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Analysis of the Nonvented Fill of a 4.96-Cubic-Meter Lightweight Liquid Hydrogen Tank

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ANALYSIS OF THE NONVENTED FILL OF A 4.96-CUBIC-METER LIGHTWEIGHT LIQUID HYDROGEN TANK

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ABSTRACT

As part of its development of cryogenic fluid management techniques for spacecraft, the NASA Lewis Research Center Cryogenic Fluid Technology Office (CFTO) is planning to perform ground tests of nonvented fill techniques on a 4.96 cubic meter light weight liquid hydrogen tank. This tank is similar in size and shape to the tankage planned for CFTO's COLD-SAT liquid hydrogen flight experiment. This paper presents the analyses used to select two injection systems for nonvented fills of this tank at design flow rates between 220 and 450 kg/hr. The first system uses multiple nozzles spraying from the top of the tank through the ullage space. This system should be capable of liquid fill levels in excess of 95 percent. The second system injects the liquid through a submerged nozzle and should produce fill levels on the order of 80 percent liquid.

NOMENCLATURE

a	tank semi-major axis
A	area
b	tank semi-minor axis
C_0	discharge coefficient
C_p	specific heat
D	diameter
G	volumetric flow rate
h	enthalpy
$\bar{h}$	heat transfer coefficient
k	thermal conductivity
L	characteristic length
M	mass
$\dot{M}$	mass flow rate
n	number of drops in spray
Nu	Nusselt Number
R	radius
t	flight time
Re	Reynolds Number
u	internal energy
v	velocity
V	volume
$\dot{w}$	rate of work
x	liquid fill height
α	thermal diffusivity
λ	scale factor
μ	viscosity
ρ	density
σ	surface tension

<u>Subscripts</u>

cond	condensation
fullscale	prototype OTV value
gas	gaseous property
H2	hydrogen property

in	inflow property
inf	interface
lg	between liquid and gas
liq	liquid property
model	value for tank being scaled to (RPM tank)
sat	saturation property
sgas	saturated gas property
system	system property
wall	tank wall property
water	water property

INTRODUCTION

On orbit transfer of cryogenic liquids is considered enabling to many future NASA missions, from space based orbital transfer vehicles (OTV) to manned mars exploration. The techniques required to transfer cryogens in low gravity are quite different from those used terrestrially. During a normal gravity fill a top vent is kept open to vent the vapor generated during the fill process thereby maintaining a low tank pressure. If the normal gravity technique is used on orbit, the uncertainty of liquid and vapor distributions in low gravity may result in dumping of large amounts of liquid overboard. The No-Vent Fill process is a methodology used to reduce fluid loss by allowing the tank vents to be kept closed while the tank is filling (Chato, 1988).

The procedure works as follows:
The tank wall is cooled to a temperature sufficient to remove most of the thermal energy from it. The tank is filled with a subcooled liquid so that the end state condition is also subcooled liquid. Sprays and Jets are used to mix the fluid and maintain conditions in the tank close to thermodynamic equilibrium. Providing the initial energy in the tanks is low enough, the equilibrium mixture keeps the tank pressure low so that the entire tank can be filled without opening the vent valve.

The concept of No-Vent Fill has been extensively analyzed (Merino et al, 1978; Willen et al, 1981), but very little testing has been done (see Fester et al, 1970 for liquid fluorine testing) and, to the authors knowledge, no liquid hydrogen data has been published. To obtain a more empirical understanding of the No-Vent Fill problem a series of liquid hydrogen experiments is to be conducted at the NASA Lewis, Plum Brook K-Site cryogenic vacuum chamber facility using an existing lightweight liquid hydrogen tank. This paper presents the rationale used to design the fill systems for the existing tankage and their predicted performance during the No-Vent Fill process.

SUMMARY OF THE EQUATIONS FOR THE NO-VENT FILL PROCESS

In his previous work (Chato, 1988) the author developed a model of the No-Vent Fill process. This model separates the fill process into two stages. The first stage is the "Liquid Flashing Stage". This stage is characterized by flashing and boiling and is modeled as an equilibrium energy balance between the hot tank wall and the incoming liquid flow. The equations for this phase are:

Wall energy balance (assuming the liquid inflow vaporizes on striking the wall but does not superheat)

$$-M_{wall}\frac{d(C_pT)}{dt} = \dot{M}_{in}(h_{sgas} - h_{in}) \quad (1)$$

Gas mass balance

$$\frac{dM_{gas}}{dt} = \dot{M}_{in} \quad (2)$$

Gas energy balance

$$M_{gas}\frac{du_{gas}}{dt} = \dot{M}_{in}(h_{sgas} - u_{gas}) \quad (3)$$

The second stage the "Vapor Condensation and Compression Stage" divides the tank into three nodes; gas, liquid, and interface. It then models the energy transport between them. The equations for this stage is as follows:

Gas mass balance

$$\frac{dM_{gas}}{dt} = -\dot{M}_{cond} \quad (4)$$

Gas energy balance (neglecting gas phase heat transfer)

$$M_{gas}\frac{du_{gas}}{dt} + \dot{M}_{cond}u_{gas} = \dot{M}_{cond}h_{gas} + \dot{w}_{lg} \quad (5)$$

Liquid mass balance

$$\frac{dM_{liq}}{dt} = \dot{M}_{in} + \dot{M}_{cond} \quad (6)$$

Liquid energy balance

$$M_{liq}\frac{du_{liq}}{dt} + u_{liq}\frac{dM_{liq}}{dt} + Q_{inf} + \dot{M}_{in}h_{in} + \dot{M}_{cond}h_{lg} = \dot{w}_{lg} \quad (7)$$

Interface mass balance (assuming an infinitely thin interface)

$$\frac{dM_{inf}}{dt} = 0 \quad (8)$$

Heat transport from bulk liquid to interface

$$Q_{inf} = \bar{h}A_{inf}(T_{sat} - T_{liq}) \quad (9)$$

Interface energy balance

$$\dot{M}_{cond} = \frac{\bar{h}A_{inf}(T_{sat} - T_{liq})}{(h_{gas} - h_{liq})} \quad (10)$$

Compression work

$$\dot{W}_{lg} = \frac{P_{gas}}{\rho_{liq}}(\dot{M}_{in} + \dot{M}_{cond}) \tag{11}$$

These equations are solved with a computer algorithm called NVFILL which uses a finite difference approximation of these equations. Refinements have been made to the NVFILL algorithm to model the test tank which will be discussed below.

EXPERIMENTAL HARDWARE

Test Tank

The test tank selected was one designed and built for use in the Research Propulsion Module (RPM) program conducted by Lewis Research Center in the early 70's (DeWitt and Boyle, 1977). The RPM liquid hydrogen tank is ellipsoidal with a 222.5 cm major diameter and a 1.2 to 1 major to minor axis ratio. The two ends are joined by a short 3.81 cm cylinder section. The tank is made of 2219 aluminum chemical milled to a nominal thickness of 2.21 mm. Thickened sections exist where required for manufacturability (mainly weld lands). The tank has a 71 cm access flange on the top. Tank weight is 149.35 kg. Tank volume is 4.96 m^3. The tank is covered with a blanket of 34 layers of multi-layer insulation (double aluminized mylar with silk net spacers) and is supported by 12 fiberglass epoxy struts. Thermal performance of the tank is documented in DeWitt and Boyle, 1977. Figure 1 shows the tank installed in its support structure. This tank has several features which make it desirable as a test bed for spacecraft technology: It is of the same lightweight chemical milled construction used in space flight tanks. It has a multi-layer insulation blanket with performance nearly identical to current insulation designs for Orbital Transfer Vehicles (OTV). In addition, the tank is similar in size and shape to the tankage planned for CFTO's COLD-SAT liquid hydrogen flight experiment. This will allow the CFTO to asses many of the problems expected to be encountered in the COLD-SAT spacecraft, even though many technologies will require flight test in low-gravity for their ultimate proof of concept.

Scaling

Work on Orbit Transfer Vehicles has resulted in the design of tankage for hydrogen with volumes of 425 m^3 and required inflow rates of 900 kg/hr. To make the testing of the 4.96 m^3 RPM tank applicable to OTV design, some scaling rationale must be used in selecting test inflow rates.

A geometrical scale factor can be defined assuming tank shape is the same for the OTV and RPM tank.

$$\lambda = \frac{R_{model}}{R_{full\ scale}} = \sqrt[3]{\frac{V_{model}}{V_{full\ scale}}} \tag{12}$$

For the RPM tank:

$$\lambda = 0.489 \tag{13}$$

Heat transfer correlations for the No-Vent Fill processes are not well developed, but a couple of scaling estimates can be made based on the assumed behavior of $\bar{h}$, the interfacial heat transfer coefficient. Defelice and Aydelott (1987) assuming a constant heat flux per unit area (and hence constant $\bar{h}$ with the same driving temperatures) give the scaling criteria for inflow as:

$$\dot{M}_{model} = \lambda^2 \dot{M}_{full\ scale} \tag{14}$$

This scaling is probably most applicable to highly mixed cases where the heat transfer coefficient asymptotically approaches some maximum value. As an alternate bound for cases where a large amount of stratification is evident an assumption of constant Nusselt numbers (Nu) seems more appropriate, where:

$$Nu = \frac{\bar{h}L}{K} \tag{15}$$

The ratio between the inflow and the condensation rate must remain constant at corresponding times in the fill process for thermodynamic similarity to be maintained, hence

$$\dot{M}_{in} \propto \dot{M}_{cond} \tag{16}$$

The characteristic length is proportional to the tank radius so:

$$L_{model} = \lambda L_{full\ scale} \tag{17}$$

For the same fluid the thermal conductivity K is constant. So:

$$Nu_{model} = Nu_{full\ scale} \tag{18}$$

Implies:

$$\bar{h}_{model} = \frac{1}{\lambda} \bar{h}_{full\ scale} \tag{19}$$

A_{inf} should be a function of tank geometry only (in the same flow regime) so:

$$A_{inf\ model} = \lambda^2 A_{inf\ full\ scale} \tag{20}$$

Substituting these relations into equation 10 for the model and the prototype and dividing through

$$\frac{\dot{M}_{in,model}}{\dot{M}_{in,fullscale}} = \frac{\bar{h}A_{model}}{\bar{h}A_{fullscale}} = \lambda \tag{21}$$

using equations 14 and 21 yield a test tank mass flow rate between 217 kg/hr and 444 kg/hr.

Spray System Selection

Two flow systems were selected for testing at K-Site. It was the objective of the spray system to bound the No-Vent Fill problem by selecting two spray systems which represent the best and the worst conditions in zero gravity. Current concepts (Chato, 1988; DeFelice and Aydelott, 1987; Merino et al, 1978) of space No-Vent Fill systems use one or more pressure atomizing spray nozzles to inject the liquid inflow as a stream of droplets through the ullage, hence, promoting condensation of the ullage gas on the droplet stream. As the tank fills, these nozzles will submerge and it is expected that the outflow will transform to a liquid jet within the bulk tank liquid. This jet will continue to promote condensation by using fluid mixing to transport colder liquid to the tank free surface. It is expected that the droplet spray will produce much higher condensation rates than the submerged jet due to the much higher surface area available for heat transfer.

Unfortunately, for the spacecraft designer, the location of the ullage bubble in zero-gravity is highly uncertain so prediction of the condition under which the spray nozzles will submerge is difficult. The selected systems for test bound the problem as follows. One is a spray system with a single spray nozzle at the bottom of the tank. This represents the worst case since it will flood soon after liquid begins to accumulate in the tank. The other spray system uses an array of 13 spray nozzles spraying from the top of the tank (13 was selected due to the availability of a commercial spray manifold with this configuration). These nozzles are located in a position such that the spray nozzles are not submerged even when the tank is 95% full of hydrogen (95% is the target fill level for OTV operation). The flow capacities of each system are sized so that (as closely as possible within the constraints of commercially available sizes) the two different flow systems have the same inflow rate for the same system pressure. Figure 2 shows the position of each system in the Tank.

The test rig is designed to operate as a blow down system with an average delta pressure of 68 KPa. The flow capacity of each system was sized to provide near the higher required inflow of 444 kg/hr liquid hydrogen at a 68 kPa pressure drop. The system is designed to operate as a pressurized transfer at a constant transfer line pressure of 207 KPa with tank pressures from vacuum to near line pressure so variable flow rates from 45 kg/hr to 771 kg/hr must be accommodated. Flow capacities for commercial nozzles are normally given in gallons per minute of water so conversion of the required hydrogen flow rate is necessary.

The continuity equation for nozzles can be used to convert a mass flow rate into a required flow velocity by the following equation;

$$v = \frac{\dot{M}}{\rho A} \tag{23}$$

For an incompressible liquid the flow velocity through a nozzle can be calculated with the following:

$$v = C_D \sqrt{\frac{2\Delta P}{\rho}} \tag{24}$$

Dombrowski and Wolfson (1972) indicate that for pressure atomizing nozzles C_D is fairly constant over a wide range of flow conditions for a specific nozzle so C_D will be assumed to be constant with flow rate and the same whether the flow is hydrogen or water. The volumetric flow rate is defined as:

$$G = Av \tag{25}$$

Using equations 23 and 25 the required G for hydrogen is

$$G_{H2} = \frac{444\ \text{kg/hr}}{70.8\ \text{kg/m}^3} = 6.26\ \text{m}^3\text{/hr} = 27.6\ \text{gpm} \tag{26}$$

Using equations 24 and 26 for the same nozzle pressure drop and size

$$\frac{G_{H2}}{G_{water}} = \sqrt{\frac{\rho_{water}}{\rho_{H2}}} = 3.76 \tag{27}$$

So the design water flow rate at 68 KPa is 1.67 m^3/hr (7.34 gpm).

For the bottom spray a commercial full cone nozzle with a flow capacity of 1.89 m^3/hr (8.3 gpm) water at 68 KPa was selected (Spray Systems Catalog). This nozzle has a nominal orifice diameter of .9525 cm and a C_D of approximately 0.6. It is shown in Figure 3. For the top spray a manifold of 13 full cone nozzles each with a flow capacity of 0.114 m^3/hr (0.50 gpm) water at 68 KPa giving a total flow of 1.48 m^3/hr were selected (Spray Systems Catalog). These have a nominal orifice diameter of 0.20828 cm and a C_D of approximately 0.8. The manifold is shown in Figure 4.

ANALYTICAL MODELS

Heat Transfer Model for the Top Spray

To model the heat transfer from the droplets of the top spray, the correlation of Brown (1951) is used. Brown gives:

$$\frac{hD}{k} = 0.667\frac{D^2}{4\alpha t}\left[\frac{1 - \frac{6}{\pi}\sum_{n=1}^{\infty}\frac{1}{n^2}e^{-(4n^2\pi^2\alpha t/D^2)}}{\frac{6D^2}{4\pi\alpha t}\left[\frac{\pi^4}{90} - \sum_{n=1}^{\infty}\frac{1}{n^4}e^{-(4n^2\pi^2\alpha t/D^2)}\right]}\right] \tag{28}$$

The spray is approximated as a monodisperse spray with a droplet diameter the same as the volume mean diameter of the spray. The manufacturers mean diameter information for water is corrected to hydrogen with the correlation of Steinmeyer (1973);

$$\frac{D_{system}}{D_{water}} = \left[\frac{\sigma_{system}}{\sigma_{water}}\right]^{0.5} \left[\frac{\mu_{system}}{\mu_{water}}\right]^{0.2} \left[\frac{\rho_{water}}{\rho_{system}}\right]^{0.3} \quad (29)$$

For hydrogen

$$\frac{D_{H2}}{D_{water}} = 0.259 \quad (30)$$

The droplet residence time is calculated as follows:

$$t = L/v \quad (31)$$

L is estimated as the centerline distance from the spray nozzle to the tank free surface. A maximum L is the distance from the spray nozzle to the tank wall. The maximum L (measured off a scale drawing of the tank) was combined with a tank fill geometry function to give the L values as the tank fills. These functions are listed in Table 1.

Table 1 Top Spray Length Functions

Top 6 nozzles L = 22.86 cm

Middle 6 nozzles L = 2×(158 cm - x),115 cm max

Centerline Nozzle L = 156 cm - x

where x is the liquid fill height

The heat transfer area is determined by calculating the number of drops in residence from the volumetric inflow rate, mean drop diameter, and flight time.

$$n = \frac{6Gt}{\pi D^3} \quad (32)$$

By multiplying by the surface area per drop;

$$A_{inf} = \frac{n\pi^2 D^2}{4} \quad (33)$$

<u>Heat Transfer from the Bottom Jet</u>

Dominik (1984) proposes for one-g transfers the following correlation for submerged jets based on rates established for jets impinging on a flat plate.

$$\frac{\bar{h}D}{K} = 0.205\ (Re)^{0.731}\ Pr^{1/3} \quad (34)$$

Where the reference diameter D is the diameter of the jet at the free surface. The angle of spread used to calculate this diameter is based on the work of Idelchik (1986), who gives spread angle as 15°. The velocity is also measured at the free surface. This is estimated as the volume mean jet velocity and is given as;

$$\frac{v}{v_o} = \frac{0.46}{\frac{0.16}{D_o} + 0.29} \quad (35)$$

The free surface area is given by the fill height x and a function of tank geometry as

$$A_{inf} = \pi a^2 \left(1 - \frac{(x - b)^2}{b} \right) \quad (36)$$

similarly fill volume as a function of fill height for the RPM tanks is

$$V_{liq} = \pi^2 a^2 x - \frac{\pi a^2 b}{3} - \frac{\pi a^2}{3b^2} (x - b)^3 \quad (37)$$

Where a = 111.25 cm and b = 92.71 cm for the RPM tank. In the actual computer algorithm this equation is iterated on to find the correct fill height given a known fill volume.

RESULTS

The NVFILL computer code was modified to include the heat transfer and area functions given in the previous sections. The solution algorithm is the same as original, except the equations for heat transfer coefficient and interface area (equations 28 and 33 for the top spray; equations 34 and 36 for the bottom jet) were used to generate values for these parameters which were then used to update equation 10 continuously. The code was run to simulate tests proposed for the RPM tank.

Figure 5 shows a comparison of pressurization rates of top spray fills at different initial tank wall temperatures. The first temperature 22.2 K is sufficiently low that almost no energy remains in the tank wall. The second temperature 77.8 K is the same as the vacuum chamber cold wall (hence easily obtained). The third temperature 100 K was picked to be about as warm a wall temperature one could select and still expect to fill the tank. As expected the pressure rises quicker at the higher wall temperatures. All three wall temperatures show acceptable pressure rise rates.

Figure 6 shows a comparison between the previous fill with a 22.2 K initial wall temperature and a bottom jet fill with the same initial wall temperature. The bottom fill is much less effective and must be terminated at only 80% full. Initially the bottom jet does a better job of filling by preventing stratification. As the fill proceeds, the heat transfer rate and available surface area decreases causing rapid pressure rise. The increasing pressure causes the flow rate to drop, decreasing the heat transfer rate, and hence even further accelerating the rate of pressure rise. Bottom jet fills at higher wall temperature were not investigated due to the poor performance of the bottom jet fill at even the lowest initial wall temperature.

Several analyses were run at a lower supply pressure in the expectation that the reduced inflow rates obtained would improve the no-vent fill process. Figure 7 shows the effect of supply pressure on the fill process for the top spray system (with an initial wall temperature of 22.2 K). Figure 8 shows the same comparison for the bottom fill. Surprisingly the reduced pressure did not have much effect on the final fill state.

CONCLUDING REMARKS

A series of tests for a 2.2 m diameter, 4.96 m^3 liquid hydrogen tank has been proposed to study the No-Vent Fill process. Analytical models to estimate No-Vent Fill performance have been derived for two spray systems. One system uses an array of 13 nozzles spraying through the ullage space. One system uses a single nozzle spraying up from the tank bottom which quickly floods producing a submerged jet. The veracity of these models will be demonstrated by a series of experimental tests planned for the summer of 1989. The results of experiments will indicate strengths and weaknesses of the proposed models and suggested areas for refinement. The continuing refinement of these models will lead to models which can be verified by the minimum amount of in-space testing which will enable the analysis of in-space operational systems.

Some areas currently under consideration for refinement include improved modelling of stratification, better simulation of bulk boiling, improved gas side heat transfer, and the addition of external heat leak effects.

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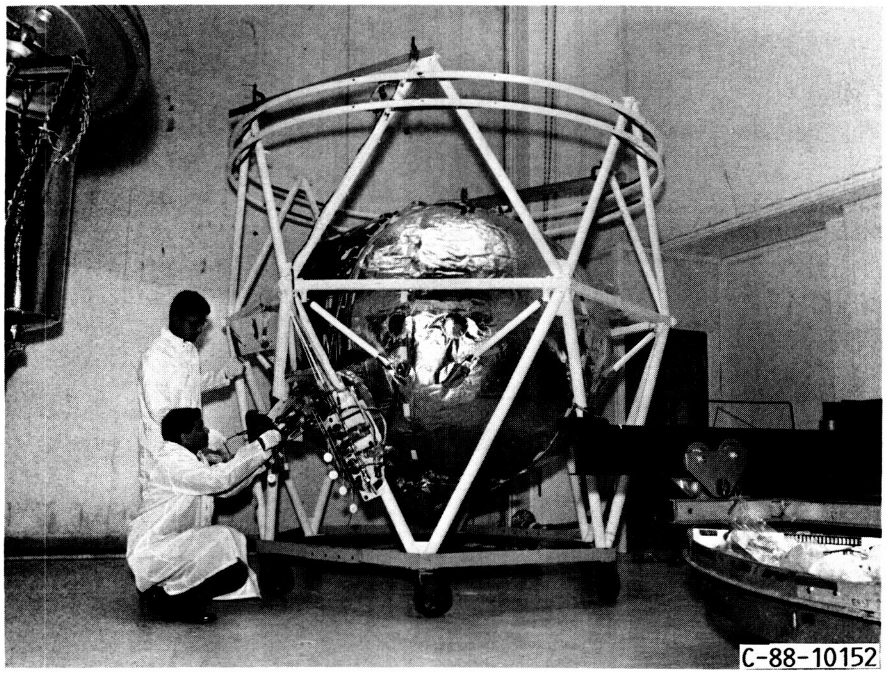

FIGURE 1. - 4.96 CU M TANK IN SUPPORT STRUCTURE.

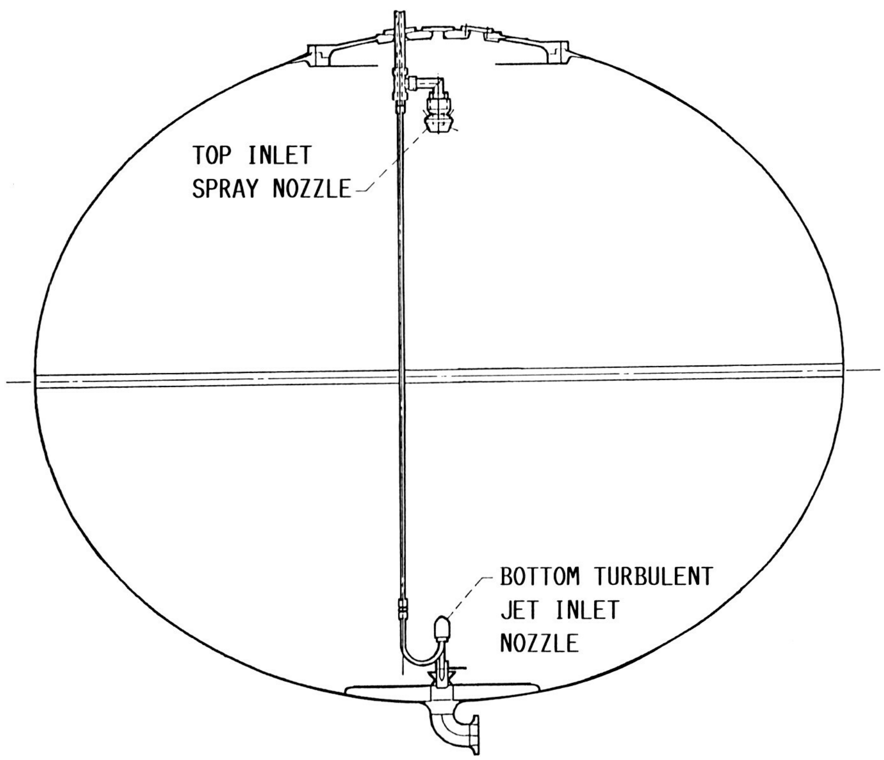

FIGURE 2. - 496 CU M TANK INTERVALS.

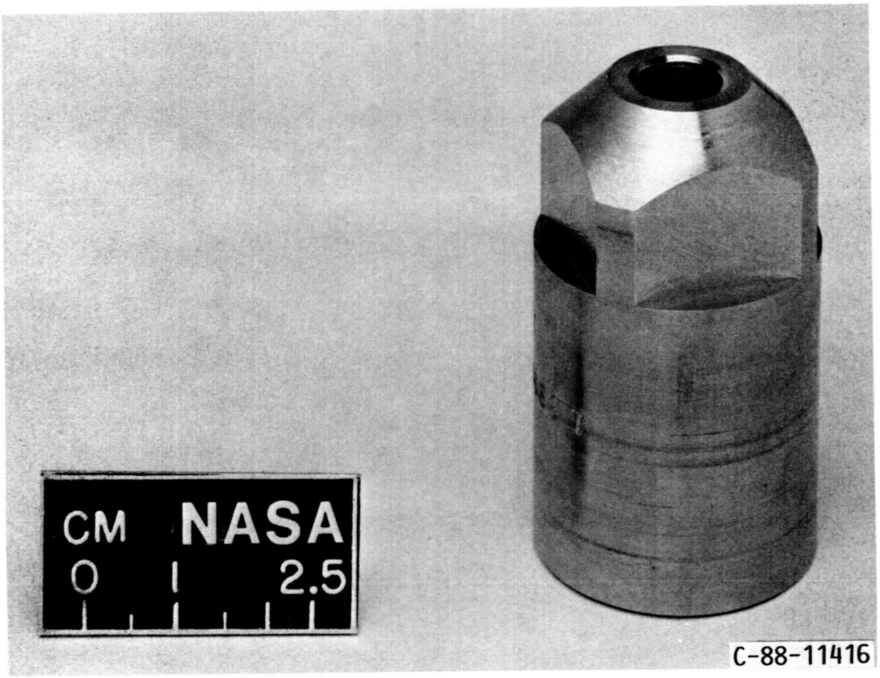

FIGURE 3. - BOTTOM JET NOZZLE.

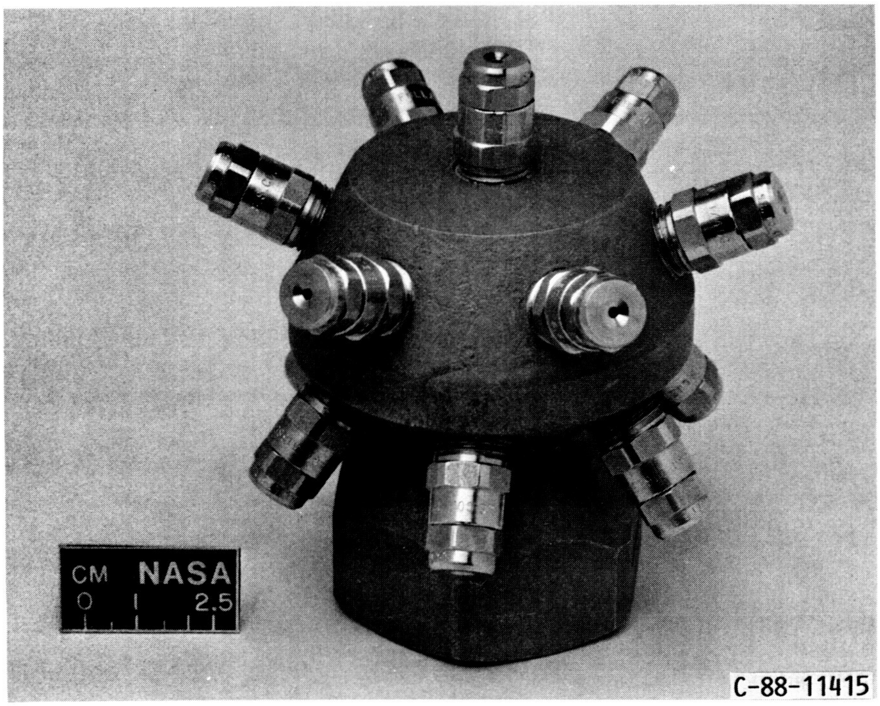

FIGURE 4. - 13 NOZZLE TOP SPRAY.

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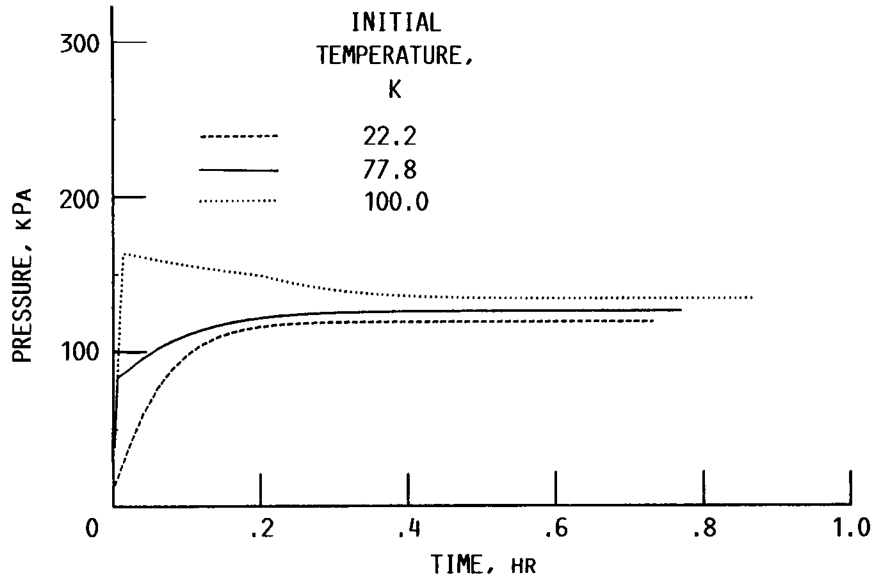

FIGURE 5. - 13 NOZZLE TOP SPRAY PREDICTED PERFORMANCE. EFFECT OF INITIAL TANK WALL TEMPERATURE SUPPLY PRESSURE, 207 kPA; INFLOW TEMPERATURE, 20.3 K, TANK VOLUME 4.96 CU M; MASS TO VOLUME RATIO, 30 KG/CU M.

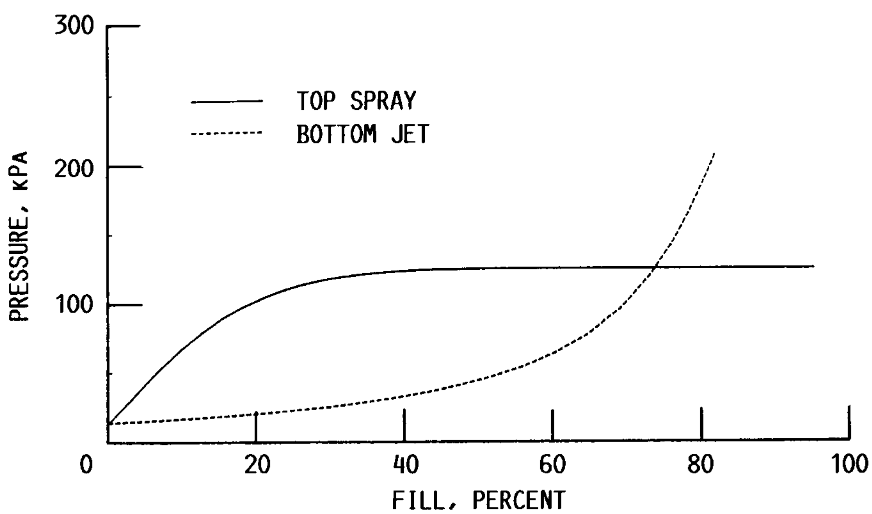

FIGURE 6. - COMPARISON OF INJECTION TECHNIQUES. SUPPLY PRESSURE = 207 kPA; INITIAL TEMPERATURE = 22.2 K, INFLOW TEMPERATURE, 20.3 K, TANK VOLUME, 4.96 CU M; MASS TO VOLUME RATIO, 30 KG/CU M.

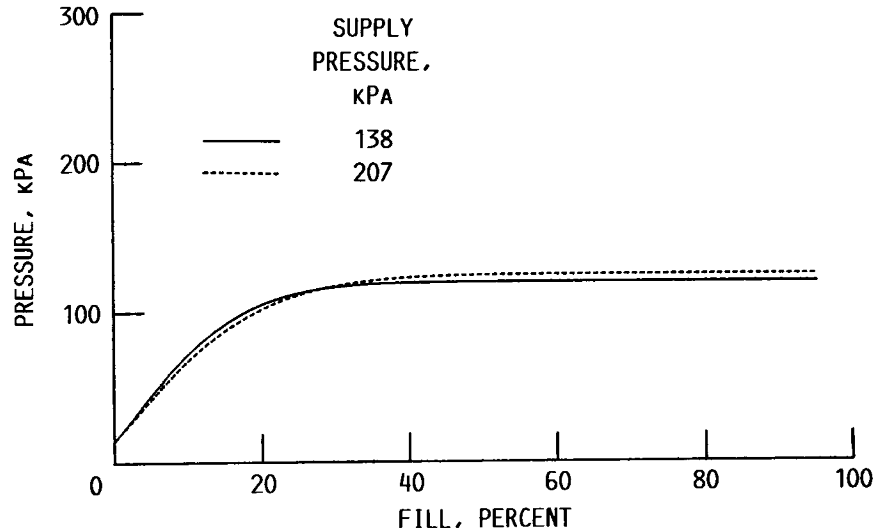

FIGURE 7. - 13 NOZZLE TOP SPRAY PREDICTED PERFORMANCE. EFFECT OF SUPPLY PRESSURE, INITIAL TEMPERATURE, 22.2 K; INFLOW TEMPERATURE, 20.3 K, TANK VOLUME, 4.96 CU M; MASS TO VOLUME RATIO, 30 KG/CU M.

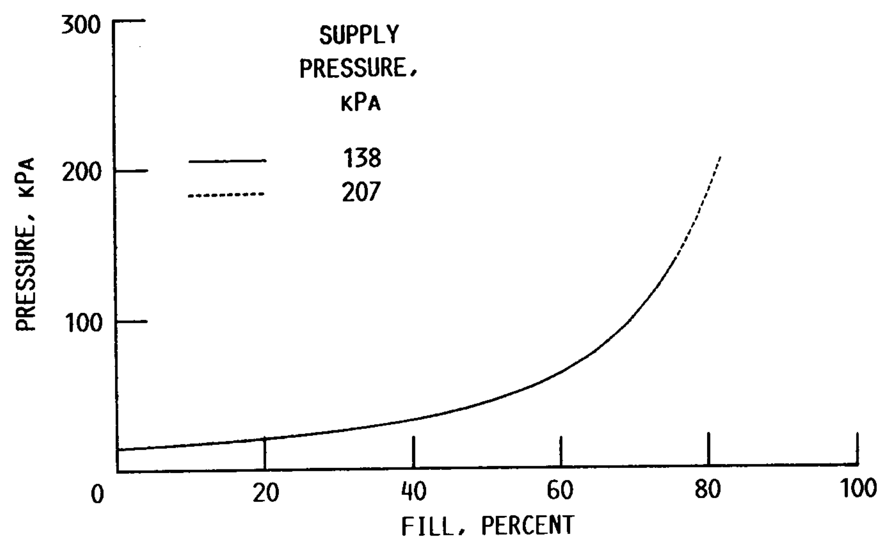

FIGURE 8. - BOTTOM JET PREDICTED PERFORMANCE. EFFECT OF SUPPLY PRESSURE, INITIAL TEMPERATURE 22.2 K; INFLOW TEMPERATURE, 20.3 K, TANK VOLUME, 4.96 CU M; MASS TO VOLUME RATIO, 30 KG/CU M.

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