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Measurements of Jet-Induced Pressure Decay in a Thermally Stratified Tank

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Forced mixing may be required for controlling pressure during in-space storage and transfer of cryogenic propellants and life support fluids. To provide applicable data on jet-induced mixing, a 1-*g* experiment was conducted in which an axial jet mixer promoted condensation of the ullage and lowered the pressure in a subscale cylindrical tank partially filled with thermally stratified Freon 11. The experiment objectives were to 1) correlate pressure reduction times with experiment parameters, 2) compare the pressure reduction times to mixing times determined by previous investigations using acid-base neutralization and dye dispersion, and 3) determine the transient condensation and heat transfer rates at the liquid surface as a function of jet parameters. Pressure reduction times were correlated with the rate of jet momentum addition to the bulk fluid. Dimensionless pressure reduction times agreed with comparable acid-base neutralization data, but not with dye dispersion data. The variation of interface heat transfer coefficient with changes in jet Reynolds number was consistent with an existing correlation based on measurements of steady-state rates of mixing-induced condensation of steam.

Nomenclature

A = area of liquid surface, in.^{2}
D = tank diameter, in.
d = nozzle diameter, in.
h = heat transfer coefficient, Btu/h-ft^2-°R
h_{fg} = latent heat of condensation, Btu/lbm
k = thermal conductivity, Btu/h-ft-°R
M_u = ullage mass, lbm
m_c = condensation rate, lbm/s
P = pressure, psi
Q = volumetric flow rate, gal/min
q_{sb} = heat transfer rate, surface to bulk liquid, Btu/h
q_{us} = heat transfer rate, ullage to surface, Btu/h
Re_n = jet Reynolds number at nozzle
s = liquid depth from surface to nozzle, in.
T_b = bulk liquid temperature, °F
T_i = layer temperature in ullage, °F
T_s = surface temperature, °F
t = pressure reduction or mixing time, s
U = average jet velocity at nozzle exit, in./s
V_i = layer volume in ullage, in.^{3}
V_l = liquid volume, in.^{3}
V_t = tank volume, in.^{3}
Θ = pressure reduction time parameter
Θ_A, Θ_O = mixing time parameters
μ = dynamic viscosity, lbm-ft-s
ν = kinematic viscosity, s/ft^2
ρ = Fluid density, lbm/ft^3

Introduction

TECHNOLOGY for managing liquid propellants and other fluids in low gravity will be critical to the success of many future space systems. Spacecraft such as Space Station Freedom, space transfer vehicles, refuelable satellites, and military space platforms will require resupply systems for propellants, reactants, coolants, and life-support fluids. Many of these applications will use cryogenic liquids, adding thermodynamic requirements to the already complex fluid dynamic issues.

One difficulty in managing cryogens in low gravity is tank pressure control. Heat leak into cryogen storage tanks increases tank pressure. Also, during transfer of a cryogen from a storage tank to a receiver tank without venting, regulation of the pressure in the receiver tank is a critical function. In both cases, conventional venting to relieve excess pressure is precluded in zero-*g* conditions by the likelihood of liquid losses (and the resulting cost penalty) caused by venting liquid along with the vapor.

Mixing in Low Gravity

A tank of cryogenic liquid in space will be subjected to nonuniform and varying sources of heat leak. Because of the absence of gravity or acceleration forces, convection will be minimal and bubbles will form and grow at these sources. Also, the resulting temperature gradients can easily be more severe in low gravity than in normal gravity. Since the tank pressure is dictated by the vapor pressure of the warmest liquid, this will result in a much higher pressure than would occur if the tank fluids were well mixed. Either a refrigeration system or a thermodynamic vent system (TVS) may be used to control the pressure, but both depend on the distribution of their cooling effect throughout the tank contents. Some fluid motion may be induced by the "*g*-jitter" caused by thruster firings, payload operations, and crew movement, but these motions will be vehicle, mission, and time dependent and cannot be relied on to provide adequate mixing for pressure control during quiescent periods. Artificial means of mixing the warmer liquid near the liquid-vapor interface with cooler liquid elsewhere in the tank will increase the surface-to-liquid heat transfer rate and the associated condensation rate, thereby providing confidence that the pressure will be controlled. Figure 1 depicts a TVS utilizing an internal heat exchanger and an axial jet mixer.

Mixing is also important for on-orbit filling of cryogen tanks using the no-vent fill process. The time required for a no-vent fill depends on the rate at which the ullage can be compressed and condensed, which in turn depends on the rate at which the heat deposited by condensation is transferred away from the liquid-vapor interface. This heat transfer rate is

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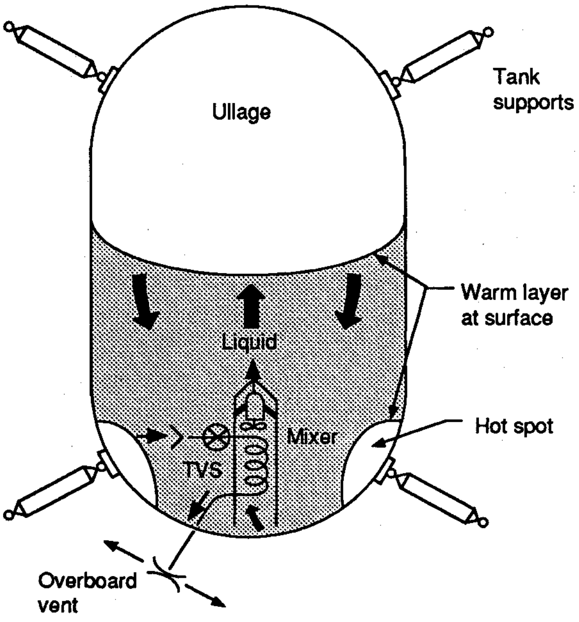

Fig. 1 **Cryogenic tank pressure control using a thermodynamic vent system with a mixer.**

small for a quiescent fluid in low gravity but is greatly enhanced by convection when fluid motion is present. During a no-vent fill, the incoming liquid can be introduced as an axial jet, which can provide the mixing necessary for inducing a high condensation rate.

Experiment Objectives

Data from a 1-*g* mixing experiment are not directly applicable to low gravity for two reasons. First, in 1 *g* even a relatively strong axial jet will not form a geyser, whereas in low gravity a less powerful jet can form a geyser that disrupts the ullage. Second, buoyancy, which can impede mixing induced by weak axial jets in 1 *g*, should not be a significant factor even for very weak jets in low gravity. However, a 1-*g* experiment provides an opportunity for detailed study of mixing and heat transfer processes that are closely related to those occurring in low-*g* mixing and pressure control.

This paper reports the results of an experimental investigation of jet-induced mixing and condensation of Freon 11 in a sealed tank in normal gravity. The objectives of this investigation were 1) to correlate pressure reduction times with the jet-mixer parameters, 2) to compare the pressure reduction times with mixing times determined by previous investigations, and 3) to determine transient heat transfer rates and condensation rates at the liquid surface as a function of jet mixer parameters. The results from this experiment will eventually be compared with data from the tank pressure control experiment,[1] which is a related low-*g* experiment that flew aboard Space Transportation System Flight STS-43 in August 1991.

Related Investigations

Several studies of fluid mixing and condensation in normal and low gravity have been conducted to identify and measure factors that will be important in designing on-orbit cryogenic fluid management systems. Dominick[2] measured steady-state heat transfer and condensation rates of Freon 11 and Freon 113 in 8- and 10-in.-diam tanks as a function of several jet parameters, including the angle of incidence of the jet to the liquid surface. He determined that an axial jet led to higher condensation rates than did jets with other orientations. Poth and Van Hook[3] measured thermal mixing times and bulk fluid motion in thermally stratified water. Among the findings was the fact that buoyancy effects can significantly impede the mixing process in normal gravity. Wollen[4] tested jet mixers in scale models of a Shuttle/Centaur hydrogen tank. These tests involved the neutralization of an acid by injecting a base into the jet mixer. Because these tests were isothermal, buoyancy effects were avoided. Bullard[5] tested a thermal conditioning unit for liquid hydrogen that employed a TVS and a jet mixer. In the only reported low-*g* studies, Aydelott[6] measured jet mixing of dye in ethanol during experiments conducted in the NASA 5-s drop tower. He was able to correlate both 1-*g* and low-*g* mixing times with a combination of the Weber and Bond numbers of the mixer jet.

Some of the mixing correlations developed in the referenced studies were derived from the results of earlier work by Okita and Oyama,[7] Fox and Gex,[8] and Fossett and Prosser,[9] who studied bulk mixing for industrial applications. One conclusion from these earlier investigations is that mixing time is approximately inversely proportional to the square root of the rate at which the mixer adds momentum to the fluid.

Thomas[10] and Sonin et al.[11] measured the steady-state rate of condensation of steam on water surfaces mixed by a submerged jet. Thomas found that the condensation rate depended on the distance from the nozzle to the liquid surface and the product of the nozzle diameter and the jet velocity at the nozzle. Sonin et al. developed a fundamental correlation between the condensation rate and the characteristic velocity of the turbulent liquid motion at the surface.

Questions remain regarding mixing and condensation in a closed tank in low gravity. Aydelott's tests provide low-*g* data, but the tests were short and the tanks subscale. Also, little of the existing data pertains to the effects of transient bulk-liquid mixing on heat and mass transfer across the liquid-vapor interface. The experiment discussed in this paper was designed to address this last issue in particular, as well as to provide some data to compare with dye dispersion and acid-base neutralization measurements.

Approach

Freon 11 was used to simulate, at room temperature, a cryogen at its boiling point. It boils at 75°F at atmospheric pressure; thus, we were able to use a glass tank and had to apply only a minimal amount of insulation. A recirculation pump external to the tank induced an axial jet directed vertically within the tank toward the liquid surface. At the start of each test, a thin circumferential heater coil just beneath the liquid surface heated the liquid and raised the pressure. Measurements of the tank pressure and temperatures provided histories of the pressure reduction and thermal destratification.

The experiment was run repeatedly with the tank at each of three different fill levels—30, 50, and 70%—and with several different mixer flow rates and several nozzle sizes. The tank pressure and the fluid temperature at numerous places in the liquid and vapor were the principal measurements.

Apparatus and Procedure

Experiment Apparatus

Figure 2 shows the experiment apparatus. The tank was a glass cylinder capped with hemispherical stainless steel domes. The jet mixer consisted of a pump, a flow meter, and a nozzle that protruded through the bottom of the tank. During operation, the Freon was withdrawn through a port in the annular region surrounding the nozzle, and then after passing through the pump, it was re-injected through the nozzle. The nozzles were sections of straight stainless steel tubing with inside diameters of 0.19, 0.28, 0.43, and 0.62 in. The outside diameters were 0.25, 0.375, 0.50, and 0.75 in., respectively. They projected 2 in. above the bottom of the tank and directed the mixer jet upward along the centerline of the tank.

Two thermocouple trees and a heater coil were suspended inside the tank by pressure-tight connectors in the upper tank dome. One of the thermocouple trees spanned the entire

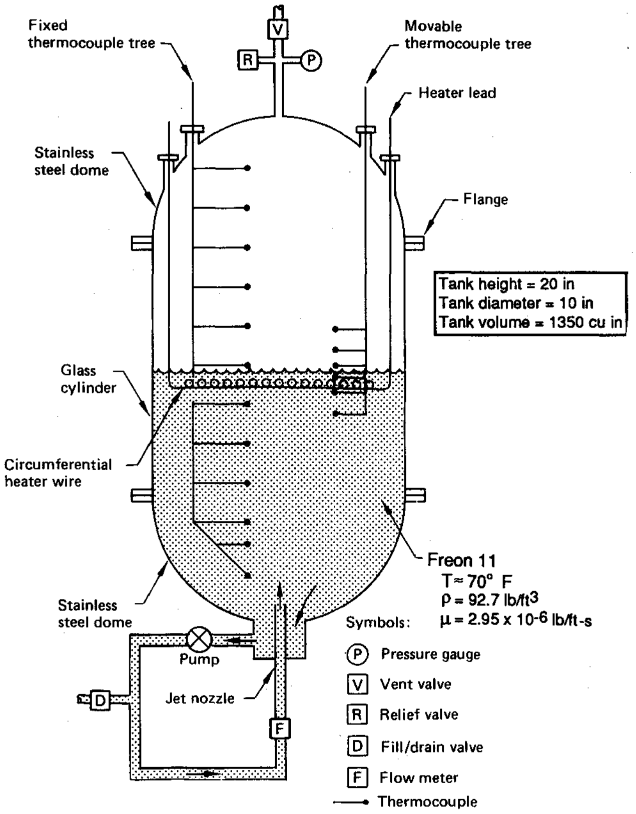

Fig. 2 Schematic of experiment apparatus.

Liquid fraction, percent	Flow rate, gpm					
	0.2	0.3	0.4	0.5	0.6	0.7
70	All	All	All	All	All	0.19" 0.28" 0.43"
50	All	All	All	All	All	0.19" 0.28" 0.43"
30	All	All	All	All	0.19" 0.28" 0.43"	None

Fig. 3 Matrix showing which of the four nozzles (0.19, 0.28, 0.43, and 0.62 in. in diameter) were tested at each flow rate and tank fill fraction.

height of the tank; it comprised 12 copper-constantan thermocouples spaced 1.5 in. apart. The second tree was a smaller, moveable one that consisted of six thermocouples located at heights of ¼, ½, and 1-⅛ in. above and below the center of the tree, which was always positioned at the liquid surface. The circumferential heater coil was similarly adjusted so that it remained ¼ in. below the liquid surface.

A variable speed gear pump provided the flow to the nozzle. The flow rate was held to prescribed levels by a feedback control loop comprising a flow meter, the experiment control computer, and a digitally controlled power supply for the pump.

The ullage pressure and the pressures upstream and downstream of the pump were measured using 0–25 psia pressure transducers. Test data were recorded by a Hewlett-Packard 3054A automatic data acquisition control system. This system recorded all temperatures and pressures, as well as the flow rate, pump power, and heater power. Forty channels of data were recorded, with a sampling interval of approximately 4 s.

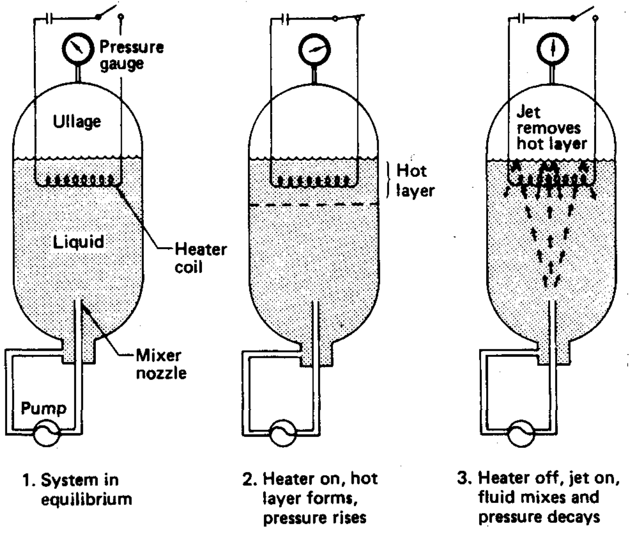

Fig. 4 Test sequence.

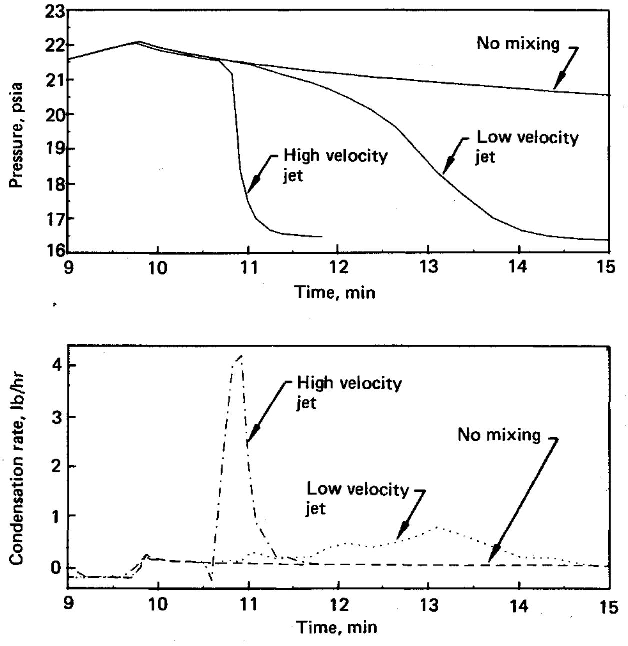

Fig. 5 Effect of jet-induced mixing on ullage pressure decay (top) and condensation rate (bottom).

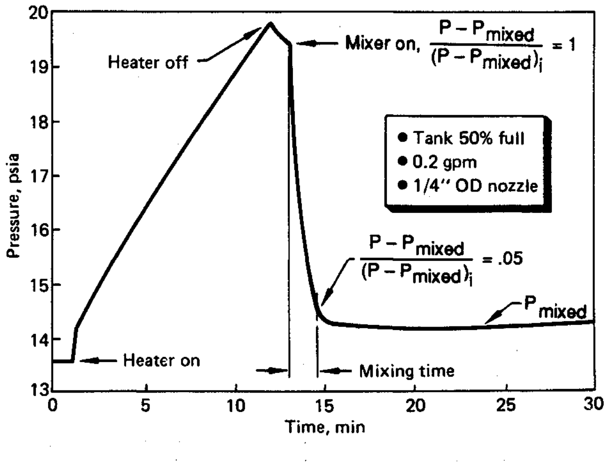

Fig. 6 Tank pressure data from a representative test run.

Test Procedure

The four nozzles were tested over the range of flow rates and tank fill fractions summarized in Fig. 3. The test procedure is depicted schematically in Fig. 4. Before the start of each test, all of the liquid and ullage temperatures were allowed to settle to within 1°F of each other. Then the heater coil was turned on to stratify the liquid and raise the tank pressure. The heating rate was 270 Btu/h (80 W), and the heater was left on until a total of 47 Btu (50 kJ) had been added to the fluid. After the heater was switched off, there was a 1-min delay before the mixer was turned on at the prescribed flow rate for that test. The delay allowed the effects due to abrupt heater shutoff to die out, which effectively isolated the pressure decay due to mixing from that due to transient heater effects. The mixer was operated continuously until the pressure had collapsed to its final value and the liquid was fully destratified.

Results

Figure 5 shows the condensation rate and ullage pressure as a function of time for three tests with different mixer conditions: 1) with a high-velocity mixer jet, 2) with a low-velocity mixer jet, and 3) with no mixing. The graph shows that even a small amount of mixing greatly increased the condensation rate and the pressure decay rate. With the high-velocity jet, the condensation rate rapidly rose to its peak value, whereas with the low-velocity jet there was a delay of several minutes before the condensation rate rose significantly. This delay was due to the time required by the low-velocity jet to penetrate the stratified liquid layer and cool the liquid-vapor interface. Buoyancy can be significant for some test conditions in a 1-*g* thermal destratification experiment such as this one, but it will be nearly absent in a low-*g* environment. We did not attempt to quantify the effects of buoyancy on pressure reduction time in these experiments but rather sought to test over a wide range of conditions so that for at least some of the tests buoyancy effects were negligible.

Dimensionless Pressure Reduction Time Correlations

The time required to reduce the tank pressure of a thermally stratified tank to near its mixed pressure can be used as a figure of merit for comparing different mixer configurations. We defined the pressure reduction time as the time required to reduce the excess tank pressure due to thermal stratification ($P - P_{\text{mixed}}$) by 95%. That is, the pressure reduction was completed when

$$\frac{P - P_{\text{mixed}}}{(P - P_{\text{mixed}})_{\text{initial}}} = 0.05$$

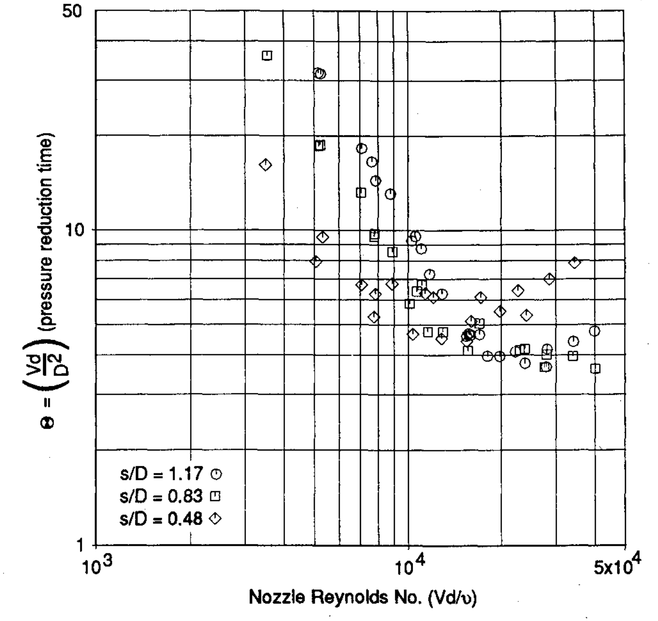

Fig. 7 Dimensionless pressure reduction time vs jet Reynolds number.

Physically, this is the time required for the mixer jet to sweep away from the liquid surface the initial stratified layer and the heat subsequently produced by ullage condensation. Figure 6 shows a representative time history of the tank pressure. The times at which the heater and mixer were turned on and off are indicated. Each test yielded a plot like this and similar ones for the temperatures.

Other studies have used dye dispersion or acid neutralization to determine the mixing time. Mixing time is similar to pressure reduction time but differs fundamentally in several ways. Since bulk mixing depends on circulating fluid throughout the entire liquid volume, bulk mixing times depend explicitly on liquid volume. It has also been shown in previous studies that bulk mixing rate is proportional to the square root of the rate of kinematic jet momentum addition, which is given by

$$(QU)^{1/2} = (\pi/4)^{1/2} Ud$$

These studies have also shown that, except for very low Reynolds numbers, bulk mixing times are independent of viscosity. In contrast to this, pressure reduction depends on jet-induced heat transfer rates at the liquid surface, which should depend on the Reynolds number (that is, it should not be independent of viscosity). Also, we do not expect pressure reduction times to depend on liquid volume to the same degree as bulk mixing times.

Using the dimensionless bulk mixing time parameter originally devised by Fossett and Prosser,[9] but replacing mixing time with pressure reduction time, we defined a dimensionless pressure reduction time parameter Θ as

$$\Theta = \frac{t\,Ud}{D^2}$$

Figure 7 shows our data plotted as Θ vs jet Reynolds number, which is defined by

$$Re_n = \frac{Ud}{\nu}$$

This plot shows that for inertia-dominated flow (in this case, $Re_n > 20 \times 10^3$) at higher fill levels the dimensionless pressure reduction times are less than 5. Note that the transition between buoyancy-dominated and inertia-dominated flow at Re_n of 20×10^3 is not a general result but instead depends on the type of fluid and the degree of stratification. Correlations exist for quantifying the boundary between these two regimes more generally.

In Fig. 7 the data are grouped according to their value of s/D, the ratio of the nozzle submergence to the tank diameter. In the buoyancy-dominated regime, the dimensionless pressure reduction time is larger for larger values of s/D because the jet dissipates more energy as it travels the longer distance to the surface and is less able to break up the buoyant layer of hot liquid. In the inertia-dominated regime, the dimensionless pressure reduction time appears to be independent of the tank fill fraction, except for $s/D = 0.48$. Since the heat input was not varied, this implies that the heat transfer rate was the same for the higher s/D tests. However, the flow patterns established at $s/D = 0.48$ apparently were not as effective in sweeping away the hot liquid layer.

Several other investigators have also correlated their data with the dimensionless group Θ. Fossett and Prosser[9] found that a value of 8 for Θ insured complete bulk fluid mixing, as

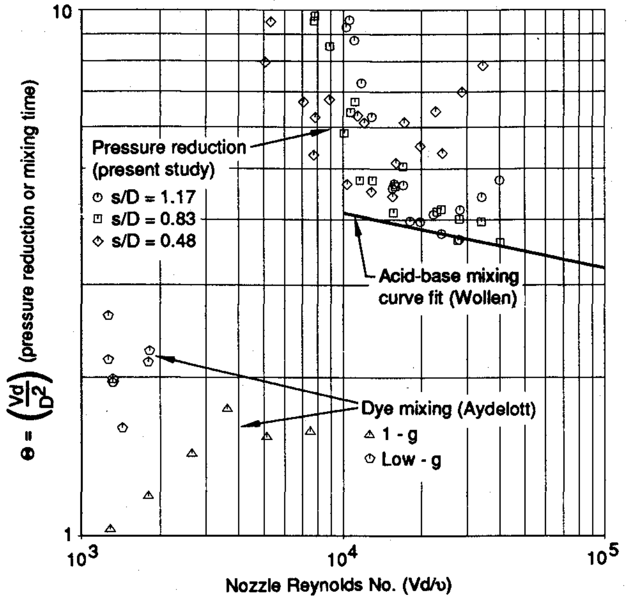

Fig. 8 Comparison of pressure reduction times with dye mixing and acid-base mixing times.

determined by measuring conductivity in sodium carbonate mixtures. Wollen's study,[4] using acid-base mixing, indicated that a lower value of Θ would be adequate. Poth and Van Hook[3] found in their investigations of dye mixing and pressure reduction that both phenomena depended in a similar fashion on the term *Ud* and thus both could be correlated in terms of Θ. They found as we did that the dimensionless pressure reduction times were largely independent of the nozzle submergence as long as buoyancy was not dominating. In addition, they determined analytically that bulk fluid mixing, in contrast, should depend on liquid height, a result that has also been presented by Okita and Oyama.[7]

Figure 8 shows our data plotted along with the data obtained by Aydelott[6] and Wollen.[4] Aydelott measured the mixing time for jet-induced dispersion of dye in small subscale vessels containing ethanol, both in normal and low gravity. We show his 1-*g* data for cylindrical tanks and some of his low-*g* data from drop-tower tests in which the jet flow did not penetrate the liquid surface and disrupt the ullage. His 1-*g* and low-*g* mixing times are lower than our pressure reduction times and Wollen's mixing times, indicating that predictions of low-*g* mixing times based on observations of dye motion in small tanks may underestimate actual low-*g* mixing times by a factor of 2 or more. Wollen conducted his experiments with both water and Freon 11 and used acid-base neutralization to give a clear indication of when mixing was completed. His data are similar to ours for $Re_n > 20 \times 10^3$, indicating that pressure reduction times based on the 5% criterion are an accurate predictor of complete mixing, at least at the values of *s/D* that Wollen tested (which are typical for propellant tank applications). Since acid-base neutralization data are not affected by buoyancy at low flow rates as our data are, this agreement indicates that acid-base mixing data may be able to give good predictions of low-*g* pressure reduction times for $Re_n < 20 \times 10^3$ as well. Any extrapolation of this data, though, to low-*g* mixing is only applicable to the case in which all of the liquid is settled over the mixer and the mixer flow remains in the liquid (i.e., no geyser is created).

Aydelott proposed another nondimensional mixing time parameter that he found to give a good correlation between his 1-*g* and low-*g* data. The definition of this parameter is the liquid-jet volumetric flow rate *Q* divided by the bulk-liquid mixing rate:

$$\Theta_A = \frac{Q}{V_l/t}$$

This definition is derived from the following expression proposed by Okita and Oyama to correlate mixing time measurements made in completely full tanks:

$$\Theta_O = \frac{tQs^{1/2}D^{1/2}}{V_t d}$$

Aydelott substituted V_l for V_t and dropped D, s, and d from the expression because in his experiments these quantities were fixed or varied only slightly. He found that he could relate his 1-*g* and low-*g* dimensionless mixing times to a jet-mixer-strength parameter based on Weber number and Bond number.

Lacking low-*g* data, we did not use Aydelott's jet-mixer-strength parameter in our data analysis, but we did assess how well his mixing time parameter Θ_A correlated our data, his data, and Wollen's data with jet Reynolds number. We found that Θ_A leaves more scatter in the data than does the parameter Θ. The form of Θ_A suggests two reasons. The jet flow rate is equal to $(\pi/4)Ud^2$, and V_l approximately equals $(\pi/4)D^2s$ for cylindrical tanks with the nozzle near the tank bottom, thus, for purposes of comparison with Θ, Θ_A may be expressed as

$$\Theta_A = \frac{tUd^2}{D^2s}$$

This parameter therefore varies in proportion to Ud^2, whereas our data, as well as the other data we have referenced, show a dependence on *Ud*. (In Aydelott's experiments *d* was fixed, and so this difference did not affect his results.) Second, Θ_A depends strongly on *s*, whereas we found only a weak dependence.

We also tried using the more complex parameter Θ_O to correlate our data but found it also was less satisfactory than Θ. Consequently, we used Θ for our analysis.

Interfacial Heat Transfer

Deriving the interfacial heat transfer rate required fitting a thermodynamic model to the data. First we divided the ullage into layers, with one layer for each thermocouple, as shown in Fig. 9. For each data sampling time, the tank pressure and layer temperatures were used to calculate the fluid density in each layer. From the densities, the fluid mass in each layer was calculated; then summing the masses yielded the total mass in the ullage. The condensation rate was taken as the rate of change of the ullage mass.

To relate the calculated condensation rate to the heat transfer rate at the surface, we assumed that during the mixing process all condensation occurred at the surface. This assumption was supported by test data that indicated that during mixing the ullage cooled more rapidly than the tank walls.

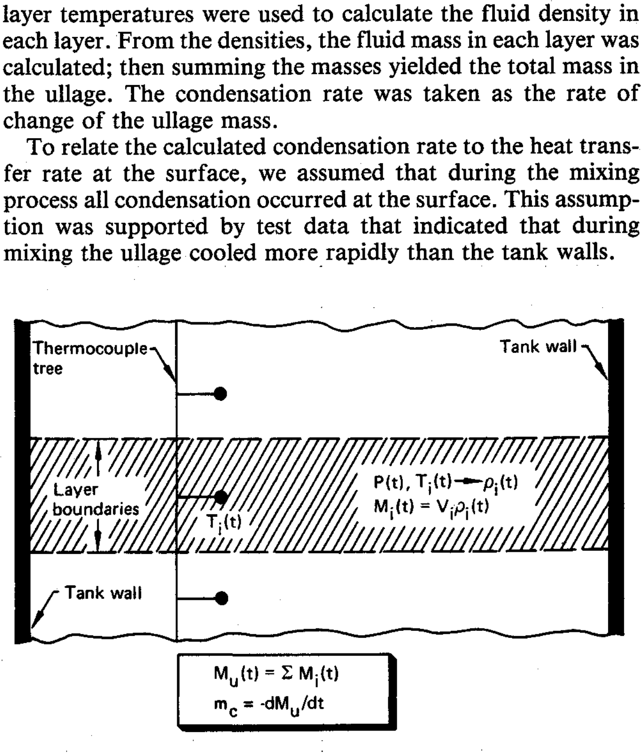

Fig. 9 Hypothetical division of ullage into layers for calculation of ullage mass.

An energy balance on an infinitesimally thin control volume drawn around the liquid surface yields

$$q_{sb} = q_{us} + m_c h_{fg}$$

We estimated the magnitude of q_{us} using free convection correlations for vapor being cooled by a horizontal flat plate[12] and determined that it is negligible.

The heat transfer coefficient for the heat flux from the surface into bulk liquid is defined by

$$h = \frac{q_{sb}}{(T_s - T_b)A}$$

For generality, the heat transfer coefficient can be put in terms of a Nusselt number based on the tank diameter:

$$Nu = \frac{hD}{k}$$

As we noted before, the condensation rate at the liquid surface is controlled by the rate at which heat is transported from the surface into the bulk liquid. This heat transfer rate was determined for each experiment run, and from that the related heat transfer coefficient h was derived. Figure 10 shows the ullage pressure and h as functions of time during a representative test. This test had a low-velocity jet that took several minutes to penetrate the stratified layer and establish a circulating flow that swept the entire liquid surface. Soon after this circulation was established, the pressure decreased rapidly, indicating a high condensation rate; then the pressure reduction rate and the condensation rate rapidly decayed. At this point in each test the heat transfer rates and temperature differences became small, and it became difficult to calculate reliable values for h. For consistency in determining a representative value of h for each run, we selected the relative maximum of h that occurred just after the peak condensation rate. For some of the tests with low Reynolds number, including all of the 0.62-in. nozzle tests, the data did not yield an unambiguous rise in condensation rate, and so heat transfer coefficients were not calculated for those tests.

Comparing the values of h obtained from many different tests, we found a significant dependence only on nozzle Reynolds number (actually Ud, since ν was not varied) and s/D. Figure 11 shows h in dimensionless form (as a Nusselt number Nu) as a function of Re_n, s/D, and d/D. The data indicate that Nu is primarily a function of Re_n and s/D for all of the tests and that for s/D equal to 0.83 and 1.17 Nu depends on $(Re_n)^{1.3}$ and increases as s/D decreases. This is a stronger dependence on Re_n than reported by Thomas,[10] who found for steady-state steam condensation that h depended on $(Re_n)^{1.1}$.

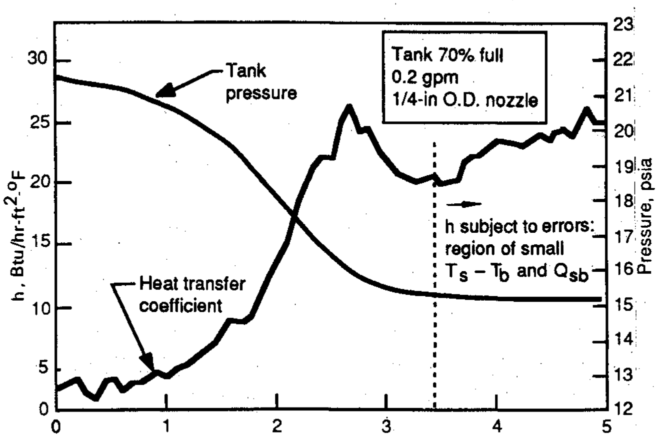

Fig. 10 Time histories of ullage pressure and surface heat transfer coefficient.

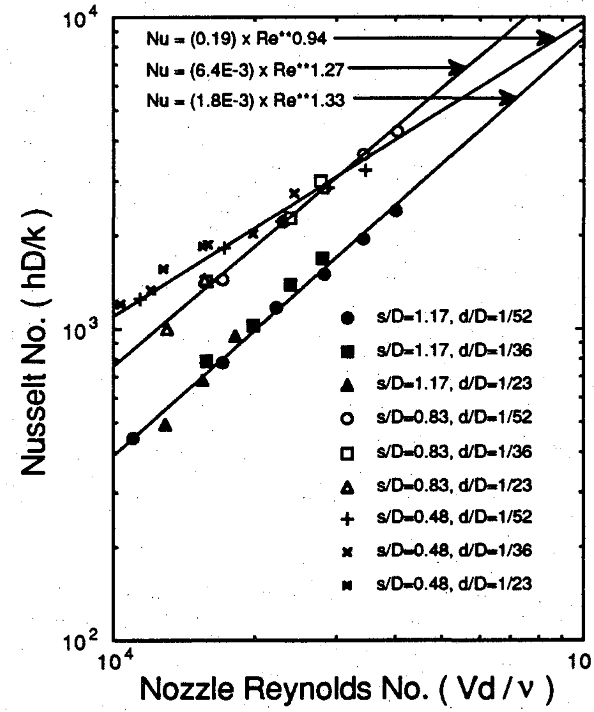

Fig. 11 Nusselt number based on surface-to-liquid heat transfer coefficient as a function of jet Reynolds number and nozzle submergence.

This difference is almost certainly due to the transient nature of our pressure reduction tests as compared with Thomas's steady-state measurements, because when we later reconfigured the experiment to allow measurement of steady-state condensation rates, we found that Nu was very nearly proportional to Re_n and that the steady-state values of Nu were as much as twice the transient values. (These data are presented in a subsequent paper.[13]) Therefore, it appears that complete jet circulation does not develop during the brief pressure reduction transient. The fact that Nu increases more rapidly with Re_n for our transient data indicates that at higher Re_n the jet circulation becomes more fully established during the transient and the Nusselt number is more nearly the steady-state value.

The results for s/D equal to 0.48 are more difficult to explain. The exponent of Re_n is less than 1, and Nu at high values of Re_n is lower than it is with s/D equal to 0.83 and 1.17. A possible explanation for this behavior is that below a certain threshold of s/D the flow characteristics change and the flow is not as effective at sweeping away the hot layer of fluid at the surface.

The dependence of Nu on the nozzle Reynolds number can be explained by the characteristics of the fluid motion just below the interface. Sonin et al.[11] showed that Nu is proportional to the liquid-side turbulence at the surface. The experiment of Sonin et al. was similar to Thomas's except that in that of Sonin et al. the nozzle was more than three tank diameters from the liquid surface, which allowed the jet to fill the tank cross section. This created isotropic turbulence at the surface, with negligible bulk circulation. Sonin et al. determined that the turbulence amplitude is a function of Re_n and s.

However, since the jet nozzle was approximately one tank diameter from the liquid surface in our experiments (and in Thomas's as well), the jet produced a radial bulk outflow in a thin, turbulent boundary layer under the surface. To apply the results of Sonin et al. strictly, the liquid-side turbulence for this flow must be integrated over the surface. Brown et al.[14] integrated the Sonin et al. correlation over a liquid surface for low-nozzle-submergence conditions relevant to our data. They found that Nu was proportional to Re_n (for fully turbulent jets and fixed D) and decreased linearly with s/D. The predictions of the Sonin-Brown model do not match the transient condensation data presented in this paper, but they do fit with the trends we later observed in steady-state tests.

Conclusions

Pressure reduction times in a sealed tank of thermally stratified Freon 11 were measured as a function of the strength of an axial mixer jet. We found that an axial jet was very effective in promoting condensation at the liquid-vapor interface and reducing the tank pressure. However, buoyancy forces were much larger than would be seen in low gravity, and so the pressure reduction times measured for low-momentum mixing jets substantially exceed what would be expected in space.

Measurements of pressure reduction time obtained when the mixing flow was inertia dominated indicate that the pressure reduction time varied in inverse proportion to the product of jet velocity and nozzle diameter, which is the same dependence found in studies of bulk-liquid mixing. We found that, for inertia-dominated flow, a dimensionless time of 5 was sufficient to reduce the tank pressure to near its destratified pressure, which compared well with the reported results of a prior experiment with isothermal acid-base neutralization in a scale model Centaur tank. This leads us to conclude that pressure reduction time as defined in this study is also a good measure of the time to mix the bulk fluid, at least in the range of liquid heights typical for propellant tanks. Dimensionless mixing times obtained in isothermal dye mixing experiments in normal and low gravity, however, were only half as large as those obtained in the present experiments and in the acid-base mixing experiments.

Also, transient condensation and heat transfer rates were measured, and values of the surface-to-liquid heat transfer coefficient were derived. The heat transfer coefficient varied proportionally with jet Reynolds number to approximately the 1.3 power. This differs slightly from correlations of steady-state mixing-induced condensation of steam.

Caution is required in applying any of the available mixing data to thermal destratification in low gravity. Trends that would appear in low gravity may be masked by gravity-dependent effects such as buoyancy. With flow rates high enough to avoid significant buoyancy effects in 1 g, the mixer jet operating in low gravity will penetrate into the ullage, creating completely different flow patterns. Experiments conducted in space are required to obtain adequate design data for future in-space fluid storage and transfer systems.

References

[1] Bentz, M. D., Meserole, J. S., and Knoll, R. L., "Tank Pressure Control Experiment—A Low-*G* Mixing Investigation," AIAA Paper 90-2376, July 1990.

[2] Dominick, S. M., "Mixing Induced Condensation Inside Propellant Tanks," AIAA Paper 84-0514, Jan. 1984.

[3] Poth, R. J., and Van Hook, J. R., "Control of the Thermodynamic State of Space-Stored Cryogens by Jet Mixing," *Journal of Spacecraft and Rockets*, Vol. 9, No. 5, 1972, pp. 332–336.

[4] Wollen, M. A., "Evaluation of Jet Induced Mixing for Centaur LH2 Tank," NASA TM-696-83333, Aug. 1983.

[5] Bullard, B. R., "Liquid Propellant Thermal Conditioning System Test Program: Final Report," NASA CR-72971, July 1972.

[6] Aydelott, J. C., "Modeling of Space Vehicle Propellant Mixing," NASA TP-2107, Jan. 1983.

[7] Okita, N., and Oyama, Y., "Mixing Characteristics in Jet Mixing," *Chemical Engineering, Japan*, Vol. 27, No. 4, 1963, pp. 252–259.

[8] Fox, E. A., and Gex, V. E., "Single-Phase Blending of Liquids," *AIChE Journal*, Vol. 2, No. 4, 1956, pp. 539–544.

[9] Fossett, H., and Prosser, L. E., "The Application of Free Jets to the Mixing of Fluids in Bulk," *Proceedings of the Institute of Mechanical Engineers*, Vol. 160, No. 2, 1949, pp. 224–232.

[10] Thomas, R. M., "Condensation of Steam on Water in Turbulent Motion," *International Journal of Multiphase Flow*, Vol. 5, No. 1-A, 1979, pp. 1–15.

[11] Sonin, A. A., Shimko, M. A., and Chum, J. H., "Vapor Condensation onto a Turbulent Liquid—I," *International Journal of Heat and Mass Transfer*, Vol. 29, No. 9, 1986, pp. 1319–1332.

[12] Holman, J. P., *Heat Transfer*, McGraw-Hill, San Francisco, CA, 1976, p. 245.

[13] Jones, O. S., Meserole, J. S., and Bentz, M. D., "Correlation of Ullage Condensation Rate with Fluid Mixing Intensity in Propellant Tanks," AIAA Paper 91-2543, July 1991.

[14] Brown, J. S., Helmick, M. R., and Sonin, A. A., "Vapor Condensation at a Turbulent Liquid Surface in Systems with Possible Space-Based Applications," AIAA Paper 89-2846, July 1989.

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