



Two-Phase Cryogenic Heat Exchanger for the Thermodynamic Vent System

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Abstract

A two-phase cryogenic heat exchanger for a thermodynamic vent system was designed and analyzed, and the predicted performance was compared with test results. A method for determining the required size of the Joule-Thomson device was also developed. Numerous sensitivity studies were performed to show that the design was robust and possessed a comfortable capacity margin. The comparison with the test results showed very similar heat extraction performance for similar inlet conditions. It was also shown that estimates for Joule-Thomson device flow rates and exit quality can vary significantly and these need to be accommodated for with a robust system design.

Introduction

Thermodynamic vent systems (TVS) are being developed to control the pressure in cryogenic propellant tanks used on spacecraft. Heat penetrates these tanks and causes the cryogenic propellant to boil-off. In order to prevent over pressurization of the tank, this excess vapor needs to be vented off. In microgravity environments it is difficult to just vent off vapor without including some of the liquid. A TVS provides a means to prevent this by tapping off liquid, vaporizing this liquid, and using the heat of evaporation to cool the tank. No liquid is vented overboard. The vapor can also be used to cool penetrations and other tanks. In liquid hydrogen propellant applications, para-to-ortho catalytic conversion can be added to increase the cooling capacity.

Description

Thermodynamic Vent System

The Thermodynamic Vent System (TVS) consists of two main parts: the Joule-Thomson device and the two-phase heat-exchanger. A schematic of the system is shown in Figure 1. There is also a pump which provides external cooling flow over the tube coils inside the heat exchanger; and control valves are provided to modulate the flow to the vent and to direct the pump discharge to either an axial jet or a spray bar. The Joule-Thomson (J-T) device is a multi-stage orifice called a ViscoJet (Lee Company) which is manufactured by the Lee Company (Ref. 1).

Operation is rather simple: the control valve on the left of Figure 1 is opened and this provides high pressure liquid cryogen to the ViscoJet (a.k.a. J-T device). The liquid in the tank is at a high pressure but there is low pressure downstream of the ViscoJet. The low pressure is typically 1 atmosphere for ground testing but would likely be vacuum in a space application. The J-T device provides a temperature change in the fluid with no heat exchange, i.e., it is isenthalpic. The fluid, now at a lower temperature and pressure, passes through the heat exchanger absorbing heat from the external fluid flow and evaporates. The vapor exits the tank through the vent system where the mass flow is measured.

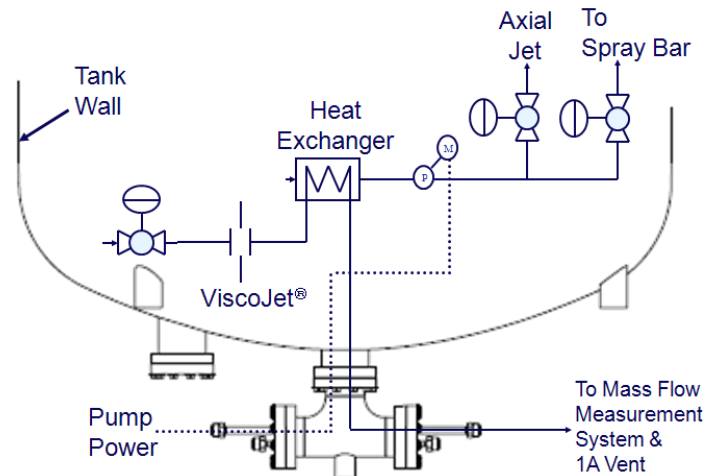


Figure 1.—Schematic of thermodynamic vent system.

The Lee ViscoJet is a multi-stage orifice which is being used as a Joule-Thomson device. This series of orifices provides relatively high pressure drops using relatively large orifice dimensions, thus low flows can be obtained without the need for very small passages which would be subject to clogging by contamination. The flow through a ViscoJet can be calculated using the term for flow resistance, LOhm (Lee Company), and the manufacture's equation (Ref. 2):

$$\dot{m} = (10,000/\text{LOhm})(\Delta P S)^{1/2}$$

where

\dot{m} = mass flow rate (lb/hr)
 LOhm = liquid resistance (gal/min/psid)
 ΔP = pressure drop (psid)
 S = specific gravity (lb/ft³)

Joule-Thomson Device

Papell, Saiyed and Nylan (Ref. 3) reported that this equation is not accurate for two-phase flows. Since cryogenic propellants are typically stored at near saturated conditions, it is likely that cavitation will occur within the ViscoJet. Therefore they conducted tests with a series of ViscoJets over a range of pressures, temperatures and vapor qualities and developed an equation that corrects for vapor formation:

$$\dot{m} = (10,000/\text{LOhm})(\Delta P S)^{1/2}(1-X)$$

where

X = vapor quality

This equation, called the Modified Lee Equation, was used in an iterative process to determine what LOhm size of ViscoJet would provide a mass flow closest to that desired. The ViscoJets are manufactured in fixed sizes and not all sizes are available from stock. Therefore the selected ViscoJet may not have the exact characteristics desired.

A spreadsheet, named LOX JT Sizer.xls, was developed to aid this iterative process. A snapshot of the spreadsheet is shown in the Appendix.

1. The user first specifies the desired heat rate (\dot{Q}) to be removed by the TVS system.
2. Using the latent heat of evaporation (h_{lv}) the desired mass flow rate (\dot{m}) is calculated: $\dot{m} = \dot{Q} / h_{lv}$
3. The user also provides the upstream total pressure (P_{total}) and the tank temperature (T_{cryo}) or, if Helium is used for pressurization, the partial pressure of the cryogen.
4. An estimate of the downstream pressure is also provided (P_{exit}). This is a function of the pressure drop through the heat-exchanger, the vent tubing and the exit pressure.
5. The spreadsheet calculates the saturation temperature (T_{sat}).
6. The upstream and downstream enthalpies are calculated and the difference between the two are used to calculate the quality: $X = (h_{liq_{inlet}} - h_{liq_{exit}}) / (h_{vap_{exit}} - h_{liq_{exit}})$. The presence of vapor increases the required pressure drop across to achieve the desired mass flow rate.
7. Use Goal Seek to adjust the LOhm value (cell C23) until the difference (cell C28) between the actual exit pressure (cell C29) and the calculated exit pressure (cell C27) equals zero. The spreadsheet calculates the necessary LOhm value to generate the necessary pressure drop based on the mass flow, temperature and quality conditions.
8. The user can then select a standard device that is close to the LOhm value calculated and adjust the initial heat input until the calculated LOhm value matches the standard size.

Two-Phase Heat Exchanger Design

The heat exchanger is a simple coil of stainless steel tubing held within two concentric cylinders as shown in Figure 2. The radial spacing between the coil and the cylinders is maintained by separators located at three locations. Suction from the centrifugal pump causes cryogenic fluid to enter the open end of the annular flow channel and creates a cross flow over the tube coils.

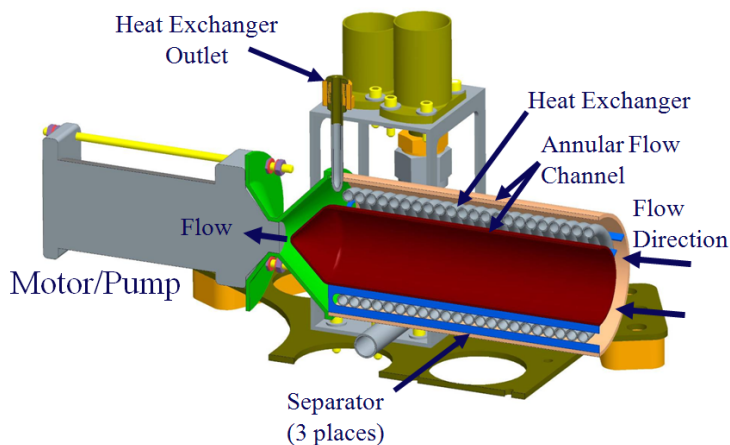


Figure 2.—Thermodynamic vent system, two-phase cryogenic heat-exchanger.

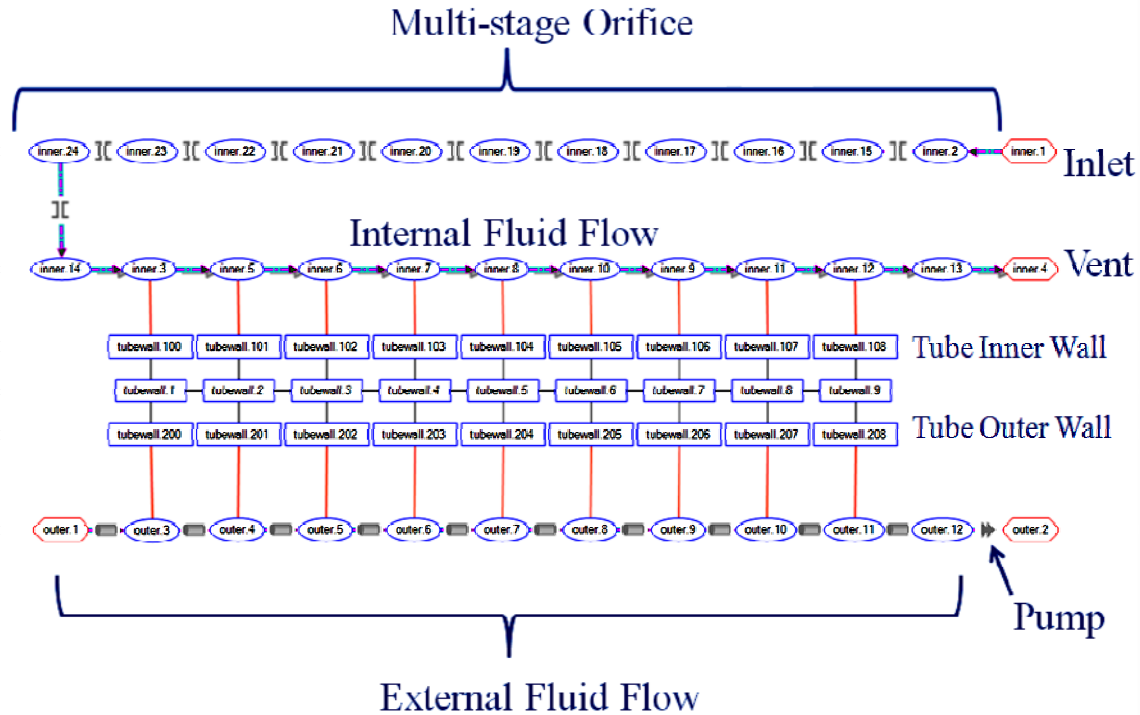


Figure 3.—SINAPS model of TVC.

To analyze the heat exchanger, a SINDA/FLUINT model was developed using the graphical user interface SINAPS (Cullimore and Ring Technologies). The system schematic is shown in Figure 3. The multi-stage orifice is located in the upper row and is modeled as a series of orifices. In this example, eleven orifices were used but this is an arbitrary number. Later in this report, the number of orifices will be reduced to one to show the effect of using one versus several.

The second row represents the internal tube fluid flow of the coil. A thermal conduction model of the tube wall is represented by the 3rd, 4th, and 5th rows and the external fluid flow is the bottom row. The fluid paths are solved by FLUINT whereas the conduction model is solved by SINDA. The FLUINT and SINDA models are coupled by Ties (shown as red lines) which represent the convective heat transfer. The pump is modeled a constant mass flow device at the end of the “External Fluid Flow.”

The “Internal Fluid Flow” is modeled with ten fluid paths in series and an eleventh path for the vent flow. Each segment of the flow path is a different length, each length being a multiple of the coil circumference which is 0.244 m (9.6 in.). The various lengths of the path segments are shown in the simplified model of Figure 4. The length of the first fluid path is 1/5 of the coil circumference. The lengths of the paths gradually increase from 1/5 of a circumference to 6 times the circumference, followed by a vent tube which is 100 times the circumference length.

The heat transfer coefficient on the outside of the cooling tubes was calculated using the empirical correlations for annular flow between concentric cylinders, see Figure 5. Although the heat transfer coefficient may be inaccurate, it will be shown later that the overall effectiveness of the heat exchanger is not very sensitive to this value. This is because the thermal resistance of the tube's internal flow heat transfer is much higher and controls the overall heat flow.

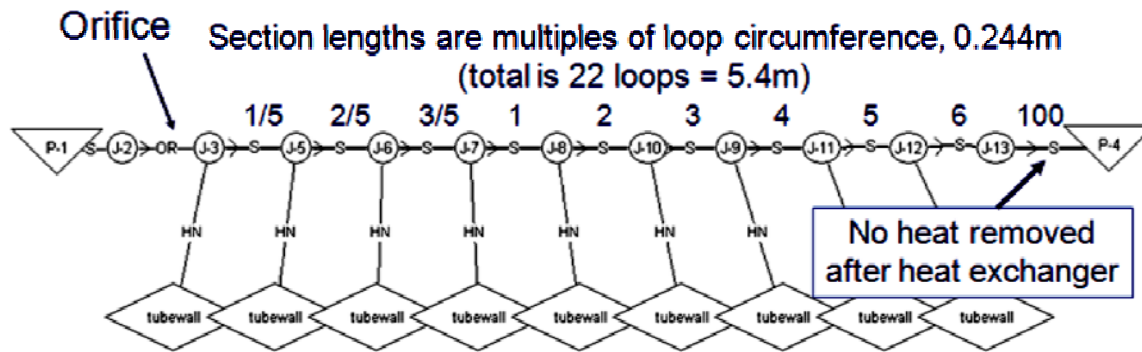


Figure 4.—Internal fluid flow paths.

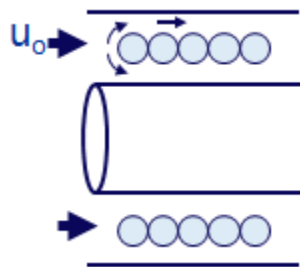


Figure 5.—External flow over tubing.

TABLE 1.—RESULTS OF SENSITIVITY TESTS

Case	ACCEL (m/s ²)	CURV (m)	Orifice twinned	Forced stratification	Lump initial condition	Quality at Junction 10
Normal	9.81	0.039	Yes	No	Vent temperature	0.2253
0	0.0	0.039	Yes	No	Vent temperature	0.2258
2	9.81	∞	Yes	No	Vent temperature	0.2267
3	9.81	0.039	No	No	Vent temperature	0.3659
4	9.81	0.039	Yes	Yes	Vent temperature	0.285
5	9.81	0.039	Yes	No	Vessel temperature	0.2169

Sensitivity and Stability

The model was subjected to various changes in geometry and boundary conditions to determine how sensitive the results are to these changes and to insure that the model is not strongly sensitive to modeling inaccuracies. The performance of the SINDA/FLUINT model might be affected by traverse acceleration due to gravity, centripetal accelerations caused by the curvature of the coil, two-phase flow in the orifice, stratification of the flow, and initial conditions. Each of these was varied to see what affect it would have on the performance of the heat exchanger. The performance was monitored by observing the vapor quality at fluid Lump #10, i.e., the sixth fluid lump in the heat exchanger's flow path. The baseline case is defined in the row titled "Normal" of Table 1.

Traverse accelerations due to gravity occur during on-ground testing but do not occur in space. Therefore the acceleration value was changed from 1 to 0G, and there was no significant effect. Next the curvature value was changed from 0.039 m (1.5 in.) to no curvature (∞) and again the effect was not significant. This model allows for two-phase flow in the orifice, but Cullimore and Ring have stated that two-phase flow has not been verified for orifices. Therefore the two-phase flow feature (a.k.a. twinned flow) was turned OFF for the orifice. When this was done, the quality at Junction 10 increased slightly from 0.2253 to 0.3659 and the mass flow decreased from 2.92 to 2.84 g/sec. Neither of these changes are of concern provided all of the liquid is vaporized before exiting the heat exchanger, which it was. To avoid this issue in the future, the ViscoJet should be modeled with just one orifice. If significant two-phase flow actually occurs within the ViscoJet, then this type of model is likely not adequate.

The flow regime is typically annular but occasionally becomes stratified. The different flow regimes are shown in Figure 6. When the flow was forced to always be stratified, the quality changed from 0.2253 to 0.2850 and the flow decreased from 2.917 to 2.808 g/sec. Indicating that a switch in flow regimes, which can occur unexpectedly, will not have a detrimental effect. Finally the initial temperature of the lumps was changed from that of the vent temperature to that of the tank temperature. This is the worst case because it generates the lowest quality, i.e. the least amount of fluid has been vaporized. Still the quality only changed from 0.2253 to 0.2169 and the mass flow was unchanged. The results are summarized in Table 1.

A few tests were performed to verify that the Fluint model and the modified Lee Equation perform similarly when flow conditions are changed; and to determine if the Fluint model requires any adjusting when these changes occur. First the inlet pressure was increased from 150 to 250 psia. Both showed the same increase in flow. Therefore, when inlet pressure is changed, it was not necessary to readjust the Fluint orifice flow area to match the flow conditions. Next the vent system back-pressure was set at three

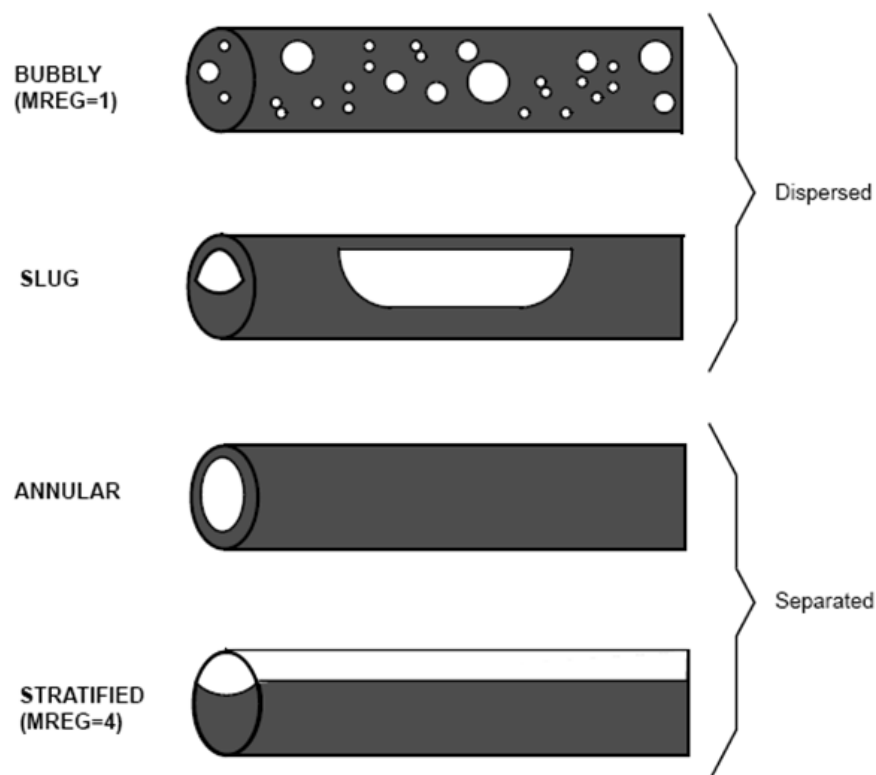


Figure 6.—Two-phase flow regimes (Ref. 4).

levels, 21, 34, and 65 psia. The minimum and maximum mass flow rates only differed by 10%, indicating that the system is relatively insensitive to changes in back-pressure. Similarly, the internal diameter of the vent tubing was varied from 12 mm to 11 mm and to 10 mm. The change from 12 mm to 11 mm actually caused a 6% increase in flow, which is not understood. The change to 10 mm caused the Fluint model to become unstable. Therefore it is recommended that the vent tubing have an inside diameter of 12 mm or larger. Finally, the orifice flow area was increased and decreased by 10%, to determine how sensitive the models are to inaccuracies in defining flow area. When the flow area was varied $\pm 10\%$ in the Fluint model, the flow increased by 16% and decreased by 4%. The pressure at the exit of the ViscoJet changed from 19.1 to 19.9 psia when the area was increased and became 18.9 psia when the area was decreased. These changes in ViscoJet exit pressures were applied to the Modified Lee Equation which showed no significant flow changes. Using compressible versus non-compressible fluid properties also showed no significant difference in the results. The above show that the two cases are stable and can be used over a reasonable range of conditions.

Next the sensitivity to the accuracy of the heat transfer coefficient on the external surfaces of the heat exchanger was examined. The heat transfer coefficients for the two extremes were determined for tubes in cross-flow (Ref. 5) and annular flow (Ref. 6). The equations are:

$$\text{Cross-Flow: } Nu = 0.27 Re^{0.63} Pr^{0.36} \quad 10^3 < Re < 2 \times 10^5$$

$$\text{Annular-Flow: } Nu = 0.023 Re^{0.8} Pr^{0.4} \quad Re > 6420$$

With a flow rate of 10 GPM, the heat transfer coefficients were found to be 3200 and 1300 W/m², respectively. These two conditions were applied to the heat exchanger. Since the performance of the heat exchanger is based on how much fluid is vaporized, the quality of the fluid at the 4th loop was examined. With the lower heat transfer coefficient the quality was 22% and with the higher coefficient it was 31%. This shows that the analysis is somewhat insensitive to the accuracy of the external heat transfer coefficient.

Sizing the Joule-Thomson Device

The required size of the Joule-Thomson device was determined using the following conditions:

Parameter	Value
Inlet pressure, psia	250
Inlet temperature, K	113
Heat to be removed, W	750
J-T device exit pressure (initial guess), psia	20

Using the latent heat of evaporation for liquid oxygen and the desired heat removal rate, the required vaporization mass rate was determined to be 3.95 g/sec. The orifice flow area in the Fluint model was adjusted until this flow rate was achieved. The Fluint results showed that the quality at the J-T device exit was 2.7% and the exit pressure was 20.7 psia. Those conditions were then applied to the Modified Lee Equation and a 'goal-seek' was done with Excel to find the closest ViscoJet size, which was found to be 5000 LOhm. Under these conditions the Modified Lee Equation, which assumes isenthalpic expansion, determined the quality to be 16%. If this is true, then the flow rate would be reduced to 3.4 g/sec and the heat absorbed by vaporization would only be 595 W when the inlet pressure is 250 psia. At lower pressures, since the flow rate is proportional to square root of the pressure difference, the heat rate will be lower. For example, at an inlet pressure of 150 psia, the 5000 LOhm device would pass enough flow to absorb 460 W of heat.

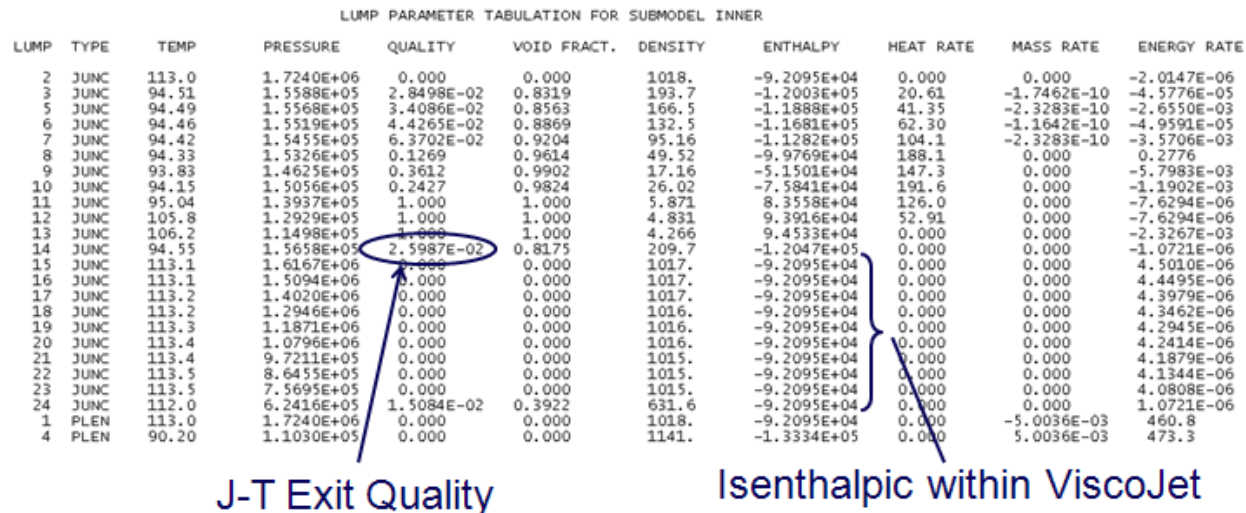


Figure 7.—Sample output of SINDA/FLUINT model.

Two-Phase Heat Exchanger Performance

After the J-T device is sized, the heat exchanger needs to be analyzed to determine if all the liquid is being vaporized in the heat-exchanger. A sample of the Fluint analysis output is shown in Figure 7. It shows that isenthalpic expansion occurs in the orifice group and the quality at the exit is 2.65%. It also shows that the quality reaches 100% by Lump #11, which is only 1/2 through the heat exchanger. Therefore all the liquid is being vaporized before it has traveled 1/2 way through the coils. This also indicates that a lower LOhm device could be used which would increase the flow rate and increase the heat removal rate. For example, a 4000 LOhm device would increase the heat absorption to 1000W at 250 psia. Or, two 5000 LOhm devices used in parallel could be used, which would provide a total fluid resistance of 2500 LOhm and would increase the flow to 8 g/sec.

Jurns (Ref. 7) found that ViscoJet flow rates differed substantially from those calculated using the Modified Lee Equations. Actual flow rates in the 20 to 100 psia range differed as much as $\pm 45\%$ from predicted values. At higher pressures the difference could be greater. These test anomalies advocate that actual flow rates need to be verified with testing. If two 5000 LOhm devices were used and if both had a flow rates greater than 25% above the predicted value, then this would exceed the capacity of the heat exchanger and liquid would enter the vent tubing.

Comparison With Test Data

The TVS system was tested by VanDresar (Ref. 8) with a 5000 LOhm ViscoJet using liquid oxygen and liquid nitrogen, and one of the LOX test points is compared with the analytical predictions in Table 2. The results show very similar heat extraction rates for similar inlet conditions. Heat leak rates of 600W are not typical for cryogenic propellant tanks. In order to accelerate testing schedules, the TVS system was oversized to accelerate the cooling and heaters were attached to the test tank to provide rapid heating. Although heaters were available for the above test, they were not used during the TVS cycles. The test was accelerated by using a small temperature control band. A more thorough study should be done by analyzing and comparing all of the test points.

TABLE 2.—COMPARISON WITH TEST DATA

	Model	Test
Tank pressure, psia	250	231
Inlet temperature, K	113	111
Heat extraction, W	595	631

Conclusions

A 5000 LOhm ViscoJet is predicted to provide a mass flow rate of 3.4 g/sec when the inlet pressure is 250 psia. This flow will be totally vaporized within the heat exchanger, providing an heat extraction rate of 595 W. When the inlet pressure is decreased to 150 psia the heat extraction rate will likewise decrease to 460 W.

It was found, that for this design, it was easy to vaporize the fluid in a short amount of tube length. For these cases, liquid oxygen at 113 K, initially between 150 and 250 psia, was being subjected to a large drop in pressure to below 20 psia while being kept in a 113 K environment. The boiling point at 20 psia is 93.2 K, therefore there is a 20 K difference between the boiling point and the environment and the fluid is easily vaporized. In all of the cases analyzed there was 100% vapor quality before the half way point in the heat exchanger.

The J-T device can be modeled as a single orifice or a multistage orifice. If vapor occurs within the J-T device, due to the limitations of Fluint, the J-T device should be modeled as a single orifice. For sizing the heat exchanger under these conditions, it was not necessary to accurately model the J-T device: Just use the modified Lee equation to find the mass flow rate and assume 0% quality exiting the ViscoJet. Iterate with the Fluint model to find the backpressure and adjust the mass flow rate, then size the heat-exchanger for those conditions. This might not be true for other conditions, for example, when the inlet and outlet pressures are closer and there is a greater opportunity for vapor to be present in the J-T device.

Appendix—LOX JT Sizer Spreadsheet

The filename for the spreadsheet is LOX JT Sizer.xls.

	A	B	C	D	E	F	G	H	I	J	K	L	M	N
1														
2	Heat consumption	Q	615 W											
3	Mass flow rate	mdot	0.003333 kg/s											
4			0.00735 lb/sec		needed for Lee equation									
5	Inlet conditions													
6	Tank pressure	Ptotal	236.4 PSIA		Total tank pressure									
7	Tank temperature	Tcryo	113 K		Actual fluid temperature									
8	Average temperature	Tavg	104 K		Vapor exiting HX is at same temperature between the tank and VJ exit									
9	Tank liquid enthalpy	hliq	-93 kJ/kg		Enthalpy of liquid in tank at tank temperature									
10	Tank vapor enthalpy	hvap	92 kJ/kg		Enthalp of vapor leaving HX assuming an average temperature									
11	Heat of evaporation	hlv	-185 kJ/kg		thus heat absorbed is a function of enthalpy changes from tank to HX exit									
12														
13	Exit conditions													
14	Guess at exit pressure	Pexit	26.1 PSIA		Replace with Fluint calculated exit pressure									
15	Exit saturation temp	Tsat	95 K											
16														
17	Liquid enthalpy at exit	hliq	-123 kJ/kg		evaluated at saturation conditions									
18	Vapor enthalpy at exit	hvap	84 kJ/kg		evaluated at saturation conditions									
19	Heat of evaporation	hlv	-207 kJ/kg		from liquid at tank temperature to vapor at exit temperature									
20	Estimate quality at exit of ViscoJet													
21	Quality	X	0.147		if process is isenthalpic									
22														
23	Select a ViscoJet		5000 LOHM											
24														
25	Pressure drop	ΔP	210 PSID		Lee equation									
26	Desired drop		210 PSID		Pin-Pout									
27	Exit pressure	Pexit	26.0											
28	Difference		0 PSID											
29														

With Goal Seek to adjust C23 until C28=0 then replace C23 with the closest available LOhm value. Then adjust the heat consumption until the pressure difference is 0. If the FLUINT model mass flow rate does match the above, adjusts the FLUINT model orifice diameter. Rerun and apply the new back pressure in C14. Repeat the process until the FLUINT and VJ Sizer match in flow rate and back pressure with the selected LOHM in C23 and 0 pressure difference in C28.

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