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SUMMARY

An experimental investigation was conducted to determine the effect of pressurant gas injector geometry on the pressurant gas (hydrogen) required during discharge of liquid hydrogen from a 29-cubic-foot cylindrical tank. Tests were conducted for a range of expulsion times at a nominal operating pressure level of 160 pounds per square inch absolute and with inlet pressurant gas temperatures between 508° and 558° R. Data were obtained using six injector geometries (cone, hemisphere, disk, radial, multiple screen, and straight pipe).

The first five injectors, which gave some degree of distribution or diffusion of pressurant gas into the top of the tank, were found to have similar pressurant gas requirements (within 10 percent) during the expulsion period. The straight pipe injector, which introduced the pressurant gas in a concentrated stream towards the liquid surface, showed a significant decrease in pressurant gas required for comparable propellant outflow rates.

The experimental results for the multiple screen, cone, and 1-inch-diameter straight pipe injectors were compared with results predicted by a previously developed analytical method. This comparison indicated that good agreement existed between the analysis and experimental data for the diffuser-type injectors. However, an altered analysis - one which would incorporate a mixing theory to account for the radial and axial temperature gradients in the tank ullage - would be required for prediction of pressurant gas requirements when using straight pipe injectors.

It was experimentally determined that the use of a floating insulation layer at the liquid-gas interface had little effect on overall gas requirements for expulsion. In fact, the use of the barrier increased pressurant gas requirements slightly.

INTRODUCTION

The ability to optimize a propellant tank pressurization system for a given vehicle mission profile largely depends upon the ability to accurately predict pressurant gas requirements. An underdesigned pressurization system could easily result in mission failure and an overdesigned system leads to reduced vehicle payload.

There exist several analyses (e.g., refs. 1 and 2), which attempt to predict (according to a selected set of assumptions) pressurant gas requirements during the pressurized discharge of a cryogenic fluid. These analyses do not consider the possible effects of gas injector design on pressurant gas requirements inasmuch as the selected set of assumptions do not consider variations in gas injection pattern. Because of the complexity that this factor introduces into the analytical representation of pressurizing and expelling a cryogenic fluid, the determination of its effect has to be largely based on correlations of experimental results.

An experimental investigation dealing with determination of the characteristics of selected pressurant gas injectors is available in the literature (ref. 3). The main objective of this work, however, was to minimize dilution of the cryogenic propellant being used (oxygen) by the pressurant gas (nitrogen). The experimental work did not provide pressurant gas requirements for the tank expulsion.

An experimental study, reported herein, was made in an effort to obtain insight into the effect of pressurant gas injector design on pressurant requirements. Six different injector geometries were evaluated in terms of pressurant gas consumption in the pressurized discharge of liquid hydrogen from a 29-cubic-foot cylindrical vacuum jacketed tank; the pressurant gas was hydrogen. The tests were conducted at a nominal tank pressure of 160 pounds per square inch absolute and inlet pressurant gas temperatures between 508° and 558° R; the variable parameter was the liquid hydrogen outflow rate (or expulsion time), which varied between 0.20 and 1.17 pounds per second. The results are presented and discussed herein to compare the different injector geometries on the basis of pressurant gas consumption over a range of outflow rates (expulsion time) and to point out reasons for differences in performance. The experimental results were also compared to the analytical results obtained using the analytical program as developed in reference 1, so as to note the influence of the injector on the applicability of the analysis.

It was further surmized that for certain pressurant-gas injectors, which directed the pressurant gas towards the liquid surface, an insulator or floating thermal barrier at the ullage liquid-gas interface would reduce overall gas consumption by reducing gas to liquid heat transfer. Therefore, tests were conducted using 1/8- to 3/8-inch-diameter preexpanded polystyrene beads as a surface insulator in the aforementioned tank. For the thermal barrier tests the nominal tank pressure was 60 pounds per square inch absolute, and the average gas inlet temperature was 537° R. The quantity of pressurant gas required for expulsion of liquid hydrogen over a range of outflow rates was determined with and without the floating thermal barrier, and the results are compared herein.

APPARATUS AND INSTRUMENTATION

Test Facility

The general schematic of the test tank and associated equipment is shown in figure 1 (p. 4). The apparatus included a 29-cubic-foot cylindrical tank, which had an inner diameter of 27 inches and a cylindrical length of 82 inches. Two 28-inch-outside-diameter flange and dished head assemblies were used as the tank ends. The tank was constructed of 5/16-inch-thick 304 stainless steel plate. The heat leak into the tank was controlled by a vacuum jacket surrounding the entire tank. A heat exchanger and blend valve subsystem capable of delivering gaseous hydrogen at a temperature of 210° to 700° R at a maximum flow rate of 0.04 pounds per second was used to control pressurant gas inlet temperature. A ramp generator and control valve were used for controlling the initial rate of pressurization of the propellant tank. A closed loop pressure control circuit was used to maintain constant tank pressure during the expulsion period. The liquid outflow rate was controlled by a remotely operated variable flow valve. The liquid hydrogen outflow from the tank was returned to the storage Dewar.

Liquid outflow rates were measured using a turbine-type flowmeter located in the transfer line; pressurant gas inlet flow rates were determined by use of an orifice located in the pressurant supply line. Tank, line, and differential pressures were measured with bonded strain-gage-type transducers.

The commercially available turbine-type flowmeter was calibrated with water at Lewis. This calibration was then analytically altered to include the effect of change of viscosity of fluid as well as thermal contraction due to the cryogenic application. Over the testing range presented herein the probable error of the meter and its associated recording channel was computed to be less than ±2.8 percent of the indicated value. (Probable error may be described as follows: There is a 50 percent probability that the error will be no larger than the value stated.)

An analysis was performed to determine the probable error expected in the computation of pressurant gas flow rate. The analysis included the effect of the complete instrumentation channels (transducers to recorders), which furnished the necessary measurements of gas temperature as well as line and orifice differential pressures. For the experimental work reported herein, the largest probable error in pressurant gas flow rate for all runs was ±2.0 percent; the average probable error was ±1.0 percent. Additionally, if the complete instrumentation channel is considered, the accuracy of tank

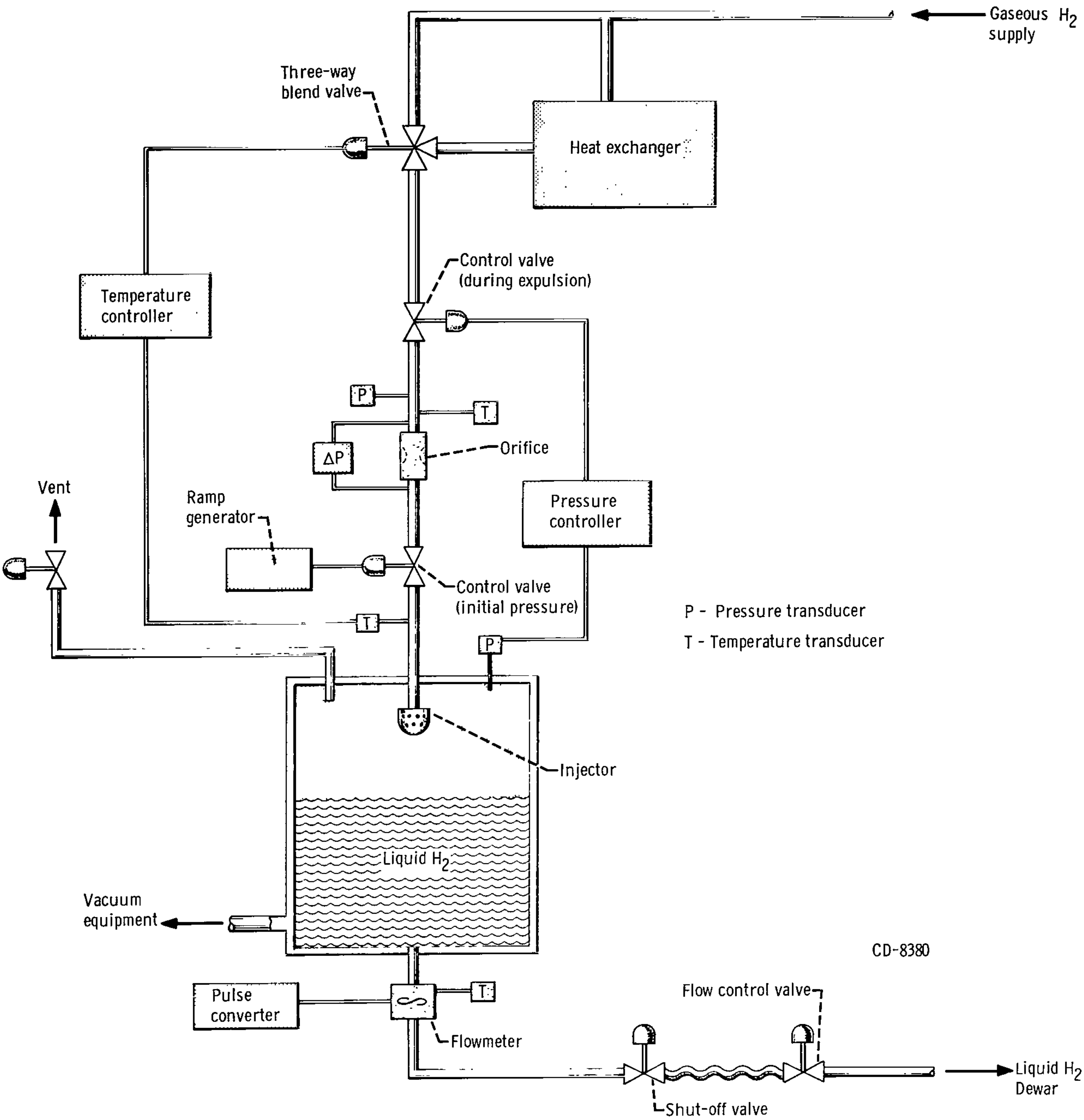


Figure 1. - General schematic of facility.

pressure was within ±1.0 percent of reported values.

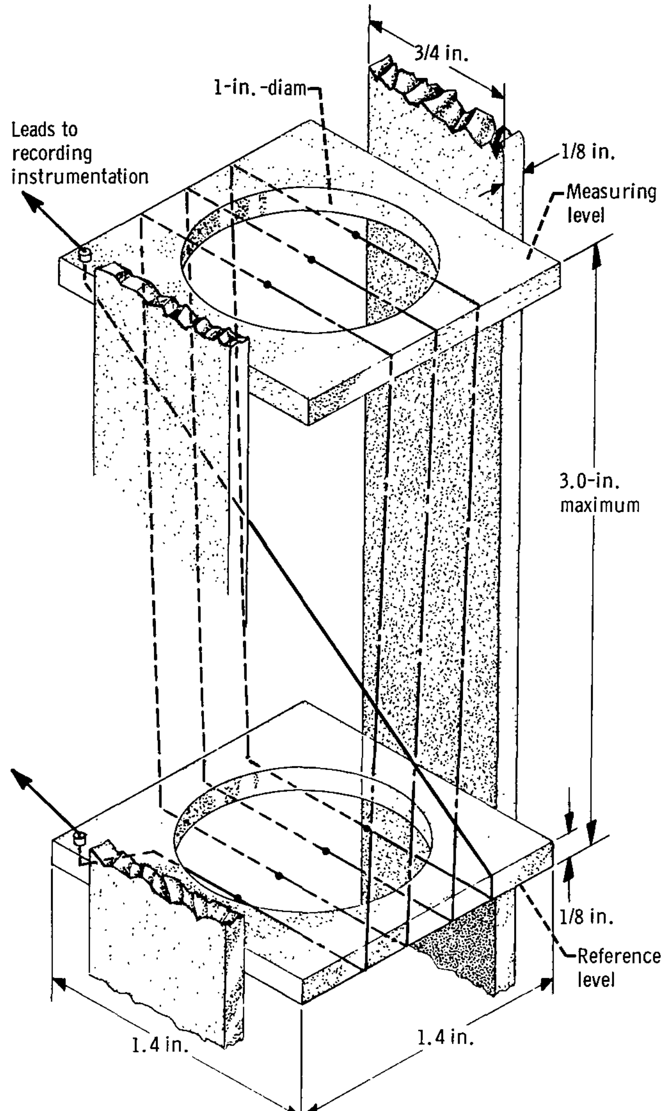
Internal Tank Instrumentation

Poor response characteristics of temperature sensors has been a major problem in the cryogenic tank pressurization area; measurement of rapid changes in temperature particularly near the moving liquid-gas interface during expulsion has been difficult. Since the response time of a temperature transducer is directly proportional to sensor mass and inversely proportional to sensor exposed surface area, low mass thermocouples were selected for this investigation. Also, since a single thermocouple has a poor signal to noise ratio at the low temperatures encountered when working with liquid hydrogen, several thermocouples were combined to form a thermopile configuration.

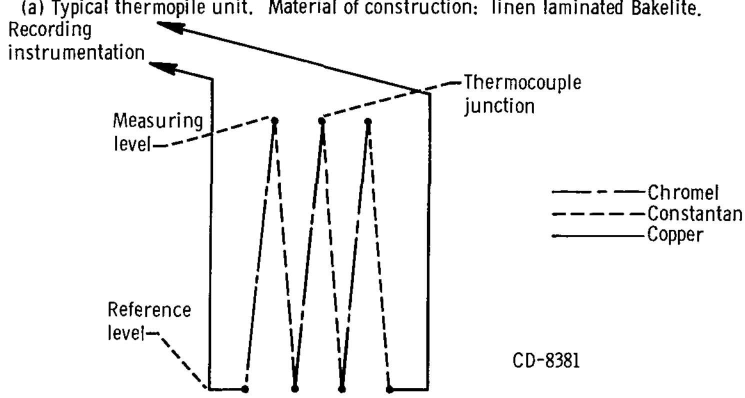
A typical three element thermopile unit and its associated wiring schematic are illustrated in figure 2 (p. 6). Chromel-constantan thermocouples were used in construction of the thermopile. Such thermopile units, used to measure temperature difference between the measurement and reference levels, were employed in the present investigation to sense ullage gas temperature in the tank. An effort was made to guard against erroneous measurements by (1) using small diameter thermocouple wire (0.008 in.), (2) designing into the thermopile unit sufficient distance between the thermocouple junctions and the Bakelite support rake to minimize thermal conduction, and (3) keeping the spacing between the reference and measuring levels of any given thermopile to a value of 3 inches or less. The latter point served to keep all thermocouple wire within the test tank and hence minimize any spurious electromotive force generated in the thermocouple lead wires as a result of the temperature differential between the thermopile measurement and reference levels.

The technique for employing the thermopiles did not employ one temperature reference for all differential measurements. Vertical and horizontal ullage gas temperature distributions in the tank were obtained by "stacking" the individual thermopile units as shown in figure 3 (p. 7). A platinum resistor temperature sensor, located at the bottom reference station of the vertical rake sensed the absolute temperature at that location and provided the basic reference for the thermopile measurements. To obtain a relatively constant reference temperature, this sensor was kept submerged in liquid hydrogen at all times during a test expulsion. The absolute temperature at any station above this platinum resistor was obtained by the summation of the individual differential voltages of each thermopile unit between the platinum resistor and the measurement station being considered.

The spacing between the reference and measuring levels for the 26 thermopiles composing the vertical rake was 3 inches except for the five units at the top of the rake which

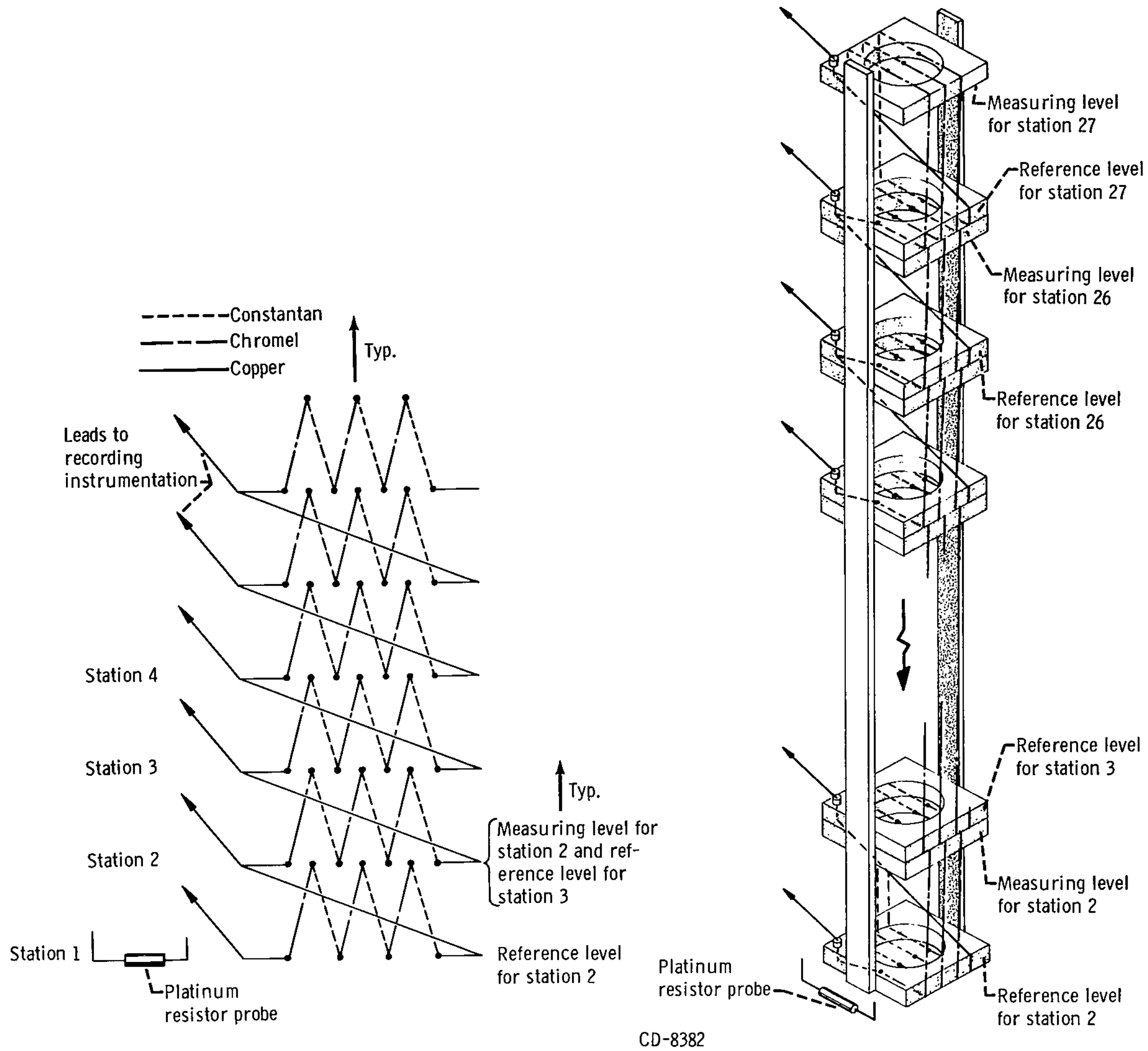


(a) Typical thermopile unit. Material of construction: linen laminated Bakelite.



(b) Wiring diagram.

Figure 2. - Three-element thermopile unit.



(a) Wiring schematic. (b) Sketch of rake.

Figure 3. - Thermopile rake.

had a 2-inch spacing. The spacing between reference and measuring levels for the 18 thermopiles composing three horizontal rakes was 2.5 inches.

A typical thermopile assembly was calibrated to determine sensor output against temperature for the range 36.4° to 98° R. This calibration, supplemented by temperature-electromotive force data for Chromel-constantan thermocouples already existing in the literature (e.g., ref. 4), was then used for test data reduction. Further, considering a single complete thermopile instrumentation channel, the probable error for a differential temperature measurement was computed to be $\pm 1.72^{\circ}$ R at a measured temperature of 57° R. This error, for one channel, however, decreases with increasing temperature (e.g., $\pm 0.36^{\circ}$ R for a differential temperature measurement at an absolute

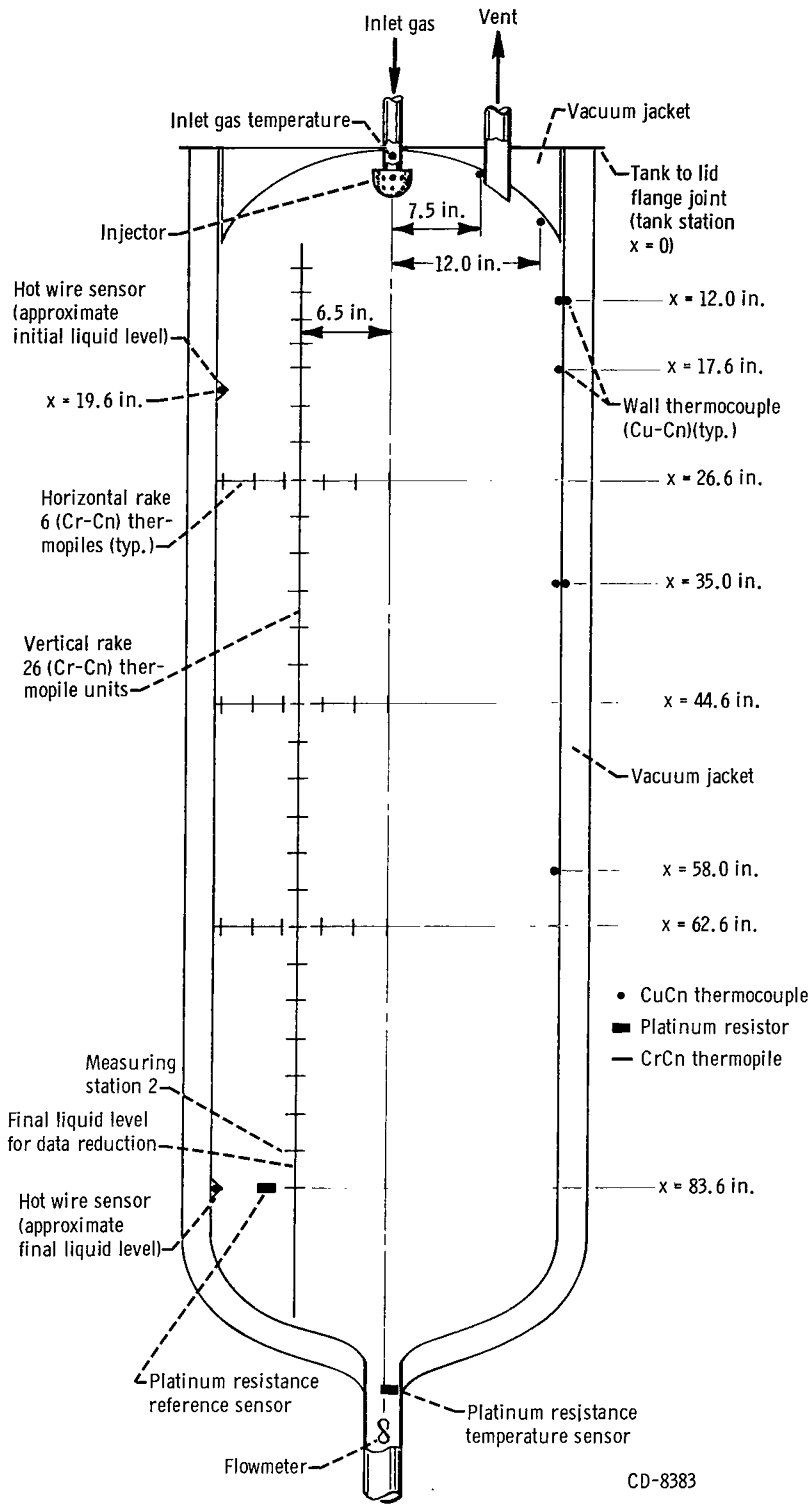


Figure 4. - Tank and instrumentation.

temperature of 500° R). It should be pointed out that the probable error of a temperature measurement taken from a station considerably removed from the platinum resistor will necessarily be in excess of the errors stated previously for a single channel. The probable error in absolute temperature measured at any station above the platinum resistor was obtained by including the probable errors on the individual voltages of each thermopile unit between the platinum resistor and the measurement station being considered. Since the probable error for any station is dependent on (1) the absolute temperature being measured at that station and (2) the probable errors present in each termopile unit between the station being considered and the platinum resistor, it can be seen that the temperature profile existing in the tank ullage gas determined the accuracy of the measuring technique. A discussion of how the accuracy of these thermopiles affected the reduced data will be given in the section DATA REDUCTION.

Internal tank instrumentation is illustrated in figure 4. Location of the vertical and horizontal ullage gas temperature rakes are indicated. Copper-constantan thermocouples were used to determine (1) wall temperatures at four locations, (2) tank lid temperatures at three locations, and (3) pressurant-gas inlet temperature. A discussion of how the accuracy of these thermocouples affected the reduced data will be given in the section DATA REDUCTION.

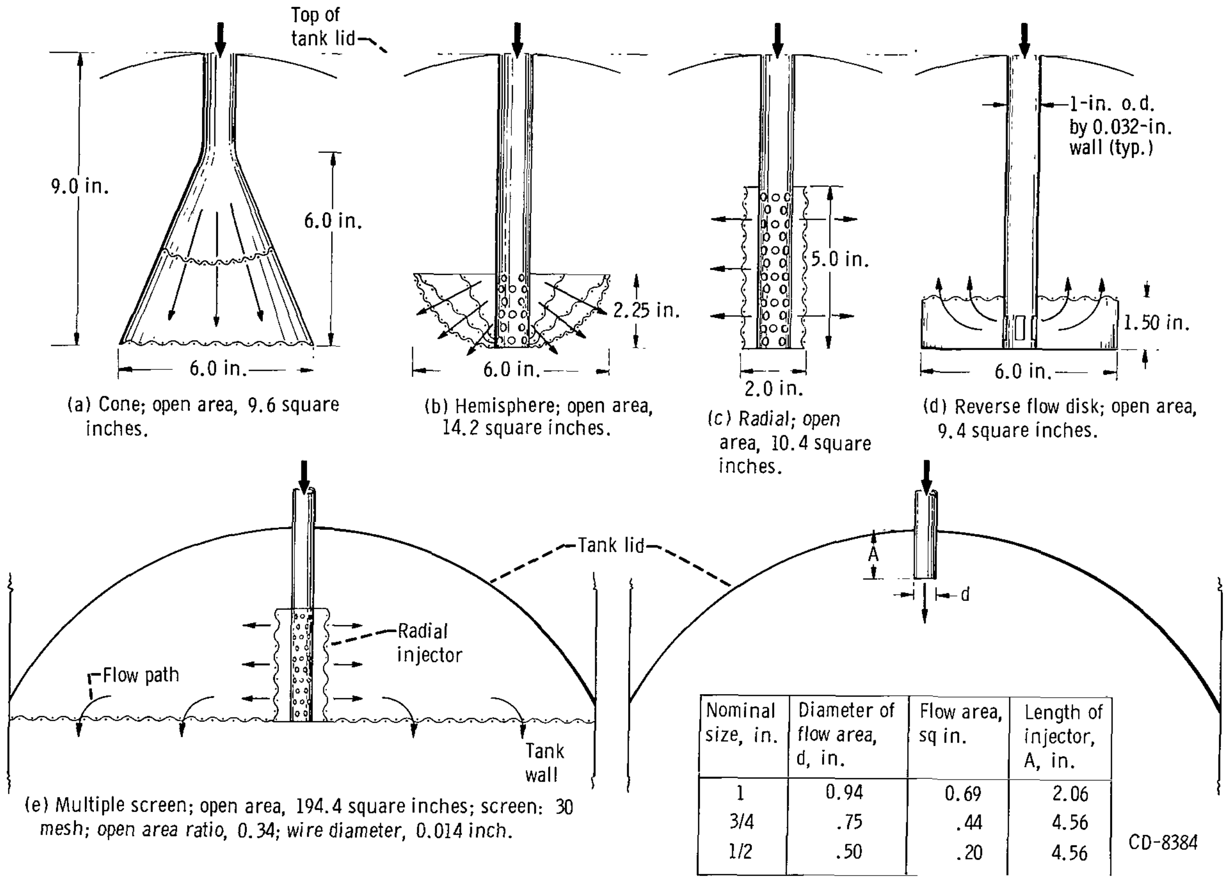
Commercially available hot wire liquid level sensors were used to determine the approximate initial liquid level as well as to initiate automatic shutdown of the rig after completion of the test expulsion.

All measurements were continuously recorded on a direct reading oscillograph and/or an automatic voltage digitizing system.

Injector Geometries

The six basic injector geometries tested are shown in figure 5 (p. 10). The first five injectors (i.e., cone, hemisphere, radial, disk, and multiple screen) were designed basically to diffuse the pressurant gas in a uniform pattern throughout the ullage volume. These units will be referred to as "conventional" or "diffuser-type" injectors. A basic design consideration with the first four diffuser-type injectors was that all should have approximately the same exit area. The multiple screen geometry, which was representative of an injector previously used in expulsion tests conducted at Lewis (ref. 5), was constructed by using the radial injector in conjunction with a spreader screen mounted normal to the tank centerline.

The remaining injector geometry - the straight pipe - injected the pressurant gas in a concentrated stream towards the liquid surface. The three configurations of the straight pipe geometry tested had nominal inside diameters of 1, 3/4, and 1/2 inch, respectively.



(a) Cone; open area, 9.6 square inches.

(b) Hemisphere; open area, 14.2 square inches.

(c) Radial; open area, 10.4 square inches.

(d) Reverse flow disk; open area, 9.4 square inches.

(e) Multiple screen; open area, 194.4 square inches; screen: 30 mesh; open area ratio, 0.34; wire diameter, 0.014 inch.

Nominal size, in.	Diameter of flow area, d, in.	Flow area, sq in.	Length of injector, A, in.
1	0.94	0.69	2.06
3/4	.75	.44	4.56
1/2	.50	.20	4.56

(f) Straight pipes.

Figure 5. - Injector geometries.

TEST PROCEDURE

The tank was initially purged with helium and then filled from the bottom with liquid hydrogen to approximately 19.6 inches from the tank-to-lid flange joint (fig. 4). The liquid was maintained at this level by a topping process.

After the tank fill was completed, the tank was pressurized at a controlled ramp rate from atmospheric to the desired operating pressure (0 to 1, fig. 6). The tank pressure was then held constant (interval 1 to 2, fig. 6) for approximately 50 seconds to stabilize internal tank temperatures. Pressurized expulsion (interval 2 to 3, fig. 6) was then initiated; during each expulsion, approximately 20.5 cubic feet of liquid hydrogen was discharged from the tank. Tank pressure and pressurant gas inlet temperature were stabilized within 10 to 15 seconds after initiation of the expulsion period. Liquid hydrogen outflow was stopped by means of a signal from the hot wire liquid level sensor located 83.6 inches from the tank-to-lid flange joint.

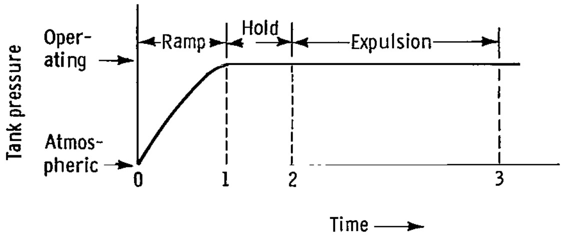


Figure 6. - Typical tank cycle.

DATA REDUCTION

Pressurant Gas Added

The main parameter used to compare the performance of the six injector geometries was the actual pressurant gas required during any or all of the time periods of the tank cycle. In general, the weight of pressurant gas used from any initial time t_i to any final time t_f was determined by numerical integration of the gas orifice equation (Symbols are defined in the appendix.)

$$m_{a,i\text{-}f} = \int_{t_i}^{t_f} KyD^2C\sqrt{\rho\,\Delta P}\,dt \tag{1}$$

Ullage Mass

The initial mass m_i and final mass m_f in the ullage volume were obtained by numerical integration of the particular density profiles as follows:

$$m_i = \int_{\mathscr{V}_i} \rho(T)\delta\mathscr{V} \tag{2}$$

$$m_f = \int_{\mathscr{V}_f} \rho(T)\delta\mathscr{V} \tag{3}$$

Radial temperature gradients encountered were included in the numerical integration of equation (3).

Liquid level for data reduction at the beginning of expulsion was determined indirectly in the following manner:

(1) After the beginning of the tank expulsion, the time of saturation temperature was determined from the continuous temperature-time recording of the first thermopile measuring station to be uncovered by the receding liquid.

(2) The amount of liquid hydrogen expelled from the tank up to that point in time was then computed from the liquid outflow rate measurements.

(3) This quantity of liquid was then converted to a corresponding tank height and combined with the known position of the thermopile transducer.

Again, by using saturation temperatures, as indicated by the vertical rake transducers, final liquid level was taken as the midpoint between the platinum resistance sensor and measuring station 2 (fig. 4, p. 8). Taking the final liquid level at this point permitted a convenient partition to be taken of the ullage volume for data reduction purposes.

Mass Transfer

A mass balance was performed on the ullage volume from an initial time t_i to a final time t_f as follows:

$$m_i + m_{a,i\text{-}f} = m_f + m_{T,i\text{-}f} \tag{4}$$

Therefore, the mass transferred was calculated as

$$m_{T,i\text{-}f} = m_i + m_{a,i\text{-}f} - m_f \tag{5}$$

If $m_{T,i\text{-}f}$ was a positive quantity, this implied mass leaving the ullage system (condensation).

Heat Balance

For an open thermodynamic system consisting of the tank ullage volume, the first law of thermodynamics for an increment of time dt may be written as

$$\delta m_a\left(u_a + P_a v_a + \frac{V_a^2}{2g} + Z_a\right) - \delta m_\ell\left(u_\ell + P_\ell v_\ell + \frac{V_\ell^2}{2g} + Z_\ell\right) = \delta Q_\ell + \delta W_\ell + d(U) \tag{6}$$

The kinetic and potential energy terms are small in comparison with the other energy terms and will be neglected in this development. If $u = h - Pv$ is substituted, equation (6) becomes

$$\delta m_a h_a - \delta m_\ell h_\ell = \delta Q_\ell + P\,d\mathcal{V} + d(U) \tag{7}$$

Further,

$$d(U) = dH - P\,d\mathcal{V} - \mathcal{V}\,dP \tag{8}$$

Hence,

$$\delta m_a h_a - \delta m_\ell h_\ell = \delta Q_\ell + dH - \mathscr{V}\,dP \tag{9}$$

For all phases of the tank cycle except the initial pressurization period, tank pressure P can be considered constant; therefore, the final form of equation (6) becomes

$$\delta m_a h_a - \delta m_\ell h_\ell = \delta Q_\ell + dH \tag{10}$$

Equation (10) can be integrated over any time period of essentially constant tank pressure (i.e., the expulsion interval 2 to 3, fig. 6). The physical interpretation of mass leaving the ullage system m_ℓ is mass transferred m_T. The physical interpretation of the remaining quantities in equation (10) are as follows:

$$\underbrace{\int_2^3 \delta m_a h_a}_{\substack{\text{Enthalpy}\\ \text{input by}\\ \text{pressurant}\\ \text{gas}}} - \underbrace{\int_2^3 \delta m_T h_T}_{\substack{\text{Enthalpy of}\\ \text{mass leav-}\\ \text{ing through}\\ \text{mass}\\ \text{transfer}}} = \underbrace{\int_2^3 \delta Q_\ell}_{\substack{\text{Heat}\\ \text{leaving}\\ \text{system}}} + \underbrace{\int_2^3 dH}_{\substack{\text{Total}\\ \text{change}\\ \text{in}\\ \text{ullage}\\ \text{enthalpy}}} \tag{11}$$

The symbol h_T represents the enthalpy that the particular quantity of mass m_T has when entering or leaving the system (i.e., for condensation h_T is taken at the saturated liquid state point; for evaporation h_T is taken at the saturated gas state point). Also, the mass transferred m_T will be a positive quantity for condensation and a negative quantity for evaporation as indicated in equation (5).

The terms of equation (11) can be grouped into three categories: enthalpy input by pressurant gas, total change in ullage enthalpy, and energy lost. They were evaluated individually in the following manner.

Enthalpy input. - The first term of equation (11) may be evaluated as follows:

$$\int_2^3 \delta m_a h_a = m_{a,2\text{-}3} h_a \tag{12}$$

The mass added during expulsion $m_{a,2\text{-}3}$ was determined from equation (1). The specific enthalpy of the inlet gas h_a was evaluated at a time-weighted average inlet temperature and pressure.

Total change in ullage enthalpy. - At constant pressure the ullage gas density and enthalpy are only functions of temperature; that is, $\rho = \rho(T)$ and $h = h(T)$. Therefore, by knowing the ullage gas temperature profiles at points 2 and 3 (fig. 6, p. 11), the total change in ullage enthalpy was evaluated according to the equation

$$\int_2^3 dH = \int_{\mathscr{V}_3} \rho(T)h(T)\delta\mathscr{V} - \int_{\mathscr{V}_2} \rho(T)h(T)\delta\mathscr{V} \tag{13}$$

Energy loss. - The two remaining terms in equation (11) can be combined to represent the total energy loss Q_L by the system during the expulsion period

$$Q_{L,2-3} = \int_2^3 \delta m_T h_T + \int_2^3 \delta Q_\ell \tag{14}$$

The quantity $\int_2^3 \delta m_T h_T$ represents the amount of energy leaving the ullage volume through mass transfer. Hence,

$$\int_2^3 \delta m_T h_T = m_{T,i-f} h_T \tag{15}$$

The other quanity in equation (14), $\int_2^3 \delta Q_\ell$, can be divided into two parts: (1) the heat lost to the tank wall and lid $\int_2^3 \delta Q_w$ and (2) the heat transferred to the bulk liquid $\int_2^3 \delta Q_b$ where

$$\int_2^3 \delta Q_b = |h_{fg} m_T| + Q_s \tag{16}$$

The integral $\int_2^3 \delta Q_w$ could be evaluated because the wall temperature profiles at points 2 and 3 in figure 6 were known. Since the tank was vacuum jacketed, external heat input was neglected. Hence,

$$\int_2^3 \delta Q_w = \sum_{i=1}^{7} \left[\delta m_w \int_{T_2}^{T_3} c_w(T_w) dT_w \right]_i \tag{17}$$

where

m_w mass of tank wall and lid

$c_w(T_w)$ specific heat of wall and lid

T_w measured wall temperature

The heat transferred to the bulk liquid $\int_2^3 \delta Q_b$ may be evaluated by

$$\int_2^3 \delta Q_b = \underbrace{\int_{m_3} h_b \delta m_b}_{\text{Final bulk liquid enthalpy}} - \underbrace{\int_{m_2} h_b \delta m_b}_{\text{Initial bulk liquid enthalpy}} + \underbrace{\int_2^3 h_b \delta m_{b\ell}}_{\text{Enthalpy of liquid expelled from tank}} - \underbrace{\int_2^3 \delta m_T h_T}_{\text{Enthalpy gained by bulk liquid through mass transfer}} \tag{18}$$

Therefore, all terms of equation (11) could be evaluated directly if the associated temperature profiles in the liquid and ullage volumes were known. The accuracy of the thermopile transducers in the liquid hydrogen, however, precluded any direct experimental measurement of the heat transferred to the bulk liquid during this investigation. Thus, the quantity could not be evaluated directly, and a complete independent check of Q_L could not be made. The total energy lost Q_L was, therefore, evaluated by subtracting the change in enthalpy of the ullage gas between initial and final states from the enthalpy added to the system during expulsion

$$Q_L = \int_2^3 \delta m_a h_a - \int_2^3 dH \tag{19}$$

TABLE I. - MASS BALANCE FOR ALL INJECTORS

Run	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
	Injector	Tank pressure, psia	Volume discharge, cu ft	Inlet gas temperature, °R	Ramp time, sec	Hold time, sec	Expulsion time, sec	Tank cycle time, sec	Ramp			Hold				Expulsion		
									Initial ullage mass, m_0, lb	Mass added during ramp, $m_{a,0-1}$, lb (a)	Mass transfer during ramp, $m_{T,0-1}$, lb	Ullage mass after ramp, m_1, lb	Mass added during hold, $m_{a,1-2}$, lb (b)	Mass transfer during hold, $m_{T,1-2}$, lb	Ullage mass after hold, m_2, lb	Mass added during expulsion, $m_{a,2-3}$, lb	Mass transferred during expulsion, $m_{T,2-3}$, lb	Final ullage mass, m_3, lb
216	Cone	160	20.09	528	33.6	48.9	130.4	212.9	0.125±0.004	0.330±0.003	-0.007±0.005	0.462±0.001	0.152±0.002	-0.001±0.003	0.615±0.002	2.569±0.02	0.329±0.022	2.855±0.010
206		159.6	20.23	532	36.4	76.8	204.6	317.8	.151± .006	.326± .003	.082± .007	.395± .001	.160± .002	-.138± .005	.693± .005	2.983± .03	.291± .043	[c]3.385
208		160	20.22	531	36.4	53.7	275.7	365.8	.144± .004	.323± .003	.080± .006	.387± .004	.188± .002	-.051± .005	.626± .002	3.341± .02	.361± .041	[c]3.606
211		160	20.09	525	35.2	48.5	335.5	419.2	.076± .001	.318± .003	.034± .006	.360± .005	.154± .002	-.006± .005	.520± .001	3.505± .03	.321± .048	[c]3.704
213		161.5	20.33	532	34.9	48.2	404.3	487.4	.104± .002	.319± .003	.084± .005	.339± .003	.138± .001	-.064± .004	.541± .002	3.700± .04	.371± .027	3.870± .027
67	Hemi-sphere	157	19.88	532	44.1	72.8	115.3	232.2	0.230±0.010	0.463±0.005	0.204±0.011	0.490±0.001	0.344±0.003	0.122±0.004	0.712±0.003	2.642±0.03	0.342±0.029	3.012±0.012
57		158	19.84	542	44.7	182.4	201.4	428.5	.267± .014	.434± .004	.215± .015	.486± .001	.426± .004	.167± .005	.745± .003	2.980± .03	.111± .047	[c]3.614
59		159	20.18	535	44.7	91.2	270.8	406.7	.161± .007	.399± .004	.126± .008	.434± .001	.367± .004	.210± .004	.591± .001	3.511± .02	.386± .042	[c]3.716
61		159.5	19.83	530	43.6	75.4	329.3	448.3	.196± .009	.361± .004	.105± .010	.452± .001	.344± .003	.115± .004	.681± .002	3.712± .03	.458± .049	[c]3.935
63		159	20.08	536	42.8	70.2	392.1	505.1	.209± .009	.362± .004	.094± .010	.477± .001	.308± .003	.016± .005	.769± .004	3.923± .03	.652± .043	4.040± .030
119	Disk	157	20.25	527	38.3	49.4	75.4	163.1	0.196±0.004	0.372±0.004	0.184±0.008	0.384±0.006	0.228±0.002	0.051±0.007	0.561±0.003	2.227±0.02	0.085±0.023	2.703±0.010
109		159.3	19.93	531	51.9	30.2	188.2	270.3	.106± .003	.395± .004	.081± .008	.420± .006	.085± .001	-.005± .006	.510± .001	3.020± .03	.227± .045	[c]3.303
111		160	20.17	529	54.9	32.0	279.8	366.7	.137± .004	.446± .004	.157± .011	.426± .009	.152± .002	.052± .009	.526± .001	3.647± .02	.408± .043	[c]3.765
113		159.5	20.23	524	51.1	35.6	304.7	391.4	.101± .002	.388± .004	.087± .007	.402± .006	.120± .001	-.019± .006	.541± .002	3.539± .02	.335± .042	[c]3.745
122		159.5	20.19	520	55.1	48.5	351.6	455.2	.093± .002	.429± .004	.108± .007	.414± .006	.147± .001	.019± .006	.542± .001	3.673± .03	.308± .040	3.907± .026
101	Radial	158.4	20.23	522	37.9	73.1	131.9	242.9	0.169±0.007	0.355±0.004	0.107±0.008	0.417±0.001	0.294±0.003	0.020±0.009	0.691±0.008	2.785±0.01	0.479±0.017	2.997±0.011
93		157.6	20.25	529	40.5	125.4	211.7	377.6	.205± .011	.357± .004	.124± .012	.438± .001	.303± .003	.079± .004	.662± .003	3.019± .03	.165± .046	[c]3.516
95		159.6	20.19	528	37.6	84.0	268.0	389.6	.191± .009	.343± .003	.133± .010	.401± .001	.237± .002	.016± .003	.622± .002	3.293± .02	.232± .042	[c]3.683
97		159.5	20.13	526	38.8	81.0	334.0	453.8	.189± .009	.361± .004	.128± .010	.422± .001	.303± .003	.133± .004	.592± .002	3.649± .03	.335± .049	[c]3.906
99		159.3	20.17	523	38.7	72.6	436.6	547.9	.150± .007	.361± .004	.102± .008	.409± .001	.301± .003	.142± .004	.568± .002	4.014± .04	.425± .005	4.157± .030
254	Multiple screen	160.7	19.93	526	38.0	46.5	124.2	208.7	0.251±0.014	0.433±0.004	0.164±0.014	0.520±0.002	0.225±0.002	0.075±0.017	0.670±0.003	2.780±0.01	0.400±0.017	3.050±0.013
243		159.3	19.89	528	35.3	41.1	200.3	276.7	.224± .010	.417± .004	.150± .011	.491± .004	.181± .002	-.044± .024	.716± .002	3.211± .03	.543± .045	[c]3.384
245		162	19.87	508	38.6	42.5	280.8	361.9	.283± .013	.477± .005	.217± .014	.543± .002	.229± .002	.116± .017	.656± .003	3.564± .05	.504± .062	[c]3.716
248		160	20.23	540	42.6	41.1	339.4	423.1	.081± .001	.706± .007	.363± .007	.424± .001	.179± .002	.113± .009	.490± .002	3.589± .03	.432± .047	[c]3.647
250		161	20.11	527	42.3	43.4	407.1	492.8	.124± .004	.432± .004	.097± .006	.459± .002	.179± .002	.071± .012	.567± .002	3.968± .04	.573± .048	3.962± .027
228	1-inch-diameter straight pipe	160	20.38	538	34.0	56.0	131.9	221.9	0.118±0.004	0.373±0.004	-0.550±0.010	1.041±0.008	0.092±0.001	0.095±0.011	1.038±0.008	1.991±0.02	0.124±0.036	2.905±0.005
218		161	20.27	523	31.5	94.5	208.0	334.0	.177± .008	.299± .003	-.785± .015	1.261± .012	.158± .002	.053± .022	1.366± .018	2.469± .04	-.095± .059	[c]3.930
226		160	20.74	521	30.1	55.2	289.7	375.0	.103± .004	.266± .002	-.506± .009	.875± .008	.068± .001	.073± .011	.870± .008	2.793± .03	.419± .045	[c]3.244
222		159.5	20.65	525	32.7	55.9	349.4	438.0	.205± .012	.259± .003	-.544± .017	1.008± .011	.124± .001	.071± .017	1.061± .013	3.142± .03	.337± .051	[c]3.866
224		160	20.68	535	39.6	47.9	411.8	499.3	.113± .003	.253± .003	-.607± .018	.973± .018	.089± .001	.354± .019	.708± .005	3.332± .02	.366± .030	3.674± .022
332	3/4-inch-diameter straight pipe	158.6	20.08	549	33.2	98.1	128.0	259.3	0.112±0.001	0.274±0.003	-1.260±0.026	1.646±0.026	0.142±0.001	0.199±0.035	1.589±0.024	1.766±0.02	0.126±0.032	3.229±0.006
320		159.3	20.03	553	85.5	48.6	207.7	341.8	.171± .006	.304± .003	-.227± .007	.702± .002	.118± .001	.002± .006	.818± .006	2.297± .04	.081± .050	[c]3.034
322		158.3	20.07	552	104.4	27.6	282.2	414.2	.242± .014	.340± .003	-.121± .014	.703± .002	.052± .001	-.010± .005	.765± .004	2.591± .02	.203± .038	[c]3.153
324		158.8	20.19	554	114.3	29.8	351.8	495.9	.190± .009	.310± .003	-.189± .010	.689± .002	.042± .001	-.011± .005	.742± .004	2.827± .03	.127± .046	[c]3.442
328		157.6	20.29	558	87.1	31.2	428.3	546.6	.237± .013	.279± .003	-.355± .014	.871± .004	.053± .001	.091± .006	.833± .005	3.069± .02	.459± .026	3.443± .016
358	1/2-inch-diameter straight pipe	159.8	19.91	538	49.0	62.3	280.0	391.3	0.206±0.015	0.479±0.005	-2.295±0.258	2.980±0.258	0.095±0.001	0.232±0.598	2.843±0.540	2.346±0.03	1.655±0.540	3.534±0.017
363		158.8	19.59	547	51.9	39.9	342.4	434.2	.202± .015	.435± .004	-1.944± .121	2.581± .180	.051± .001	-.205± .376	2.837± .330	2.529± .04	1.944± .053	[c]3.422
351		160.1	19.84	547	75.6	64.6	395.6	535.8	.237± .013	.293± .003	-.850± .026	1.380± .022	.071± .001	.024± .030	1.427± .020	2.493± .05	.278± .059	3.642± .017

[a] 1 percent error was assumed on these values when calculating probable error in $m_{T,0-1}$ (i.e., col. 11).

[b] 1 percent error was assumed on these values when calculating probable error in $m_{T,1-2}$ (i.e., col. 14).

[c] 1 percent error was assumed on these values when calculating probable error in $m_{T,2-3}$ (i.e., col. 17).

Error Analysis

An analysis was performed to determine the magnitude of probable error which would be present in the ullage gas mass integrations of equations (2) and (3). This analysis considered the errors that the thermopile transducers as well as the tank pressure sensor introduced into the ullage mass integration. These calculations were performed for all runs at time 0, 1, and 2 (fig. 6, pg. 11), as well as at time 3 for the fast and slow expulsions for each injector. For the six basic injector geometries (i.e., cone, hemisphere, disk, radial, multiple screen, and straight pipe), the errors are shown in table I.

In table II, the average as well as the largest probable errors in the ullage mass integrations at the tank cycle times of 0, 1, 2, and 3 are shown in percent for all injectors tested.

The mass transferred during any interval was calculated using both the integrated ullage contents at t_i and t_f and the mass of pressurant gas added during the interval (i.e., according to eq. (5)). A probable error for mass transferred was computed for all runs reported herein by using the error values stated in table I for the ullage gas mass integrations and also the inherent error in the orifice measuring system (stated previously). For the six basic injector geometries (i.e., cone, hemisphere, disk, radial, multiple screen, and 1-inch-diameter straight pipe), the largest probable errors in mass transferred were ±0.02 pound for both the ramp and hold intervals and ±0.06 pound for the expulsion interval. For the 3/4-inch-diameter straight pipe, the largest probable errors for the ramp, hold, and expulsion periods were ±0.03, ±0.04, and ±0.05 pound, respectively. For the 1/2-inch diameter straight pipe, errors of ±0.26, ±0.60, and

TABLE II. - AVERAGE AND LARGEST PROBABLE ERRORS IN ULLAGE MASS INTEGRATIONS

Injector geometry	Tank cycle time							
	0		1		2		3	
	Average error, percent	Largest error, percent	Average error, percent	Largest error, percent	Average error, percent	Largest error, percent	Average error, percent	Largest error, percent
All diffuser-type injectors and 1-inch-diameter straight pipe	±3.7	±5.9	±1.0	±2.5	±0.8	±1.3	<±1	<±1
3/4- and 1/2-inch-diameter straight pipes	±5.1	±7.4	±0.6	±1.6	±4.4	±19	<±1	<±1

±0.54 pound were calculated for the three successive intervals of the tank cycle.

In addition, an analysis was conducted to determine the magnitude of error that could be present in the quantities $\int_2^3 \delta m_a h_a$ (eq. (12)), Q_L (eq. (19)), and $\int_2^3 \delta Q_w$ (eq. (17)). This analysis considered the errors that the thermopile transducers, the wall and lid thermocouples, and the pressurant inlet thermocouple introduced into the enthalpy calculations. The probable error was computed for the longest and shortest expulsion period for the cone, multiple screen, and 1-inch-diameter straight pipe. The largest probable error computed, during the expulsion period only, for the quantity $\int_2^3 \delta m_a h_a$ was ±1.1 percent. For the same time period, the total energy lost Q_L (as determined by subtracting the change in enthalpy of the ullage gas between initial and final states from the enthalpy added to the system during expulsion) could be in error by as much as ±2.4 percent. Heat lost to the tank wall $\int_2^3 \delta Q_w$ had a probable error of as great as ±9.7 percent.

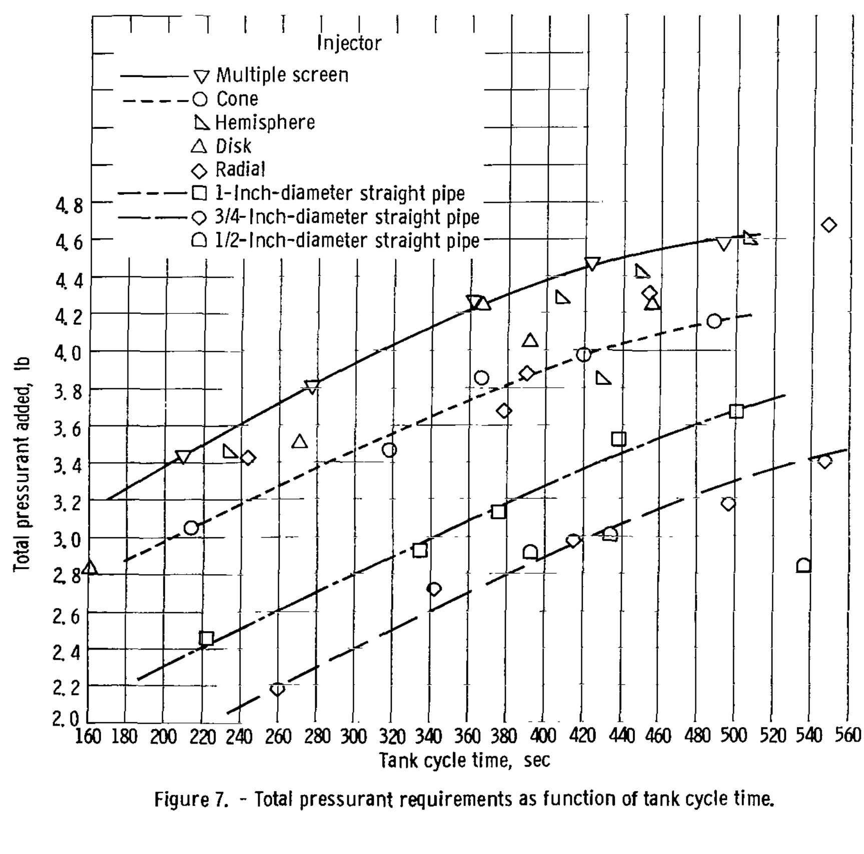


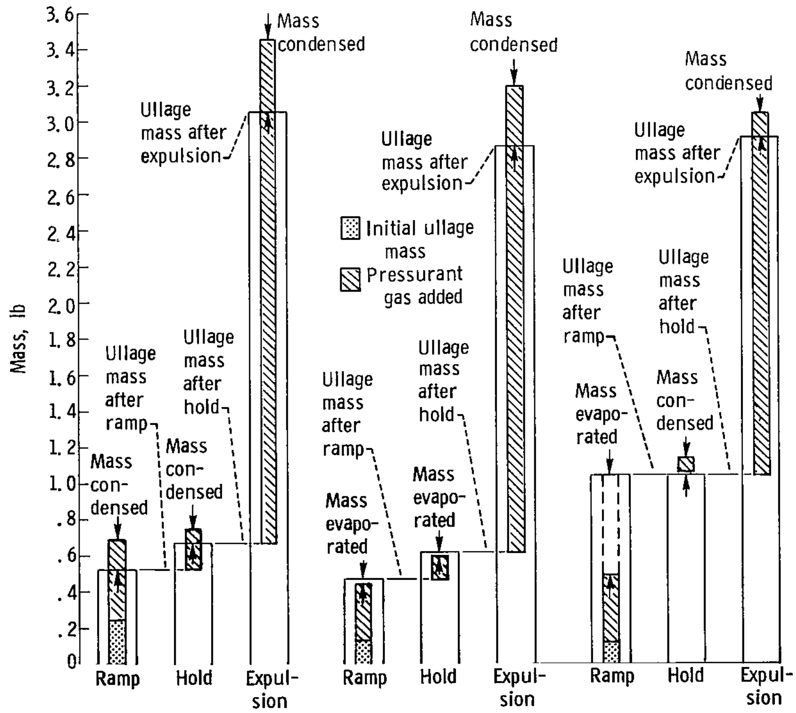
Figure 7. - Total pressurant requirements as function of tank cycle time.

RESULTS AND DISCUSSION

Experimental Testing of Pressurant-Gas Injector Configurations

The operating parameters and major experimental results are summarized in table I. There exist slight differences in the quantities "volume discharged," "tank pressure," and "inlet gas temperature" among the data in table I. These differences were due to facility limitations, which prevented exact repeatability of test conditions. However, maximum deviations in volume discharged and tank pressure level were only 0.97 and 2.02 percent, respectively; average inlet gas temperature for the range of the first six series of tests was $525 \pm 17^{\circ}$ R.

Trends during overall tank cycle. - As mentioned previously, the tank cycle (fig. 6, p. 11) was divided into the ramp, hold, and expulsion time periods. The various gas injector configurations are compared in figure 7 on the basis of the weight of pressurant gas added to the tank during the combined ramp, hold, and expulsion intervals as a function of total cycle time. As can be seen in the figure, the total pressurant gas added when using the diffuser-type injectors was significantly greater than the pressurant re-



(a-1) Multiple screen injector.

(a-2) Cone injector.

(a-3) 1-Inch-diameter straight pipe.

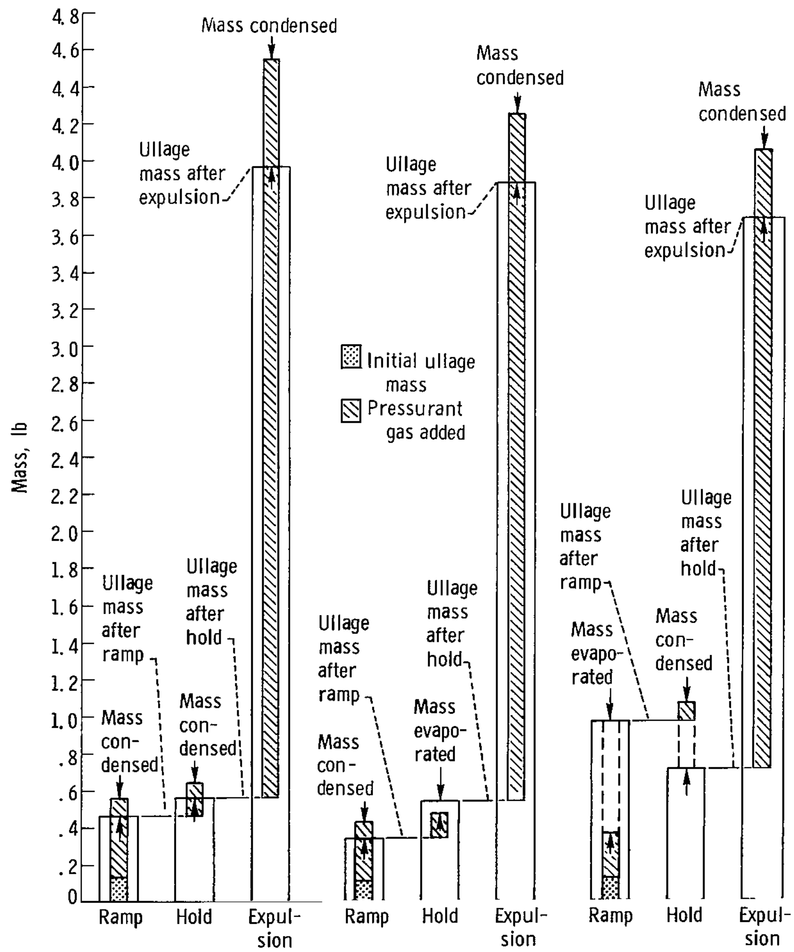
(a) Tank cycle, ~215 seconds.

Figure 8. - Ullage mass history.

quired when using any of the straight pipe configurations. Further, considering only the diffuser-type injectors, the pressurant gas added was generally a maximum for the multiple screen configuration and a minimum for the cone injector.

To obtain a better understanding of pressurant-gas injector performance during the tank cycle, a mass balance comparison was made for total cycles of approximately 215 and 493 seconds for the multiple screen, cone, and 1-inch-diameter straight pipe geometries. Net ullage system mass histories for these three injectors are plotted in figure 8(a) for a total cycle time of approximately 215 seconds. For the same injectors figure 8(b) gives a mass history for total cycle time of approximately 493 seconds.

It can be seen from figure 8 that the mass of gas in the tank ullage at the completion of the ramp period for the straight pipe injector was in general more than two times that



(b-1) Multiple screen injector.

(b-2) Cone injector.

(b-3) 1-Inch-diameter straight pipe.

(b) Tank cycle, ~493 seconds.

Figure 8. - Concluded.

of the diffuser-type configurations. The mass transfer encountered with the diffuser-type injectors during the ramp period was generally condensation (for all tests, see table I (p. 16), col. 11), whereas the straight pipe injector produced evaporation. During the hold period, the net mass transfer encountered was random. At the end of the hold period (and prior to expulsion) the difference in ullage volume mass content between the straight pipe and the diffuser-type injectors was reduced. From all the runs made for the three injectors being considered average ullage mass after hold was 0.609 pound for the diffuser-type injectors as compared to 1.008 pound for the straight pipe geometry. Considering the tank expulsion period, it can be shown in figure 8 that pressurant gas requirements were greater for the diffuser-type injectors as compared with the straight pipe geometry. Further, considering only the diffuser-type injectors, the pressurant gas added was a maximum for the multiple screen geometry and a minimum for the cone injector (absolute quantities are listed in table I, col. 16). Mass transfer encountered

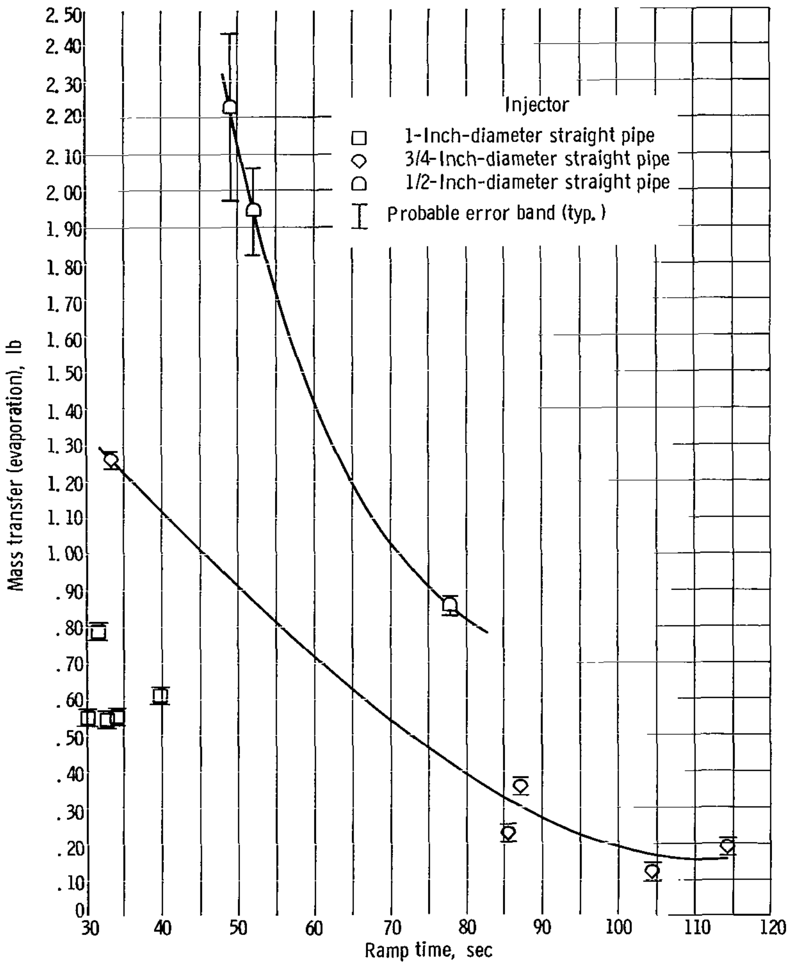


Figure 9. - Mass transfer by evaporation; ramp period as function of ramp time.

during the expulsion period was condensation for all injectors.

In comparing the diffuser-type injectors with the three straight pipe injector configurations, it is to be noted that the tank ullage mass is not identical for all injectors at the beginning of the expulsion period. This ullage condition was determined primarily by the ramp period, particularly for the straight pipe injectors (compare cols. 12 and 15 in table I). The effect of variation in the ramp time on the mass transferred (evaporation) for the three straight pipe configurations is shown in figure 9. (Ramp time was not varied to any extent for the diffuser-type injectors.) Although ramp time also was not varied appreciably for the 1-inch-diameter straight pipe, the general trend for all straight pipe configurations is increasing evaporation as ramp time is decreased; it is believed that this is due to the more forceful impingement of the incoming pressurant gas stream onto the liquid propellant as will be described later in the section Tank pressure dropoff with 1/2-inch-diameter straight pipe injector. The same effect is also obtained by decreasing the injector diameter for any given ramp time. It should be noted that these results were obtained for initial tank ullages between 17.5 and 21.6 percent and may change markedly for other values of this variable.

Comparison of first six injectors during expulsion period. - Even though there exist varying ullage gas conditions prior to expulsion, it is a major objective of this report to compare injector performance for the expulsion period during which an average of 84 percent of the total pressurant gas required was used. For any chosen mission profile having a single burn period the expulsion time becomes a fixed value whereas the ramp and hold times are generally much shorter and usually require a much smaller quantity of pressurant gas than the burn period.

The actual weight of pressurant gas added to the ullage during the expulsion period for the six basic gas injector geometries is presented in figure 10 as a function of expulsion time. The performance of the 3/4- and 1/2-inch-diameter straight pipe injectors is omitted and will be treated later in the report. When the diffuser-type injectors were used, the pressurant gas requirements for the expulsion period were a maximum with the multiple screen injector and a minimum with the cone injector. The difference between these limiting boundaries represents a decrease in pressurant gas requirements of 8.9 percent for a 130-second and 6.9 percent for a 400-second expulsion for the cone injector relative to the multiple screen injector. The remaining configuration, the 1-inch-diameter straight pipe, had the lowest pressurant requirement of the six configurations over equivalent expulsion periods; this difference ranged from 29.7 to 16.8 percent less than the pressurant required for the multiple screen injector at the expulsion times of 130 and 400 seconds, respectively.

To gain an insight into the reasons for the variations in pressurant-gas requirements when using injectors of various geometries, temperature profiles measured in the ullage gas immediately after 130- and 407-second expulsions are presented in figures 11 and 12.

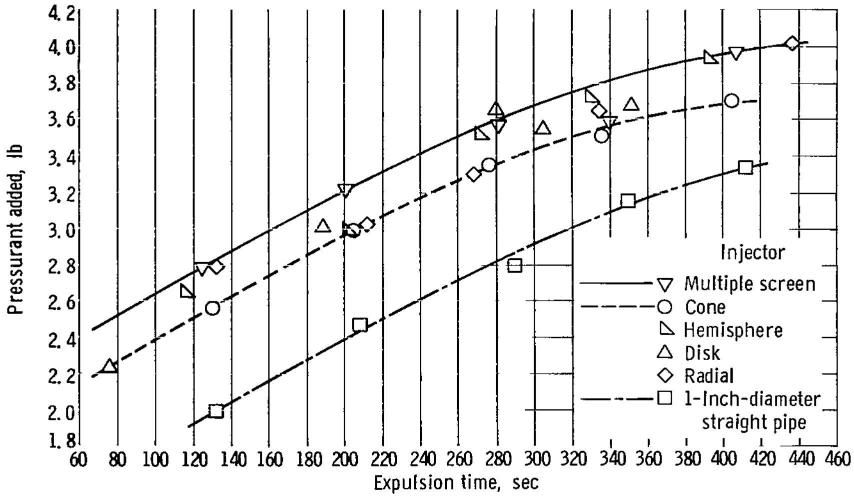
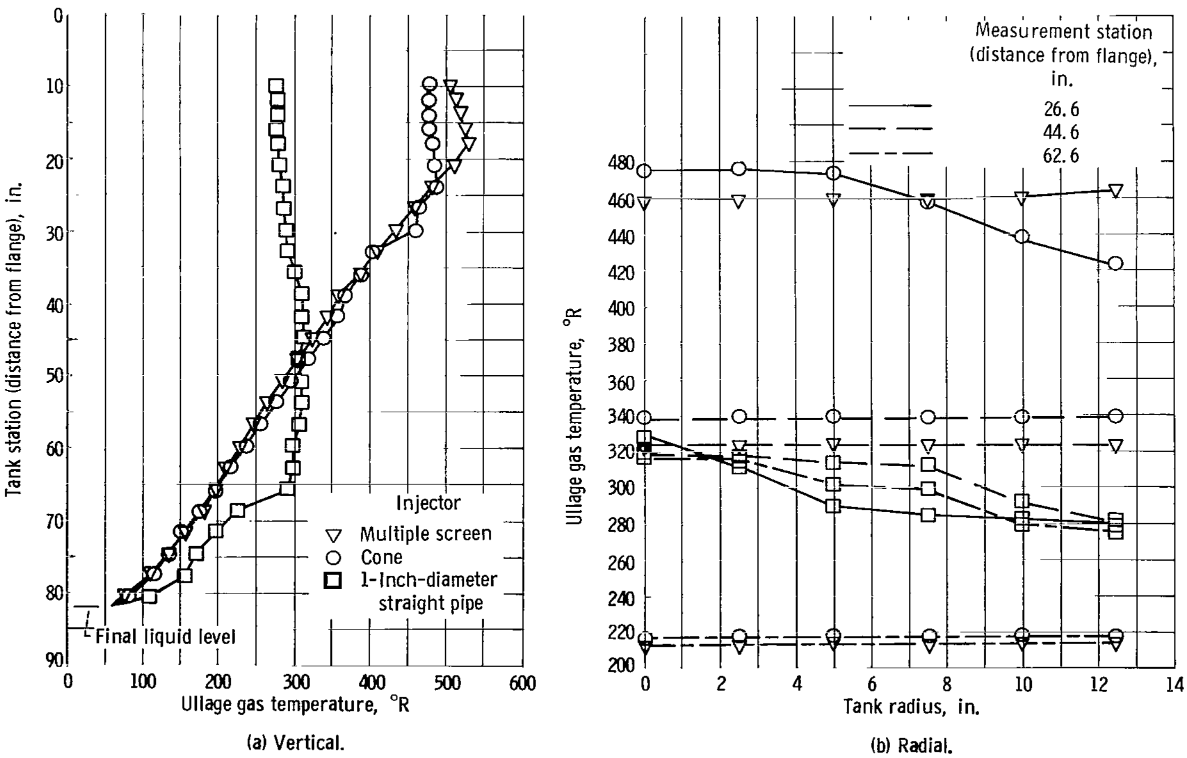


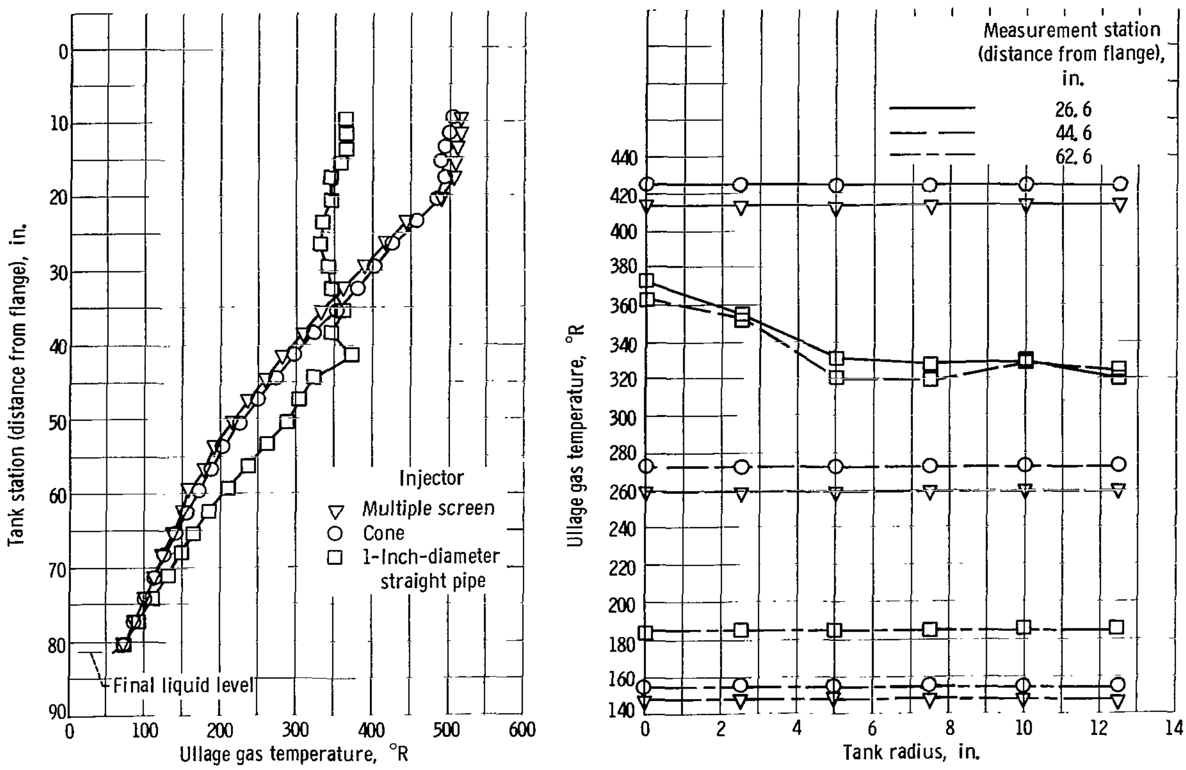
Figure 10. - Pressurant gas added to tank during expulsion interval. Tank pressure, ~160 pounds per square inch absolute; inlet temperature, ~525±17° R.



(a) Vertical.

(b) Radial.

Figure 11. - Temperature profiles in ullage gas at end of 130-second expulsion.



(a) Vertical.

(b) Radial.

Figure 12. - Temperature profiles in ullage gas at end of 407-second expulsion.

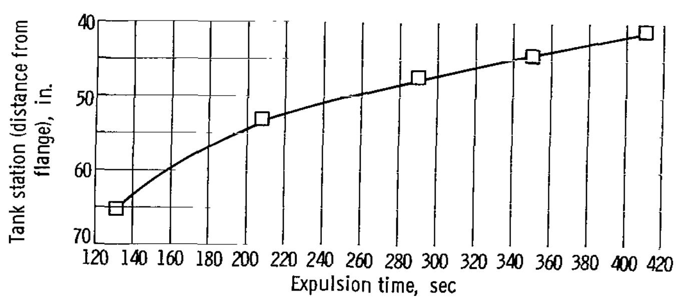


Figure 13. - Depth of vertical zone of essentially constant temperature as function of expulsion time for 1-inch-diameter straight pipe.

Both vertical and radial temperature profiles are shown; injector geometries include the multiple screen, cone, and 1-inch-diameter straight pipe. It can be observed that the diffuser-type injectors generally provided (1) vertical temperatures at the rake location, which increased almost linearly from near the liquid surface through much of the ullage, and (2) constant radial temperatures except in the upper ullage for fast expulsions. In contrast, the straight pipe injector provided (1) vertical temperatures at the rake location that were constant through a portion of the upper ullage and (2) radial temperatures that were lowest near the tank wall and highest at the center of the tank. The depth of the vertical constant temperature zone increased as expulsion time was decreased, as shown in figure 13 for all expulsions. It is apparent that with the straight pipe injector the ullage gas temperatures were warmer near the liquid surface and cooler in the upper ullage than when the diffuser-type injectors were used. Even though the temperature profiles are

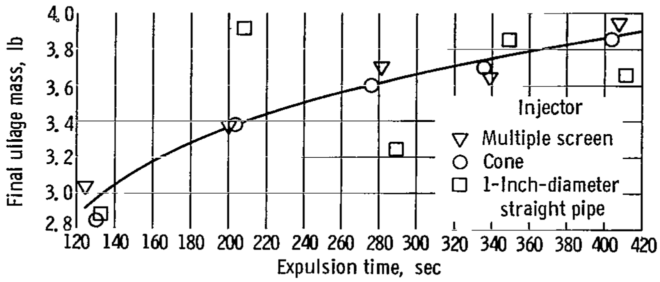
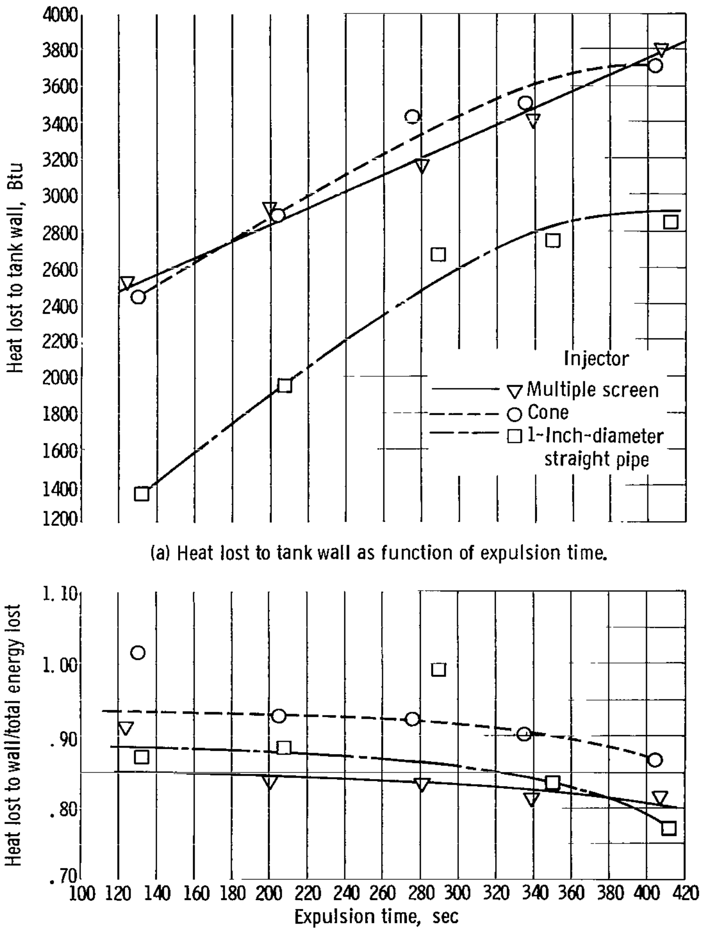


Figure 14. - Integrated final ullage mass as function of expulsion time.



(a) Heat lost to tank wall as function of expulsion time.

(b) Ratio of heat lost to tank wall to total energy lost by incoming pressurant gas as function of expulsion time.

Figure 15. - Heat lost to tank wall and ratio of heat lost to tank wall to total energy lost by pressurant gas as functions of expulsion time.

TABLE III. - EXPERIMENTAL HEAT BALANCE AND RESULTS OF ANALYTICAL PROGRAM

Run	Injector	Expulsion time, sec	Enthalpy input by pressurant gas, $Q_{a,2-3}$, Btu	Total energy lost, $Q_{L,2-3}$, Btu	Heat lost to tank wall and lid, $Q_{w,2-3}$, Btu	Change in total system enthalpy, $\int_2^3 dH$, Btu	Predicted mass requirements, $m_{a,2-3}$, lb (a)	Predicted heat lost to tank wall, $Q_{w,2-3}$, Btu (a)
216	Cone	130.4	4557±33	2404±34	2439±137	2153	2.686	2331
206		204.6	5342	3120	2886	2222	2.999	2832
208		275.7	5973	3716	3431	2257	3.262	3194
211		335.5	6193	3896	3507	2297	3.382	3325
213		404.3	6627±72	4286±73	3710±127	2341	3.575	3610
254	Multiple screen	124.2	4921±33	2764±33	2521±127	2157	2.680	2287
243		200.3	5706	3521	2936	2185	2.879	2797
245		280.8	6080	3803	3169	2277	3.284	2917
248		339.4	6532	4201	3417	2331	3.392	3433
250		407.1	7039±70	4669±71	3803±140	2370	3.598	3599
228	1-inch-diameter straight pipe	131.9	3610±37	1557±37	1355±131	2053	2.578	2312
218		208.0	4345	2195	1947	2150	2.913	2660
226		289.7	4894	2702	2676	2192	3.294	3150
222		349.4	5548	3301	2750	2247	3.368	3365
224		411.8	6007±46	3706±48	2856±137	2301	3.585	3677
358	1/2-inch-diameter straight pipe	280.8	4244	2479	1552	1765	-----	----
363		342.4	4661	2717	1768	1944	-----	----
351		395.6	4597	2542	2193	2055	-----	----

[a]Using analysis in reference 1.

considerably different, the integrated final ullage mass for any expulsion time for the multiple screen, cone, and 1-inch-diameter straight pipe injectors is approximately the same (fig. 14).

Figure 15(a) presents a comparison of heat input to the tank wall for the multiple screen, cone, and 1-inch-diameter straight pipe injectors as a function of expulsion time (data presented in table III). The heat input to tank wall when using the 1-inch-diameter straight pipe was 45 percent less when compared to the diffuser-type injectors at an expulsion time of 130 seconds and 21 percent less for a 400-second expulsion period. One of the factors leading to the different wall heating rates was the difference in the ullage gas temperatures near the wall. This temperature established an upper limit for the rising temperature of the propellant tank wall. The lower, nearly constant, temperature zone provided by the straight pipe injector resulted in a smaller quantity of heat lost to the tank wall.

The fact that the tank wall is the major heat sink of the system is shown in figure 15(b), where the ratio of heat lost to the tank wall to the total energy lost by the incoming pressurant gas is presented as a function of expulsion time for the multiple screen, cone, and straight pipe injectors. Of the total heat that was lost by the pressurant gas, between 77 and 93 percent was lost to the tank wall.

Since the wall heating represents a large percentage of the total heat lost by the pressurant gas, a significant decrease in its value will ultimately result in a significant decrease in pressurant gas consumption. It should be noted that changes in the tank material of construction, thickness, and surface-to-volume ratio, among other variables, will significantly influence the quantity of heat lost by the pressurant gas to the tank wall.

Another item of importance is probably the direct impingement of the incoming pressurant gas stream onto the liquid propellant as will be discussed later in the section Tank pressure dropoff with 1/2-inch-diameter straight pipe injector. It is believed that the resulting liquid hydrogen vaporization encountered with the 1-inch-diameter straight pipe during the ramp period, which contributed significantly to the initial ullage mass (prior to expulsion), also contributed to the lower pressurant requirements of that injector during the expulsion period.

Hence, the reduced pressurant requirements measured when using the straight pipe injector as opposed to the multiple screen injector were primarily attributed to reduced tank wall heating rates; however, it is believed that the increased initial ullage mass encountered with the straight pipe (due to the initial vaporization of liquid hydrogen by direct impingement of the pressurant gas on the liquid interface) may also have contributed to the reduced pressurant requirements during expulsion.

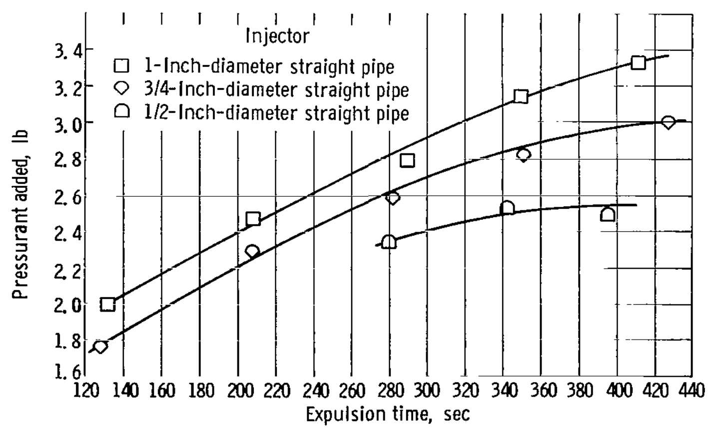


Figure 16. - Pressurant added to tank during expulsion interval for three straight pipe configurations.

3/4- and 1/2-inch-diameter straight pipe injectors during expulsion period. - The effect of reducing the diameter of the straight pipe injector on pressurant gas requirements during the expulsion period is presented in figure 16. Slightly higher pressurant gas inlet temperatures existed during these experimental runs as compared to the expulsions made with the first six injector configurations.

As can be seen in the figure, further reduction in pressurant gas requirements was evident as injector diameter was decreased. At a common expulsion time, a comparison between the ullage gas radial and vertical temperature profiles exhibited by the 1-inch-diameter straight pipe with the profiles exhibited by the smaller diameter injectors showed that, as injector diameter was reduced, (1) the radial gradients became steeper and (2) the vertical temperatures at the rake location became warmer near the liquid surface and cooler in the upper ullage (fig. 17). The reduction in diameter, however, led to two distinct limitations that must be recognized when employing injectors which give rise to high velocity pressurant gas streams impinging directly

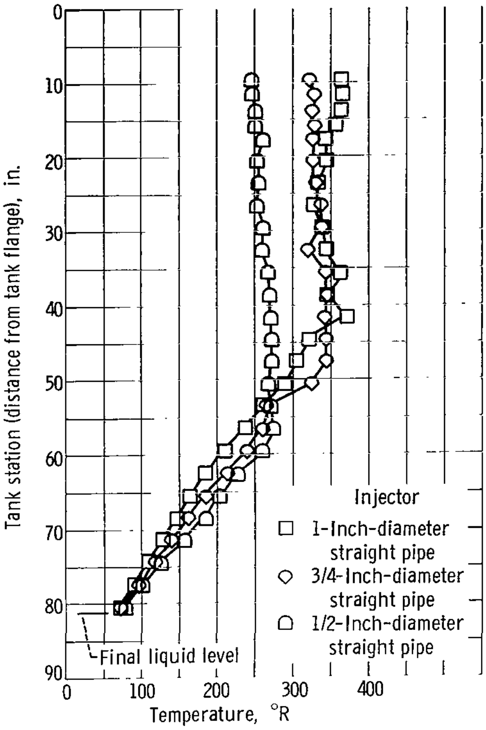


Figure 17. - Ullage gas vertical temperature profiles for straight pipe injectors at end of nominal 407-second expulsion.

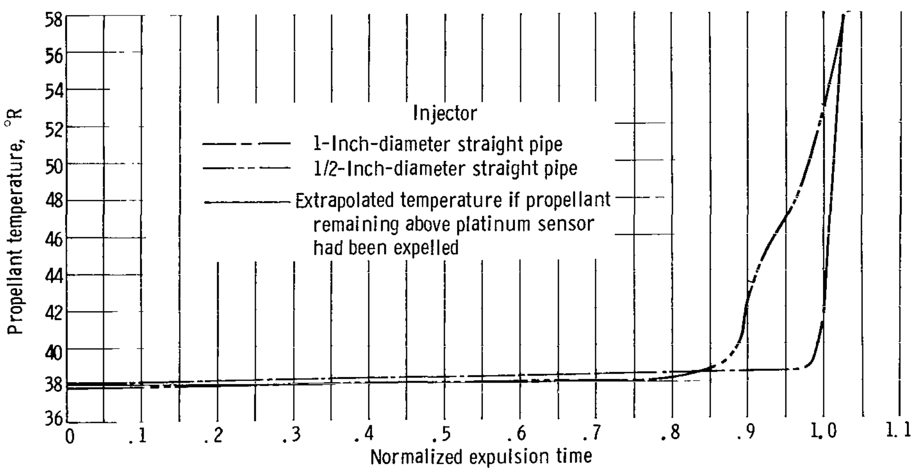


Figure 18. - Propellant temperature as measured by vertical rake platinum temperature sensor as function of normalized expulsion time. Expulsion time, ~285 seconds.

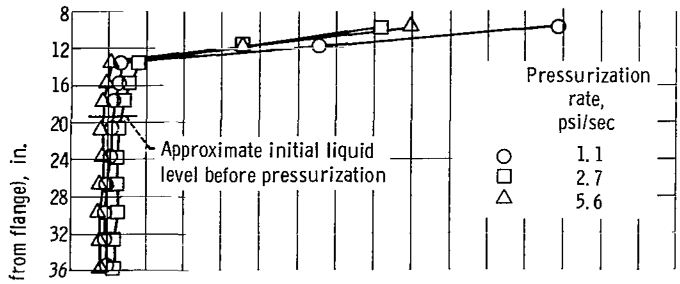
on the liquid-ullage interface. Both limitations were observed only with the 1/2-inch-diameter injector.

The first limitation was that the heat gained by the bulk liquid during the tank cycle, due to (1) direct heat transfer from the ullage gas as well as (2) pressurant-gas condensation, was observed to be greater with the small diameter injector than with any of the other geometries tested. This increased propellant heating was qualitatively demonstrated by the increased propellant temperatures recorded near the end of the expulsion period at the platinum resistor temperature sensor located at the base of the vertical rake. A typical increase in propellant temperature is plotted in figure 18 for an expulsion time of approximately 285 seconds. As can be seen in this figure, when using the 1/2-inch-diameter straight pipe the markedly increased propellant temperatures were measured during approximately the last 15 percent of the expulsion period. This bulk heating observed when using the small diameter straight pipe will generally lead to increased propellant tank outage for most cryogenic liquid rocket flow systems. The liquid propellant in a saturated, or near saturated condition, is definitely undesirable in most cryogenic flow systems inasmuch as it generally leads to propellant cavitation problems.

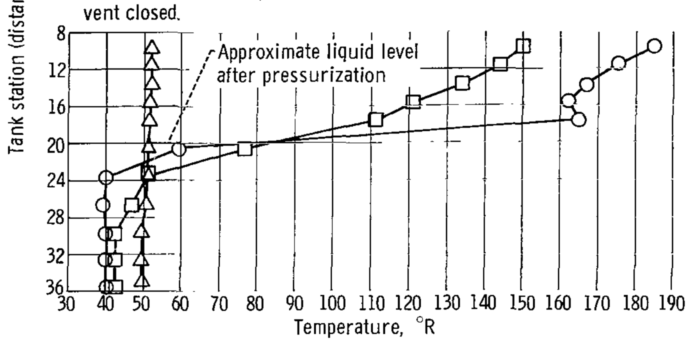
The second limitation was the tank pressure dropoff encountered at the beginning of the expulsion period for expulsion times of less than 280 seconds. This dropoff will be discussed later in this section.

Mass transfer during expulsion period for all injectors. - For all but one of the test runs, the type of net mass transfer encountered during expulsion was condensation (table I, col. 17). For the first six basic injector geometries and the 3/4-inch-diameter straight pipe, net mass transfer ranged from a high of 0.65 pound condensed to a low of 0.09 pound evaporated. For the 1/2-inch-diameter straight pipe the high value of pressurant mass condensed was 1.94 pounds; the low value was 0.28 pound.

Tank pressure dropoff with 1/2-inch-diameter straight pipe injector. - Data for the 1/2-inch-diameter straight pipe injector were not attainable for expulsion times lower than 280 seconds because of a tank pressure dropoff encountered at the beginning of the expulsion period. Tank pressure dropoffs from the nominal operating pressure of 160 pounds per square inch absolute to values as low as 40 pounds per square inch absolute were observed before the recovery of pressure back to the original operating level could be achieved by the control system. In order to obtain qualitative data as to the reason for these tank pressure dropoffs, viewports were added to the lid of the test tank to facilitate use of a movie camera. Photographic records were made of the tank ullage volume during the initial ramp period, the hold period, and the beginning of expulsion for several pressurization rates and liquid outflow rate settings. For rapid pressurization of the tank (i.e., tank pressure rise rates greater than approximately 2 psi/sec) the photographs showed considerable mixing in the ullage (i.e., pronounced liquid droplet splashing through the visible ullage volume). Ullage gas temperature profiles immedi-



(a) Profiles before ramp. Tank pressure, 0 pound per square inch gage; vent closed.



(b) Profiles after ramp. Tank pressure, 157±1 pounds per square inch absolute; gas inlet temperature, 516±10° R.

Figure 19. - Ullage gas vertical temperature profiles for several pressurization rates using 1/2-inch-diameter straight pipe injector.

iately before and after the ramp period for these runs may be seen in figure 19. It can be seen that the ullage gas temperatures at the end of the ramp period were lower for the greater pressurization rates, or similarly, as mixing in the initial ullage became more pronounced. During the hold period little pressurant gas was used; the velocity of the incoming pressurant was negligible compared to that during the initial pressurization period, and hence, little or no mixing was visible. However, once the expulsion period began the demand for pressurant gas was increased and again violent mixing in the ullage occurred. The net result was that ullage gas temperatures dropped, inlet pressurant gas velocity increased and more liquid splashing was encountered. This process ultimately ended when the ullage gas was at approximately saturation temperature corresponding to the operating pressure level of the tank.

No reliable pressurization gas flow rate data were obtained during the tank pressure dropoff period due to the high gas flow rates encountered. As a result of the qualitative photographic and quantitative temperature data obtained, it can be stated that liquid penetration by the incoming pressurant, as well as excessive cooling of the ullage gas due to the violent mixing, were definitely occurring. This resulted in a rapid reduction in tank pressure.

Comparison of Analytical and Experimental Results for Expulsion Period

In order to obtain a direct comparison between analytical predictions and experimental results, the tank pressurization analysis described in reference 1 was used to predict (1) the heat transferred to the tank wall as well as (2) the quantity of pressurant gas required for three of the sets of experimental runs. The injectors under consideration were the multiple screen, the cone, and the 1-inch-diameter straight pipe. It is to be noted that the multiple screen and the 1-inch-diameter straight pipe represented the best and worst agreement, respectively, with the following basic assumption stated in reference 1: "The ullage gas velocity is everywhere parallel to the tank axis and does not vary radially or circumferentially."

Input data for the computer calculations were supplied from actual experimental conditions (e.g., (1) ullage gas and tank wall temperature profiles prior to expulsion, (2) initial liquid level, (3) inlet gas temperature, tank pressure, outflow rate, and liquid surface temperature as a function of expulsion time, (4) total expulsion time, and (5) geometry of the propellant tank and the physical properties of its material of construction). In addition, the option of using the heat transfer coefficient calculation routine was employed.

The values of heat transferred to the tank wall and the pressurant gas required, obtained both analytically and experimentally, are plotted in figure 20 as a function of the total expulsion time (tabulated values appear in table III, p. 26). For the multiple screen injector the predicted values of heat transferred to the tank wall were from 0 to 9 percent less than those obtained experimentally. For the cone injector the predicted values were also less than those experimentally obtained ranging from 2 to 7 percent lower. However, for the straight pipe injector the predicted values of heat transferred were from 18 to 71 percent greater than those experimentally measured.

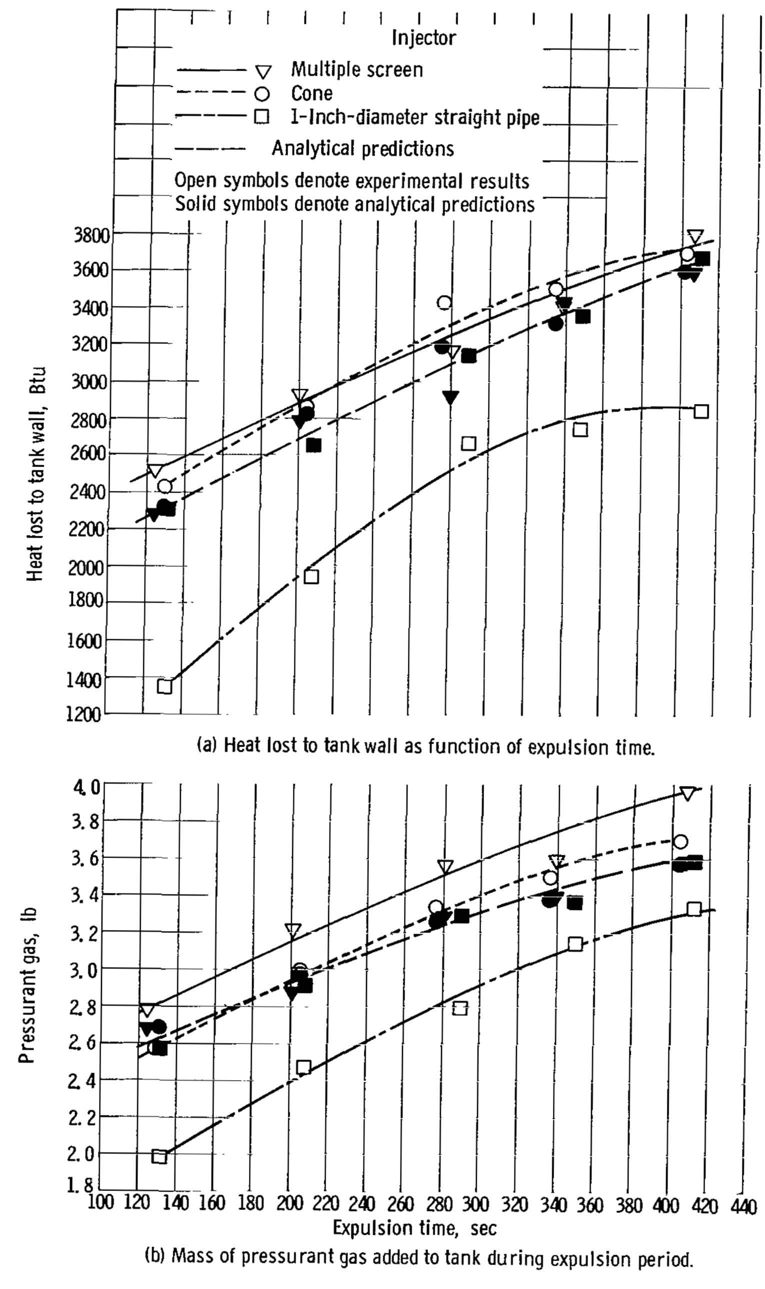
Correspondingly, for the multiple screen injector the predicted quantities of pressurant gas required were from 3 to 10 percent less than actually used; for the cone injector the analysis predicted from 4.3 percent more to 4 percent less than required; for the straight pipe injector predicted requirements were from 29 to 8 percent greater than experimental results.

It is to be noted that for all three injectors the values of heat transferred to the tank wall as predicted by the analysis are approximated by a single curve. The same fact is true for the analytically predicted pressurant-gas requirements.

The comparison between the analytical and experimental results indicates that the analytical program and assumptions are adequate to allow prediction of pressurant gas requirements when using a diffuser-type injector. Conversely, the program assumptions are not valid if the analysis is used for prediction of gas requirements when employing a straight pipe injector. Finally, an altered analysis that would incorporate a mixing theory to account for the radial and vertical temperature gradients in the tank ullage would be required for prediction of the pressurant gas requirements when using straight pipe injectors.

Experimental Evaluation of Floating Thermal Barrier

As stated in the INTRODUCTION, it was surmized that for certain injectors which directed the pressurant gas towards the liquid surface an insulator or floating thermal



(a) Heat lost to tank wall as function of expulsion time.

(b) Mass of pressurant gas added to tank during expulsion period.

Figure 20. - Comparison of analytical program predictions and experimental results.

barrier at the gas-liquid interface would reduce the overall gas consumption by reducing the gas to liquid heat transfer.

Employing the hemisphere injector, tests were conducted using a layer approximately 1 inch thick of preexpanded polystyrene beads (1/8 to 3/8 inch in diameter) as a floating thermal barrier. Propellant tank operating pressure was limited to a nominal value of 60 pounds per square inch absolute. At pressures greater than 60 pounds per square inch absolute the closed cell structure within the beads tended to collapse resulting in an effective bead density greater than that of the liquid hydrogen propellant.

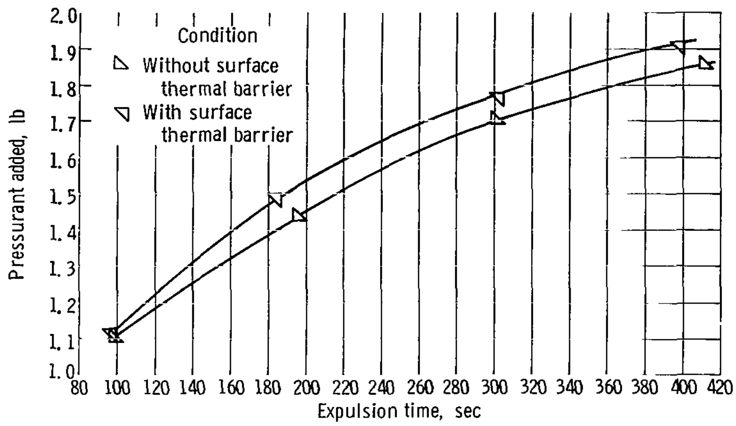


Figure 21. - Comparison of pressurant required during expulsion with and without surface thermal barrier. Tank pressure, ~60 pounds per square inch absolute; inlet temperature, 537±12 °R.

TABLE IV. - MASS BALANCE FOR FLOATING THERMAL BARRIER TESTS

Run	(1) Injector	(2) Tank pressure, psia	(3) Volume discharge, cu ft	(4) Inlet gas temperature, °R	(5) Expulsion time, sec	(6) Ullage mass after hold, m_2, lb	(7) Mass added during expulsion, $m_{a,2-3}$, lb	(8) Mass transferred during expulsion, $m_{T,2-3}$, lb	(9) Final ullage mass, m_3, lb
80	Hemisphere	54.9	20.36	542	98.0	0.278±0.002	1.101±0.006	0.177±0.011	1.202±0.009
71	without	56.9	20.27	549	194.8	.379± .005	1.433± .019	.083± .026	[a]1.729
73	beads	58.0	20.31	548	300.9	.332± .004	1.700± .016	.058± .026	[a]1.974
78		58.2	20.22	543	411.1	.271± .002	1.856± .014	.047± .024	2.080± .019
85	Hemisphere	54.6	20.21	527	98.1	0.409±0.008	1.116±0.007	0.083±0.018	[a]1.442
87	with beads	58.4	20.31	529	185.1	.312± .002	1.489± .009	.084± .019	[a]1.717
89		59.2	20.44	530	303.1	.292± .003	1.765± .021	.034± .029	[a]1.923
91		59.5	20.38	528	398.3	.310± .003	1.908± .013	.063± .026	[a]2.155

[a]1.0 percent error was assumed on these values when calculating probable error in $m_{T,2-3}$ (i.e., col. 8).

The actual pressurant gas required as a function of expulsion time is shown in figure 21 for a set of runs made using the floating barrier and a duplicate set employing no barrier (tabulated data appear in table IV). The addition of the floating thermal barrier resulted in an increase in pressurant-gas requirements of from 1 to 5 percent as compared to the expulsions made employing no barrier. No explanation is advanced for this phenomena.

SUMMARY OF RESULTS

Tank pressurization and propellant expulsion tests were conducted using a 29-cubic-foot vacuum jacketed cylindrical tank (length to diameter ratio $L/D \cong 3$) constructed of 5/16-inch-thick 304 stainless steel plate. Initial tank ullage values for all tests were between 17.5 and 21.6 percent. The majority of tank expulsions were conducted at a nominal pressure of 160 pounds per square inch absolute; expulsion tests employing a floating thermal barrier were conducted at a nominal pressure of 60 pounds per square inch absolute. Pressurant-gas inlet temperatures for all tests were between 508° and 558° R. The results of data obtained for six injector geometries (cone, hemisphere, disk, radial, multiple screen, and straight pipe) are as follows:

1. The first five injector geometries evaluated were designs that basically tended to diffuse the pressurant gas in a uniform pattern throughout the ullage volume. The pressurant gas requirements for the expulsion of the propellant when using these injectors were a maximum for the multiple screen injector and a minimum for the cone. Gas requirements for the cone were from 8.9 percent less for a 130-second expulsion to 6.9 percent less for a 400-second one than those for the multiple screen injector.

2. The 1-inch-diameter straight pipe, which injected the pressurant gas in a concentrated stream towards the liquid surface, showed a significant decrease in pressurant gas required during the expulsion period. Reductions ranging from 29.7 to 16.8 percent below the quantities required by the multiple screen injector were measured. The ullage mass prior to beginning expulsion was greater, and the quantity of heat lost to the tank walls was less with the straight pipe injector as compared to the multiple screen (or diffuser-type) injector. The reduced pressurant requirements measured when using the straight pipe injector as opposed to the multiple screen injector were attributed to reduced tank wall heating rates; however, it is believed that the increased initial ullage mass encountered with the straight pipe (due to the initial vaporization of liquid hydrogen by direct impingement of the pressurant gas on the liquid interface) may also have contributed to the reduced pressurant requirements during expulsion.

3. Reduction of the diameter of the straight pipe injector resulted in additional reductions of pressurant gas requirements. However, there was a limit in pipe size inas-

much as disruption of the liquid-gas interface and gas mixing (at the resulting higher pressurant gas injection velocities) led to tank pressure dropoffs at the beginning of the expulsion period.

4. Further, reduction of the diameter of the straight pipe injector led to an observed increase in the heat gained by the bulk liquid. This was observed even for the expulsions that did not have a tank pressure dropoff.

5. The experimental results for the multiple screen, cone, and 1-inch-diameter straight pipe injectors were compared with results predicted by the analysis of NASA TN D-2585. This comparison indicated that good agreement existed between the analysis and experimental data for the diffuser-type injectors and also that an altered analysis - one which would incorporate a mixing theory to account for the radial and axial temperature gradients in the tank ullage - would be required for prediction of pressurant gas requirements of the straight pipe injectors tested.

6. The results of expulsions conducted both with and without a floating layer of pre-expanded polystyrene beads showed that the insulation had little effect on overall gas requirements. In fact, when employing the floating insulation, pressurant gas requirements were increased slightly.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, December 30, 1965.

APPENDIX - SYMBOLS

C	orifice coefficient of discharge
c	specific heat of tank wall and lid, Btu/(lb)(^{O}R)
D	diameter, in.
H	enthalpy, Btu
h	specific enthalpy, Btu/lb
i	index (1 to 7)
K	conversion factor, 0.525 $ft^{3/2}$/(in.)(sec)
L	length
m	mass, lb
P	tank pressure, lb/sq ft
ΔP	differential pressure, lb/sq in.
Q	heat, Btu
Q_L	total energy lost by system, Btu
T	temperature, ^{O}R
t	time, sec
U	total internal energy, Btu
u	specific internal energy, Btu/lb
V	velocity, ft/sec
	ullage volume, cu ft
v	specific volume, cu ft/lb
W	work, P dv, Btu
y	net expansion factor for compressible flow through orifices
Z	potential height above reference, ft
ρ	density, lb/cu ft
δ	finite increment

Subscripts:

a	added to system
b	bulk liquid
c	cone injector
f	final time
fg	heat of vaporization
i	initial time
i-f	initial to final time interval
ℓ	leaving system
s	sensible heat
T	transferred
w	wall and/or lid
0	prior to ramp period
1	completion of ramp (or prior to hold period)
2	end of hold (or prior to expulsion period)
3	completion of expulsion

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