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INVESTIGATION OF A SHELL AND TUBE EXCHANGER IN LIQUEFIED NATURAL GAS VAPORIZATION SERVICE

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ABSTRACT

Liquefied natural gas (LNG) is commonly converted from liquid to vapor for gas distribution. One of the methods for vaporizing LNG is to use a shell and tube heat exchanger. Water is used on the shell side to provide the heat source and LNG is then vaporized through the tube side passages of the exchanger. In many of these applications, the LNG is at a high pressure on the tube side while the water is at a lower pressure than the LNG as it flows through the shell side. The industry consensus document API 521[1] "Guide to Pressure Relieving and Depressuring Systems," Fourth Edition, paragraph 3.18 "Heat Transfer Equipment Failure" states that a complete tube rupture is to be considered for the possible overpressure of the equipment. The typical shell and tube exchanger application described above has rupture discs on the shell body to protect the shell from being over-pressured due to a tube rupture scenario. The possible freezing of the water in the shell due to mixing with cryogenic LNG is a concern. The issue to consider is whether freezing will occur before the rupture discs can safely relieve a possible over- pressure condition of the shell. A numerical analysis of the condition was performed using Computational Fluid Dynamics (CFD) software. The exchanger service, the analysis procedure and the conclusions found are detailed in this paper.

INTRODUCTION

This paper contains the results of analyses performed on a shell and tube heat exchanger designed as an LNG vaporizer. The purpose of the analyses was to evaluate the effect that a double guillotine break in one of the LNG tubes would have on the surrounding fluid. In particular, the analyses evaluated the probability that the cryogenic natural gas fluid/gas mixture would freeze the shell-side propylene-glycol/water (glycol/water) mixture resulting in the relief valves plugging and not venting.

To perform the analyses, two different Computational Fluid Dynamics (CFD) models of the vaporizer were constructed. These models were then run to study the effects of a tube break on the performance of the exchanger. The two models were examined to determine:

- the size of the freeze front after a tube break, and
- the time for the pressure pulse caused by the break to propagate through the vaporizer.

It should be noted that results found through this study cover a specific set of design conditions that were evaluated and a specific set of assumptions about the system's performance after the rupture occurred.

The characteristics associated with LNG vaporizers vary widely based on the application. This paper is not intended to be an all-encompassing study for various conditions in an LNG shell and tube vaporizer. The findings reached in this paper are based on a specific set of conditions for both sides (shell and tube) of the heat exchanger. This paper is specific to this data alone. Further investigation would be required for changes in these conditions. . Specifically, changes in LNG pressure or water temperature or quality would impact results observed from the model. The specific set of conditions evaluated for this paper can be seen in the shell and tube data contained in Table 1 below.

	Shell Side - Glycol / Water	Tube Side - LNG
Mass flow rate (lb/hr)	1,996,049	220,976
Inlet Temperature (deg. F)	200	-253.9
Inlet Pressure (psia)	104.7	349.7

Table 1 – Summary data for LNG vaporizer studied

PROCEDURES

Two analyses were performed on the vessel: an analysis to determine the size of the freeze front that could be expected to form after a tube rupture occurred, and an analysis to determine the time for the pressure pulse caused by the break to reach the rupture discs in the vessel.

To evaluate the size of the freeze front that would occur after a tube break, a half-symmetry model of the vaporizer was constructed. This model contained 549,056 cells and was composed of hex and wedge elements. The baffles inside of the shell and tube exchanger were modeled using 2-dimensional surfaces. An image of the model is contained in Figure 1 below.

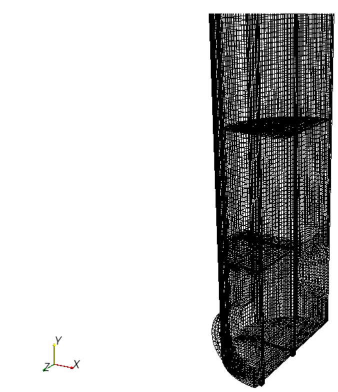

Figure 1 – Computational grid used for freeze front analysis

The water inlet flow rate (3988 GPM) and temperature (200 °F) values were the selected process design conditions for this particular application as illustrated in the shell and tube data. As the pumping system dynamics were not known, the flow rate into the vaporizer was assumed to be constant throughout the event.

Because the tubes were not explicitly modeled, no heat transfer between the tubes and water was accounted for in the model. In order to reduce the overall model size, the individual tubes were not included in the model. To account for the tubes' impact on the flow, a porous media was used with the value of the counter flow friction factor taken from empirical data. Based on the average Reynolds number in the exchanger, a friction factor of 0.8 was used [2].

A double-ended break, as specified in API 521, was assumed to occur in one LNG tube at the inlet to the exchanger. The tube dimensions were based on a 1.0" outside diameter (OD) tube size. The mass flow rate of the flow entering the exchanger due to this break was determined using the procedures shown in the Appendix. The calculated flow rate from the double guillotine break was approximately 164,700 lb/hr. This represents nearly 75% of the total LNG flow rate for this exchanger. Thus, a break should be detected quickly by the operator due to a loss of pressure on the tube side of the vaporizer.

The break was assumed to occur at the LNG inlet tubesheet. This location was chosen because the LNG is at its coldest temperature at this location, so the predicted freeze volume of the shell-side fluid should be at a maximum at that location. Using this assumption, the LNG enters into the vaporizer at a temperature of -253.9°F. Three break locations at the inlet tubesheet were considered for the analysis, a break occurring near the water inlet, a break occurring at the center of the vaporizer and a break occurring near the outside wall of the vaporizer. Eash of these were considered to be the worst case cryogenic condition. The LNG properties used for the analysis are shown in Figures 2 and 3 below.

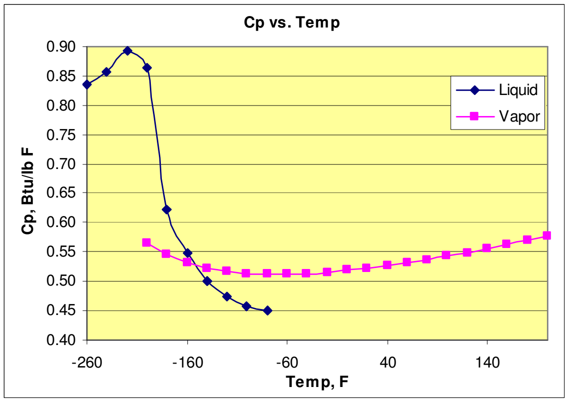

Figure 2 – Specific heat versus temperature values used for the natural gas during the freeze front analysis

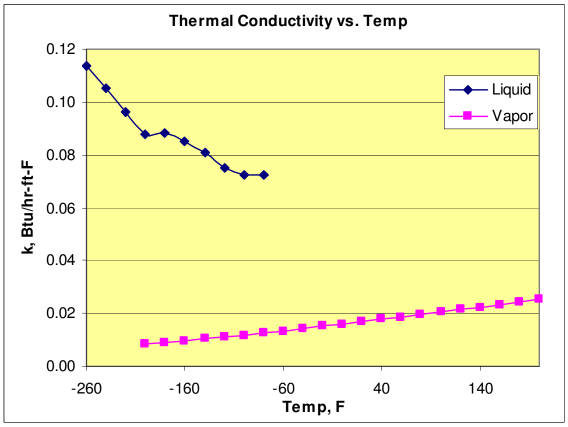

Figure 3 – Thermal conductivity versus temperature values used for the natural gas during the freeze front analysis

As can be seen in these figures, no information for the vapor phase below -200 °F was available. For this reason, the LNG was assumed to vaporize at this temperature. The heat of vaporization (HOV) of the LNG was taken to be 260 BTU/lb. The HOV was accounted for by modifying the C_p of the fluid to account for the required vaporization energy over a 1°F interval. The vaporization interval occurred between -200 and –199 °F and was represented as a spike in the C_p values for the stream between these two temperatures. This accounted for the vaporization of the LNG, as any of the LNG stream that rose to or above -199 °F was required to absorb the energy associated with the HOV from the glycol/water mixture.

The glycol/water properties were defined using information for a 50% mixture of propylene-glycol and water. These properties are contained in Table 2 below.

Temperature (F)	*Specific Heat (BTU/lb-F)*	*Thermal Conductivity (BTU/hr-ft-F)*
-20	0.8	0.177
30	0.825	0.19
80	0.85	0.201
130	0.875	0.208
180	0.901	0.214
230	0.926	0.215

Table 2 – Water/Glycol properties used for CFD analysis

The glycol/water mixture was assumed to freeze at -27 °F based on the data sheet provided by the glycol supplier. As no better data was available, the heat of formation was taken to be that of water. The formation of ice within the glycol/water mixture was accounted for using the same C_p modification technique that was used for the LNG, with the C_p spike occurring between -27 °F and -28 °F.

Transient analyses were used to allow for modeling the specific heat, and therefore the phase changes, within the system. Because the analyses were transient, they were conducted for a long enough period of time that the predicted freeze volume reached a quasi-steady-state value. The freeze front analyses were conducted using Star-CCM+ [3].

To conduct the analysis of the time for the pressure pulse to propagate through the vaporizer, a full 3-dimensional model was constructed. A full model was required because a symmetry plane within the model would have caused spurious reflections of the pressure pulse. A far-field boundary applied to the symmetry cut plane would be non-conservative in the energy dissipated by pulses reaching this location. The mesh employed for this analysis consisted of 50,064 elements. The model is shown in Figure 4.

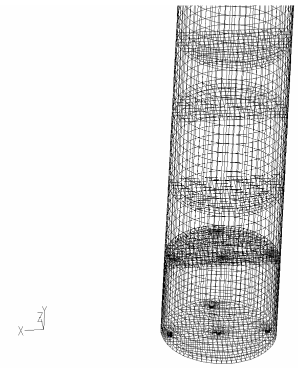

Figure 4 – Mesh used for pressure pulse analysis

To conduct the analysis, the system was set to an initial operating pressure of 104.7 psi. At time t = 0, an instantaneous pressure of 349.7 psi was applied at the bottom of the vessel over an area the size of 1 tube. The shell-side fluid was treated as a compressible fluid with a bulk modulus of 816 MN/m^2. A transient analysis was conducted to track the pressure pulse along the outer wall of the shell and tube exchanger. No consideration was given to the pressure relief that would occur once a rupture disc was activated. Instead, the analysis only sought to determine the amount of time required for the vaporizer pressure to reach a value where the disc would rupture. The pressure pulse analysis was conducted using Fluent [4].

RESULTS

Several methods were employed to determine the location of the freeze front, including querying the distributions of the specific heat, the thermal conductivity and the temperature of the fluid within the domain. The temperature results gave both good visual reference of the freeze front and also correlated well with the volumes represented by the numerical data for the other parameters queried from the model. For this reason, areas with a temperature less than -27 °F were assumed to freeze for visualization purposes.

Figure 5, illustrates a cross section of the model on the symmetry plane, the blue and dark green regions are where freezing is indicated.

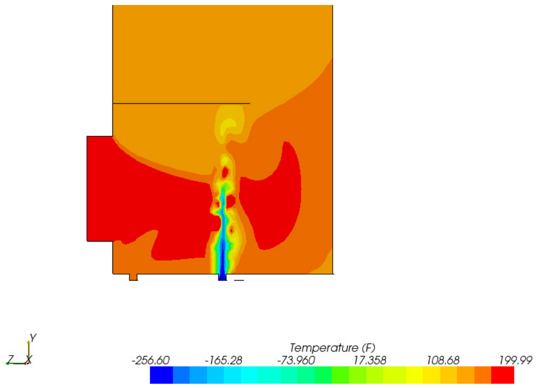

Figure 5 – Contours of temperature showing the freeze front within the vaporizer

Figure 6, shows an isosurface that was created to highlight the regions with indicated temperatures below the freezing temperature.

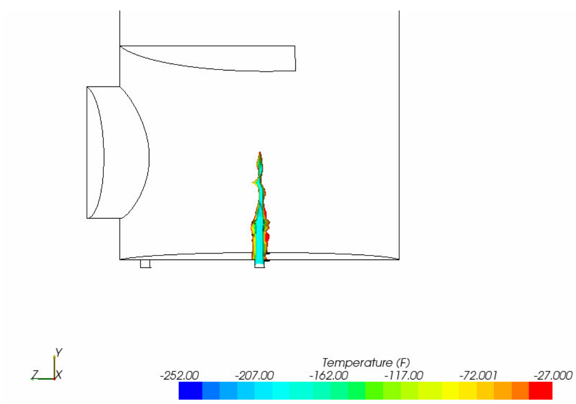

Figure 6 – Isosurface showing areas within the vaporizer subject to freezing

Figure 7 shows the temperature distribution at the outlet of the vaporizer. This value was queried to ensure that the proper energy balance within the exchanger was being maintained. The Appendix shows the calculations that were used to predict the outlet temperature of the vaporizer after the tube rupture occurred.

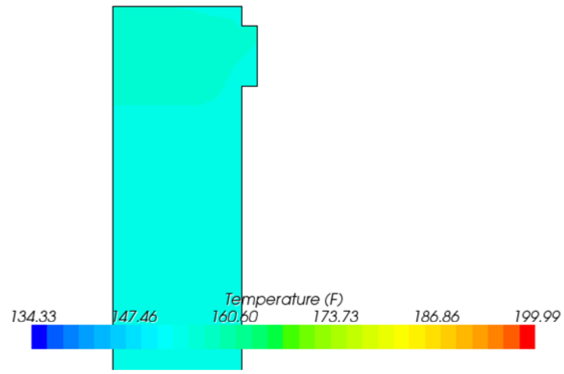

Figure 7 – Temperatures at the vaporizer water/glycol exit

Figure 8 contains the average pressure on the exterior wall of the vaporizer computational domain versus time for the pressure wave analysis.

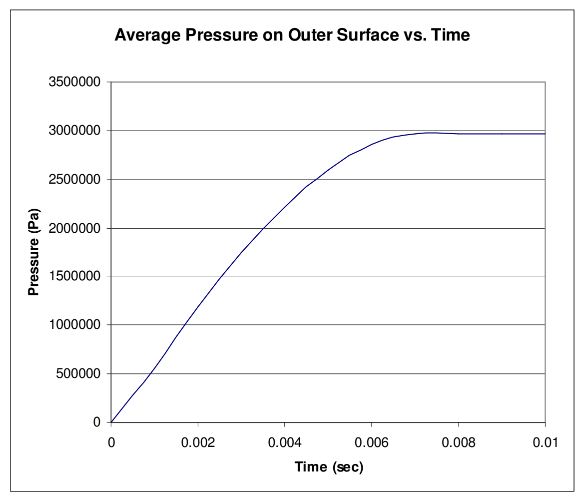

Figure 8 – Average outer wall pressure versus time for the transient pressure pulse analysis

DISCUSSION OF RESULTS

Evident in Figures 5 and 6 is the fact that the water freezing is confined to a localized area around the break. Due to the influx of the warm glycol/water mixture, only a small pocket of ice forms instead of spreading throughout a majority of the vessel. This indicates that very large pieces of ice capable of blocking the rupture discs will likely not form and that the rupture discs should function as expected.

As can be seen in Figure 7, the bulk exit temperature for the water is somewhat higher than the temperature predicted using the empirical techniques contained in the Appendix. Since the empirical techniques did not include the latent heat required to freeze the water (due to the volume of ice formed being an unknown quantity in the empirical calculations), the fact that the CFD temperature is somewhat above the empirical calculation is reasonable.

As can be seen from the pressure trace in Figure 8, the entire exchanger experiences the pressure pulse within 0.007 seconds. The theoretical time for the pulse to traverse the vessel at the speed of sound in water is 0.0054 seconds. This indicates that the baffles only increase the travel time of the wave through the exchanger by approximately 30%.

CONCLUSIONS

Using the boundary conditions and assumptions outlined in this paper, it can be shown that after the break event, the region where freezing occurs is limited to a small area downstream of the break and is terminated at the first baffle. This indicates that a large volume of ice is not formed within the vaporizer and that the probability of plugging the rupture disc should be low.

The pressure pulse associated with a double guillotine break will traverse the extent of the vaporizer in approximately 0.007 seconds. This indicates that the disc rupture is an almost instantaneous event after the tube rupture occurs. Since this rupture will occur almost instantaneously, some degree of pressure relief on the shell side will occur simultaneously with the tube rupture event. As discussed below, the amount of relief associated with the disc rupture would need to be evaluated using a more sophisticated modeling approach.

One fundamental assumption made to support these conclusions is that the glycol/water on the shell side continues to flow at a continuous rate. That is, the glycol/water flow must be able to overcome the pressure pulse without drastically changing the water flow rate into the vaporizer. If the inflow rate of the glycol/water mixture is affected by the pressure pulse or by the characteristics of the pressure until relieved within the vaporizer, the warm water mixture flow rate would need to be modified from the value used in this model.

One of the tenets of this assumption is the fact that water flow continues and the system pumps do not trip due to the pressure pulse. To determine if a pump trip is possible, analyses would have to be conducted on the piping system, from the pumps to the vaporizer shell-side inlet, to determine the pulse characteristics at the pump. These characteristics could then be compared to the pump’s operational characteristics to determine if a trip event would be initiated. The results of this analysis would also provide a time-history of the glycol/water flow rates into the vaporizer. These values could then be incorporated into additional analyses.

Additionally, further studies should be conducted to assess the specific quantity, location and size of the relief devices (rupture discs) required for this application. These studies would need to simultaneously take into account the dynamics of the pumping system (variable flow rates of glycol/water feed) and the dynamics of the flow occurring within the vaporizer and at the rupture discs. To accomplish this, an iterative analysis procedure using a piping model for the upstream piping network and a CFD model of the vessel would need to be employed. In this case, the predicted pressures within the vaporizer would be used as inputs for the outlet pressure of the piping model. The piping model would then be used to conduct a new time-history analysis of the vaporizer inlet flow. This inlet flow would then be input into the CFD model for a new analysis. This procedure would be repeated until an acceptable level of convergence is achieved between the transferred boundaries.

To consider the flow dynamics within the vaporizer, it is necessary to account for the change in volume due to the vaporization of the LNG. This vaporization would displace a large amount of fluid that could only exit the vaporizer at the rupture disc locations and at the glycol/water inlet and outlet locations. This condition would appear to be the limiting basis for sizing the desired capacity of the rupture disc. If the disc capacity is not sufficient, imposing a flow restriction at these locations, or if the flow path to the rupture disc is too restricted, then it is probable that even with an actuated rupture disc the vaporizer would be subjected to pressures higher than the operational pressure. This would be caused by the high liquid flow rates at the outlets that would be required due to the volume displaced by the gasified LNG. It is expected that these higher pressures within the system would continue until the vaporized gas stream reached an exit location. This operation at a higher pressure would likely reduce the water flow rate into the vaporizer, and in the worst case, would cause the flow at the shell-side inlet to reverse and leave the vaporizer at this location. In these situations, the amount of ice formed within the vaporizer could be considerably more than the amounts predicted in these analyses.

REFERENCES

[1] API 521, Guide to Pressure-Relieving and Depressuring Systems, Fourth Edition, March 1997, Paragraph 3.18

[2] Hodge, B.K., Analysis and Design of Energy Systems, Second Edition, Prentice Hall, Englewood Cliffs, New Jersey, 1990, Figure 2-18 Tube-bank friction and correction factors for staggered tubes

[3] Star-CCM+ v. 2.04.001, CD-Adapco, Melville, NY

[4] Fluent v. 6.0, Lebanon, NH

Appendix

Preliminary Heat Transfer Calculations

$D_o := 1 \cdot in$	Tube OD	$t := .065 \cdot in$	Wall thickness (BWG 16)
$D_i := D_o - 2 \cdot t$	Tube ID	$\rho_w := 62 \cdot \frac{lbf}{ft^3}$	Density of water
$A := \pi \cdot \frac{D_i^2}{4}$	Open area	$C_d := 0.82$	Thick orifice
$P1 := 350 \cdot psi$	Tube pressure	$P2 := 104.7 \cdot psi$	Shell pressure
$h := \frac{P1 - P2}{\rho_w}$	Head on orifice		
$Q_v := C_d \cdot A \cdot \sqrt{2 \cdot g \cdot h}$	Volumetric flow rate	$Q_v = 0.648 \frac{ft^3}{s}$	
$\rho_{lng} := 26.8 \cdot \frac{lb}{ft^3}$	Density of LNG	$C_{plng} := .8 \cdot \frac{BTU}{lb \cdot F}$	
$H_{lng} := 260 \cdot \frac{BTU}{lb}$	Latent heat of LNG		
$Q_m := 2 \cdot Q_v \cdot \rho_{lng}$	Mass flow rate of LNG		
$Q_w := 1975802 \frac{lb}{hr}$	Mass flow rate of water		
$C_{pw} := 0.9 \cdot \frac{BTU}{lb \cdot F}$	Specific heat of water		
$\Delta T := \frac{Q_m \cdot H_{lng}}{Q_w \cdot C_{pw}}$	Mixing temperature change of water - due to latent heat only		$\Delta T = 18.287 F$

Due to mixing, we get:

$\Delta T_{lng} := \frac{200 \cdot F + 273 \cdot F}{1 + \left(\frac{Q_m \cdot C_{plng}}{Q_w \cdot C_{pw}}\right)}$	Change in LNG temperature due to mixing with water	$\Delta T_{lng} = 447.803 F$
$T_{lng} := -253.4 F + \Delta T_{lng}$	LNG temperature	$T_{lng} = 194.403 F$
$T_{mix} := T_{lng} - \Delta T$	Final mixture temperature	$T_{mix} = 176.117 F$