³. The cylinder valves were removed and 19.1 mm I.D. tubing leading to a large-orifice solenoid valve was used to deliver the make-up nitrogen. The valve was a piston-plunger type solenoid valve which required a pressure difference of 30 kPa to open after the solenoid was activated. The opening time is on the order of 50 ms for nitrogen flow at test condition pressures. The make-up nitrogen entered the vessel from an access hole in the top flange. There were two reasons for using a solenoid valve here: first, the bottle contents must be separated from the mitrogen make-up tanks or else agent vapor will mix with the nitrogen, and second, it was desired to shut down the nitrogen flow after the agent discharge to avoid losing most of the make-up nitrogen and pressurizing the entire system.

The piping was drawn stainless steel tubing. Two diameters were investigated, 15.9 mm and 9.5 mm inner diameter with tube wall thicknesses of 3.2 mm. Brass pressure taps were constructed with the same inner diameter as each of the pipes (Figure 100). These taps allowed the pipe pressure transducers to be installed flush to the tube wall. The transducers were mounted and maintained in the horizontal orientation. An extra access port was provided for thermocouple installation. The piping sections were connected with threaded fittings for easy reconfiguration, and rigidly attached to a bench to limit motion during agent discharge.

The recovery tanks contained the agent after it flowed through the piping. They consisted of four separate high pressure sample cylinders with a combined volume of 15×10^{-3} m³, located in a commercial freezer and surrounded by dry ice. This allowed most of the agent to condense in the sample cylinders after a test. Piping connected to the bottom of the cylinders led to a transfer cylinder which was used to transfer the agent back to the discharge vessel for subsequent tests. Usually, 80 % to 90 % of the agent was recovered with this system after a given test. This significantly reduced the total amount of agent required to perform the series of tests.

8.6.5.2 Experimental Measurements. The liquid and gas temperatures in the vessel were recorded by type E sheathed thermocouples prior to a test. Temperatures were not taken during tests because the thermal time lag through the sheath was longer than the discharge time. Attempts to measure the temperature of the fluid in the piping during discharge were unsuccessful because the unsheathed fine wire (13 μ m diameter) thermocouple probes would break in the flow.

The pressure in the storage vessel was recorded with a strain-gage-type transducer (Druck Model 330) with a total accuracy of 0.75 % and an operating and compensated temperature range of



Figure 98. Schematic of the pressure vessel.





Figure 100. Schematic of pipe tap for pressure transducer and thermocouple.

- 54 °C to 150 °C. The transducer's pressure range was 0 MPa to 14 MPa with a response time on the order of 1 ms. Pressure in the piping was recorded by strain-gage type, flush-mount pressure transducers (Omega Model PX-600) with a diaphragm face diameter of 7.9 mm. The transducers had a nominal range of 0 MPa to 3.4 MPa or 0 MPa to 6.9 MPa, depending on the gage. The stated accuracy was 1 % over the compensated temperature range of 16 °C to 71 °C. The operable temperature range was - 54 °C to 150 °C; a temperature correction must be applied to achieve the nominal accuracy outside the compensated temperature range. The gage response time was on the order of 1 ms. A fast-response (2 kHz) signal conditioner/amplifier provided the boosted voltage signal that was recorded by the data acquisition system. In early trial experiments, piezoelectric-type dynamic transducers were used in the piping. Pipe pressure traces from the dynamic transducers and the strain-gage type transducers were essentially the same for nitrogen flow, but for agent flow the results deviated near the end of the liquid two-phase expulsion. The deviation was attributed to a significant non-compensated temperature effect experienced by the dynamic transducers. That problem eliminated dynamic transducers from consideration for use.

Initially, the pipe pressure transducers were calibrated against the more accurate and stable storage vessel transducer. This was done by filling the vessel and the piping with nitrogen to a number of pressures and recording the output from each pipe transducer and comparing it to the vessel transducer output. Periodic re-calibrations indicated zero shifts for the pipe pressure transducers on the order of \pm 30 kPa, though very little change in calibration-line slopes was evident. The pipe pressure transducers were subjected to water-hammer-type shock waves during flow start-up, which can sometimes cause a zero-shift. Therefore, the pipe pressure traces are offset to initial values of 0.101 MPa in the reported results.

Transparent acrylic tube test sections with the same inner diameter as the larger diameter pipe (15.9 mm) were fabricated and one transparent test section was hydrostatically tested to failure at a pressure of 11 MPa. A section was connected to the piping at the upstream location right after the valve reducer and was replaced after one or two tests. A high speed movie camera (Photec IV) operating at 500 frames per second recorded the flow through the transparent section. Kodak Ektachrome high-speed daylight film with intense lighting at a 90° angle to the camera was used. The camera was turned on approximately 1 second before the discharge valve was activated to allow the camera motor to come up to the stated framing rate. The liquid two-phase flow portions of selected tests were filmed and analyzed.

Data were acquired with a Strawberry Tree Flash-12 Model 1 data acquisition board installed in a DOS-based personal computer. This board can scan a single channel at a rate of 1 MHz. Here the multiple channels were scanned at 1 kHz since the response time of the pressure transducers is only 1 ms. A timing circuit triggered the data acquisition and sent signals to solid state relays to activate the valves and movie camera if a high speed movie was taken. The raw data were stored in the computer for subsequent analysis.

8.6.5.3 Test Matrix. The series of tests included a number of transient discharges with various initial vessel conditions. The effects of agent temperature, fill volume, and vessel pressure were investigated. Piping configurations included 3.5 m of straight pipe, straight pipe with a 90° angle bend, tee, pipe expansion, and pipe contraction. Figure 101 details the piping configurations. Constant-head tests were performed with fixed 3.5 m of piping at 3 constant vessel pressures: 3.0 MPa, 2.5 MPa, and 2.0 MPa. Table 34 details the nominal conditions for the separate tests. A total of 21 different tests were performed for each agent. A limited number of repeated tests at slightly different fill conditions were also performed. The initial pipe and recovery tanks pressure were at atmospheric pressure except for one test where the pipe and recovery tanks were evacuated.



Figure 101. Piping configurations. Pressures were recorded at tap locations marked with a "p".

	Pipe ge	cometry	Initial storage vessel conditions			
Parameter tested	Diameter (mm)	Nominal length (m)	Pressure (MPa)	Liquid fill volume (%)	Temperature (°C)	
Constant-head	9.5 and 15.9	3.0	2.0, 2.5 and 3.0	N.A.	23	
Straight pipe	9.5 and 15.9	3.0	2.75 and 4.12	50 and 67	23	
Storage temperature	15.9	3.0	4.12	50	- 45 and 71	
90° Bend	9.5 and 15.9	3.0 and 4.0	4.12	50	23	
Tee	15.9	4.0	4.12	50	23	
Expansion	9.5 to 15.9	3.0	4.12	50	23	
Contraction	15.9 to 9.5	3.0	4.12	50	23	

Table 34. Test Matrix

Pressure taps with installed transducers were placed at four locations in the piping. The first pipe pressure tap was in the 15.9 mm diameter section of the valve reducer. For straight piping the pressure was recorded at 1.00 m intervals from the valve reducer. Since the distance from the valve reducer to the flow split in the recovery tanks was 3.50 m, the last pressure recording was at the 3.00 m location. For the 3.50 m of piping with a 90° angle bend, the taps were located at 1.00 m intervals starting from the valve reducer. For the 4.50 m of piping with a 90° angle bend, the taps were located at the taps were located at the valve reducer. For the 4.00 m downstream. For the flow-splitting tee experiments, the taps were located at the valve reducer and at 2.00 m downstream before the tee. The tee was located 2.10 m downstream, and the last two taps were place in the opposing pipes 0.50 m from the tee. Pressure in the recovery tanks was monitored in about half of the tests.

Cold storage vessel tests were performed with the large diameter straight piping configuration. An insulated sheet-metal sleeve was placed around the vessel and filled with dry ice to cool the liquid contents to about -45 °C (Figure 102). First the vessel was filled to 1/2 liquid volume at 23 °C and pressurized with nitrogen to a total pressure of 4.12 MPa, then dry ice was added to the cooling jacket. When the temperature of the liquid reached about -45 °C, the agent was discharged.

High storage vessel temperature tests were also performed with the large diameter straight piping configuration. Electrical heating tape was wrapped around the vessel to heat up the vessel and contents (Figure 102). First, the vessel was filled to 1/2 liquid fill volume at 23 °C and pressurized with nitrogen to 4.12 MPa, then power was supplied to the heating tape. When the temperature of the liquid reached about 70 °C, the agent was discharged. Pipe pressures were recorded at the valve reducer section and the last pressure tap 3.00 m downstream. The pipe pressure transducers for the elevated temperature discharge have a nominal range of 0 MPa to 6.9 MPa.

8.6.6 Results and Discussion. The initial fill conditions for each test are given in Tables B-1 to B-5 (see **Appendix B**). The temperatures of the liquid and gas phase in the storage vessel were between 21 °C and 23° C (23 °C being the ambient room temperature) for the room temperature fill conditions. The mass of agent and pressure were recorded prior to each test. The combined standard uncertainty in the mass determination is estimated as ± 0.01 kg based on the load cell uncertainty of ± 0.005 kg



Figure 102. Agent cooling and heating techniques for the cooled and heated agent discharge tests.

and the fact that typically three agent transfers were required to fill the storage vessel. The combined standard uncertainty in the tabulated pressures is ± 0.005 MPa due to rounding off the third digit past the decimal and not due to the stated uncertainty of the transducer. The computer program "PROFISSY" was used to estimate the fill condition and mass of the liquid and gas phase assuming equilibrium conditions were achieved at a temperature of 23 °C.

Temperature equilibration was normally achieved in about 1 hour. Slow bubbling of the nitrogen through the liquid phase (typically 30 min in duration) greatly facilitated nitrogen dissolution. For some storage vessel fills, the agent/nitrogen mixture was allowed to sit for 60 h or longer. The pressure drop over such a long time is indicative of the level of nitrogen saturation achieved by the bubbling technique. Two equilibrium calculations (using PROFISSY) were performed based on the agent mass and the pressure achieved just after bubbling and after a long time storage. Assuming the system had equilibrated after the long storage, the nitrogen saturation level achieved from bubbling alone was approximately (90 \pm 5) % of the mass required for saturation. HFC-125 achieved the lowest level of nitrogen saturation from the bubbling technique, while HFC-227ea and CF₃I nearly achieved nitrogen saturation from bubbling alone. The test results did not appear to be strongly influenced by differences in nitrogen saturation level on the order of 10 %.

Pressure traces for all tests are shown in Appendix B. The results from the different configurations are discussed below.

8.6.6.1 Constant-Head Discharge. The three selected pressures (3.0 MPa, 2.5 MPa and 2.0 MPa) cover the bottle pressure range where most of the liquid would be discharged from the bottle for a typical transient discharge. Figure 103 (Figure B-2 in **Appendix B**) is a typical constant-head pressure trace for the small diameter piping. The agent was halon 1301 with the vessel pressure fixed at 2.5 MPa, and the pipe pressure was initially atmospheric. At approximately 1000 ms the discharge valve opened. The vessel pressure dropped sharply then recovered due to the nitrogen flow from the make-up tanks. The make-up tank solenoid valve was set to open 20 ms after the discharge valve opened (the solenoid valve requires a pressure difference to open). The make-up tanks were filled to the initial vessel pressure and quickly re-pressurized the vessel. The vessel pressure remained nearly constant until the make-up nitrogen valve was closed about 2.5 s after it was opened. The pipe pressure traces rose sharply after the discharge valve opened, reaching a maximum about 100 ms later and remained nearly constant until the liquid ran out of the vessel. As the liquid runs out of the vessel, the pressure starts to rise in the pipe, then falls as the pipe empties the two-phase fluid contents. After the make-up tank valve was closed, the bottle and pipe pressures continue to drop until the pressure equilibrated about 3.5 s after the beginning of the discharge.

Figure 104 (Figure B-14 in **Appendix B**) is a typical constant-head pressure trace for the large diameter piping. The agent was halon 1301 with a vessel pressure fixed at 2.0 MPa, and the initial pipe pressure was atmospheric. The pipe pressure traces rise fast and take about 300 ms to reach steady values. The crossover of the third pressure tap's trace with that of the fourth tap's trace is due most likely to an incorrect time response of the gage at the third tap during the transient pipe filling period. The recovery tank pressure starts to climb much slower than the pipe pressures and ends up at the final pressure. Notice that the increasing recovery pressure has no noticeable effect on the pipe pressure trace until after the two-phase fluid runs out of the bottle and gas is being discharged. The pressure equilibrated at about 2.0 MPa.

Experimental time-averaged mass flow rates were estimated by dividing the initial liquid-phase mass (calculated by PROFISSY) by the time increment from the initial rise in the pipe pressure to the first indication that the liquid has run out of the vessel. The combined standard uncertainty in the experimental mass flow rate was estimated by the RSS method (root-sum-of-the-squares) for the standard uncertainty of the time increment and the liquid mass. Here the standard uncertainty of the









liquid-phase mass is estimated to be equal to the standard uncertainty of the total agent mass. The results, including the uncertainty estimates are reported in Table 35.

The large diameter pipe mass flow rates are 3 to 4 times greater than the small diameter pipe results. Theoretically, during the filling and emptying of the pipe, the mass flow rate should be higher than the steady value; there is a shorter length of pipe for the contents to flow through. Since the filling and emptying times of the large pipe tests are a significant fraction of the total flow time, the estimated average flow rate over the entire test will be higher than the flow rate during the steady-flow portion of the test.

The constant-head discharge tests yield steady pipe flow conditions from 0.5 s to over 4 s, depending on the fill condition and pipe diameter. There are no observed effects of the recovery tanks pressure on the two-phase flow during the tests. The flatness of the pressure traces suggests that transient heat transfer from the pipe walls to the two-phase fluid was not significant. Nonlinearity of the pressure drop in the piping is indicated by the variation in the pressure difference between the equally spaced pipe taps, which implies compressible flow. Another indication that the flow is compressible is that if the flow is assumed to be incompressible, calculated mass flow rates from Equation (37) using the initial densities are on the order of 2-3 times greater than the experimentally determined mass flow rates.

8.6.6.2 Transient Discharge. The transient discharge tests simulated two-phase pipe flow of realistic systems. A typical ambient temperature, straight piping test result is shown in Figure 105 (Figure B-41 in Appendix B). Initially, the vessel was filled with HFC-125 to 1/2 liquid fill volume, pressurized to 4.12 MPa with the pipe (0.0159 m I.D.) and recovery tanks at atmospheric pressure. At approximately 1000 ms the discharge valve was opened, and the vessel pressure started to drop while the pipe and recovery tanks pressures started to rise. The pipe pressures peak, then start to fall with the vessel pressure. Notice that one trace, representing the second transducer, crossed over the third and fourth traces. As stated above, this effect is most likely due to that transducer's incorrect response to the highly transient conditions; it is apparent in the large diameter tests, though not apparent in the small pipe diameter tests. The vessel and pipe pressures stop decreasing after about 1400 ms, achieve a local maximum, and then resume decreasing. This pressure recovery is thought to be due to degassing of nitrogen in the storage bottle. At about 1800 ms the vessel pressure starts decreasing at a faster rate, which is attributed to the discharge of the ullage contents after the two-phase mixture leaves the vessel. At the same time, pipe pressures start to increase, peak, then decrease at a much faster rate which is indicative of the two-phase mixture leaving the piping followed by discharge of the ullage contents. All pressure traces equilibrate to about 1.2 MPa. The final equilibrated pressure for the transient tests ranged from 0.5 MPa to 1.5 MPa, depending on the agent, fill conditions, and piping configuration.

Experimental liquid discharge times were estimated for each test as the time interval from the initial vessel pressure drop to the final hump in the last pipe pressure trace. There is some uncertainty in the liquid discharge time since the last amount of liquid draining from the vessel will not flow out as a plug, but will be entrained into the discharging ullage gas. The bracketed time is estimated to represent lower and upper limits of the liquid discharge time, with the best estimate given by the mean value, and the standard uncertainty given by the half width of the time interval divided by the square root of 3 (Taylor and Kuyatt, 1994). The results for the experimental liquid discharge times are presented in Tables 36-39. The discharge times for the large diameter pipe were approximately 2.5 times shorter than the small diameter pipe.

The effects of the increasing recovery tanks pressure on the transient flow are identifiable, given the pipe and recovery tanks pressure traces. If the pipe exit pressure is above the discharge pressure (recovery tanks pressure), the flow remains choked, and the flow rate is independent of the discharge

Agent/ figure number	Pressure (± 0.005 MPa)	Pipe I.D. (mm)	Liquid phase mass (± 0.01 kg)	Flow time (± 0.025 s)	Mass flow rate [*] (kg/s)	Calculated mass flow rate
						(kg/s)
CF ₃ Br/B1	2.00	9.5	3.00	2.95	1.02 (0.01)	1.09
CF ₃ BR/B2	2.50	9.5	3.15	2.50	1.26 (0.01)	1.28
CF ₃ Br/B3	3.00	9.5	3.09	2.00	1.55 (0.02)	1.50
CF ₃ Br/B14	2.00	15.9	3.87	1.00	3.89 (0.10)	3.39
CF ₃ Br/B15	2.50	15.9	3.57	0.85	4.20 (0.12)	3.92
CF ₃ Br/B16	3.00	15.9	4.23	0.93	4.55 (0.12)	4.51
HFC 125/B26	2.00	9.5	2.31	2.05	1.13 (0.01)	1.06
HFC 125/B27	2.50	9.5	2.75	2.50	1.10 (0.01)	1.14
HFC 125/B28	3.00	9.5	2.32	1.80	1.29 (0.02)	1.33
HFC 125/B37	2.00	15.9	2.54	0.77	3.30 (0.11)	3.07
HFC 125/B38	2.55	15.9	3.15	0.70	4.50 (0.16)	3.51
HFC 125/B39	3.00	15.9	2.63	0.60	4.38 (0.18)	3.98
HFC 227ea/B51	2.00	9.5	3.24	2.45	1.32 (0.01)	1.36
HFC 227ea/B52	2.50	9.5	3.10	2.05	1.51 (0.02)	1.51
HFC 227ea/B53	3.00	9.5	3.16	2.00	1.58 (0.02)	1.70
HFC 227ea/B62	2.00	15.9	3.68	0.70	5.25 (0.19)	4.25
HFC 227ea/B64	2.50	15.9	3.59	0.70	5.13 (0.18)	4.81
HFC 227ea/B65	3.00	15.9	3.49	0.65	5.37 (0.21)	4.93
CF ₃ I/B77	2.00	9.5	5.32	3.50	1.52 (0.01)	1.61
CF ₃ I/B78	2.50	9.5	5.50	3.00	1.83 (0.02)	1.95
CF ₃ I/B79	3.00	9.5	5.13	2.60	1.97 (0.02)	2.11
CF ₃ I/B88	2.00	15.9	3.92	0.60	6.53 (0.27)	5.16
CF ₃ I/B89	2.50	15.9	4.10	0.60	6.83 (0.29)	5.94
CF ₃ I/B91	3.00	15.9	4.20	0.60	7.00 (0.29)	6.45

Table 35. Constant-head mass flow rates

*Bracketed numbers are combined standard uncertainties (2 σ) for experimental mass flow rates.

pressure. If, however, the pipe exit pressure is equal to the discharge pressure, then the flow rate depends on the discharge pressure, and it will not achieve the maximum choked value. For these tests, the pipe exit pressure is somewhat lower than that recorded by the last pressure tap (the pipe terminates at the recovery tanks some distance downstream from the last pressure tap), so if that trace follows the trend of the recovery tanks pressure it is evidence that the flow is not choked at the exit.



Figure 105. Large diameter piping (15.9 mm I.D.) transient discharge test. The configuration was straight piping. The vessel was filled to 1/2 liquid fill volume with HFC-125 and pressurized to 4.12 MPa.

	1						
Pipe ge	ometry	Fill conditions		Figure number	t_d (exp.) (s)	(ca	t_d llc.) (s)
Piping config.	Diameter (mm)	Pressure (MPa)	Liquid fill vol. (%)			Equil.	Non-Eq.
Straight	9.5	2.77	46.6	B-4*	3.3 - 3.7	3.41	3.41
	15.9	2.75	51.1	B-17*	1.4 - 1.7	1.46	1.46
	9.5	4.12	50.4	B-8*	2.5 - 2.8	2.79	3.02
	15.9	4.12	49.6	B-19	1.0 - 1.2	1.03	1.08
	9.5	4.12	65.7	B-9*	3.6 - 4.0	4.23	4.48
	15.9	4.12	65.5	B-20 [*]	1.4 - 1.6	1.55	1.66
3.5 m,90°	9.5	4.12	48.9	B-10*	2.9 - 3.1	2.75	2.95
Bend	15.9	4.12	48.7	B-21	1.1 - 1.3	1.02	1.05
4.5 m, 90°	9.5	4.12	49.1	B-11*	3.0 - 3.4	3.09	3.32
Bend	15.9	4.12	48.7	B-22	1.1 -1.3	1.14	1.16
Expansion	9.5/15.9	4.12	48.3	B-13	2.1 - 2.4	1.84	1.98
Contraction	15.9/9.5	4.12	48.3	B-12	2.4 - 2.8	2.30	2.43
tee	15.9	4.12	47.9	B-23	1.0 - 1.2	1.00	1.05
straight	15.9	2.20^{+}	41.8	B-24	1.4 - 1.5	1.09	1.10

Table 36. Liquid discharge times for halon 1301

^{*}Two-phase flow appears effected by recovery tanks pressure [†]Cold temperature discharge

To simulate the flow of the two-phase mixture discharging to the atmosphere, it is only necessary to delay the effect of the recovery tanks pressure increase until the ullage contents start to discharge, ensuring that the two-phase flow is always choked. In some cases it appears that the recovery tanks pressure is high enough to affect the two-phase flow from the piping, occurring at the later stage of the two-phase discharge. Figure 106 (Figure B-54 in **Appendix B**) is an example of this effect. The agent is HFC-227ea and the fill conditions are 1/2 liquid fill volume, pressurized to 2.75 MPa. The configuration was straight, small diameter piping. The pressure trace for the last tap location shows a gradual pressure increase starting at about 4500 ms. This behavior is most evident for the following conditions: high fill volume, small diameter piping and agents with lower saturation vapor pressures (HFC-227ea and CF₃I). If the flow is not always choked at the pipe exit, the flow time will be slightly longer than the flow time for an atmospheric discharge. The tests that appear to be affected by the finite size of the recovery tanks are indicated in Tables 36-39. This effect probably results in not more than a 10 % increase in flow time for any affected test.

In some experiments, like the one described in Figure 105, the storage vessel pressure experiences a recovery to a local maximum. It is most pronounced for halon 1301 and HFC-125 2/3 fill

Pipe geo	Pipe geometry Fill conditions		Figure number	t _d (exp.) (s)	t _d (calc.) (s)		
Piping config.	Diameter (mm)	Pressure (MPa)	Liquid fill vol. (%)			Equil.	Non-Eq.
Straight	9.5	2.76	50.4	B-30*	3.1 - 3.3	3.35	3.48
	15.9	2.75	54.2	B-40	1.3 - 1.5	1.40	1.43
	9.5	4.13	56.4	B-31*	2.7 - 2.9	2.79	3.34
	15.9	4.12	52.7	B-42	0.9 - 1.1	0.99	1.08
	9.5	4.12	69.8	B-32	3.4 - 3.7	4.19	4.65
	15.9	4.12	65.9	B-44*	1.4 - 1.5	1.39	1.59
3.5 m,90°	9.5	4.12	47.1	B-33	2.6 - 2.8	2.28	2.53
Bend	15.9	4.12	48.8	B-46	0.9 - 1.0	0.90	0.99
4.5 m, 90°	9.5	4.12	48.1	B-34	2.6 - 2.8	2.63	2.92
Bend	15.9	4.12	46.6	B-47	0.9 -1.0	0.95	1.02
Expansion	9.5/15.9	4.12	49.6	B-35	1.9 - 2.1	1.67	1.91
Contraction	15.9/9.5	4.12	49.8	B-36	2.2 - 2.4	2.10	2.34
tee	15.9	4.12	54.4	B-48	0.9 - 1.0	1.04	1.18
straight	15.9	2.55 [†]	39.1	B-49	0.9 - 1.0	0.80	0.82

Table 37.Liquid discharge times for HFC-125

*Two-phase flow appears effected by recovery tanks pressure

[†]Cold temperature discharge

conditions. It is not strong for HFC-227ea and CF_3I and is only indicated in one trace each for those agents. No visual observations of the fluid inside the vessel were made during these tests. It has been surmised that the pressure recovery is due to nitrogen coming out of a supersaturated solution, which in turn increases the liquid two-phase volume compressing the ullage gas and raising the vessel pressure. The exact mechanism(s) for bubble formation is not known and it is possibly a combination of heterogeneous and homogenous nucleation. There is scatter in the lowest pressure reached before the pressure rise for any agent. It appears to depend on the fill volume, vessel pressure and pipe diameter. If it was purely homogenous nucleation, it should only depend on the initial pressure. This degassing phenomenon has implications for the discharge time and is explored in the model calculations to follow.

8.6.6.3 Straight Pipe Configuration. The straight pipe configuration was the baseline for the tests at different fill conditions and vessel pressures. Figure 101(a) shows the configuration. Typical results are shown in Figures 105 and 106. Two fill conditions (nominally 1/2 and 2/3 liquid filled) at 4.12 MPa and two vessel pressures (4.12 MPa and 2.75 MPa) at 1/2 liquid filled were examined. The liquid discharge time was always longer for the 2/3 liquid filled condition compared to the 1/2 liquid

Pipe geo	ometry	Fill conditions		Figure number	t _d (exp.) (s)	(ca (<i>t_d</i> .lc.) (s)
Piping config.	Diameter (mm)	Pressure (MPa)	Liquid fill vol. (%)			Equil.	Non-Eq.
Straight	9.5	2,75	56.4	B-54*	4.2 - 4.4	3.44	3.81
	15.9	2.75	54.1	B-66	1.2 - 1.4	1.19	1.27
	9.5	4.12	53.7	B-55*	2.9 - 3.1	2.43	2.63
	15.9	4.12	49.4	B-68	0.8 - 0.9	0.80	0.84
	9.5	4.12	70.0	B-57*	4.1 - 4.4	3.90	4.80
	15.9	4,12	69.4	B- 70 [*]	1.7 - 1.9	1.40	1.64
3.5 m,90°	9.5	4.12	49.2	B-58	2.5 - 2.7	2.19	2.31
Bend	15.9	4.12	49.4	B-7 1	0.8 - 1.0	0.82	0.86
4.5 m, 90°	9.5	4.12	49.2	B-59	2.6 - 2.9	2.47	2.61
Bend	15.9	4.12	48.8	B-72	0.9 - 1.0	0.90	0.94
Expansion	9.5/15.9	4.12	49.0	B-61	1.8 - 2.0	1.46	1.55
Contraction	15.9/9.5	4.12	49.0	B-60	2.1 - 2.3	1.81	1.90
tee	15.9	4.12	48.4	B-73	0.8 - 0.9	0.79	0.82
straight	15.9	3.00 [†]	43.5	B-74	0.9 - 1.0	0.75	0.76
	15.9	5.95 [‡]	59.8	B-76	0.8 - 0.9	0.90	1.03

Table 38. Liquid discharge times for HFC-227ea

*Two-phase flow appears effected by recovery tanks pressure

[†]Cold temperature discharge

[‡]Hot temperature discharge

filled condition. The relative increase depends on the particular agent. The liquid discharge times for the 2.75 MPa charge pressures were 30 % to 50 % greater than the 4.12 MPa charge pressures

8.6.6.4 Piping with a 90° Angle Bend. Tests with 3.5 m of piping with a smooth-bend-radius 90° angle bend located 2.5 m downstream, and 4.5 m of piping with a smooth-bend-radius 90° angle bend located 3.5 m downstream (see Figures 101(b) and (c)) were performed. Comparing the straight piping results to the 3.5 m piping with a 90° angle bend, the liquid discharge times and the pressure drops are longer and lower, respectively, for the configurations with the elbow compared to the straight piping results. The 4.5 m piping with a 90° angle bend displayed still longer liquid discharge times, due to the effects of the bend and the extra 1.00 m of piping.

8.6.6.5 Straight Pipe with a Sudden Expansion. The sudden expansion configuration consisted of 1.00 m of straight 9.5 mm I.D. piping connected to 2.5 m of straight 15.9 mm I.D. piping (see Figure 101(d)). Figure 107 (Figure B-13 in Appendix B) is a typical pressure trace. Following the

Pipe geo	ometry	Fill co	Fill conditions		t _d (exp.) (s)	t _d (calc.) (s)	
Piping config.	Diameter (mm)	Pressure (MPa)	Liquid fill vol. (%)			Equil.	Non-Eq.
Straight	9.5	2.75	49.7	B-80 [*]	4.1 - 4.3	3.35	3.57
	15.9	2.80	49.6	B-92	1.3 - 1.4	1.26	1.32
	9.5	4.13	50.2	B-81*	3.4 - 3.6	2.61	2.76
	15.9	4.12	48.7	B-93	1.0 - 1.1	0.94	0.98
	9.5	4.12	67.9	B-83*	5.1 - 5.3	4.38	5.10
	15.9	4.12	68.2	B-94 [*]	1.9 - 2.1	1.62	1.84
3.5 m,90°	9.5	4.12	52.0	B-84*	3.6 - 3.8	2.84	3.02
Bend	15.9	4.12	49.4	B-95	1.0 - 1.1	0.98	1.02
4.5 m, 90°	9.5	4.12	51.2	B-85*	3.8 - 4.0	3.12	3.29
Bend	15.9	4.12	55.1	B-96	1.2 -1.4	1.29	1.35
Expansion	9.5/15.9	4.12	53.8	B-87	2.5 - 2.6	2.01	2.15
Contraction	15.9/9.5	4.12	52.4	B-86	2.9 - 3.1	2.40	2.53
tee	15.9	4.12	49.9	B-97	0.8 - 1.0	0.98	1.02
straight	15.9	3.05 [†]	44.2	B-98	1.1 - 1.2	0.95	0.96
	15.9	5.70 [‡]	56.7	B-99	0.9 - 1.0	1.00	1.09

Table 39. Liquid discharge times for CF₃I

*Two-phase flow appears effected by recovery tanks pressure

[†]Cold temperature discharge

[‡]Hot temperature discharge

first pipe tap's pressure trace, after it initially rises, it tracks the vessel pressure trace very closely, with a small pressure difference between the two. This is common to all tests where a section of small piping follows the valve reducer. The reason these two pressure traces track so closely is because the first pressure tap is located in the valve reducer where the inner diameter is 15.9 mm and most of the flow resistance occurs in the small diameter piping located 0.05 m downstream. There is a large pressure drop from the valve reducer to the next pressure tap 1.00 m downstream. The next two pressure records 2.00 m and 3.00 m downstream are only slightly different, and after some time they follow the recovery tanks pressure trace. These observations suggest that the flow does not choke at the recovery tanks entrance, but at the expansion point. This implies that liquid discharge time to the end of the first 1.00 m of piping would be equivalent to the liquid discharge time through a 1.00 m section of piping discharging to atmosphere.

8.6.6.6 Straight Pipe with a Sudden Contraction. The sudden contraction configuration consisted of 1.00 m of straight 15.9 mm I.D. piping connected to 2.00 m of straight 9.5 mm I.D.



Figure 106. Small diameter piping (9.5 mm I.D.) transient discharge test. The configuration was straight piping. The vessel was filled to 1/2 liquid fill volume with HFC-227ea and pressurized to 2.75 MPa.




piping, then connected back to 0.5 m of 15.9 mm I.D. piping leading to the recovery tanks (see Figure 101(e)). Figure 108 (Figure B-35 in **Appendix B**) is a typical trace. The first two pipe tap pressure traces rapidly rise to just below the bottle pressure; they then track the bottle pressure for the rest of the test. The first two traces follow the bottle pressure because the taps are located in the 15.9 mm I.D. valve reducer and piping which are before the contraction. The next two traces show significant pressure drop from the bottle pressure. These traces are for the taps 2.00 m and 3.00 m downstream located in the 9.5 mm I.D. piping. The liquid discharge time for this configuration should be approximately equivalent to the discharge time for a 2.00 m section of 9.5 mm I.D. piping leading from the vessel to the atmosphere.

8.6.6.7 Tee Configuration. The tee configuration consisted of 2.10 m of 15.9 mm I.D. piping leading to a tee fitting, connected to two legs, one 1.80 m long and the other 2.05 m long, both with 2 smooth radius 90° angle bends leading to the recovery tanks (see Figure 101(f)). Figure 109 (Figure B-73 in **Appendix B**) is a typical example. The first pipe tap's pressure trace rises quickly then follows the vessel pressure with some pressure difference. The second pipe tap's pressure trace (the tap is located just before the tee) rises quickly, then falls much slower than the vessel or the first tap's pressure trace until the hump signifying the liquid emptying from the pipe. The pressure traces for the taps located in the tee legs 0.50 m downstream of the tee essentially follow the second pipe tap's trace with some pressure drop. The slight difference in the two tee leg traces is most likely due to the slight difference in length to the recovery tanks.

8.6.6.8 Cooled Agent Discharge. Cooled agent discharges were performed with 3.5 m of straight 15.9 mm I.D. piping (Figure 101(a)). Table B-1 details the initial fill conditions, final test pressures, and the equilibrium conditions. Thermal equilibrium was never achieved between the liquid and gas phases due to conduction of heat from the gas phase. Liquid phase temperatures ranged from $-44 \,^{\circ}C$ to $-50 \,^{\circ}C$, while gas phase temperatures were around $4 \,^{\circ}C$. Though thermal equilibrium was not achieved, the final pressures were only approximately 10 % higher than predicted equilibrium pressures based on the liquid-phase temperature for all of the agents. Figure 110 (Figure B-98 in **Appendix B**) is a typical trace. The pressure trace results appear similar to ambient temperature discharges at low storage vessel pressure. The longer liquid discharge times for the cooled agent discharge appear to be due to the lower initial vessel pressure.

The first pipe tap's pressure trace shows a temperature effect. The pressure tap is located in the valve reducer which is in contact with the vessel via the discharge valve. Cold ambient temperature affects the zero and the slope of the pressure transducer calibration. (Calibration of the gage recording the recovery tanks pressure was done at the cold ambient temperature). The other pressure taps are located far downstream and do not appear to be cooled below ambient conditions appreciably during the agent chilling period.

8.6.6.9 Heated Agent Discharge. High temperature agent discharges were performed with 3.5 m of straight piping 15.9 mm I.D. (Figure 101(a)). Table B-1 details the initial fill conditions, the final temperatures and pressures, and calculated equilibrium conditions based on the final temperature. The final temperature achieved in all of the tests was approximately 70 °C. PROFISSY calculations indicate that two of the agents (halon 1301 and HFC-125) are single-phase fluids at this elevated temperature: the bottle conditions are given in Table B-1. The temperatures inside the bottle as measured with the two installed thermocouples were within 5 °C. Temperature differences could be due to non-equilibrated gas and liquid phases or a temperature gradient in the fluid. The results are shown in Figures 111-114 (Figures B-25, B-50, B-75 and B-99 in **Appendix B**) for halon 1301, HFC-125, HFC-227ea and CF₃I respectively. The pipe pressure traces for halon 1301 and HFC-125







Figure 109. Transient discharge test; tee configuration. The vessel was filled to 1/2 liquid fill volume with HFC-227ea and pressurized to 4.12 MPa.







Figure 111. Transient heated agent discharge test; straight piping. The vessel was filled to 1/2 liquid fill volume with halon 1301, pressurized to 4.12 MPa, then heated to 70 °C prior to discharge.







Figure 113. Transient heated agent discharge test; straight piping. The vessel was filled to 1/2 liquid fill volume with HFC-227ea, pressurized to 4.12 MPa, then heated to 70 °C prior to discharge.



Figure 114. Transient heated agent discharge test; straight piping. The vessel was filled to 1/2 liquid fill volume with CF₃I, pressurized to 4.12 MPa, then heated to 70 °C prior to discharge.

do not show characteristic signs of liquid run out. This observation can be explained by the prediction of single-phase halon 1301/nitrogen and HFC-125/nitrogen mixtures at the recorded elevated temperatures. A single-phase compressible fluid is discharging in those cases. HFC-227ea and CF₃I pipe pressure traces do show liquid run out. These liquid discharge times are not significantly less than the discharge times if the agents had been discharged at 23 °C at the lower initial bottle pressures. One effect that compensates for the higher discharge pressure is the increase in the initial liquid-phase volume in the storage vessel and consequently a decrease in the ullage volume.

8.6.6.10 Visual Observations. The transparent test section was installed and high speed movies taken of a limited number of experiments. CF_3I attacked the plastic pipe strongly and would probably shatter the tube consistently, and therefore, it was the only agent not filmed. Although the test section was hydrostatically tested to failure at 11 MPa, during testing a number of tube failures occurred at around 1 MPa. These failures were attributed to temperature induced failure and/or chemical attack on the plastic tube. All failures occurred at the end of the test after the pressures had equilibrated.

The location of the sight tube was at the exit of the vessel just after the valve reducer section. The initial vessel conditions were ambient temperature, 4.12 MPa charge pressure, and 1/2 liquid fill volume. In addition, one HFC-227ea test with the same fill conditions had the sight tube located 2.8 m downstream from the valve reducer. One constant-head test of HFC-227ea at 2.5 MPa was filmed with the sight tube located just downstream from the valve reducer. The most important observation for all of these tests was that a cloudy two-phase fluid was observed immediately after the vessel valve was opened, and it persisted until essentially all of the liquid contents of the vessel had emptied. This implies that bubbles nucleate somewhere before this location, and two-phase flow is always present during the liquid discharge period. Figure 115 shows four frames of the movie taken of the HFC-125 discharge. The first picture is of the frame just before the arrival of the agent. The next picture is of the flow during the middle of the two-phase flow time and is indicative of the frames throughout the two-phase flow period. The last picture is of a frame near the end of the two-phase flow time. The tube is noticeably clearing up.

The time interval between the initial cloudy flow and until it starts to clear up is indicative of the liquid discharge time and it roughly corresponds to the time from the initial pipe pressure rise to the end of the hump in the pipe pressure traces. Table 40 gives the time interval from the first cloudy frame to the point where the flow is clearing up.

8.6.7 Transient Discharge Model. The experimental study yielded a great deal of quantitative information on the flow characteristics of the alternative agents and halon. The effects of piping configuration and initial fill conditions were shown to have significant impact on the flow characteristics. Even though a number of tests were performed, a very limited range of conditions and piping configurations were explored. Modeling the pipe flow would allow other configurations and conditions to be examined. With this in mind, a computer model was developed that predicts the transient discharge of an alternative agent superpressurized with nitrogen from a storage bottle through piping. The model assumptions and equations are presented below.

The model assumes homogeneous, equilibrium two-phase compressible flow. The homogeneous assumption specifies that the gas and liquid phase are well mixed and traveling at the same velocity. This is a good assumption for bubbly flow at low void fractions. Bubbles tend to coalesce at higher void fractions; the transition from bubbly flow to plug flow occurs around a void fraction of 0.3 (Whalley, 1990). At high liquid flow rates though, large bubbles tend to be broken up. The fluid is assumed to be in thermodynamic equilibrium at all locations. Adiabatic, isenthalpic flow is assumed, greatly simplifying the analysis. The change in entropy is attributed to lost work due to frictional



t = 0 ms

t = 2 ms



t = 1200 ms

t = 1400 ms

Figure 115. A sequence from a high-speed movie (500 frames/s) of the flow of HFC-125 in 15.9 mm I.D. piping just after the valve reducer.

Agent	Figure number	Pressure (MPa)	Cloudy flow interval (± 0.05 s)	<i>t_d</i> (exp.) (s)
Halon 1301	B-18	4.12	1.2	1.0 - 1.2
HFC-125	B-41	4.12	1.2	0.9 - 1.0
HFC-227ea	B-67	4.12	1.1	0.8 - 0.9
HFC-227ea	B-68	4.13	0.9	0.8 - 0.9
HFC-227ea	B-63	2.5	1.1	0.6 - 0.7

Table 40. Two-phase time interval from high speed movies

losses. The agent flow in the piping is always assumed to be two-phase flow for initially nitrogensaturated bottle conditions. If the liquid agent is not saturated with nitrogen, the flowing agent is a single-phase liquid until the pipe pressure drops to the equilibrium nitrogen saturation pressure, and at that point degassing occurs. The liquid in the bottle can either follow the isenthalpic pressure/density path, or be frozen (constant density, single-phase liquid) until a critical pressure is reached. There is no mass transfer across the initial liquid-gas boundary in the bottle; bubbles that form in the liquid phase are assumed to stay in the liquid phase. The ullage gas expands adiabatically and isentropically, and is treated as an ideal gas. The ratio of specific heats $C_p/C_v = \gamma$ is a constant value.

Figure 116 shows schematically the bottle and pipe fluid contents for the modeled conditions. Pipe filling and emptying, and ullage discharge are calculated differently than the pipe full case.

8.6.7.1 Model Equations. The foundation for the flow prediction is an application of the steadystate mechanical energy balance and continuity equation for pipe flow. An important assumption is that the process is quasi-steady, *i.e.*, the upstream stagnation properties (the bottle conditions) are fixed over a time increment, and a steady mass flow rate is calculated. The small out-flow of mass changes the bottle conditions for the next time increment. The steady-state differential mechanical energy balance is

$$u\,du\,+\,\frac{dP}{\rho}\,+\,\delta LW\,=\,0\tag{38}$$

where δLW , the lost work term for pipe flow is given by

$$\delta LW = \frac{2fu^2}{D}dL \tag{39}$$

The equation of continuity for steady flow is



Figure 116. Schematic showing stages of agent discharge. The model calculation scheme is different for the various stages.

$$\rho u = G = const. \tag{40}$$

where G is the mass velocity.

Equation (38) can be re-arranged, substituting for u with the continuity equation, and integrated over the pipe length to yield

$$\int_{P_1}^{P_2} \rho \, dP + G^2 \ln(\frac{\rho_1}{\rho_2}) + \frac{2fG^2L}{D} = 0 \tag{41}$$

States I and 2 are the initial (upstream) and final (exit) states respectively. This equation is applicable to steady compressible flow in pipes with diameter D. If the pipe length is known, G can be calculated, though the calculation is not explicit since the pipe exit pressure is not specified and the flow may be choked. The solution is the pipe exit pressure that maximizes the value of G in Equation (41). Re-arranging Equation (41) so that it is explicit in G, a one-dimensional search is performed to obtain the value of the exit pressure that maximizes G.

If the discharge pressure (the pressure of the receiving vessel, or atmospheric pressure if discharged to the open) is above some critical value, the pipe exit pressure is at the discharge pressure for G to be maximized. If the discharge pressure is below some critical value, the flow in the pipe is choked, and the pipe exit pressure is higher than the discharge pressure.

In the above formulation, it is necessary to have the density of the two-phase mixture expressed explicitly as a function of pressure. Recall that the two-phase flow is assumed to expand adiabatically at constant enthalpy. The Gibbs phase rule for non-reacting systems is

$$F = c - \phi + 2 \tag{42}$$

where F is the degrees of freedom, c is the number of components, and ϕ is the number of phases. In a two-phase, two-component mixture, there are two degrees of freedom. Therefore, with enthalpy fixed, density can be expressed as a function of pressure. An appropriate equation of state is all that is needed to generate the functional relationship between density and pressure.

For any upstream condition, the mass velocity and pipe exit pressure can be calculated. This alone is not enough though for a prediction of the transient bottle discharge. Over each time increment, some mass exits the bottle and the pipe (after the pipe has filled). The difference between the mass leaving the bottle and that leaving the pipe is the change in the mass of the pipe contents. A mass balance must be applied to ensure that the mass exiting the pipe and the change in the mass of the pipe contents is equal to the mass leaving the bottle. This allows for a time increment to be calculated. The calculation scheme is described below.

Assuming the pipe is already filled at a given bottle pressure, a mass velocity and pipe exit pressure are calculated. From the mass velocity a mass flow rate is obtained; this is the mass flow rate leaving the bottle over a short time period. The mass in the pipe at that time is obtained by using a variation of Equation (41) and solving for pressure as a function of distance along the pipe. Since density is a known function of pressure, the density along the pipe is known. Therefore, the mass in the pipe is given by

$$Mp = \frac{\pi}{4}D^2 \sum_{x=0}^{L} \rho_{avg}(x) \Delta L$$
(43)

where Mp is the mass in the pipe, the sum of ΔL is the total length L, and $\rho_{avg}(x)$ is the mean density at the pipe location (x). The mass lost from the pipe is given by the following "liquid" mass balance, (a mass balance applied to the original liquid phase contents)

$$Mlp = (Mvl_{+} + Mp_{+}) - (Mvl_{+++} + Mp_{++++})$$
(44)

where Mlp is the mass lost from the pipe over the time increment Δt . Mvl is the liquid mass in the vessel. The liquid mass is obtained from the liquid volume multiplied by the liquid density at the vessel pressure. The liquid volume is the difference between the total volume and the ullage volume. The ullage expands isentropically and is assumed to behave as an ideal gas so the following relationship holds

$$P(\frac{V_u}{M_g})^{\gamma} = Const.$$
⁽⁴⁵⁾

where P is the bottle pressure, V_{μ} is the ullage volume and M_{g} is the mass of the ullage contents.

The time increment (Δt) over which the pressure in the bottle drops from P_t to $P_{t+\Delta t}$ is calculated from

$$\Delta t = \frac{Mlp}{(W_t + W_{t+\Delta t})/2} \tag{46}$$

where W_t and $W_{t+\Delta t}$ are the mass flow rates at times (t) and $(t+\Delta t)$ respectively. The bottle pressure is then stepped down and the process repeated. A pressure increment of 5 kPa was found to be sufficient for calculation accuracy.

The filling and emptying of the pipe with the two-phase mixture is approximated using a different method. In each case, the effective pipe length is segmented, and the calculation proceeds by the filling or emptying of the segments. Eight segments provided sufficient accuracy for the calculations described here.

In the filling case, the addition of each segment represents an increase in the pipe length for the calculation of pipe end pressure and mass velocity. The flow is assumed to be two-phase and choked at the "liquid front" which is located at the end of the particular segment of interest. For the first time increment, the pipe length is equal to the length of a segment. At this point, the bottle pressure, pipe end pressure, mass velocity and mass in the pipe are all unknown. The solution is obtained by guessing a bottle pressure, solving for the remaining unknowns and then comparing the "liquid" mass balance equation to that solution. The "liquid" mass balance reduces to

(47)

$$Mvl_t = Mvl_0 + Mp_t$$

during the pipe filling stage. Mvl_0 is the initial liquid-phase mass. Once the exit pressure and mass velocity are obtained for the guessed bottle pressure, the mass in the pipe can be calculated. The value of Mvl_t is used to calculate a new liquid volume in the bottle, and an ullage volume is obtained by difference. This ullage volume is compared to the ullage volume calculated from the isentropic expansion relationship using the initial bottle pressure guess. Successive guesses of bottle pressure are required until the ullage volumes converge. The time to fill the first section of pipe is estimated as the mass in the pipe divided by the mass flow rate. The time to fill the pipe up to the *j*th segment is estimated by

$$t_{fill, j} = \sum_{i=1}^{j} \frac{M p_i - M p_{i-1}}{W_i}$$
(48)

When the bottle is emptied of its originally liquid contents, a calculation similar to the one above is performed. To start, the first segment is empty of the two-phase mixture, and the bottle pressure is calculated from the isentropic expansion relationship, with the ullage volume equal to the bottle volume plus the empty pipe segment volume. Isentropic expansion of the ullage gas into the pipe is justified due to the relatively slow velocity of the two-phase mixture/gas front which acts as a slowmoving piston allowing the gas to expand essentially reversibly. The pipe length is equal to the effective length minus the segment. The pipe exit pressure is solved using Equation (41) and maximizing the mass velocity. The mass in the pipe is then calculated. The time increment for the liquid two-phase/gas front to travel to the end of the *j*th segment is estimated by

$$t_{empty, j} = \sum_{i=1}^{j} \frac{Mp_{i-1} - Mp_i}{W_i}$$
(49)

Emptying of the ullage contents (assumed to behave as an ideal gas) is calculated in a similar quasi-steady fashion. Assuming adiabatic flow of an ideal gas, the following relation holds (Balzhiser *et al.*, 1972)

$$G^{2} = \frac{0.5(1 - \frac{\rho_{2}^{2}}{\rho_{1}^{2}})}{\frac{\gamma - 1}{4\gamma}(\frac{\rho_{2}^{2}}{\rho_{1}^{2}} - 1) + (\frac{\gamma - 1}{2\gamma} - 1)\ln\frac{\rho_{2}}{\rho_{1}} + \frac{2fL}{D}}$$
(50)

After the two-phase slug leaves the pipe, the gas in the pipe accelerates to its maximum mass velocity given by the vessel pressure, assuming no ullage mass has left the piping. This creates a discontinuous step down to the exit pressure for the gas flow as a result of the approximation that the gas isentropically expands into the piping while the two-phase mixture is leaving the piping. In reality, there is a smooth, steep decrease in the exit pressure as the two-phase mixture leaves and the ullage begins to discharge.

The calculation proceeds in the following manner. The mass flow rate of the gas exiting the piping is multiplied by a small time increment. This mass is subtracted from the mass balance for the next time step. Successive guesses of the vessel pressure are made until the guessed pressure and the calculated pressure from isentropic expansion of the vessel contents converge. A new mass flow rate is obtained, and the process is repeated.

The pressure recovery in the bottle is thought to be due to nitrogen coming out of solution from the supersaturated liquid. In these tests there was no direct visual confirmation (*i.e.*, visualization of the formation of bubbles in the vessel liquid) proving that to be the case, but it is assumed to be the correct interpretation. For some agents and fill conditions the pressure recovery is more pronounced, whereas in others it is not evident. Pipe flow visualization with high speed movies does confirm the two-phase nature of the fluid exiting the valve. To simulate the retardation of the nitrogen degassing in the bottle, the calculation initially proceeds with the bottle liquid phase density fixed at the initial density. When the bottle pressure drops below a critical degassing pressure, nitrogen comes out of solution. The critical degassing pressure used is an experimentally inferred value for a particular agent and fill condition, and it is the lowest pressure achieved before the observed pressure rise. There is a lot of scatter in these data for a given agent; a mean value was selected. For HFC-227ea and CF_3I the calculation assumes no degassing in the bottle, since no characteristic pressure recovery was noticed for these agents (except one HFC-227ea and CF₃I test at 2/3 fill volume). When the bottle pressure drops below the degassing pressure, the liquid density decreases due to bubble formation. The liquid (two-phase) mass is fixed, thus its volume must increase. The increase in the liquid (two-phase) volume compresses the ullage gas, and the pressure rises. The mass velocity at the pipe exit does not instantaneously increase to a value given by an elevated bottle pressure. There must be a pressure wave moving down the length of the pipe (similar to the pipe filling case), since after the bottle pressure rises, more mass can be stored in the pipe. This transient is approximated in the following manner. Over the transient time Δt_{trans} , the mass lost from the pipe is

$$Mlp = W_t \Delta t_{trans} \tag{51}$$

Time (t) is at the beginning of the transient. The transient time is equal to the sum of the mass lost from the pipe and the change in the pipe mass divided by an average mass flow rate as shown below. The average mass flow rate is the mean value of the mass flow rate after the transient and the mass flow rate before the transient.

$$\Delta t_{trans} = \frac{Mp_{t+\Delta t} - Mp_t + Mlp}{(W_{t+\Delta t} + W_t)/2}$$
(52)

or substituting for *Mlp*

$$\Delta t_{trans} = \frac{M p_{t+\Delta t} - M p_t}{(W_{t+\Delta t} - W_t)/2}$$
(53)

To calculate Δt_{trans} , a bottle pressure at the end of the transient is guessed, $W_{t+\Delta t}$ and $Mp_{t+\Delta t}$ are calculated, then Δt_{trans} is calculated from above. The bottle liquid content mass at the end of the transient is

$$Mvl_{t+\Lambda t} = Mvl_{t} + Mp_{t} - Mp_{t+\Lambda t} - Mlp$$
(54)

From this bottle two-phase mass and the two-phase density based on the guessed pressure, the liquid volume and then the ullage volume is calculated. Bottle pressure is calculated from the isentropic relationship of the ullage gas, compared to the initial guessed pressure and, subsequently, new guesses are made until the two pressures converge.

8.6.7.2 Fill Conditions and Thermodynamic Inputs. Specific fill conditions, *i.e.*, mass in the liquid and gas phase and fill volume are calculated by the program PROFISSY. The thermodynamic relationships are obtained from a variation of the program PROFISSY that performs isentropic or isenthalpic expansion calculations. While the process for obtaining the thermodynamic inputs is not automatic, the inputs are valid for any fill condition at a given charge pressure and temperature. First, PROFISSY is used to calculate the mass of agent and nitrogen in the liquid and gas phases for a given fill condition and temperature. An isenthalpic expansion of the liquid phase and an isentropic expansion of the gas phase calculation is performed. For the isenthalpic expansion, at the initial storage pressure a single phase exists. During the expansion a two-phase gas/liquid mixture is formed. The average density, temperature, and mass fraction in the liquid phase are obtained at each pressure step. These results are fitted to third or fourth order polynomial functions of pressure. Appendix C shows some plotted results. The coefficients for the pressure/density for each of the agents at various pressures are given in Table C-1. The isentropic expansion of the gas phase begins with initially gaseous agent and nitrogen; upon expanding, some agent condenses. The mean density is plotted as a function of pressure, then fitted by a power law equation. The inverse of the exponent is defined as the ideal γ coefficient. Results are shown in Appendix C for agent/nitrogen mixtures initially at 4.12 MPa. The power-law exponent was insensitive to initial pressure, so γ was fixed for each agent/nitrogen mixture.

8.6.7.3 Friction Factor. Since the flow is described as a homogeneous two-phase flow, the friction factor analysis will follow the method for established for single phase flows. The Colebrook equation (see Perry *et al.*, 1984), shown below, relates the Fanning friction factor to pipe diameter, roughness, and Reynolds number (Re) for turbulent flows as

$$\frac{1}{\sqrt{f}} = -4\log_{10}\left(\frac{\varepsilon}{3.7D} + \frac{1.256}{Re\sqrt{f}}\right)$$
(55)

which requires an iterative scheme to solve for f. The Reynolds number is defined as

$$Re = \frac{uD\rho}{\mu}$$
(56)

At high Re, Equation (55) reduces to von Kármán's equation (see Perry et al., 1984)

$$\frac{1}{\sqrt{f}} = -4 \log_{10} \left(\frac{\varepsilon}{3.7D}\right)$$
(57)

Haaland (see Welty *et al.*, 1984) has shown that for $4 \ge 10^4 \le Re \le 10^8$ and $\varepsilon/D \le .05$ the friction factor can be expressed (within ± 1.5 %) as

$$\frac{1}{\sqrt{f}} = -3.6 \log_{10} \left[\frac{6.9}{Re} + \left(\frac{\varepsilon}{3.7D}\right)^{\frac{10}{9}} \right]$$
(58)

which allows explicit calculation of f. Moody (see Perry *et al.*, 1984) reports the value of surface roughness for drawn tubing as 0.00152 mm, and it is the value used here.

For the flows encountered in this study, the density and viscosity of the mixture are functions of pressure and temperature. Therefore the Reynolds number, and thus the friction factor, is a function of pipe position, and in addition, a function of time for the transient discharge. In the model, the friction factor is assumed to be constant. The validity of this assumption is demonstrated by the following analysis.

The NIST program REFPROP (Gallagher *et al.*, 1993) was used to obtain the viscosity of the liquid and gas phase for halon 1301, HFC-125, and HFC-227ea as a function of temperature; nitrogen viscosity as a function of temperature was also obtained. The viscosity of the homogeneous two-phase mixture was estimated from the functional form suggested by Dukler *et al.* (1964).

$$\mu_{h} = \mu_{g} \frac{y \rho_{h}}{\rho_{g}} + \mu_{l} \frac{(1-y) \rho_{h}}{\rho_{l}}$$
(59)

where y is the mass fraction of the gas phase, and the subscripts l, g and h stand for liquid, gas, and homogeneous respectively. The gas phase mixture viscosity at a given temperature was estimated as the sum of the agent and nitrogen gas viscosities multiplied by their respective volume fractions in the gas phase. Table C-2 in Appendix C gives the calculated results, expressed as polynomial fits, at initial pressures of 2.75 MPa and 4.12 MPa at 20 °C. The first step in the analysis was to calculate a

friction factor from the von Kármán equation, then simulate the flow. From the calculations, the velocity, density, and viscosity are known for any location and time in the pipe, thus local Reynolds numbers are calculated, and Equation (58) is used to obtain the local friction factor. The mean friction factor at a specified time is given by

$$f_m = \frac{1}{L} \sum_{i=1}^{L} \frac{f_i + f_{i-1}}{2} (x_i - x_{i-1})$$
(60)

where f_m is the mean friction factor, and x is a particular pipe location. Plots of the Reynolds number and mean friction factor versus time for a 4.1 m straight pipe with two different internal piping diameters (15.9 mm and 9.5 mm) are shown in Figures 117 to 122. In these figures, the thick dashed lines are the high Reynolds number limits for the friction factors given by the von Kármán equation. The mean Reynolds number is shown along with two curves representing $\pm 2\sigma$. The standard deviation of the mean friction factor is on the order of 10^{-6} and is not shown on the plots. Although the Reynolds number decreases over the flow time, the mean friction factor rises only slightly. The mean friction factor and the friction factor calculated from the von Kármán equation differ by 3 % for the larger pipe and 6 % for the small pipe. The main conclusion is that for the conditions here, the Reynolds number is high enough that for all practical purposes viscous effects can be ignored, allowing the von Kármán equation to be used.

Pressure drops across bends, valves, and other piping elements can be included by treating those items in terms of an effective pipe length. This technique is well established in hydraulic calculations. Here the flow elements of interest are the discharge valve and the reducer, 90° angle bends, expansions, contractions, and tees. The effective lengths for the discharge valve and valve reducer section were determined from the experimental pressure drops. The effective length was varied until the calculated pressure drop from the bottle pressure to the first pressure tap reproduced the experimental pressure drop. For the large diameter piping, an effective length equivalent to the open volume of the valve and the reducer ($1.25 \times 10^{-4} \text{ m}^3 \pm 2.5 \text{ cm}^3$) divided by the pipe cross-sectional area gives 0.63 m and approximates the experimental pressure drop. For the small pipe, the pressure drop across the valve and reducer was very small (most of the pressure drop occurs in the piping), and the effective length was estimated to be 0.1 m for the constant head tests. For the transient tests the pressure drop across the valve and reducer was ignored, and the open volume of the valve and reducer was added to the vessel volume. When the valve opens, it is assumed the fluid fills that volume and the ullage isentropically expands. The effective bottle volume becomes the bottle volume plus the valve and reducer volume.

The effective length for smooth bends in the piping was estimated from the following equation (SFPE, 1988)

$$\frac{2fL_{eff}}{D} = [0.13 + 1.85(\frac{r}{R})^{3.5}] (\frac{\alpha^{\circ}}{180^{\circ}})^{0.5}$$
(61)

where α° is the bend angle, R is the bend radius, and r is the pipe inner radius. With the friction factor given by the von Kármán equation, the equivalent length for the small pipe with a 50 mm radius bend was 0.25 m and for the large pipe with a 150 mm radius bend was 0.47 m.



Figure 117. Mean friction factor and Reynolds number as a function of time for a simulated transient discharge of halon 1301: 15.9 mm I.D. piping was specified, and the vessel was at 1/2 liquid fill volume and pressurized to 4.12 MPa.



Figure 118. Mean friction factor and Reynolds number as a function of time for a simulated transient discharge of halon 1301: 9.5 mm I.D. piping was specified, and the vessel was at 1/2 liquid fill volume and pressurized to 4.12 MPa.



Figure 119. Mean friction factor and Reynolds number as a function of time for a simulated transient discharge of HFC-125: 15.9 mm I.D. piping was specified, and the vessel was at 1/2 liquid fill volume and pressurized to 4.12 MPa.







Figure 121. Mean friction factor and Reynolds number as a function of time for a simulated transient discharge of HFC-227ea: 15.9 mm I.D. piping was specified, and the vessel was at 1/2 liquid fill volume and pressurized to 4.12 MPa.



Figure 122. Mean friction factor and Reynolds number as a function of time for a simulated transient discharge of HFC-227ea: 9.5 mm I.D. piping was specified, and the vessel was at 1/2 liquid fill volume and pressurized to 4.12 MPa.

For the contraction run, the pressure drop in the first 1.00 m of large piping was very small, therefore, the pipe volume was added to the valve and reducer volume which increased the effective bottle volume. For the expansion run, the flow was assumed to be choked at the expansion, therefore, the downstream pressure does not affect the mass flow rate. The calculations assume a 1.00 m length of small piping.

The effective length for the tee configuration was the sum of the effective length of the valve and reducer (0.63 m) plus the length of piping to the tee (2.1 m), plus the effective length of the tee element (0.9 m), plus the length of one leg to the recovery tanks (1.2 m).

The input requirements for a piping element are the effective length and the location of the piping element. The location of the piping element is important because the mass in the pipe is based on the sum of the mass in a number of small portions of the physical pipe. At a piping element with an effective pipe length, a step change in the density is required for the pipe mass calculation.

8.6.8 Model Calculations

8.6.8.1 Constant-Head Conditions. The constant-head tests were simulated by fixing the upstream conditions and calculating the steady mass flow rate and the pipe pressures at the tap locations. The effective length of the 15.9 mm diameter pipe was specified as 4.13 m, 3.50 meters of pipe from the valve reducer to the recovery tanks plus a 0.63 m effective length for the valve and valve reducer section (which essentially reproduces the experimental pressure drop from the bottle to the first pressure tap location). For the 9.5 mm pipe an effective length of 3.10 m was specified. There are 3.00 m of physical pipe length until the expansion to the larger pipe diameter which leads to the recovery tanks, and 0.10 m was chosen as an effective pipe length for the valve and reducer.

Table 35 compares the calculated steady mass flow rate with the experimentally determined average mass flow rate. Figure 123 is a plot of experimentally determined flow rates versus the calculated flow rates for all of the tests. The calculated results for the small diameter tests are within 10 % of the mean experimental value. For the large diameter tests, the experimental values are larger than the calculated values. Given the uncertainty in the experimental values and the probability that the calculated steady values should underestimate the large diameter values, the results compare favorably.

Calculated pipe pressures are shown in Figures 124-131. The pressure predictions show good agreement with the experimental pressures and illustrate the validity of the flow assumptions. Deviations from the experimental values may be due to metastable flow, heat transfer, and/or departures from the assumed homogeneous flow condition.

8.6.8.2 Transient Conditions. Each transient test was simulated by the model. Experimental fill conditions were used. It was assumed that the mass of dissolved nitrogen was the equilibrium amount. Calculations were performed for the cases with the bottle liquid contents in thermodynamic equilibrium and repeated with a specified degassing pressure for each agent at the given storage pressure.

Figures 132 and 133 are the simulated pressure traces for one test (halon 1301 at 1/2 liquid fill volume and 4.12 MPa storage pressure flowing through 3.5 m of straight piping). The results from Figure 132 are for equilibrium bottle conditions, while the results for Figure 133 are for "frozen" non-equilibrium bottle conditions until the critical degassing pressure (in this case 2.5 MPa) is reached. Figures 134 and 135 compare selected calculated pressure traces to experimental pressure traces. The





Figure 123. Constant-head mass flow rate. The dashed lines represent \pm 10 % deviation of predictions. Estimated measurement errors are indicated by error bars.

7

6

5







Figure 125. Pipe pressures for constant-head tests: halon 1301 and 4.13 m effective length of 15.9 mm I.D. piping. Symbols: calculations, and vertical lines with error bars: mean experimental pressures $\pm 2\sigma$ uncertainty.



















Figure 130. Pipe pressures for constant-head tests: CF_3I and 3.1 m effective length of 9.5 mm I.D. piping. Symbols: calculations, and vertical lines with error bars: mean experimental pressures $\pm 2\sigma$ uncertainty.



Figure 131. Pipe pressures for constant-head tests: CF_3I and 4.13 m effective length of 15.9 mm I.D. piping. Symbols: calculations, and vertical lines with error bars: mean experimental pressures $\pm 2\sigma$ uncertainty.


Figure 132. Simulated pressure traces for transient halon 1301 discharge: 1/2 liquid fill volume, pressurized to 4.12 MPa, and 3.5 m of 15.9 mm I.D. piping. Vessel liquid contents are assumed to be in equilibrium at all times.



Figure 133. Simulated pressure traces for transient halon 1301 discharge: 1/2 liquid fill volume, pressurized to 4.12 MPa, and 3.5 m of 15.9 mm I.D. piping. Vessel liquid contents are in a frozen, non-equilibrium state until degassing occurs.









non-equilibrium bottle condition calculations yield vessel and pipe pressure results that compare more favorably to experiments than do the equilibrium bottle condition calculations.

The model describes flow conditions which may be of interest. Figure 136 is a plot of the mean density of the fluid at the pipe exit (or at the liquid two-phase front before the pipe is filled). Figure 137 is the mass flow rate. Figure 138 is the exit velocity for the two-phase mixture. Figure 139 is the exit temperature of the two-phase mixture. Figure 140 is the exit quality of the two-phase mixture. Figure 141 is the mass fraction discharged from the pipe.

The results for the calculated and the experimental liquid discharge times are presented in Tables 36-38. Figures 142-145 show the experimental liquid discharge times versus the calculated discharge times. In these figures, the dashed lines represent \pm 10 % deviation from the experimental discharge times, and the error bars indicate 2σ uncertainty in the experimental values. The liquid flow time calculations based on equilibrium bottle conditions tend to be shorter than the calculations based on a fixed degassing pressure or no bottle degassing. For HFC-227ea and $CF_{3}I$, there appears to be a systematic trend for the cases with the small diameter piping. For most cases, the calculated liquid discharge time is less than 90 % of the experimental discharge time. These deviations may be due to the recovery tanks pressure build-up during those tests which decreases the flow rate below the choked condition for a period of time. Another systematic trend appears to be an overprediction in the liquid discharge time for the 2/3 fill, small piping cases. One possible explanation may be that significant heat transfer to the ullage contents may raise the pressure above the isentropic expansion condition. Since the ullage can expand by a factor of 3 before the liquid runs out, the adiabatic temperature of the ullage will be lower than the 1/2 fill case, which favors heat transfer from the vessel walls. The two-phase flow time is the longest for these tests which also favors heat transfer. The cold temperature discharge simulation of halon 1301 significantly underpredicts the liquid discharge time, which suggests that heat transfer from the pipe walls may be increasing the fluid expansion which slows the flow down.

There is no obvious reason why the model cannot be extended to higher bottle pressure and larger pipe diameter predictions. Testing at higher bottle pressures and larger pipe diameters would confirm the model's ability to predict liquid discharge times under those conditions and add confidence to extending the predictive capabilities.

8.6.9 Hardware Design Scheme. Since the model predicts the experimental discharge times within ± 10 %, it is assumed that the model can be used to calculate the liquid discharge times at higher pressures and larger pipe diameters, though it should be validated for those cases. A design exercise for hardware and fill specifications for each of the three alternative agents is detailed below. The exercise illustrates the model's potential usefulness in selecting hardware for a replacement agent.

Assume that a given halon fire suppression system is designed to protect an engine nacelle and meets the following specifications: 3.37 kg of halon 1301 superpressurized with nitrogen to 4.12 MPa at 23 °C in a 0.0042 m³ bottle, and 4 m of 0.016 m diameter piping from the bottle to the injection port. The calculated liquid discharge time is 1.0 s which meets the duration of discharge criterion in MIL-E-22285 section 3.9 which states "The period of time required to discharge the calculated amount of agent shall be one second or less, measured from the time the agent starts to leave the tubing ends until the required amount of agent has been discharged." The "calculated amount" means the mass stored in the liquid phase which is 90 % of the total agent mass in this case; the discharge time of the agent vapor in the ullage would take another second for this configuration. For a given alternative agent, the mass of agent, bottle size, storage pressure, fill condition, and pipe diameter that fulfills the discharge requirement of a one second liquid discharge time needs to be determined.

Realizing that a number of conditions may meet the discharge requirement, an "optimized" system requires knowledge of the economics of specific configurations and the space limitations for a given



Figure 136. Calculated mean density of the two-phase fluid at the pipe exit (or at the two-phase front if it has not reached the pipe exit): halon 1301, 1/2 liquid fill volume, pressurized to 4.12 MPa, and 3.5 m of straight 15.9 mm I.D. piping.



Figure 137. Calculated mass flow rate of the two-phase fluid: halon 1301, 1/2 liquid fill volume, pressurized to 4.12 MPa, and 3.5 m of straight 15.9 mm I.D. piping.



Figure 138. Calculated exit velocity of the two-phase fluid: halon 1301, 1/2 liquid fill volume, pressurized to 4.12 MPa, and 3.5 m of straight 15.9 mm I.D. piping.







Figure 140. Calculated exit flow quality of the two-phase fluid: halon 1301, 1/2 liquid fill volume, pressurized to 4.12 MPa, and 3.5 m of straight 15.9 mm I.D. piping.







Figure 142. Experimental and calculated liquid discharge times for halon 1301. Calculations assuming equilibrium and non-equilibrium bottle conditions.



Figure 143. Experimental and calculated liquid discharge times for HFC-125. Calculations assuming equilibrium and non-equilibrium bottle conditions.



Figure 144. Experimental and calculated liquid discharge times for HFC-227ea. Calculations assuming equilibrium and non-equilibrium bottle conditions.





Agent	Mean volume factor	Liquid density at 25 °C (kg/m ³)	Equivalent mass stored in liquid phase (kg)		
Halon 1301	1	1538	3.02		
HFC-125	3.0	1190	7.02		
HFC-227ea	2.5	1395	6.86		
CF ₃ I	0.8	2016	3.31		

Table 41. Agent mass requirements for a retro-fit system

bottle location. Such a design optimization is well beyond the scope of this study, but for the sake of this exercise, the following considerations are probably close to an actual optimal design scheme. First, assume that the preferred case would be if the original bottle size and pressure could be used and that the piping remains the same for an alternative agent. This would represent a "drop-in" replacement for halon 1301. The next best case would be if the only change was an increase in the storage pressure. This implies that the bottle can withstand the elevated pressure. If the discharge time cannot be achieved with the original bottle size, a larger bottle size and increased storage pressure then is checked. Finally, the piping can be replaced with larger diameter piping to achieve the desired discharge time.

The pressure range searched was from 4.12 MPa to 8.24 MPa in increments of 1.37 MPa. Three liquid fill volumes where included; 1/3, 1/2, and 2/3 filled, as were piping diameters of 15.9 mm and 25.4 mm. The calculations assume equilibrium bottle conditions. The specific amount of each alternative agent equivalent in suppression capability to halon 1301 for engine nacelle protection was estimated from the results reported in NIST SP 861 (Grosshandler *et al.*, 1994) on the volume factor. The volume factor (VF) is a measure of the liquid volume of the agent in the storage vessel normalized by the liquid volume of halon 1301 required to suppress a flame under identical conditions. The average value of the volume factors for the opposed flow diffusion burner, cup burner, spray burner, and deflagration tube was used. Table 41 shows the average volume factor and the required amount of agent for equivalent protection. A major assumption here is that this average volume factor prescribes the proper amount of alternative agent for the engine nacelle case. This must be validated by actual engine nacelle testing, or a more appropriate relationship should be used (see Section 9).

From the replacement agent liquid densities and the mass in the liquid phase required, only CF_3I has a chance to fit in the original bottle and therefore is a candidate for "drop-in" replacement. From PROFISSY, 3.5 kg of CF_3I in a 0.0042 m³ liter vessel pressurized to 4.12 MPa will yield 3.34 kg of agent in the liquid phase with a liquid fill volume of 40.1 % at 23 °C. The liquid discharge time through 4 m of 15.9 mm diameter piping is calculated to be 0.71 s. So, CF_3I can be used as a direct replacement of halon 1301. No more calculations with CF_3I were performed.

HFC-125 and HFC-227ea will require a larger bottle size than the 0.0042 m³ halon bottle. Since the search includes 3 liquid fill volumes and 4 pressures, a total of 12 bottle conditions were examined. The exact bottle size is fixed by the liquid fill volume and pressure for each of the agents. If bottle sizes are limited to off-the-shelf items, then a search of bottle size and pressure could be performed. The bottle size and the total agent mass tends to increase as the pressure increases for a fixed liquid fill volume. Tables 42 and 43 give the fill conditions for the various pressures and liquid fill volumes and the calculated liquid discharge times for the simulations with the two pipe diameters.

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Storage pressure	Total mass of agent	Mass of agent inFill volumeStorage vesselliquid phase (kg)(%)volume (m³)	Fill volume	Storage vessel	Liquid discharge time (s)		
(MPa)	(kg)		(m ³)	pipe I.D. 15.9 mm	pipe I.D. 25.4 mm		
4.12	8.60	7.10	32.5 (1/3)	0.0200	2.00	0.75	
4.12	7.80	7.08	49.9 (1/2)	0.0130	2.37	0.90	
4.12	7.30	6.95	67.1 (2/3)	0.0095	2.86	1.10	
5.49	8.75	6.97	33.0 (1/3)	0.0205	1.66	0.62	
5.49	7.85	6.98	50.3 (1/2)	0.0135	1.98	0.74	
5.49	7.40	6.98	67.9 (2/3)	0.0100	2.46	0.94	
6.87	9.30	7.05	33.3 (1/3)	0.0220	1.52	0.56	
6.87	8.20	7.02	48.7 (1/2)	0.0150	1.75	0.66	
6.87	7.60	7.04	66.5 (2/3)	0.0110	2.18	0.82	
8.24	9.95	7.01	32.9 (1/3)	0.0240	1.40	0.52	
8.24	8.50	7.02	49.4 (1/2)	0.0160	1.65	0.61	
8.24	7.80	7.06	66.3 (2/3)	0.0120	2.03	0.77	

 Table 42.
 Fill conditions and liquid discharge times for HFC-125

The liquid discharge times are also plotted against bottle pressure for the different fill conditions in Figures 146 and 147. A lower liquid fill volume and higher bottle pressure decrease the liquid discharge time for a given pipe diameter. The rate of decrease in the liquid discharge time slows as the bottle pressure is increased at any one fill condition.

The liquid discharge times for HFC-125 through the 15.9 mm diameter piping are not under one second for any fill condition and pressure, which necessitates an increase in piping diameter. At a piping diameter of 25.4 mm, all liquid discharges are under 1 s except for the highest liquid fill volume and lowest bottle pressure (2/3 fill and 4.12 MPa). For HFC-227ea, the liquid discharge times through the 15.9 mm diameter piping also are not under 1 s, but at the lowest liquid fill volume and highest pressure, (1/3 filled and 8.24 MPa) the discharge time is nearly 1 s, which might warrant a closer look. The penalty for the high pressure and low liquid fill volume is a relatively large bottle volume. Discharge times through the 25.4 mm diameter piping are all below 1 s for HFC-227ea.

The above exercise illustrates the vastly different hardware requirements for HFC-125 and HFC-227ea versus halon 1301 and CF_3I . The results are indicative only of this particular case; other cases require separate study, hence the importance of a general calculation scheme. Once the particular alternative agent is selected and the design agent mass specified (using the volume factor approach, or other suitable approach), this design scheme can be repeated for any number of retro-fits or new systems.

8.6.10 Conclusions and Recommendations. The experimental study and modeling effort of the transient discharge of agent through piping presented here should assist in the design of engine nacelle

Storage pressure	orage Total mass Mass of Fill Storessure of agent agent in volume very (kg) liquid (%) vo phase (kg) (hg)	Mass of agent in	Fill volume	Storage vessel	Liquid discharge time (s)		
(MPa)		(m ³)	pipe I.D. 15.9 mm	pipe I.D. 25.4 mm			
4.12	7.40	6.88	33.9 (1/3)	0.0150	1.46	0.56	
4.12	7.15	6.88	50.0 (1/2)	0.0105	1.73	0.67	
4.12	7.00	6.86	66.3 (2/3)	0.0079	2.13	0.84	
5.49	7.50	6.87	33.6 (1/3)	0.0160	1.27	0.48	
5.49	7.25	6.92	49.2 (1/2)	0.0110	1.48	0.57	
5.49	7.05	6.89	66.6 (2/3)	0.0081	1.84	0.72	
6.87	7.70	6.90	32.7 (1/3)	0.0170	1.14	0.43	
6.87	7.30	6.90	49.7 (1/2)	0.0112	1.33	0.51	
6.87	7.10	6.91	67.0 (2/3)	0.0083	1.66	0.65	
8.24	7.85	6.88	32.9 (1/3)	0.0175	1.07	0.40	
8.24	7.35	6.88	50.0 (1/2)	0.0115	1.25	0.48	
8.24	7.10	6.87	67.6 (2/3)	0.0085	1.55	0.60	

 Table 43.
 Fill conditions and liquid discharge times for HFC-227ea

fire protection hardware systems which specify a replacement candidate. The observed flow characteristics for all of the replacement candidates are similar to halon 1301. Non-linear pressure drop versus pipe length increment indicates the high compressibility of the flows, while high speed movies of transparent piping sections validate the two-phase nature of the flow after it leaves the storage bottle. Discharge times for the initially liquid phase in the bottle for fixed fill volume, pressure, and piping condition did not vary widely for the three alternative agents and halon 1301. The amount of mass discharged did vary from agent to agent due mostly to the density differences. Liquid flow times from the small pipe diameters were approximately a factor of 2.5 greater than those for the large pipe diameter. There was significant scatter in the observed bottle pressure where the pressure recovery occurred (the lowest bottle pressure achieved before it began to rise) for any given agent. In most cases, model calculations of the steady-state mass flow rates and transient discharge times are within ± 10 % of the experimental values, suggesting that the model can be used as a design tool for hardware selection of an alternative agent fire suppression system. A hardware design scheme for retro-fitting a halon 1301 system illustrated the model's utility.

Some areas of further research for transient discharge would include the behavior of a squib activated discharge, its interaction with nitrogen degassing, and the degassing phenomena in general. Discharges at elevated bottle pressure and larger piping diameters should be performed to validate the model for those conditions. Unbalanced flow networks should be studied if they are important for engine nacelle application.



Figure 146. Calculated liquid discharge times for HFC-125 for a various liquid fill volumes and storage pressures. The mass in the liquid-phase was held constant. Calculations assume equilibrium bottle conditions.



Figure 147. Calculated liquid discharge times for HFC-227ea for a various liquid fill volumes and storage pressures. The mass in the liquid-phase was held constant. Calculations assume equilibrium bottle conditions.

8.7 Concluding Remarks

Several important aspects related to agent storage and its subsequent discharge have been studied in detail. They are summarized in the following.

- 1. A computer program (PROFISSY) for calculating binary vapor-liquid equilibrium has been developed to aid the bottle designers by providing useful and reliable data on the pressure-temperature characteristics of the selected agent/nitrogen mixtures.
- 2. The dissolved nitrogen in the liquid agent was found to play a very important role in the discharge whether in a dry bay or in a piping system.
- 3. In simulated dry bay discharges, CF₃I has the fastest volumetric discharge rate but the slowest dispersion velocities.
- 4. A discharge model which incorporates nitrogen degassing has also been developed. The model predicts agent discharge times within a factor of four (usually much better) without any *a priori* information other than the initial conditions of the vessel.
- 5. The exploratory study using complex computer codes (*e.g.*, KIVA-II) has identified the specification of the initial spray condition and the rapid evaporation of the droplets as problem areas. Methods need to be developed further to handle the highly transient flashing process in a computationally efficient manner before the detailed velocity, temperature, and concentration field predictions which have been made can be properly interpreted.
- 6. A computer model based on a homogeneous two-phase flow has also been developed to facilitate transient pipe-flow calculations for engine nacelle applications. The model has been validated by the experimental results. A design exercise was given to illustrate the usefulness of the model in system hardware selection for a halon alterative.

Figure 148 typifies a generic flow chart for discharge system optimization. To optimize a new discharge system, system operational requirements (*e.g.*, amount of agent required to suppress a fire) and constraints (*e.g.*, vessel growth potential, operating temperature) have to be identified or provided to the system designers. Once these constraints are known, the designers can then proceed to size the bottle using thermodynamic data of the agent (*e.g.*, PROFISSY). After the bottle size and the allowable nitrogen pressurization have been determined, the discharge time of the agent can be estimated by using a discharge model (*e.g.*, the one developed above) for dry bay applications. For engine nacelle fire protection, the pipe-flow model discussed above can be used to size the pipe and to predict agent discharge time. These calculation processes may have to be iterated several times until all the calculated parameters meet the system constraints and operational requirements.



Figure 148. A generic flow chart for discharge system optimization.

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Appendix A. A User's Guide to PROFISSY

There are two files (PROFISSY.EXE and PROFLIB) in the disc. The current version of PROFISSY is Version 0.08. In the following, the **bold** type style represents any text one must type. Sentences in *italic* represent the instructions given by the computer. To start a run, simply type **PROFISSY**, followed by hitting the <enter> key, from the drive or directory where the program is installed. The screen will appear as follows.



Type y and hit <enter> if you want to change the default units. The Property Unit Menu will appear on the screen.

			Property U	nit Menu		• •	
Property	******	******	*********opti	ons*********	*****	current	
Temperature	(K	,R	,C	,F	,	,)-	F
Pressure	(atm	,bar	,MPa	,kpf	,Psia	,mmHG)-	Psia
Volume	(liter	,m**3	,cm**3	,in**3	,ft**3	,)-	ft**3
Energy	(cal	,J	,btu	,kcal	,kJ	,)-	btu
Mass	(mol	,lb-mol	,kg	,g	,lb	,)-	lb
Velocity	(m/s	,cm/s	,ft/s	,in/s	, '	,)-	ft/s
Viscosity	(uP	,cP	,uPa.s	,lb/ft.s	,lb/ft.h	,)-	lb/ft.s
Thm. Cond.	W/m.K	,cal/cm.s.K	,btu/ft.s.F	,btu/ft.h.F	,mW/m.K	,)-	btu/ft.h.F
default set options							
(1) Scientific	(K ,atm	,liter ,cal ,	,mol ,cm/s	,uP ,cal	l/cm.s.K)		
(2) S.I.	(K ,MPa	,m**3 ,kJ	,kg ,m/s	,uPa.s	,W/m.K)		
(3) Engineering	(F ,Psia	,ft**3 ,btu	,lb ,ft/s	,lb/ft.s ,btu/ft.	.h.F)		
(4) Mixed	(K ,bar	,liter ,kJ ,1	mol ,m/s	,uP ,mW	√/m.K)		
Enter the new unit or	default optic	on (X to exit) ?				·	

Type **C,MPa,cm**3,g,x** and hit <enter> if you want temperature in C, pressure in MPa, volume in cm^3 , and mass in g. The screen will appear as follows.

Please input the fire suppression agent. Choose from the following list:	
Name	Synonym
CF3I R218 R125 R227ea R13b1	trifluoroiodomethane perfluoropropane pentafluoroethane heptafluoropropane trifluorobromomethane

Type **cf3i** and hit <enter> if you want to run CF_3I . Follow the instructions given by the computer to input the data.

Input the amount (in g) of fire suppression agent.700 <enter>

Input the container volume (in cm**3)500 <enter>

Do you want to input system pressure (P) or N2 mass (M) $?\mathbf{p} < \text{enter} >$

Input the pressure (in MPa)4.12 <enter>

Input the fill temperature (in C)22 <enter>

Do you want to specify the temperature limits for the table? (Y,N) If NO, limits will be generated automatically y < enter>

If the input is **n**, the computer will generate the table from -60 to 150 °C with 10 °C increment.

Input the Tmin, Tmax, and T increment (in C)20,40,10 <enter>

The computer will response: please wait, computations in progress

After the computation, the screen will appear as follows.

CF3I with nitrogen su Pressure - Temperatu	uperpressure re Characteristics			
Agent mass	700.000(g)			
Nitrogen Mass	15.522(g)			
Container Volume	500.000(cm**3)			
Fill Temperature	22.000(C)	Fill Pressure	4.120(MPa)	
Bubble point temp.	92.006(C)	Bubble point pres.	6.784(MPa)	
Press Enter to Contin	iue.			

After hit <enter>, the computer will list the calculated results on the screen. The calculated bubble point pressure and temperature are obtained at the fill conditions.

Temp. (C)	Vol. Pct. Liq. Fill	Mass Liquid CF3I (g)	Nitr xN2	ogen wN2	Mass liquid	N2 (g) vapor	Pressure MPa	
20.0	68.99	0.6910E+03	0.085	0.013	0.9129E+01	0.6393E+01		4.1
30.0	70.83	0.6890E+03	0.088	0.014	0.9558E+01	0.5964E+01		4.3
40.0	72.90	0.6869E+03	0.093	0.014	0.1005E+02	0.5473E+01		4.6
Do you want another run? (Yes,No)								

If you do not want another run, type n and press <enter>.

Goodbye !

"Vol. Pct. Liq. Fill" represents the volume percent of liquid fill in the container. "xN2" and "wN2" represent the mole and mass fractions of nitrogen in the liquid mixture respectively.

Compilation of Results. For quick references, three cases with initial *pure* (before nitrogen pressurization) liquid agent volume fractions of 1/3, 1/2, and 2/3 (see Section 8.3.2.1) have been run for all the agents which the computer code currently supports and are summarized in Figures A-1 to A-15. In the calculations, the initial fill temperature is 22 °C and the three initial fill pressures are 2.75 MPa, 4.12 MPa, and 5.50 MPa.







Figure A-2.

Calculated pressure as a function of temperature for HFC-227ea/N₂ mixture under three initial fill conditions and $P_i = 4.12$ MPa.



Figure A-3.

Calculated pressure as a function of temperature for HFC-227ea/N₂ mixture under three initial fill conditions and $P_i = 5.50$ MPa.



Figure A-4.

Calculated pressure as a function of temperature for CF_3I/N_2 mixture under three initial fill conditions and $P_i = 2.75$ MPa.



Figure A-5.

Calculated pressure as a function of temperature for CF_3I/N_2 mixture under three initial fill conditions and $P_i = 4.12$ MPa.



Figure A-6.

Calculated pressure as a function of temperature for CF_3I/N_2 mixture under three initial fill conditions and $P_i = 5.50$ MPa.


Figure A-7.

Calculated pressure as a function of temperature for FC-218/N₂ mixture under three initial fill conditions and $P_i = 2.75$ MPa.



Figure A-8.

Calculated pressure as a function of temperature for FC-218/N₂ mixture under three initial fill conditions and $P_i = 4.12$ MPa.



Figure A-9.

Calculated pressure as a function of temperature for FC-218/N₂ mixture under three initial fill conditions and $P_i = 5.50$ MPa.



Figure A-10.

Calculated pressure as a function of temperature for HFC-125/N₂ mixture under three initial fill conditions and $P_i = 2.75$ MPa.



Figure A-11.

Calculate pressure as a function of temperature for HFC-125/N₂ mixture under three initial fill conditions and $P_i = 4.12$ MPa.



Figure A-12.

Calculated pressure as a function of temperature for HFC-125/N₂ mixture under three initial fill conditions and $P_i = 5.50$ MPa.



Figure A-13.

Calculated pressure as a function of temperature for CF_3Br/N_2 mixture under three initial fill conditions and $P_i = 2.75$ MPa.



Figure A-14.

Calculated pressure as a function of temperature for CF_3Br/N_2 mixture under three initial fill conditions and $P_i = 4.12$ MPa.



Figure A-15.

Calculated pressure as a function of temperature for CF_3Br/N_2 mixture under three initial fill conditions and $P_i = 5.50$ MPa.

Appendix B. Tabulated fill Conditions and Pressure Traces for All Pipe Flow Discharge Tests

This Appendix lists the experimental conditions and results for all the pipe flow tests performed in this study. Table B-1 tabulates the temperatures and fill conditions for hot and cold agent discharge tests. Tables B-2 to B-5 tabulate the experimental conditions for halon 1301, HFC-125, HFC-227ea, and $CF_{3}I$, respectively. The fill voulmes were estimated by the computer code, PROFISSY. The experimental results for halon 1301, HFC-125, HFC-227ea, and $CF_{3}I$ are shown in Figures B-1 to B-25, Figures B-26 to B-50, Figures B-51 to B-76, and Figures B-77 to B-99, respectively. The individual pressure traces are not labeled on each figure, but follow the logical sequence of the vessel pressure being the highest, then followed in descending order by the pipe pressure traces for locations successively farther downstream. Finally, the pressure trace of the recover tank is the lowest. In some tests, not all pressures at each tap location were recorded. In those cases, the location of the tap where data was not taken is noted.

Agent	Figure number	Initial fill volume (%)	Liquid temp. (°C)	Gas temp. (°C)	Final pressure (± 0.05 MPa)	Equilibirum fill volume (%)	Equilibrium pressure (MPa)
CF ₃ Br	B-24	48.9	-50	4	2.20	41.8	2.0
	B-25	49.1	70	66	8.00	Single phase	7.4
HFC-125	B-49	46.4	-44	4	2.55	39.1	2.3
	B-50	49.3	68	73	7.40	Single phase	7.4
HFC-227ea	B-74	49.2	-45	4	3.00	43.5	2.8
	B-75	48.8	68	72	6.00	58.4	5.5
	B-76	49.0	70	68	5.95	59.8	5.6
CF ₃ I	B-98	50.4	-46	4	3.05	44.2	2.7
	B-99	49.7	70	70	5.70	56.7	5.7

Table B-1	Temperature	and fill	conditions	for <i>k</i>	not and	cold	agent	discharge	tests
	TOUDDIAGU		VOIGIUUU	101 1	101 0010	0010			

Storage pressure (± 0.005 MPa)	Agent mass (± 0.01 kg)	Fill volume (%)	Diameter (mm)	Configuration	Figure number
2.00	3.30	N.A.	9.5	Straight pipe	B-1
2.50	3.11	N.A.	9.5	п	B-2
3.00	3.40	N.A.	9.5	"	B-3
2.77	3.26	46.6	9.5	"	B-4
4.12	2.88	40.8	9.5		B-5
4.15	3.00	43.1	9.5		B-6
4.12	3.15	45.9	9.5	н	B-7
4.12	3.39	50.4	9.5	u	B-8
4.12	4.21	65.7	9.5	31	B-9
4.12	3.31	48.9	9.5	3 m, 90° bend	B-10
4.12	3.32	49.1	9.5	4 m, 90° bend	B-11
4.12	3.28	48.3	15.9/9.5	Contraction	B-12
4.12	3.28	48.3	9.5/15.9	Expansion	B-13
2.00	4.09	N.A.	15.9	Straight pipe	B-14
2.50	3.83	N.A.	15.9	.**	B-15
3.00	4.43	N.A.	15.9	"	B-16
2.75	3.52	51.1	15.9	"	B-17
4.12	3.25	47.8	15.9	. 11	B-18
4.12	3.35	49.6	15.9	"	B-19
4.12	4.20	65.5	15.9	"	B-20
4.12	3.30	48.7	15.9	3 m, 90° bend	B-21
4.12	3.30	48.7	15.9	4 m, 90° bend	B-22
4.12	3.26	47.9	15.9	"T" flow split	B-23
4.12	3.31	48.9	15.9	Cold	B-24
4.12	3.32	49.1	15.9	Hot	B-25

Table B-2. Test fill conditions for halon 1301

Storage pressure (± 0.005 MPa)	Agent mass (± 0.01 kg)	Fill volume (%)	Diameter (mm)	Configuration	Figure number
2.00	2.51	N.A.	9.5	Straight pipe	B-26
2.50	2.92	N.A.	9.5	н	B-27
3.00	2.53	N.A.	9.5		B-28
4.12	3.69	N.A.	9.5		B-29
2.76	2.62	50.4	9.5	10	B-30
4.13	2.78	56.4	9.5	. 11	B-31
4.16	3.33	69.9	9.5		B-32
4.12	2.40	47.1	9.5	3 m, 90° bend	B-33
4.12	2.44	48.1	9.5	4 m, 90° bend	B-34
4.12	2.50	49.6	15.9/9.5	Contraction	B-35
4.12	2.51	49.8	9.5/15.9	Expansion	B-36
2.00	2.72	N.A.	15.9	Straight pipe	B-37
2.55	3.29	N.A.	15.9	••	B-38
3.00	2.81	N.A.	15.9	. *	B-39
2.75	2.79	54.2	15.9	11	B-40
4.12	2.50	49.6	15.9	Π.	B-41
4.12	2.63	52.7	15.9	11 -	B-42
4.12	2.64	53.0	15,9		<u>B-43</u>
4.12	3.17	65.9	15.9	11	<u>B-44</u>
4.14	3.82	81.8	15.9	11 U	<u>B-45</u>
4.12	2.47	48.8	15.9	3 m, 90° bend	B-46
4.12	2.38	46.6	15.9	4 m, 90° bend	B-47
4.12	2.70	54.4	15.9	"T" flow split	B-48
4.12	2.37	46.4	15.9	Cold	B-49
4.12	2.49	49.3	15.9	Hot	B-50

Table	B-3	Test fill	conditions	for halon	HFC-1	25
raute	D -J.	I COL IIII	COnditions	ion nuton		

Storage pressure (± 0.005 MPa)	Agent mass (± 0.01 kg)	Fill volume (%)	Diameter (mm)	Configuration	Figure number
2.00	3.31	N.A.	9.5	Straight pipe	B-51
2.50	3.18	N.A.	9.5	u	B-52
3.00	3.24	N.A.	9.5	"	B-53
2.75	3.26	56.5	9.5	"	B-54
4.12	3.05	53.7	9.5	11	B-55
4.12	3.36	60.2	9.5	u	B-56
4.12	3.91	70.0	9.5	t1	B-57
4.12	2.81	49.2	9.5	3 m, 90° bend	B-58
4.12	2.81	49.2	9.5	4 m, 90° bend	B-59
4.12	2.80	49.0	15.9/9.5	Contraction	B-60
4.12	2.80	49.0	9.5/15.9	Expansion	B-61
2.00	3.74	N.A.	15.9	Straight pipe	B-62
2.50	2.73	N.A.	15.9	"	B-63
2.50	3.65	N.A.	15.9	11	B-64
3.00	3.56	N.A.	15.9		B-65
2.75	3.13	54.1	15.9	**	B-66
4.12	2.77	48.4	15.9	н	B-67
4.12	2.82	49.4	15.9	n	B-68
4.12	3.23	57.1	15.9	n .	B-69
4.12	3.88	69.4	15.9	11	B-70
4.12	2.82	49.4	15.9	3 m, 90° bend	B-71
4.12	2.79	48.8	15.9	4 m, 90° bend	B-72
4,12	2.77	48.4	15.9	"T" flow split	B-73
4.12	2.81	49.2	15.9	Cold	B-74
4.12	2.79	48.8	15.9	Hot	B-75
4.12	2.80	49.0	15.9	Hot	B-76

Table B-4. Test fill conditions for halon HFC-227ea

Table B	-5. T	est fill	conditions	for	CF ₁

Storage pressure (± 0.005 MPa)	Agent mass (± 0.01 kg)	Fill volume (%)	Diameter (mm)	Configuration	Figure number
2.00	5.40	N.A.	9.5	Straight pipe	B-77
2.50	5.58	N.A.	9.5	"	B-78
3.00	5.22	N.A.	9.5	u	B-79
2.75	4.34	49.7	9.5	#	B-80
4.13	4.31	50.2	9.5	u	B-81
4.12	5.57	65.8	9.5	11	B-82
4.12	5.74	67.9	9.5	17	B-83
4.12	4.46	52.0	9.5	3 m, 90° bend	B-84
4.12	4.39	51.2	9.5	4 m, 90° bend	B-85
4.12	4.49	52.4	15.9/9.5	Contraction	B-86
4.12	4.60	53.8	9.5/15.9	Expansion	B-87
2.00	4.03	N.A.	15.9	Straight pipe	B-88
2.50	4.21	N.A.	15.9	На п	B-89
2.50	6.27	N.A.	15.9	11	B-90
3.00	4.32	N.A.	15.9		B-91
2.80	4.33	49.6	15.9		B-92
4.13	4.19	48.7	15.9	**	B-93
4.20	5.76	68.2	15.9	n	B-94
4.17	4.24	49.4	15.9	3 m, 90° bend	B-95
4.14	4.71	55.2	15.9	4 m, 90° bend	B-96
4.12	4.29	49.9	15.9	"T" flow split	B-97
4.19	4.32	50.4	15.9	Cold	B-98
4.12	4.27	49.7	15.9	Hot	B-99





Figure B-2. Halon 1301 test. Refer to Table B-2 for experimental conditions.





Figure B-4. Halon 1301 test. Refer to Table B-2 for experimental conditions.



Figure B-5. Halon 1301 test. Refer to Table B-2 for experimental conditions.





Figure B-7. Halon 1301 test. Refer to Table B-2 for experimental conditions.







Figure B-9. Halon 1301 test. Refer to Table B-2 for experimental conditions.



683



Halon 1301 test. Refer to Table B-2 for experimental conditions.

Figure B-12.



685



Figure B-13.



Halon 1301 test. Refer to Table B-2 for experimental conditions.

687



Halon 1301 test. Refer to Table B-2 for experimental conditions.



Figure B-16.









Figure B-20.



Figure B-21.


Figure B-22. Halon 1301 test. Refer to Table B-2 for experimental conditions.



Figure B-23. Halon 1301 test. Refer to Table B-2 for experimental conditions.



Halon 1301 test. Refer to Table B-2 for experimental conditions.



Figure B-25. Halon 1301 test. Refer to Table B-2 for experimental conditions.



Figure B-26.



Figure B-27.







HFC-125 test. Refer to Table B-3 for experimental conditions.

703



Figure B-31.

HFC-125 test. Refer to Table B-3 for experimental conditions.



HFC-125 test. Refer to Table B-3 for experimental conditions.

705





HFC-125 test. Refer to Table B-3 for experimental conditions.

707





Figure B-36.



Figure B-37.





Figure B-39.

HFC-125 test. Refer to Table B-3 for experimental conditions.



Figure B-40.





Figure B-42.



Figure B-43.







Figure B-46.





Figure B-48.

HFC-125 test. Refer to Table B-3 for experimental conditions.

721







Figure B-51. HFC-227ea test. Refer to Table B-4 for experimental conditions.



Figure B-52. HFC-227ea test. Refer to Table B-4 for experimental conditions.



Figure B-53. HFC-227ea test. Refer to Table B-4 for experimental conditions.



Figure B-54. HFC-227ea test. Refer to Table B-4 for experimental conditions.



Figure B-55.





HFC-227ea test. Refer to Table B-4 for experimental conditions.

Figure B-57.




Figure B-59.





Figure B-61. HFC-227ea test. Refer to Table B-4 for experimental conditions.

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Figure B-63.



Figure B-64.



Figure B-65.





Figure B-67. HFC-227ea test. Refer to Table B-4 for experimental conditions.





Figure B-69.





Figure B-71.

HFC-227ea test. Refer to Table B-4 for experimental conditions.



Figure B-72.





Figure B-74.



Figure B-75.

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Figure B-76. HFC-227ea test. Refer to Table B-4 for experimental conditions.

749











Figure B-81. $CF_{3}I$ test. Refer to Table B-5 for experimental conditions.



Figure B-82. CF_3I test. Refer to Table B-5 for experimental conditions.



Figure B-83. $CF_{3}I$ test. Refer to Table B-5 for experimental conditions.



 CF_3I test. Refer to Table B-5 for experimental conditions.

Figure B-84.



Figure B-85. CF₃I test. Refer



Figure B-86.



Figure B-87.





8. OPTIMIZATION OF SYSTEM DISCHARGE





Figure B-91.



Figure B-92.




CF₃I test. Refer to Table B-5 for experimental conditions.

767



Figure B-95.

 CF_3I test. Refer to Table B-5 for experimental conditions.

8. OPTIMIZATION OF SYSTEM DISCHARGE



 CF_3I test. Refer to Table B-5 for experimental conditions.



CF₃I test. Refer to Table B-5 for experimental conditions.



CF₃I test. Refer to Table B-5 for experimental conditions.



 CF_3I test. Refer to Table B-5 for experimental conditions.

Appendix C. Thermodynamic and Transport Properties of Agent/Nitrogen Mixtures

Coefficients to polynomial fits of calculated thermodynamic and transport properties of agent/nitrogen mixtures during isenthalpic expansion are tabulated in Tables C-1 and C-2. Figures C-1 to C-6 show the polynomial fits to the density, temperature and mass fraction results at initial pressures of 2.75 and 4.12 MPa. Figure C-7 shows the pressure/density relationships for isentropic expansion of gas-phase agent/nitrogen mixture initially at 20 °C. The curves through the data represent power-law fits to the points. In general, the exponent was not a function of initial pressure, thus the pseudo-ideal γ was taken as a constant for each agent. The values obtained for halon 1301, HFC-125, HFC-227ea, and CF₃I were 1.07, 1.12, 1.01, and 0.99 respectively.

Agent	P _{initial} (MPa)	$\rho_{\text{mean}} (\text{kg/m}^3) = a_0 + a_1 * P + a_2 * P^2 + a_3 * P^3$ $P \text{ in kPa}$				
		. a ₀	a ₁	a ₂	a ₃	
CF ₃ Br	2.00	-7.815	231.5	-46.81	160.8	
	2.50	14.77	79.10	184.6	11.39	
	2.75	18.49	66.39	189.5	-5.087	
	3.00	16.36	68.25	180.4	-12.23	
	4.12	3.334	118.4	81.99	-5.742	
HFC-125	2.00	15.55	29.97	185.7	47.50	
	2.50	17.09	5.372	226.7	-17.28	
	2.75	14.32	21.18	198.0	-19.27	
	3.00	9.038	47.35	153.9	-14.11	
	4.12	3.586	76.32	81.13	-8.235	
	5.49	-0.9375	88.697	40.26	-3.845	
	6.87	-5.628	92.75	18.41	-1.511	
	8.24	-7.401	85.04	7.405	-0.5212	
HFC-227ea	2.00	-46.28	375.9	478.8	-159.2	
	2.50	-50.25	407.1	222.5	-65.64	
	2.75	-49.05	402.7	174.3	-50.13	
	3.00	-56.03	418.8	98.02	-28.24	
	4.12	-50.43	362.2	28.29	-8.473	
	5.49	-44.59	297.1	3.781	-2.334	
	6.87	-35.94	241.5	-1.419	-0.8257	
	8.24	-30.56	193.6	-1.828	-0.3262	
CF ₃ I	2.00	-52.59	467.7	893.7	-303.6	
	2.50	-73.51	613.6	402.7	-124.6	
	2.75	-80.37	648.9	269.3	-82.36	
	3.00	-98.79	690.8	151.0	-49.62	
	4.12	-87.52	607.0	22.44	-11.49	

Table C-1. Mean agent/nitrogen mixture densities

8. OPTIMIZATION OF SYSTEM DISCHARGE

Agent	P _{initial} (MPa) @ 20 °C	$\mu_{\text{mean}} (g/\text{cm s}) = a_0 + a_1^* P + a_2^* P^2 + a_3^* P^3$ P in MPa				
		a ₀	a ₁	a ₂	a ₃	
Halon 1301	2.75	1.456 x 10 ⁻⁴	7.500 x 10 ⁻⁵	1.461 x 10 ⁻⁴	6.043 x 10 ⁻⁶	
	4.12	1.294 x 10 ⁻⁴	1.192 x 10 ⁻⁴	5.682 x 10 ⁻⁵	6.407 x 10 ⁻⁸	
HFC-125	2.75	1.238 x 10 ⁻⁴	6.513 x 10 ⁻⁵	1.865 x 10 ⁻⁴	-1.171 x 10 ⁻⁵	
	4.12	1.111 x 10 ⁻⁴	1.194 x 10 ⁻⁴	6.457 x 10 ⁻⁵	-3.316 x 10 ⁻⁶	
HFC-227ea	2.75	2.783 x 10 ⁻⁵	7.386 x 10 ⁻⁴	2.947 x 10 ⁻⁴	-8.376 x 10 ⁻⁵	
	4.12	3.707 x 10 ⁻⁵	6.770 x 10 ⁻⁴	4.709 x 10 ⁻⁵	-1.312×10^{-5}	

Table C-2. Mean agent/nitrogen mixture viscosities



Figure C-1. Mean density as a function of pressure for initially liquid-phase agent/nitrogen mixture undergoing isenthalpic expansion. Initial pressure and temperature are 2.75 MPa and 20 °C.



Figure C-2. Temperature as a function of pressure for initially liquid-phase agent/nitrogen mixture undergoing isenthalpic expansion. Initial pressure and temperature are 2.75 MPa and 20 °C.



Figure C-3. Liquid-phase mass fraction as a function of pressure for initially liquid-phase agent/nitrogen mixture undergoing isenthalpic expansion. Initial pressure and temperature are 2.75 MPa and 20 °C.



Figure C-4. Mean density as a function of pressure for initially liquid-phase agent/nitrogen mixture undergoing isenthalpic expansion. Initial pressure and temperature are 4.12 MPa and 20 °C.



Figure C-5. Temperature as a function of pressure for initially liquid-phase agent/nitrogen mixture undergoing isenthalpic expansion. Initial pressure and temperature are 4.12 MPa and 20 °C.



Figure C-6. Liquid-phase mass fraction as a function of pressure for initially liquid-phase agent/nitrogen mixture undergoing isenthalpic expansion. Initial pressure and temperature are 4.12 MPa and 20 °C.



Figure C-7. Mean density as a function of pressure for initially gas-phase agent/nitrogen mixture undergoing isentropic expansion. Initial pressure and temperature are 4.12 MPa and 20 °C.

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