A Case Study of Pipework Fracture due to Hydraulic Shock in an Ammonia System

March 7, 2008

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Abstract

In June 2007 a deflagration occurred at a frozen food factory. The incident investigation identified a ruptured coil located in a spiral freezer as the source of the ammonia leak. Within a 20 minute time frame, the ammonia concentration in the room surrounding the freezer enclosure increased to a flammable level and subsequently a deflagration occurred. The arcing found in a wire inside a drinking fountain was the likely ignition source. This freezer design, including the control valve group and liquid transfer vessel, is very typical for the refrigeration industry. The freezer and its associated refrigeration infrastructure were in operation for more than 10 years and underwent 3 Process Hazard Analyses. The incident investigation concluded that a hydraulic shock caused by a vapor propelled liquid slug generated enough of a transient pressure spike to cause coil rupture. A detailed analysis of the control valve group and liquid transfer vessel design and system dynamics, in conjunction with a metallurgic fracture analysis, was used to develop a mathematical model to describe and reconstruct the mechanism of this incident. The results are quite startling and support the fact that a seemingly insignificant system upset has the potential of leading to a catastrophic event. The incident investigation identified additional safeguards that could have prevented this incident.

Presented at the 2009 IRC Forum, Madison, Wisconsin

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Introduction

Hydraulic shock is a phenomenon that frequently occurs in industrial refrigeration. However, most hydraulic shock events result in transient pressure spikes well below the bursting pressure of the connected equipment. In many circumstances, it takes two or three unusual events to occur simultaneously for a transient pressure spike to develop that exceeds the bursting pressure of the equipment leading to a catastrophic failure. What is not commonly known is the magnitude of the driving force of the hydraulic shock and the mechanisms leading to equipment failure. Many incidents in the past have not been thoroughly documented. It is very fortunate that the incident described here provided enough data and clues to allow an exact replication of the events leading up to this catastrophic event.

A common belief within industrial refrigeration is that deflagrations are very uncommon. The cause of deflagrations was commonly thought to be oil mixed with the ammonia, thereby lowering the concentration range for the flammability of the ammonia. However, this incident proved that a deflagration of a pure air-ammonia mixture can easily occur if an ignition source with sufficient energy is present.

The equipment that enabled the hydraulic shock in the incident was a valve group in combination with a liquid transfer vessel serving a low temperature freezer. This valve group was designed without any safeguards, consistent with most valve groups in the industry. For example, within this industry a certain type of main suction valve is still commonly being used which “slams” into the open position upon power failure or without positive feedback that the differential pressure has been lowered to a safe level. Many of the older valve stations do not have a “soft” hot gas valve, allowing a gradual pressure build up before the main hot gas valve opens. Most new valve stations that do have a “soft” hot gas valve do not use any positive feedback that the pressure has increased to a sufficient level before the main hot gas valve is opened. Upon high liquid level due to a valve or control failure, commonly no alarms are provided and the defrost sequence is commonly not aborted. Most valve groups designed in the past and still being designed today do not include any safeguards that protect the system in case of a system upset or failing equipment such as sticking solenoid valves, burned coils, failing level controllers, oil fouling, etc. This paper is intended to provide insight and to provide cause for implementing additional safeguards, especially in the design of large valve group stations and when large evaporator coils are involved.

Incident Summary as Reported to the NRC/EPA

Incident Type:	Gas Release, Liquid Spill/Evaporation, Deflagration
Name of Chemical Released:	Ammonia
Total Quantity Released:	2,423 lb_m
Primary Source of Release:	Evaporator
Initiating Event:	Equipment Failure
Causes Contributing to Release:	Equipment Failure, Over Pressurized Equipment, Hydraulic Shock
Types of Changes Recommended to Prevent Recurrence:	Improved or Upgraded Equipment
Wind Speed and Direction:	5 mph SW
Temperature:	74 °F
Stability Class:	E or F
Precipitation Present:	None
Weather Conditions:	Partly Cloudy
On-Site Impacts	
Number of Deaths:	0
Number of Injuries:	0
Number Evacuated:	150 (Associates and Contractors)
Number Sheltered in Place:	0
Property Damage:	+$7M
Known Off-Site Impacts	
Number of Deaths:	0

Number of Hospitalizations:	0
Number Receiving Other Medical Treatment:	0
Number Evacuated:	150 (Neighboring Facilities)
Number Sheltered in Place:	0
Property Damage:	0
Environmental Damage:	None
Notification of Offsite Emergency Responders:	Offsite Responders Notified and Responded

Sequence of Events

Approximately 1:20 AM to 1:25 AM:

Refrigeration technician initiates hot gas defrost sequence on Line 4 spiral freezer.

Approximately 1:45 AM to 1:48 AM:

A severe ammonia leak is detected in the Line 4 spiral freezer area (lower level) by the refrigeration technician who initiated the defrost.

Approximately 1:55 AM:

Factory evacuation of 150 people is initiated and followed per factory emergency response procedures.

Approximately 2:05 AM:

Refrigeration technicians wearing canister-type respirators measure ammonia concentrations below 300 ppm in upper level hallway. Upon entering Line 4 spiral freezer discharge area, an ammonia concentration of 14,000 ppm is measured using a portable ammonia detector. The refrigeration mechanics retreat immediately and initiate emergency shutdown of the ammonia refrigeration system.

Approximately 2:08 AM:

A deflagration occurs. Local and county Fire and HAZMAT crews respond to the scene. Per protocol, a Situation Control Incident Commander, in collaboration with the command of the local emergency authorities takes charge of the incident.

Description of Line 4 Spiral Freezer and Valve Group Design

The Line 4 spiral freezer was installed and placed in service in 1993. The freezer is located on the lower level of the two story factory. Since the elevation difference of the suction riser between the outlet of the evaporator and the main suction header is more than 40 ft, performance penalties were noted shortly after the freezer was placed in operation. To improve the performance of the freezer, a liquid transfer vessel was installed at the evaporator outlet of the freezer with the suction line sloping from the evaporator outlet to the liquid transfer vessel. The purpose of the liquid transfer vessel was to separate liquid from the vapor and pump it, via a mechanical pump, to the main suction header located on the roof. A schematic of this arrangement is shown in Figure 1. Upon detecting an operating level above the set point, a liquid level controller (LC) energizes the liquid transfer solenoid valve located between the pump and the main suction header to permit pumping the liquid to the main suction header located on the roof. When the operating level of the vessel is below the set point, a liquid bypass solenoid valve located between the pump opens to bypass liquid back to the liquid transfer vessel.

The Line 4 spiral freezer evaporator consists of three individual coil sections, each with its own liquid feed header and suction header. The material of construction of the coil tubing and headers is galvanized steel. The three individual coil suction headers are connected to a 10” main suction header. The Line 4 spiral freezer typically operated with an evaporating temperature between -50 °F and -45 °F.

Description of the Defrost Sequence in an Ideal Scenario:

See Figures 1 and 2 for reference.

a) Time: 0 minutes: Upon initiation of the defrost cycle, the “liquid feed solenoid valve” closes. This initiates a pump out cycle. Ideally, the pump out cycle is initiated well before the last product leaves the freezer so the residual product can provide an adequate heat load to evaporate some of the ammonia remaining in the coils. The system pressure is 14.3 inHg (7.66 psia or -7 psig).

b) Time: 23 minutes: The "main suction valve" closes, the fans turn off and the "soft start hot gas valve" opens. The system pressure in the evaporator coil, liquid transfer vessel and associated piping is 14.3 inHg (7.66 psia or -7 psig).

c) Time: 23 minutes to 25 minutes: The system pressure in the evaporator coil, liquid transfer vessel and associated piping increases from 14.3 inHg (7.66 psia or -7 psig) to 40 psig.

d) Time: 25 minutes: The "main hot gas valve" opens. The system pressure in the evaporator coil, liquid transfer vessel and associated piping is 40 psig.

e) Time 25 minutes to 33 minutes: Actual defrosting begins as the pressure in the evaporator coil, liquid transfer vessel and associated piping pressure is increased from 40 psig to 70 psig.

f) Time: 33 minutes: Most of the ice is melted. Water defrost is initiated and city water (approximately 55 °F) is sprayed over the coils to rinse off ice residue and to clean the coils. The system pressure is 70 psig.

g) Time: 35 minutes: Both hot gas solenoid valves close and the water continues to spray over the coils. The system pressure remains at 70 psig.

h) Time: 38 minutes: The water defrost solenoid valve closes. Fan #1, Fan #2 and Fan #3 are turned on to spin off any residual water. The "pressure bleed valve" opens. The system pressure is 70 psig.

i) Time: 41 minutes: Fans are stopped. System pressure is 70 psig.

j) Time: 46 minutes: The system pressure has been reduced from 70 psig to 5 psig. The "pressure bleed valve" closes and the "main suction valve" opens.

k) Time: +46 minutes: The system pressure has been reduced from 5 psig to 14.3 inHg (7.66 psia or -7 psig). The freezer is now in "standby" mode and is waiting for the "freezer ready" mode to be initiated.

Description of Coil Rupture

A visual inspection of the Line 4 spiral freezer evaporator coils confirmed that two out of the three suction manifolds had ruptured. A schematic of the evaporator coil indicating the location of the ruptures and photos of the actual ruptures are provided in Figure 3. A visual examination of the left suction header (#1 in Figure 3) confirmed that it had experienced circumferential cracking in close proximity to the left end plate. This cracking extended approximately 240° around the circumference of the header. The cracking stopped progressing when it intersected the header-to-tube welds. This detail is provided in Figure 4. The right end plate was completely detached from the body of the header and wedged between the left and middle suction headers as shown in Figure 3. Figure 5 provides a close-up photograph.

A visual examination of the center suction header (#2 in Figure 3) confirmed that the branch connection joining the branch to the suction header had experienced cracking. This cracking, which extended over the full length of the branch connection, is shown in the close-up photographs provided in Figure 6. This cracking was oriented roughly parallel with the longitudinal axis of the branch connection. As it approached the upper flange connection it bifurcated, resulting in the detachment of a triangular segment.

A visual examination of the right suction header (#3 in Figure 3) and the 10” main suction header did not reveal any noticeable defects. There was no evidence of cracking or significant dimensional changes.

The results of the preliminary visual examination indicated that the cracking in both of the failed headers was the result of a sudden overload pressure condition. There was no evidence of significant weld defects, pre-existing cracking or other significant service-related deterioration that could have caused or contributed to the failure of these headers.

Initial Findings

The sudden overload condition that caused the header ruptures could only have been generated from a hydraulic lockup of the evaporator coils or a hydraulic shock. An examination of the valve group design did not reveal any possibilities for hydraulic lockup under normal operating conditions. The defrost condensate back pressure regulator and safety relief valve mounted on the liquid transfer vessel (the safety relief valve is not shown in Figure 3) would have provided sufficient relief points to relieve any excessive pressure built up on a more gradual time interval. None of the stop valves were closed at the time of the coil rupture. All automatic valves were removed from the system and tested with compressed air. None of the solenoid valves, including connected check valves, appeared to be in the stuck open or closed position. All solenoid coils tested in working order and appeared to be strong enough to open the solenoid valves using compressed air as a test medium. However, all pressure regulator pilot valves and pressure regulators failed when tested using compressed air. A further analysis, which included the water defrost sequence, ruled out a sudden overload condition caused by hydraulic lockup.

Focusing on the liquid transfer vessel design, it became apparent that this vessel could create a liquid trap during abnormal operating conditions, e.g. if the vessel was overfilled. This potential liquid trap is shown in Figure 7. With a large enough driving force, such as a large hot gas valve opening, this trapped liquid could form a vapor propelled liquid slug, which upon impact with an end cap, could generate a hydraulic shock.

Upon examination of the liquid level control system, the following was found:

- The terminal strip as shown in Figure 8 was not pushed all the way in. In the position shown, neither the liquid transfer solenoid valve nor the bypass solenoid valves could be activated. When pushing the terminal strip upwards by approximately 1/64” the liquid transfer solenoid valve would open when simulating a high level. To activate the bypass solenoid valve the terminal strip needed to be pushed further upward. After the terminal strip was pushed all the way onto the socket, a screw driver was required to break the strip loose from the socket again. The possibility that the terminal strip became loose from the deflagration was ruled out.
- The high level relay was not used or interlocked with any other devices.

- The low level relay was not interlocked with the refrigerant pump. Even when there was no liquid in the vessel, the pump would operate when turned on.

Therefore, it was reasonable to assume that the liquid transfer vessel was not transferring the liquid during the time of the incident, i.e. the vessel and associated piping would have been flooded at the time of the coil rupture.

An interview with the night shift refrigeration technician revealed the following:

- On several occasions in the past, the refrigeration pump had failed leaving the Line 4 spiral freezer without an operational liquid transfer vessel for weeks. This situation would have caused the same scenario as described above, i.e. the vessel and associated piping would have over-filled with ammonia liquid. When this situation occurred in the past, the equipment did not experience any unusual operational problems.
- The night of the incident the refrigeration technician was attending to a problem on the Line 6 spiral freezer. This prevented him from initiating the hot gas defrost on the Line 4 spiral freezer right after the last product left the freezer. Instead, the hot gas defrost was initiated 20 – 30 minutes after the last product left the freezer. This delay means ammonia liquid was being supplied to the freezer and the fans were running with no product in the freezer.
- The approximate time frame when the refrigeration technician initiated the hot gas defrost was between 1:20 AM and 1:25 AM.

Based on the ammonia concentration measured in the waste water treatment holding tank, it was determined that at least 1,900 lb_m of ammonia flowed down the drain.

The hand metering valves feeding the Line 4 spiral freezer were inspected. Valve #1 was 2¼ out of 7 turns open, valve #2 was 6 out of 7 turns open and valve #3 was 3½ out of 7 turns open. The nominal pipe size and the size of the hand metering valves is 1½". Both hand metering valves down stream of both hot gas solenoid valves were fully open.

Incident Analysis

Based on the above findings it is reasonable to assume that the root cause for the coil rupture was a severe hydraulic shock generated by a vapor-propelled liquid slug originating from the liquid transfer vessel which had, effectively, created a liquid trap. The hot gas defrost was initiated between 1:20 AM and 1:25 AM. The coil rupture occurred between 1:45 AM and 1:50 AM. Inserting these times onto the time chart provided in Figure 2, this sequence puts the opening of the large hot gas valve around the same time the coil rupture occurred.

If a vapor-propelled liquid slug was the initiating factor for this coil rupture, several contributing factors were likely to have been present simultaneously:

- Liquid was likely present in the 10” main suction header as a source for the liquid slug.
- The evaporator coil was likely filled to the top to transmit a shock wave, through the liquid, to the manifolds that ruptured.
- The hot gas introduced into the system likely provided enough driving force to propel and to accelerate a liquid slug to a high enough velocity, whereupon impact of the liquid slug with the end cap the resulting pressure excursion (hydraulic shock) likely exceeded the bursting pressure of the ruptured manifolds.

Analysis of the Liquid Slug Source

As discussed previously, the liquid transfer vessel created an ideal trap for a liquid slug to form. This equipment is shown in Figures 1 and 7.

Analysis of Evaporator Coil Flooding

The Line 4 spiral freezer was rated at 320 TR and the design overfeed ratio of the coil was 6:1. The evaporator coil contained a total volume of 84.14 ft^3 for all three coils. At an evaporating temperature of -50 °F this would correspond to a liquid mass of 3,662 lb_m when completely filled to the top. Based on engine room log sheets and post incident testing, the pressure differential across the hand metering valves was determined to be approximately 20 psi. Based on the opening degree of each hand metering valve, the following c_v values were determined using the manufacturer's product manual: hand metering valve #1: c_v value = 2.37, hand metering valve #2: c_v value = 14.0, and hand metering valve #3: c_v value = 5.74. Based on the above c_v values and the differential pressure, the following mass flow rates were calculated using standard fluid dynamic formulas: hand metering valve #1: 1.23 lb_m/s, hand metering valve #2: 7.27 lb_m/s, and hand metering valve #3: 2.98 lb_m/s. This corresponds to a total liquid mass flow of 11.48 lb_m/s through all three coil sections combined.

The Line 4 spiral freezer had 3 centrifugal fans each with 50 HP motors. Post incident testing determined that the actual fan motor power consumption was 34.0 kW_e with no product in the freezer. Assuming that 100% of the fan motor power consumption is absorbed as heat by the evaporator coil, the heat load at the end of the freezing cycle is equivalent to 29.0 TR. This is equivalent to a refrigerant mass flow evaporation rate of 0.160 lb_m/s.

To put the above into perspective, if the Line 4 spiral freezer would have been in freeze mode without any product, at temperature, but the coil completely empty of liquid, it would have only taken 5 min 23 sec to fill the coil completely with liquid. Conclusively, the evaporator and associated piping were severely over-filled at the time the hot gas valve opened.

B. Wiencke [IIAR (2000) and IIAR (2002)] presented a useful set of formulas to compute the void fraction of evaporators and wet return lines. With only the fans providing a heat load, the individual evaporator coils would have been filled 57%, 70% and 64% respectively, i.e. the evaporator coil would have contained a total liquid mass of 2,331 lb_m. The total associated suction pipe from the evaporator outlet to the main suction valve was estimated at 46.27 ft^3. Based on the above load condition the suction pipe would have been filled 82% with liquid, i.e. the suction pipe would have contained a total liquid mass of 1,650 lb_m.

When the hot gas defrost cycle was initiated thereby starting the pump-out cycle, the liquid feed solenoid valve closed, and with the main suction valve remaining in the open position, the fans kept running for 20 minutes providing heat load to the evaporator coil. During this pump-out cycle 192 lb_m of ammonia would have evaporated. After the fans stopped and with no heat load in the freezer, the liquid in the evaporator coil and associated pipe work would have settled with most of the liquid filling the evaporator coil. With 2,331 lb_m of liquid initially contained in the evaporator, 1,650 lb_m initially contained in the associated piping system, 192 lb_m evaporating during the pump-out cycle, and 107 lb_m remaining in the liquid trap, 3,685 lb_m of liquid ammonia would have settled in the evaporator coils and its 10" main header. Since the evaporator coils were capable of only holding 3,662 lb_m, this means that the evaporator coils were completely filled with liquid and the remaining 23 lb_m would have likely settled in the 10" main header. As with all two-phase flow calculations, especially at low load conditions, the computed results should not be viewed as 100% accurate, but instead an error of margin of ±30% should be accounted for. In summary, based on the above analysis, it is very likely and plausible that the evaporator coil was completely filled with liquid when the pump out cycle was terminated.

A Simplified Model Describing the Mechanism Leading to the Coil Rupture

The last contributing factor to be analyzed is the driving force propelling the liquid slug. This requires an in-depth analysis and a simplified model to describe the mechanisms leading to the coil rupture. It was suggested above that the coil rupture must have been caused by an excessive pressure excursion resulting from the hydraulic shock. This hydraulic shock has two components and it is believed that it was initiated by a combination of a "vapor-propelled liquid slug" (VPLS) and "condensation induced shock" (CIS). A "vapor-propelled liquid slug" is the movement of a liquid refrigerant slug propelled at high velocity by high pressure vapor in pipes. In "condensation induced shock," a vapor void surrounded by sub-cooled surfaces such as liquid or piping suddenly collapses (condenses). The in-rushing liquid filling the void generates a shock as it collides with pipe walls and liquid surfaces.

In order for a hydraulic shock to occur, liquid and vapor must be present and liquid at high velocity must come to rest in a very short time interval (nearly instantaneously). All recorded and known cases of hydraulic shocks that have occurred within the ammonia refrigeration industry occur at evaporating temperatures below -20 °F.

Figure 9 is a simplified illustration of the actual piping arrangement. In order to physically describe and understand the mechanisms that lead to the pipe rupture, a few assumptions and calculations are required. As a first step it must be assumed that the pressure excursion caused by the hydraulic shock exceeded the bursting pressure of the components affected by the shock.

Determining the Bursting Pressure

To determine the bursting pressure of the piping and manifolds affected by the hydraulic shock, Barlow's formula [Lester, C. B. (1958)] is used. Barlow's formula is commonly used to predict the bursting pressure of ductile thin wall [$(t_w/D) < 0.1$] piping and tubing under ideal conditions using the ultimate tensile strength:

$$P = 2 \cdot S \cdot t_w / D$$

where:

P = bursting pressure [psi]

S = minimum tensile strength [psi]

t_w = wall thickness of pipe [in]

D = outside pipe diameter [in]

The minimum tensile strength of ASTM Specification A-53 Grade B and A-106 Grade B pipe is 60,000 psi. The minimum tensile strength of ASTM Specification A-214 ERW pipe is 47,000 psi (Source: ASME B31.5-2001). The calculated bursting pressures using Barlow's formula are shown in Table 1.

Since the liquid and piping material was operating at a temperature of -50 °F it must be assumed that temperature embrittlement reduced the actual bursting pressure to a value below the values listed in Table 1. This will be addressed in more detail in a later section.

Nominal Pipe Size	10”	8”	5”	1”
Schedule	40	40	40	-
Pipe OD [in]	10.75	8.625	5.563	1.000
Wall Thickness [in]	0.365	0.322	0.258	0.049
Minimum Tensile Strength [ksi]	60	60	60	47
Bursting Pressure [psig]	4,074	4,480	5,565	4,606

Table 1: Bursting Pressure Calculations for Various Pipe Sizes

Estimating the Minimum Liquid Slug Velocity Required to Burst the Pipe

As mentioned previously, a hydraulic shock occurs when a liquid slug moving at high speed through a vapor space comes to rest suddenly. This would happen if the liquid slug impacts a wall or another liquid surface that is at rest. Joukowski’s (1898) formula allows the computation of the pressure rise (transient pressure) due to the hydraulic shock:

$$dp = \rho_l \cdot a_l \cdot v_l / (144 \cdot g_c)$$

solving for velocity:

$$v_l = 144 \cdot dp \cdot g_c / (\rho_l \cdot a_l)$$

where:

v_l = liquid slug velocity [ft/s]

dp = pressure rise due to hydraulic shock [psi]

g_c = gravitational constant at 32.2 $ft \cdot lb_m/(lb_f \cdot s^2)$

ρ_l = liquid density [lb_m/ft^3]

a_l = speed of sound in liquid within the pipe [ft/s]

The speed of sound in liquid within a pipe is calculated as follows:

$$a_l = \frac{a_o}{\sqrt{1 + \frac{d}{t_w}\frac{K}{E}}}$$

where:

a_o = speed of sound in liquid in a perfectly rigid pipe [ft/s]

d = inside pipe diameter [in]

t_w = wall thickness [in]

K = bulk modulus of elasticity of the liquid [psi]

E = modulus of elasticity of pipe material [psi]

At -50 °F the properties of saturated liquid ammonia and carbon steel pipe material are: $\rho_l = 43.5\ lb_m/ft^3$, $a_o = 6092$ ft/s, $K = 0.348 \cdot 10^6$ psi and $E = 28.7 \cdot 10^6$ psi (source: NIST, "Thermophysical Properties of Fluid Systems").

The results for calculating the liquid slug velocity are listed in Table 2.

Nominal Pipe Size	10"	8"	5"	1"
Pipe ID [in]	10.02	7.981	5.047	0.902
Wall Thickness [in]	0.365	0.322	0.258	0.049
Speed of sound [ft/s]	5,277	5,342	5,477	5,508
Bursting Pressure [psig]	4,074	4,480	5,565	4,606
Liquid Slug Velocity [ft/s]	81.8	89.4	108.3	89.1

Table 2: Minimum Liquid Slug Velocities for Various Pipe Sizes and Bursting Pressures.

The Effect of Pressure on Vapor Void Collapse (Condensate-Induced Shock)

For a vapor void to collapse it must be in contact with sub-cooled surfaces such as liquid or piping. The open literature dealing with industrial refrigeration offers little insight concerning the magnitude of sub-cooling that is required for a vapor void to collapse. L. Loyko [IIAR (1989) and IIAR (1992)] has suggested that 60 F of sub-cooling is required for this phenomenon to occur. The saturated pressure of ammonia at -50 °F is 7.66 psia and at 10 °F it is 38.5 psia.

C. Glennon and R. Cole [IIAR (1998)] suggested that much less sub-cooling is required for a vapor void to collapse. They base their conclusion on the critical pressure ratio of the vapor. However, neither a source nor any explanation for the basis of the assumption is provided. For this scenario at the conditions described the critical pressure ratio of ammonia is 0.536 (source: NIST, "Thermophysical Properties of Fluid Systems") with a corresponding pressure of 14.3 psia. The corresponding saturated temperature is -29 °F, which is equivalent to sub-cooling of 21 F. The suggestion by C. Glennon and R. Cole could be considered a lower limit for the sub-cooling required and the suggestion by L. Loyko an upper limit for the sub-cooling required.

The mechanism for the condensation induced shock to occur would be as follows. Initially the liquid slug is at rest and the vapor is at a pressure of 7.66 psia. As the liquid slug moves out of its resting position towards the evaporator coils and end cap, the vapor space is reduced and consequently the pressure within the space increases to 14.3 or 38.5 psia respectively before the vapor void collapses.

The equivalent pipe length of the vapor space is 35 ft. A 35 ft long 10" pipe holds a volume of 19.2 ft^3 and a total of 0.579 lb_m of ammonia vapor. Assuming the vapor remains saturated during the compression cycle (due to the surrounding cold liquid and metal surfaces), the vapor space is reduced to 10.7 ft^3 when the pressure reaches 14.3 psia and to 4.23 ft^3 when the pressure reaches 38.5 psia. Therefore, the liquid slug would have to travel 15.5 ft to compress the vapor from 7.66 psia to 14.3 psia and 27.3 ft to compress the vapor from 7.66 psia to 38.5 psia. After the liquid slug traveled 15.5 ft or 27.3 ft respectively, the vapor void collapses and the liquid slug is further accelerated to a higher velocity.

Estimating the Hot Gas Mass Flow

The hot gas valve station consisted of one ½" and one 2½" solenoid valve with flow coefficients (c_V-values) of 3.3 and 77, respectively. The main function of the small solenoid valve is to slowly increase the pressure to a level where it is safe to activate the large solenoid valve. The time delay between the large and small solenoid valve was set to

2 minutes. Due to the large volume of the system and the excessive liquid accumulated in the system, it can be assumed that the small solenoid did not build up any significant pressure due to vapor condensing on the cold liquid surface and metal surfaces. The refrigeration technicians familiar with the dynamics of the Line 4 spiral freezer confirmed that after 2 minutes of opening the small hot gas valve, the pressure never exceeded 0 psig in the past. The pressure usually built to approximately 12.2 psia (5 inHg) by the time the large hot gas valve opened. Post incident readings on two other spiral freezers (Line 6 and Line 7) also equipped with a liquid transfer vessel and virtually the same configuration as Line 4 Spiral Freezer confirmed that after 2 minutes the hot gas pressure did not exceed 0 psig. Due to the excessive amount of liquid accumulation inside the evaporator coil and its associated piping, the assumption is made that the pressure did not increase significantly above 7.66 psia. Therefore, a pressure of 7.66 psia shall be used the moment the large hot gas valve opens.

Based on historical data and post incident readings, it was estimated that the hot gas supply pressure upstream of the hot gas solenoid valves was approximately 135 psig (149.7 psia) at a temperature of approximately 110 °F. Since the pressure ratio of the down stream pressure to upstream pressure is less than the critical pressure ratio, choked flow through the valve can be assumed:

$$\frac{p_b}{p_1} \leq \left[\frac{2}{k+1}\right]^{k/(k-1)}$$

where:

p_b = back pressure at outlet of valve [psia]

p_1 = pressure at inlet of valve [psia]

k = isentropic exponent at the valve inlet [-]

Inserting the actual values into the above equation yields:

$$\frac{7.66}{149.7} \leq \left[\frac{2}{1.43+1}\right]^{1.43/(1.43-1)} \quad \Rightarrow \quad 0.051 << 0.523$$

In this scenario, choked flow means that the mass flow through the valve will remain constant as long as the back pressure at the outlet of the valve is less than 78 psia.

The mass flow through the valves can be calculated with the following formula [API 520-2000]:

$$W = A \cdot C \cdot K \cdot P_1 \cdot K_b \cdot \sqrt{\frac{M}{T \cdot Z}}$$

where:

W = capacity [lb_m/h]

A = effective discharge area [in^2]

C = coefficient determined from the specific heats of ammonia

K = effective coefficient of discharge of valve

P_1 = pressure upstream of valve [psia]

K_b = capacity correction factor due to back pressure

M = molecular weight of ammonia

T = absolute temperature of the fluid at the valve inlet [R], (R = °F + 460)

Z = compressibility factor

Based on the manufacturer's data, the effective discharge area for the small and large valves is 0.196 in^2 and 4.91 in^2 respectively. The effective coefficient of discharge for the type of valve used is: K = 0.46 for the small valve and K = 0.42 for the large valve. With an inlet condition of 149.7 psia and 110 °F, the specific values are: C = 347, K_b = 1 for choked flow, M = 17.03, T = 570 R, Z = 0.9075.

For the small valve the capacity is calculated as follows:

$$W = 0.196 \cdot 347 \cdot 0.46 \cdot 149.7 \cdot 1 \sqrt{\frac{17.03}{570 \cdot 0.9075}}$$

$$W = 849.8 \text{ lb}_m/\text{h or } W = 0.2361 \text{ lb}_m/\text{s}$$

For the large valve the capacity is calculated as follows:

$$W = 4.91 \cdot 347 \cdot 0.42 \cdot 149.7 \cdot 1\sqrt{\frac{17.03}{570 \cdot 0.9075}}$$

$$W = 19{,}437\ lb_m/h \text{ or } W = 5.399\ lb_m/s$$

Estimating the Driving Force to Accelerate the Liquid Slug

The following fundamental equations can be used to calculate the acceleration of the liquid slug and the time traveled:

$$v_l = \frac{dx}{dt}$$

$$a = \frac{dv_l}{dt}$$

where:

v_l = velocity of liquid slug [ft/s]

x = distance traveled [ft]

t = time traveled [s]

a = acceleration [ft/s^2]

assuming that x = 0 and v = 0 at t = 0:

$$a = \frac{1}{2}\frac{v_l^2}{x}$$

$$t = \frac{v_l}{a}$$

To simplify this problem statement, the acceleration of the liquid slug is assumed to be constant while traveling from its origination point (x = 0 ft) to the point of impact (x = 35 ft).

Newton's 2nd Law (law of acceleration) can be used to estimate the driving force, F for the slug:

$$F = \frac{d(m \cdot v_l)}{dt}$$

with m being constant and $v_l = 0$ at $t = 0$ the equation can be integrated to the following expression in IP-units:

$$F = a \cdot m / g_c$$

where:

F = driving force for the slug [lb_f]

a = acceleration of slug [ft/s^2]

m = mass of slug [lb_m]

g_c = gravitational constant at 32.2 $ft \cdot lb_m/(lb_f \cdot s^2)$

By dividing the equation with the cross sectional area of the pipe and omitting any friction losses, the driving pressure for the slug can be estimated as:

$$\Delta p = \frac{a \cdot m}{g_c \cdot A}$$

Substituting *a* with the above velocity term, and considering that the pipe diameter is constant, the equation can be rewritten as:

$$\Delta p = \frac{v_l^2 \cdot \rho_l \cdot L_s}{2 \cdot 144 \cdot x \cdot g_c}$$

where:

Δp = driving pressure for the slug [psi]

ρ_l = liquid density of slug [lb_m/ft^3]

L_s = length of slug [ft]

To calculate the mass flow rate entering the driving space upstream of the liquid slug and to determine the final temperature and specific volume of the driving space, an iterative process using conservation of mass and conservation of energy must be used.

Mass Balance:

$$\frac{dm_d}{dt} = \dot{m}_{in}$$

integrating and rearranging yields:

$$m_d = m_1 + \dot{m}_{in} \cdot \Delta t$$

Energy Balance:

$$\frac{dU_d}{dt} = \dot{m}_{in} \cdot h_{in}$$

$$\Downarrow$$

$$\frac{d(m_d \cdot u_d)}{dt} = \dot{m}_{in} \cdot h_{in}$$

$$\Downarrow$$

$$u_d \frac{dm_d}{dt} + m_d \frac{du_d}{dt} = \dot{m}_{in} \cdot h_{in}$$

substitution yields:

$$u_d \cdot \dot{m}_{in} + m_d \frac{du_d}{dt} = \dot{m}_{in} \cdot h_{in}$$

$$\Downarrow$$

$$\frac{du_d}{dt} = \frac{\dot{m}_{in}}{m_d}\left(h_{in} - u_d\right)$$

integration yields:

$$\Delta u_d \approx \frac{\dot{m}_{in}}{m_d}\left(h_{in} - u_d\right) \cdot \Delta t$$

or:

$$u_d \approx \frac{\dot{m}_{in} \cdot h_{in} \cdot \Delta t + u_1 \cdot m_d}{\dot{m}_{in} \cdot \Delta t + m_d}$$

where:

h_{in} = specific enthalpy of ammonia hot gas entering driving space [Btu/lb_m]

m_1 = mass of ammonia vapor in driving space at t = 0 s [lb_m]

m_d = mass of ammonia vapor in driving space after t seconds [lb_m]

$\dot{m}_{in}$ = hot gas mass flow entering the driving space [lb_m/s]

Δt = travel time of liquid slug traveling a distance x [s]

u_1 = specific internal energy of ammonia vapor in driving space at t = 0 s [Btu/lb_m]

u_d = specific internal energy of ammonia vapor in driving space after t seconds [Btu/lb_m]

The above energy balance does not consider any heat transfer to the surrounding pipe and liquid. Since the travel time for the liquid slug is very short (less than one second) it can be considered negligible. Also, to simplify this problem statement no condensation is considered within the driving space.

Possible Scenarios Leading to the Coil Rupture

To obtain a better understanding about the mechanics leading to the coil rupture, it is sometimes useful to work with a range of values or upper and lower limits. In the following scenarios a bursting pressure of 3,500 psi and 5,000 psi are used. A value of 3,500 psi would consider a derating factor due to temperature-induced embrittlement. Figure 10 illustrates the following scenarios with the slug at its initial resting location and Figure 11 illustrates the following scenarios with the liquid slug at its final resting position.

Scenario One:

This simplified scenario assumes that the vapor condenses as the liquid slug moves towards the end cap. The pressure remains constant at 7.66 psia. To simplify the matter, it is assumed that the vapor condenses at the same rate as the liquid slug travels downstream in the pipe. Using the previously introduced equations, the following values are computed:

Bursting pressure at 3,500 psig:

v_l = 70.7 ft/s

a = 71.4 ft/s^2

Δp = 3.015 psi

p = 10.674 psia

T = 42.5 °F

m_d = 1.492 lb_m

ν = 29.354 ft^3/lb_m

t = 0.990 s

$\dot{m}_{in}$ = 0.7559 lb_m/s (hot gas mass flow introduced into driving vapor space)

Bursting pressure at 5,000 psig:

v_l = 101 ft/s

a = 145.7 ft/s^2

Δp = 6.152 psi

p = 13.811 psia

T = 55.6 °F

m_d = 1.886 lb_m

ν = 23.232 ft^3/lb_m

t = 0.693 s

$\dot{m}_{in}$ = 1.648 lb_m/s (hot gas mass flow introduced into driving vapor space)

Conclusion: The large hot gas valve itself would be capable of producing this mass flow, but the small valve would not be capable of producing the required mass flow.

Scenario Two:

This scenario assumes that the liquid slug compresses the vapor to 14.3 psia and it then condenses at this pressure as the liquid slug moves towards the end cap. To simplify the matter, it is assumed that the vapor condenses at the same rate the liquid slug travels downstream in the pipe, i.e. the pressure remains at 14.3 psia.

Bursting pressure at 3,500 psig:

v_l = 70.7 ft/s

a = 71.4 ft/s^2

Δp = 3.015 psi

p = 17.315 psia

T = 64.2 °F

m_d = 2.330 lb_m

ν = 18.799 ft^3/lb_m

t = 0.990 s

$\dot{m}_{in}$ = 1.602 lb_m/s (hot gas mass flow introduced into driving vapor space)

Bursting pressure at 5,000 psig:

v_l = 101 ft/s

a = 145.7 ft/s^2

Δp = 6.152 psi

p = 20.452 psia

T = 69.7 °F

m_d = 2.729 lb_m

ν = 16.052 ft^3/lb_m

t = 0.693 s

$\dot{m}_{in}$ = 2.865 lb_m/s (hot gas mass flow introduced into driving vapor space)

Conclusion: The large hot gas valve itself would be capable producing this mass flow.

Scenario Three:

This scenario assumes that the liquid slug compresses the vapor to 38.5 psia and it then condenses at this pressure as the liquid slug moves towards the end cap. To simplify the matter, it is assumed that the vapor condenses at the same rate the liquid slug travels downstream in the pipe, i.e. the pressure remains at 38.5 psia.

Bursting pressure at 3,500 psig:

v_l = 70.7 ft/s

a = 71.4 ft/s^2

Δp = 3.015 psi

p = 41.515 psia

T = 87.8 °F

m_d = 5.426 lb_m

ν = 8.0735 ft^3/lb_m

t = 0.990 s

$\dot{m}_{in}$ = 4.729 lb_m/s (hot gas mass flow introduced into driving vapor space)

Bursting pressure at 5,000 psig:

v_l = 101 ft/s

a = 145.7 ft/s^2

Δp = 6.152 psi

p = 44.652 psia

T = 89.3 °F

m_d = 5.831 lb_m

ν = 7.5124 ft^3/lb_m

t = 0.693 s

$\dot{m}_{in}$ = 7.341 lb_m/s (hot gas mass flow introduced into driving vapor space)

Conclusion: The large hot gas valve itself would be capable of producing the lower mass flow. The large hot gas valve and small hot gas valve combined would not be able to produce the larger mass flow.

Employing the above formulas and scenarios, the actual bursting pressures using the total mass flow rate of both hot gas valves can be calculated using the above introduced equations and an iterative process. The total hot gas mass flow rate is 5.6351 lb_m/s.

Computed values based on scenario 1, i.e. the vapor void collapses at 7.66 psia:

v_l = 163.6 ft/s

a = 382.1 ft/s^2

Δp = 16.132 psi

p = 23.792 psia

T = 74.0 °F

ν = 13.883 ft^3/lb$_m$

t = 0.428 s

Estimated bursting pressure: 8,096 psig

Computed values based on scenario 2, i.e. the vapor void collapses at 14.3 psia:

v_l = 142.6 ft/s

a = 290.4 ft/s^2

Δp = 12.258 psi

p = 26.558 psia

T = 76.7 °F

ν = 12.477 ft^3/lb$_m$

t = 0.491 s

Estimated bursting pressure: 7,058 psig

Computed values based on scenario 3, i.e. the vapor void collapses at 38.5 psia:

v_l = 81.9 ft/s

a = 95.8 ft/s^2

Δp = 4.042 psi

p = 42.542 psia

T = 88.0 °F

ν = 7.8767 ft^3/lb$_m$

t = 0.855 s

Estimated bursting pressure: 4,053 psig

In conclusion, the above model and computed values support the theory that sufficient hot gas was available to be the driving force for a vapor-propelled liquid slug.

Conclusions Drawn from Modeling

The above analysis and model demonstrate that all criteria would have been met for a hydraulic shock with subsequent coil rupture to occur. However, it does not explain why the ruptures occurred in the suction header of the first coil and the riser of the second coil rather than in the main header, which according to the bursting pressure calculations would have been the weakest component. Based on the above data and analysis, one would have expected that the impact of the liquid slug would have detached the end cap from the main header and would not have generated a shock wave causing fractures in three different locations. Most catastrophic failures caused by hydraulic shock typically exhibit fractures in one location only. To gain additional insight into the fracture mechanics of this coil rupture, a testing laboratory was consulted. The testing laboratory conducted metallurgic testing and performed a fracture analysis.

Metallurgic Evaluation of Coil Headers

The results of the metallurgical evaluation confirmed that the suction header of the first coil and the riser of the second coil had failed catastrophically in a brittle manner as a result of a sudden internal over-pressurization. There was no evidence of significant design flaws, fabrication defects or pre-existing service-related deterioration that could have caused the failure of these headers.

Mechanical Testing of Headers

The results of the mechanical testing confirmed that the coil manifolds (suction headers and risers) exhibited tensile properties consistent with ASTM Specification A-53, Grade B. There were no deviations in the tensile properties of the suction headers and risers of the first and second coil that could have caused or contributed to their failure. However, the results of the mechanical testing suggested that the risers of all three coils were completely brittle at an operating temperature of -55 °F. The same was true of the suction header of the third coil. The suction headers of the first and second coils had somewhat better toughness but were still primarily brittle. The results of the mechanical testing confirmed that the 10” main suction header was ductile at an operating temperature of -55 °F. The results of the mechanical testing indicated that it would have taken a considerable amount of energy to initiate cracking in all four headers. However, once that cracking had initiated, very little energy would be required for crack propagation in the suction header of the first coil and the riser of the second coil. If these manifolds had been operating at temperatures above the Ductile-to-Brittle Transition Temperature (DBTT), significant additional energy would have been required for crack propagation.

For the suction header of the first coil the tensile strength was measured to 76,000 psi, the recorded thickness was 0.323”, and the outside diameter was 8.625”. This condition would result in a rupture pressure (due to ductile failure) of 5,675 psig.

For the riser of the second coil the tensile strength was measured to 72,250 psi, the recorded average thickness was 0.258”, and the outside diameter was 5.563”. This condition would result in a rupture pressure (due to ductile failure) of 6,702 psig.

For the 10” main suction header the tensile strength was measured to 67,750 psi, the recorded thickness was 0.350”, and the outside diameter was 10.824”. The estimated rupture pressure (due to ductile failure) is 4,530 psig.

Finite Element and Fracture Mechanics Analysis of Manifolds

In and effort to determine the approximate pressure at which the manifolds ruptured, a Finite Element Analysis (FEA) using ANSYS software and Fracture Mechanics using NASCRAC software was performed. Based upon the results of these analyses and the results of the metallurgical evaluation and mechanical testing performed, the suction header of the first coil likely ruptured at a pressure of around 3,500 psig. The riser of the second coil likely ruptured at a pressure around 2,450 psig. The results from the FEA, fracture mechanics and mechanical testing are summarized in Figure 12.

Ammonia Deflagration

As will be shown in a later section, it was estimated that approximately 5,300 lb_m of ammonia in liquid and vapor form escaped the Line 4 spiral freezer just before the deflagration. The author assumes that most of the liquid escaped the Line 4 spiral freezer through the scuppers. It then entered the floor drains. Since the freezer man-doors were left open to aid the defrost most of the ammonia vapor escaped into the lower level space where the spiral freezer was located. When the spiral freezer fans were turned on by the defrost control sequence ammonia vapor was also pushed out of the out-feed conveyor opening into other areas of the factory. The highest ammonia concentration would have accumulated outside the freezer man-doors. A post incident examination of the area adjacent to the spiral freezer revealed numerous tell-tale signs of an ammonia deflagration. Burn marks were found on plastic PVC piping, a paper towel dispenser and other objects. The characteristics and pattern of the burn marks pointed towards a drinking fountain that was installed approximately 2 weeks prior to the deflagration. Upon further examination it was determined that the arcing found in a wire inside the drinking fountain was the likely ignition source. The room layout and the location of the drinking fountain are shown in Figure 13. A photograph of the drinking fountain is provided in Figure 14. Also shown in the photograph is the paper towel dispenser and PVC piping showing burn marks. The flame propagation was limited to the room where the drinking fountain was located. It propagated around 10 ft east towards the Line 4 spiral freezer wall and around 25 ft south just beyond the conveyor belt feeding the spiral freezer. The approximate ignition area is also shown in Figure 13.

The pressure wave from the deflagration propagated through the entire building with most of the damage occurring near the spiral freezer. The suspended ceiling above the spiral freezer was completely destroyed, wall panels were blown out, portions of the roof were destroyed, numerous light fixtures were damaged beyond repair, and numerous pieces of equipment were damaged. Surprisingly, the Line 4 spiral freezer only sustained damage to the enclosure; the evaporator coil and the control panel only sustained water damage.

Based on telltale signs found in the immediate deflagration area, the area that was exposed to the flame front and combustion gas can be estimated to 532 ft^2. The ceiling height in this area is 14.5 ft, i.e. the equivalent room volume is 7,714 ft^3. A simplified calculation estimated the total amount of ammonia that ignited in the range of 17 lb_m to 54 lb_m.

Note: Ammonia explosions are technically termed deflagrations because the speed of the combustion wave propagates at subsonic speed with respect to the reactants. A detonation or explosion generates a combustion wave that propagates at supersonic speed with respect to the reactants, i.e. the detonation front propagates into unburned gas at a velocity higher than the speed of sound in front of the wave. The gas ahead of a detonation (explosion) is therefore undisturbed by the detonation wave. Due to its subsonic velocity a combustion wave originating from a deflagration does not have the characteristic of a shock wave and cannot be designated as such.

Summary of Events

Strong evidence suggests that a vapor-propelled liquid slug generated a hydraulic shock leading to the coil rupture. The sequence of events can be summarized as follows:

- Due to a failed liquid level controller and delayed initiation of the defrost cycle, the Line 4 spiral freezer, liquid transfer vessel and associated piping were filled with liquid in excess. See Figure 15.
- During the pump-out mode only a very small amount of ammonia evaporated. The pump-out mode is shown in Figure 16.
- At the end of the pump-out cycle, liquid settled in the system. The evaporator coil filled to the top and the trap formed by the liquid transfer vessel filled with liquid. The main suction line contained mainly vapor. See Figure 17.
- The soft hot gas valve opened, but was unable to build any significant amount of pressure. Most of the hot gas immediately condensed within in the system. See Figure 18.

- The large hot gas valve opened and the hot gas introduced into the system imposed a force on the trapped liquid. A liquid slug began to form and subsequently started to leave the liquid trap. See Figure 19.
- A liquid slug formed and was propelled by the hot gas down the main header towards the end cap. This is commonly referred to as a “vapor-propelled liquid slug”. See Figure 20.
- As the liquid slug traveled down the main header, vapor downstream of the slug was compressed and as the vapor reached a critical condition the vapor collapsed. This is commonly referred to as “condensation induced shock”. See Figure 21.
- The vapor propelled liquid slug passed the riser of the third coil and impacted with the end cap. Based on the results of the metallurgic testing, finite element and fracture mechanics analysis, the resulting hydraulic shock generated a transient pressure between 3,500 psig and 4,530 psig. The model and calculations presented in this paper would suggests a bursting pressure of 4,053 psig. The shock wave was transmitted through the header system and caused the suction header of the first coil to rupture in two places and the riser of the second coil to rupture in one place. See Figure 22.
- The resulting ammonia release occurred in a multi step sequence as the spiral freezer controller continued to cycle through the defrost cycle steps. During the first and initial step, the liquid contained in the liquid slug and the ruptured manifolds was released. The amount of ammonia contained in the liquid slug and ruptured manifolds was estimated to be approximately 107 lb_m and 60 lb_m respectively. See Figure 23.
- After the liquid from the liquid slug was released, the hot gas valves continue to supply hot gas into the system, which was consequently released through the ruptured manifolds. The amount of ammonia released during this 8 minute time frame was approximately 2,705 lb_m. See Figure 24.
- Eight minutes after the manifold rupture, the water defrost was actuated. This introduced a very high heat load into the evaporator coils for 5 minutes. The liquid ammonia inside the evaporator coils began to swell and evaporate violently. It is

reasonable to assume that 70% of the liquid contained in the evaporator coil escapes through the ruptured manifolds, which is equivalent to 2,500 lb_m. See Figure 25.

- Thirteen minutes after the manifold rupture, the evaporator fans were turned on for 3 minutes. This pushed more ammonia out of the freezer and into the factory. This allowed the ammonia concentration to build up to a flammable mixture outside the freezer. See Figure 26.
- After 18 to 22 minutes, the deflagration occurs. The arcing found in a wire inside a drinking fountain was the likely ignition source. The location of the drinking fountain in relation to the Line 4 spiral freezer and the freezer man-doors is shown in Figure 13.

Conclusions and Recommendations

The results of the mechanical testing identified that the ASTM A-53, Grade B piping used for the coil exhibited “complete brittle” and “primarily brittle” behavior at the operating temperature. The ASTM A-106, Grade B piping used for the main header exhibited “ductile” behavior at operating temperature. Based on the computed numbers in this incident analysis, this incident most likely could have been prevented if the coil manufacturer would have used ASTM A-106, Grade B material instead of ASTM A-53, Grade B. The difference in the impact properties between the two piping materials is of great interest and requires further explanation.

Based on the metallurgic testing conducted on the manifolds no “stress corrosion cracking” was present in any of the materials. The testing lab also confirmed that no “liquid metal embrittlement” was present in any of the galvanized piping materials which would have been ASTM A-53, Grade B. Liquid metal embrittlement shares many characteristics with stress corrosion cracking. The difference in the impact properties between the 10” main suction header and the coil manifolds is due primarily to differences in the manufacturing processes for the ASTM A-106 pipe and the ASTM A-53 pipe. Specifically, the steel used to manufacture the ASTM A-106 pipe has to be killed (de-oxidized) whereas the steel used to manufacture the ASTM A-53 pipe does not. Moreover, the ASTM A-106 pipe has to be either hot-finished or heat-treated to a temperature of 1200 °F or higher. ASTM A-53 does not not contain heat treating requirements. The microstructures exhibited by the 10" main suction header indicated that it was normalized whereas the coil manifolds were not. If the difference in the impact properties identified in this incident analysis is not unique to this incident, but representative of all ASTM A-53, Grade B and ASTM A-106, Grade B piping used in low temperature applications, this industry should consider prohibiting the use of ASTM Specification A-53, Grade B piping and materials with similar characteristics for applications at temperatures below -20 °F. It is interesting to note that both piping materials are treated equally in the ASME Boiler and Pressure Vessel Code, ASME B31.5 and ASME B31.3 which are applicable to this industry. This industry should conduct further studies and research to substantiate or negate the above theory and recommendation.

The above model describing the mechanisms leading to the coil rupture in conjunction with the results of the metallurgic testing, finite element and fracture mechanics analysis supports the suggestion that a minimum of 60 F sub-cooling is required for a "condensation induced shock" to occur. The calculated transient pressure based on the available hot gas being introduced into the driving space was computed to 4,053 psig. The results of the metallurgic testing, finite element and fracture mechanics analysis suggests that the transient pressure resulting from the hydraulic shock must have been within 3,500 psig and 4,530 psig. If only 21 F sub-cooling would be required, the transient pressure of 7,058 psig resulting from the hydraulic shock would have exceeded the bursting pressure of the 10" main header which was estimated at 4,530 psig. It is very likely that the end cap would have then separated from the pipe. The finite element and fracture mechanics analysis determined that the suction header of the first coil ruptured at 3,500 psig and the riser of the second coil ruptured at 2,450 psig. In conclusion, the model introduced in this paper in conjunction with the metallurgic testing and fracture analysis appears to accurately describe and replicate the mechanism that led to the hydraulic shock and coil rupture.

The ammonia refrigeration system analyzed in this incident investigation underwent three Process Hazard Analyses (PHA's) by three different reputable companies. A review of the

PHA's performed revealed that none of them addressed or identified the "latent defect" leading to the coil rupture or any other significant potential problems with the valve group and piping design.

Although the valve group and piping arrangement design does not deviate from current industry standards, this industry should take several lessons from this incident. In addition, this industry should start reconsidering some of the standards currently employed. The lessons to be learned from this incident and the recommendation arising from this incident can be summarized as follows:

1. The possibility of a vapor-propelled liquid slug was inherent in the design of the liquid transfer vessel. It was evident from the design that a liquid trap could easily be formed.

2. Use of ASTM A-106 or of ASTM A-333 piping material for the coil manifolds could minimize risks.

3. The equipment design lacked numerous safeguards. These safeguards, which could have been easily implemented, could have prevented this incident.

    i. The high level alarm of the liquid level controller connected to the liquid transfer vessel should have initiated an alarm. In addition, the high level alarm should have been interlocked with hot gas defrost sequence, i.e. the hot gas valves should have not been allowed to open upon high level.

    ii. The "soft hot gas" feature was controlled via timer settings. The timer setting of two minutes was too short of a period to build up hot gas pressure to a safe level of approximately 35 to 40 psig.

    iii. Most importantly, a pressure transducer or pressure switch should have been included in the design of the valve group and interlocked with the large hot gas solenoid valve. Had this safeguard been installed, it would assure that the large hot gas solenoid valve would not open if the pressure would had not reached 35 to 40 psig, thus preventing the incident.

iv. The main suction butterfly valve was of the “fail open” type. This means that upon a power failure or upon a failure of the bleed solenoid valve to open and release the pressure, this valve could have opened at defrost pressure (70 psig). This could have caused a vapor-propelled liquid slug in the main suction header servicing the entire low temperature system. A better choice of valve would be a valve that remains in its last position upon loss of power. A pressure transducer or pressure switch should have been interlocked with this valve to prevent it from accidentally opening if the differential pressure across the valve was above 10 to 12 psig.

4. An ammonia detector installed in the proximity of the spiral freezer could have shut down the spiral freezer to prevent the fans from blowing the ammonia into the factory and shutting down the power to the factory.

5. The design of the liquid transfer vessel did not consider the whole system. The pump used to transfer liquid from the liquid transfer vessel to the wet suction main header could have been used to recirculate the liquid through the evaporator without any additional cost. This concept would have given dual benefits of more efficient operation because the suction header would be dry and hence pressure drop would be reduced, and more reliable operation would result because the liquid level in the liquid transfer vessel would be controlled in a fail-safe manner.

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Figure 1: Schematic of evaporator, valve group and liquid transfer vessel arrangement of Line 4 Spiral freezer.

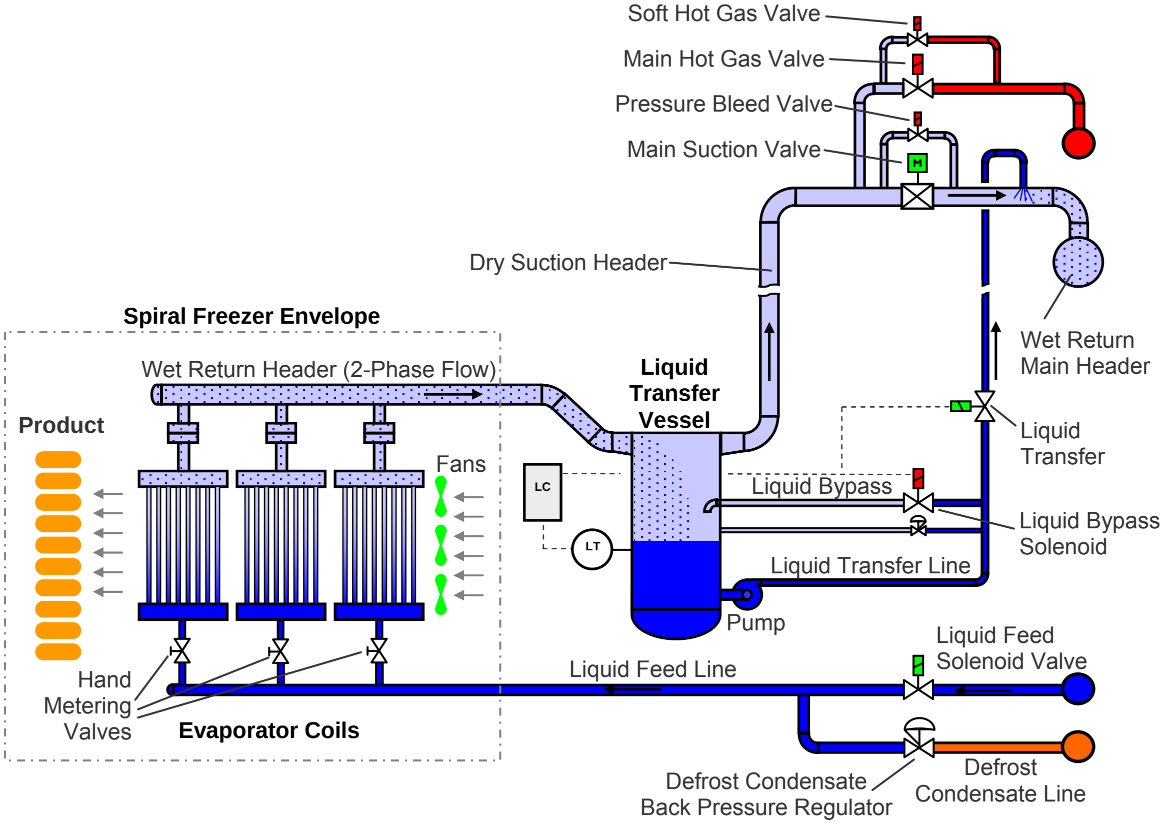


Figure 2: Defrost sequence of Line 4 spiral freezer.

Hot Gas Defrost Cycle

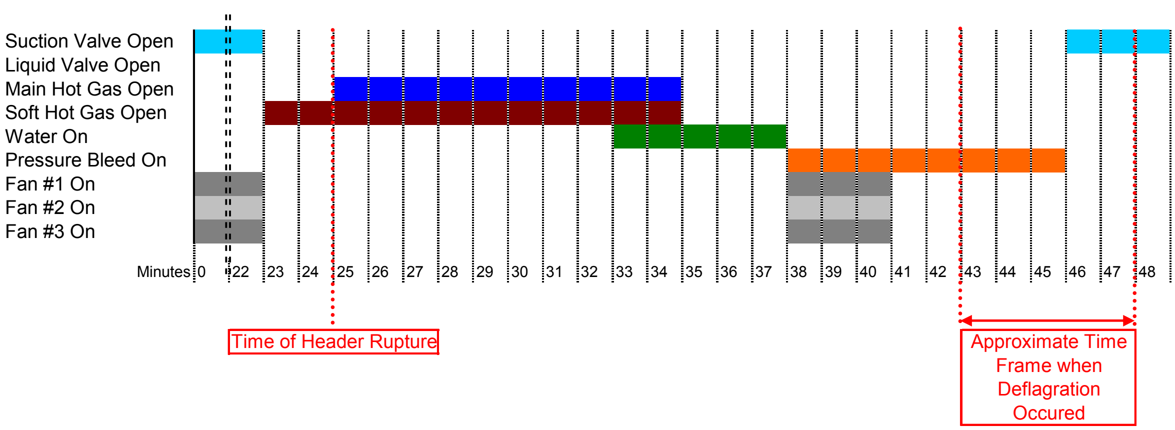


Figure 3: Location of coil ruptures and photographs of actual manifolds showing cracking.

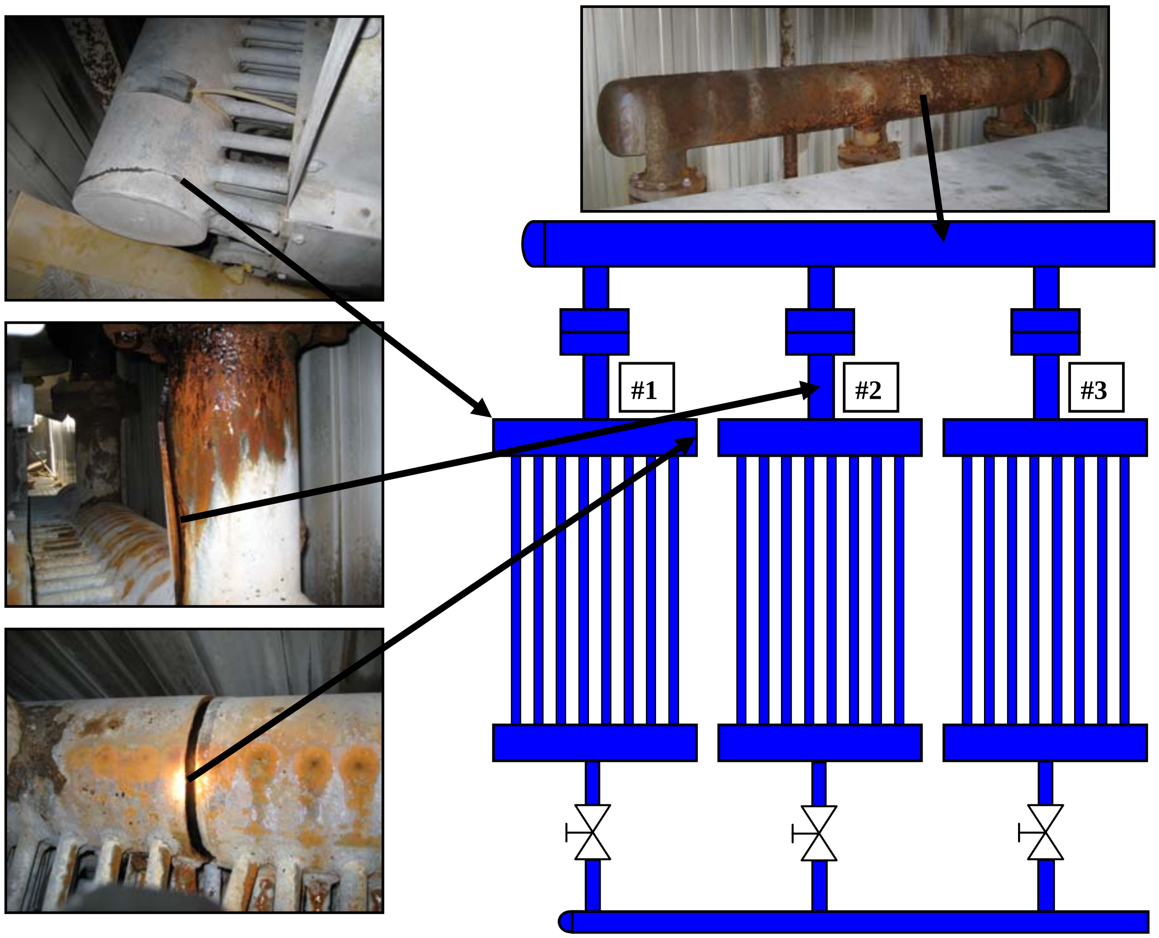


Figure 4: Close-up photographs of cracking adjacent to left end plate of suction header #1.

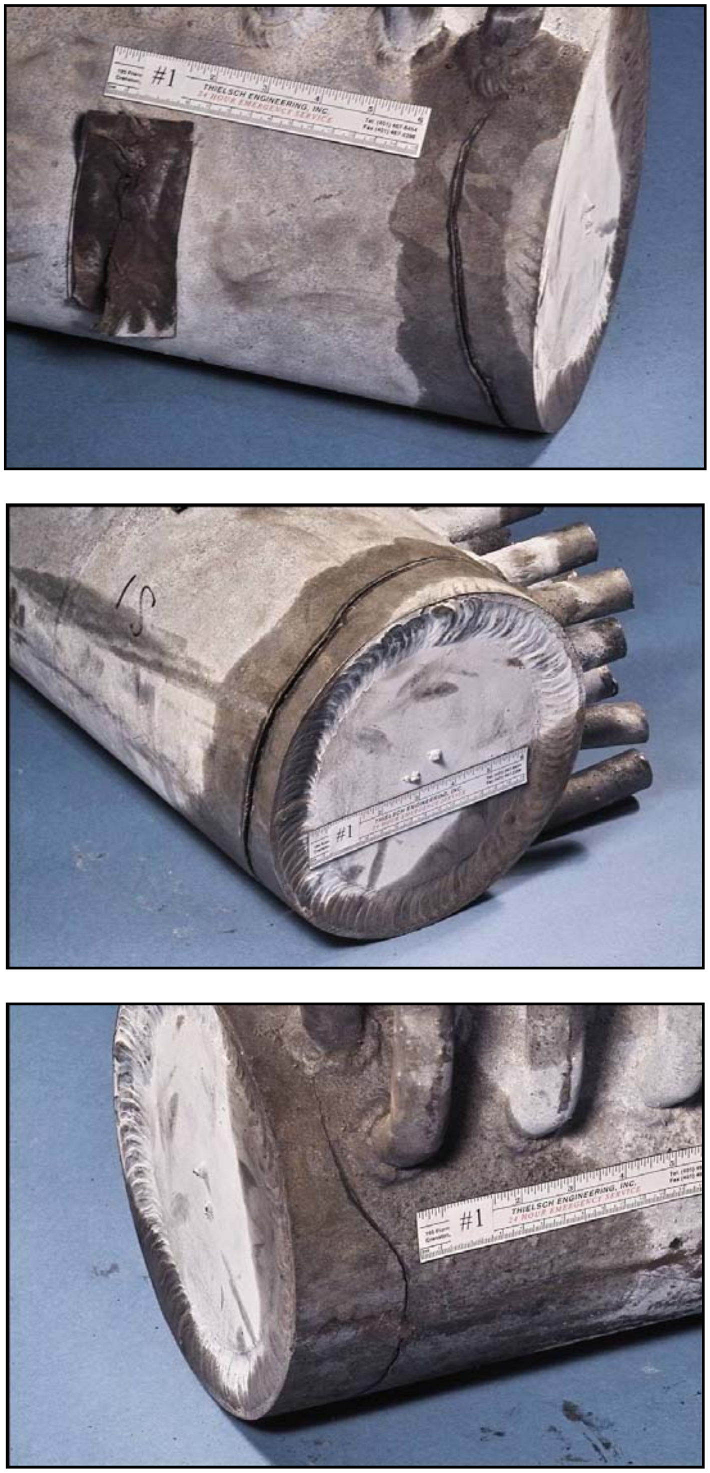


Figure 5: Close-up photographs of cracking adjacent to right end plate of suction header #1.

0°

180°

Figure 6: Close-up photographs of cracking in branch connection of suction header #2.

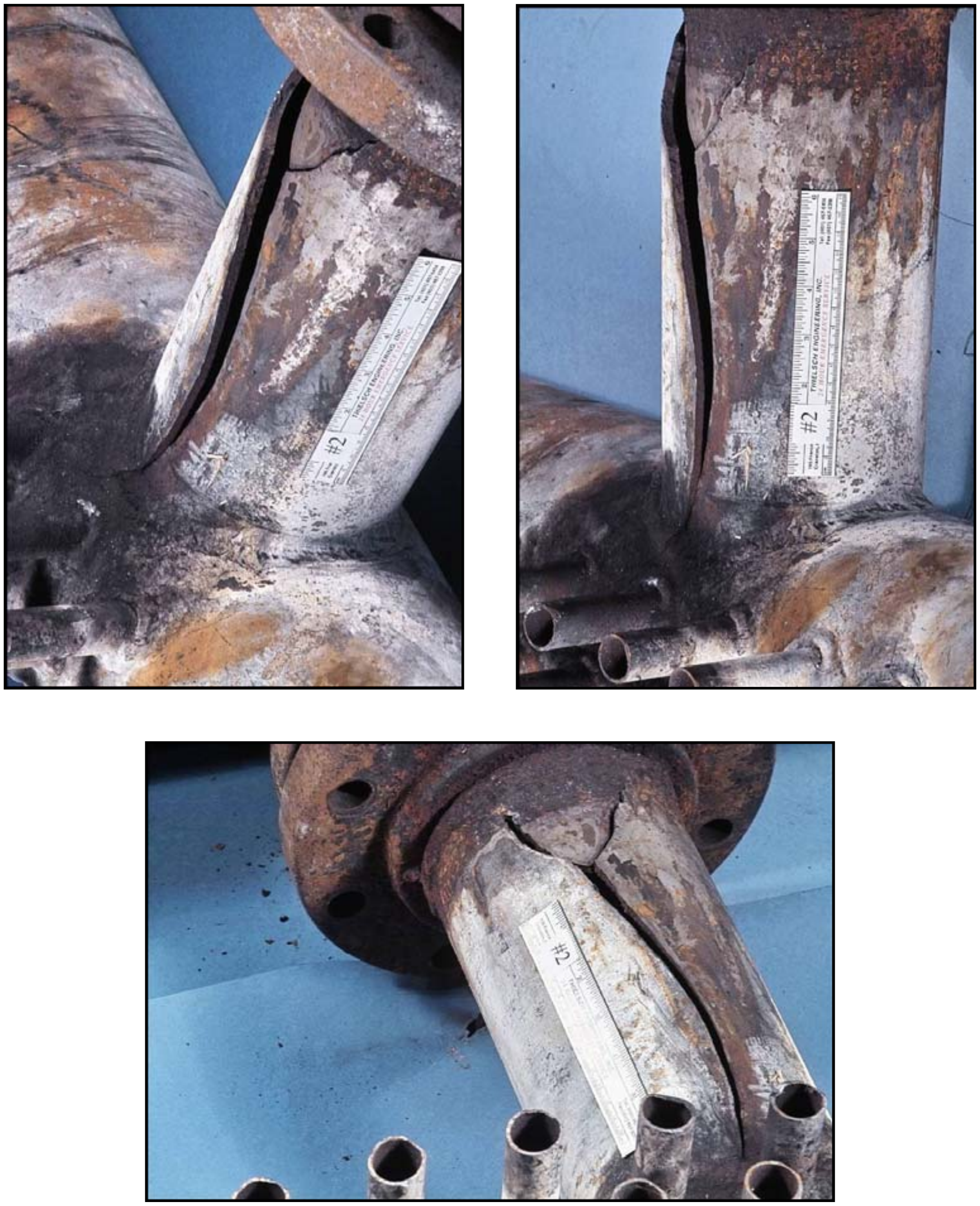


Figure 7: Post Incident photograph of liquid transfer vessel with insulation removed.

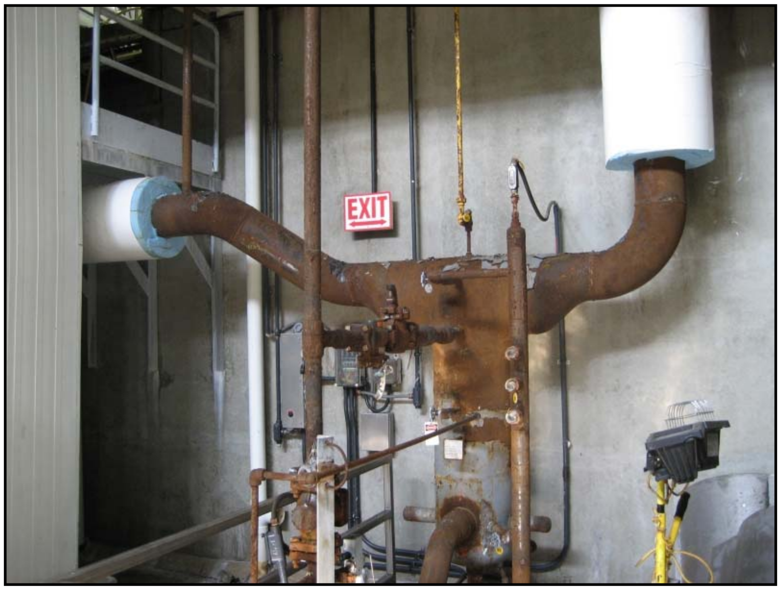


Figure 8: Loose terminal strip on liquid level controller.

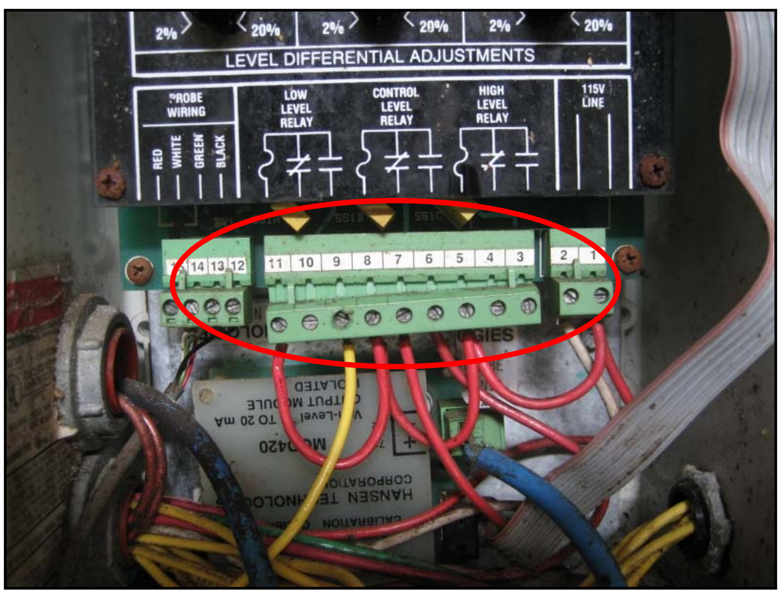


Figure 9: Simplified schematic of actual piping arrangement. Trapped liquid is at rest.

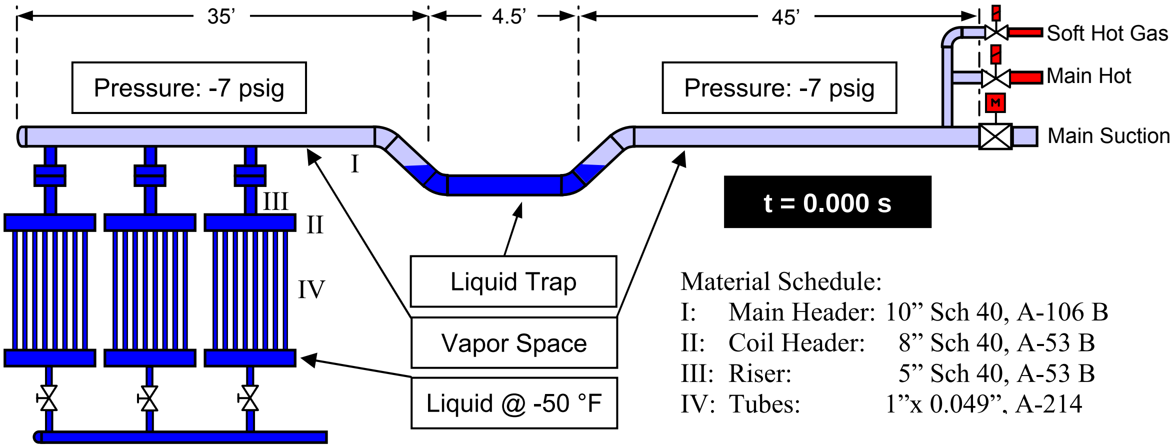


Figure 10: Liquid slug at its initial resting position with a vapor pressure of 7.66 psia (-7 psig).

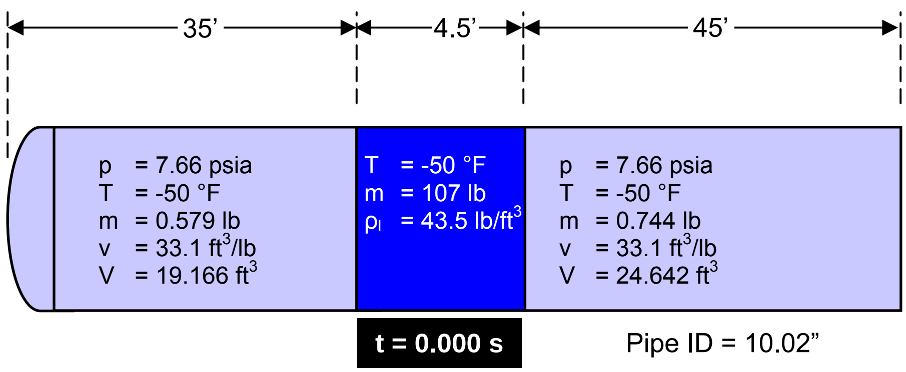


Figure 11: Liquid slug at its final resting position after traveling 35 ft.

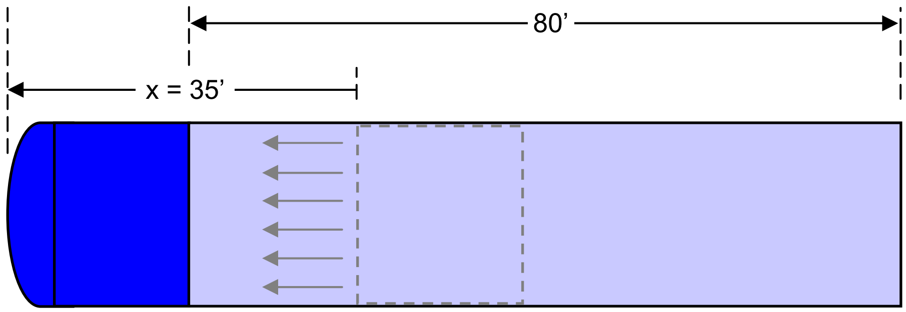


Figure 12: Material behavior of manifolds and actual bursting pressures.

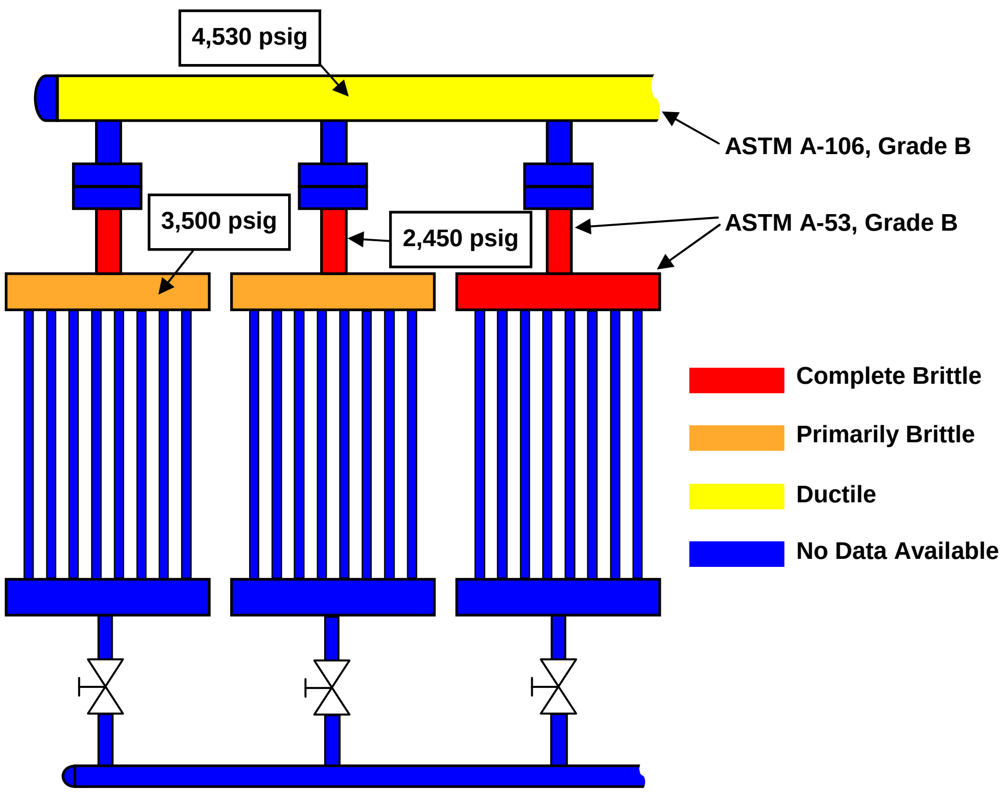


Figure 13: Schematic of Line 4 spiral freezer layout and location of drinking fountain.

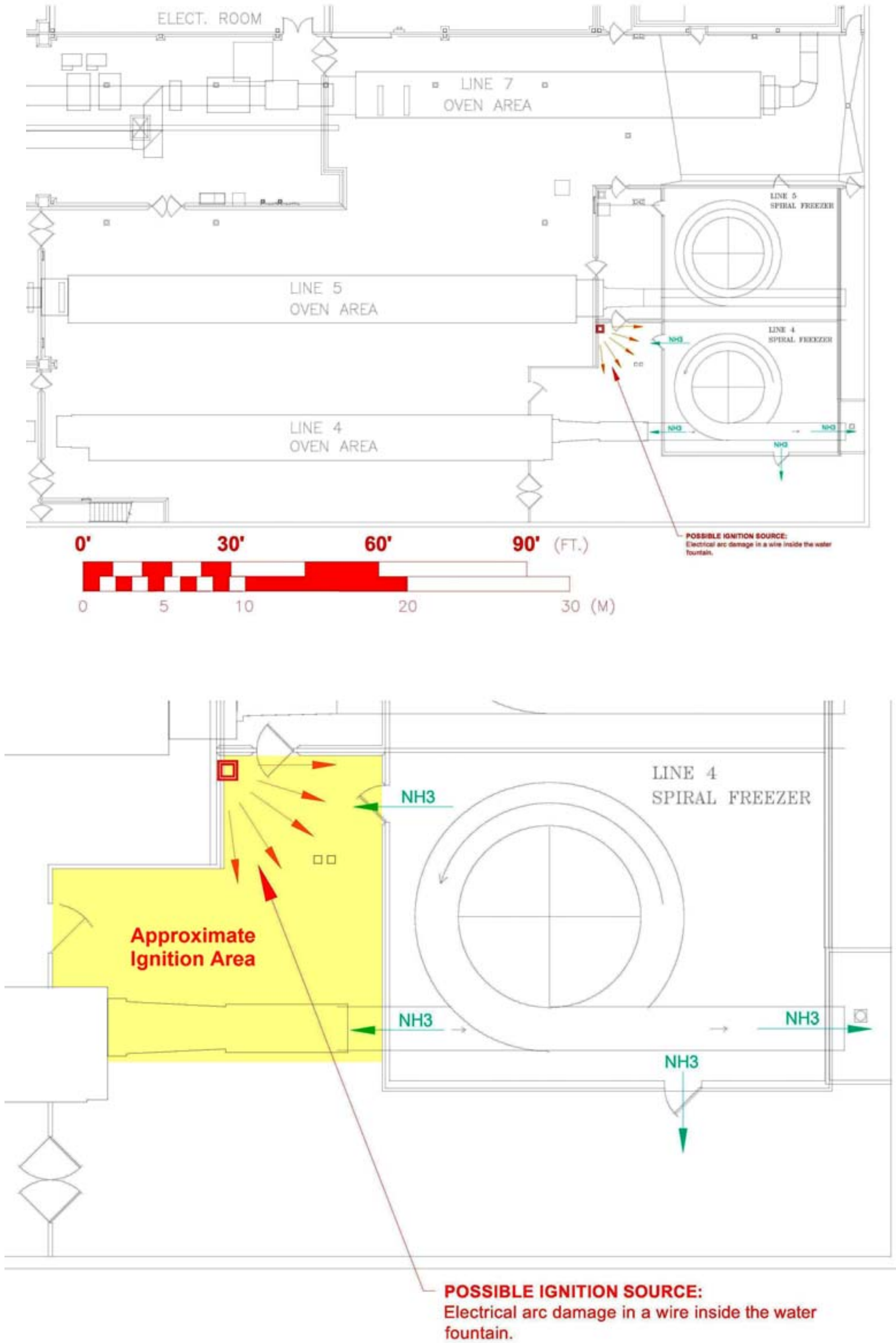


Figure 14: Photograph of drinking fountain. Also shown, burn marks on paper towel dispenser and PVC piping.

Figure 15: Schematic showing initial condition of spiral freezer just before initiating hot gas defrost. Freezer coils, liquid transfer vessel and associated piping are filled with liquid in excess.

Operating Mode: Freeze Mode

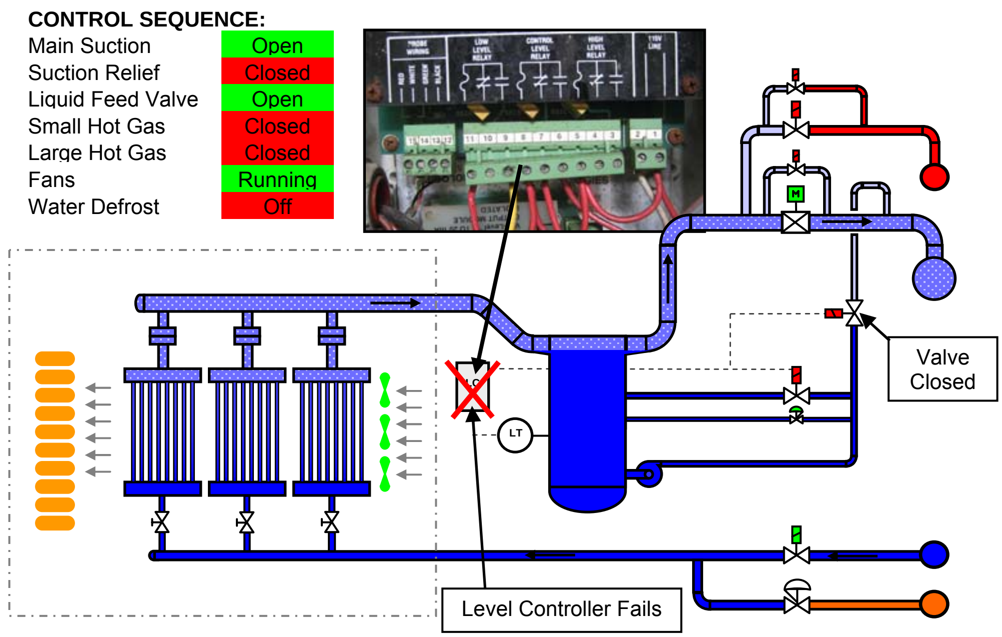


Figure 16: Schematic showing spiral freezer in pump out mode. Only a very small amount of ammonia evaporates in the evaporator coils.

Operating Mode: Defrost – Pump Out Mode

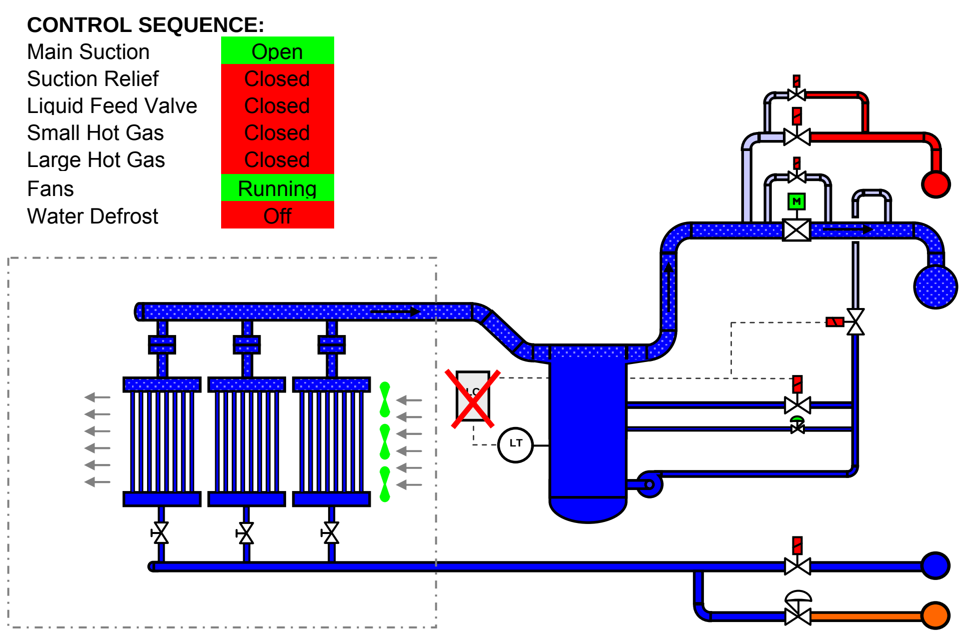


Figure 17: Schematic showing spiral freezer at the end of pump out cycle. Liquid settles in evaporator coils and liquid transfer vessel with liquid being trapped.

Operating Mode: Defrost – End of Pump Out Mode

CONTROL SEQUENCE:	
Main Suction	Closed
Suction Relief	Closed
Liquid Feed Valve	Closed
Small Hot Gas	Closed
Large Hot Gas	Closed
Fans	Off
Water Defrost	Off

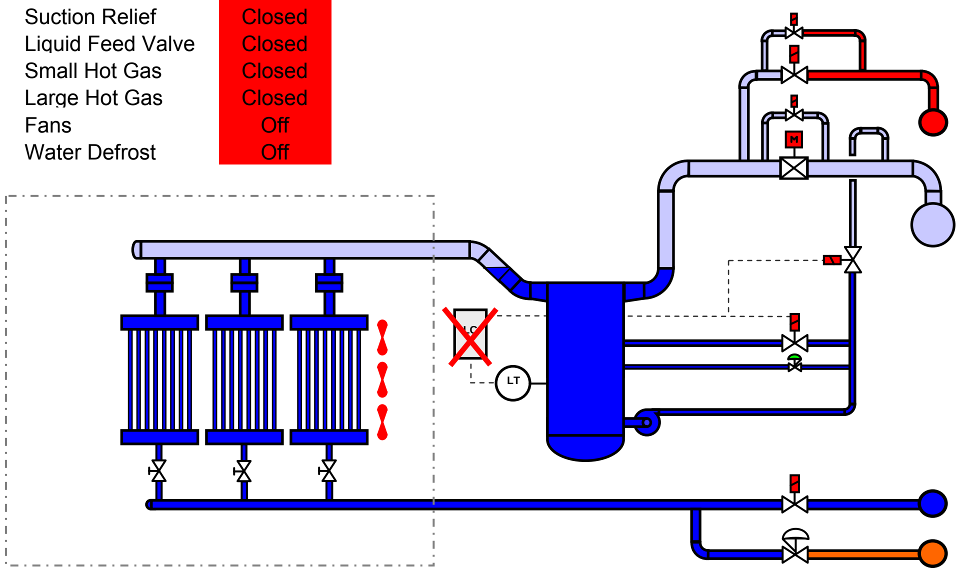


Figure 18: Schematic showing spiral freezer during soft hot gas defrost cycle.

Operating Mode: Defrost – Soft Hot Gas Defrost

CONTROL SEQUENCE:	
Main Suction	Closed
Suction Relief	Closed
Liquid Feed Valve	Closed
Small Hot Gas	Open
Large Hot Gas	Closed
Fans	Off
Water Defrost	Off

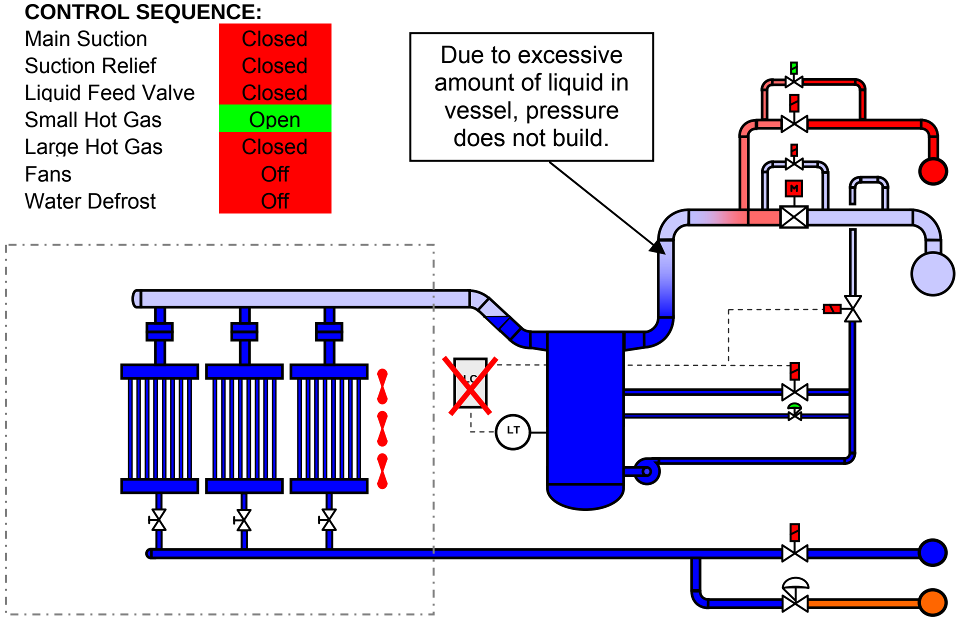


Figure 19: Schematic showing spiral freezer at the beginning of the hot gas defrost cycle with liquid slug beginning to form.

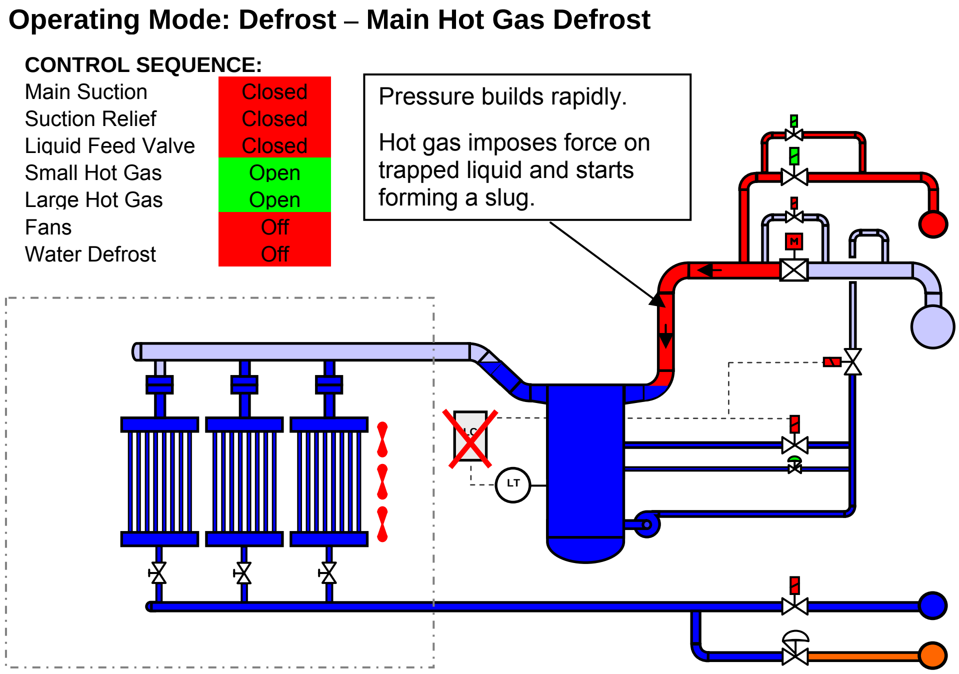


Figure 20: Schematic showing spiral freezer during the hot gas defrost cycle with "vapor propelled liquid slug" traveling down the main header towards the end cap.

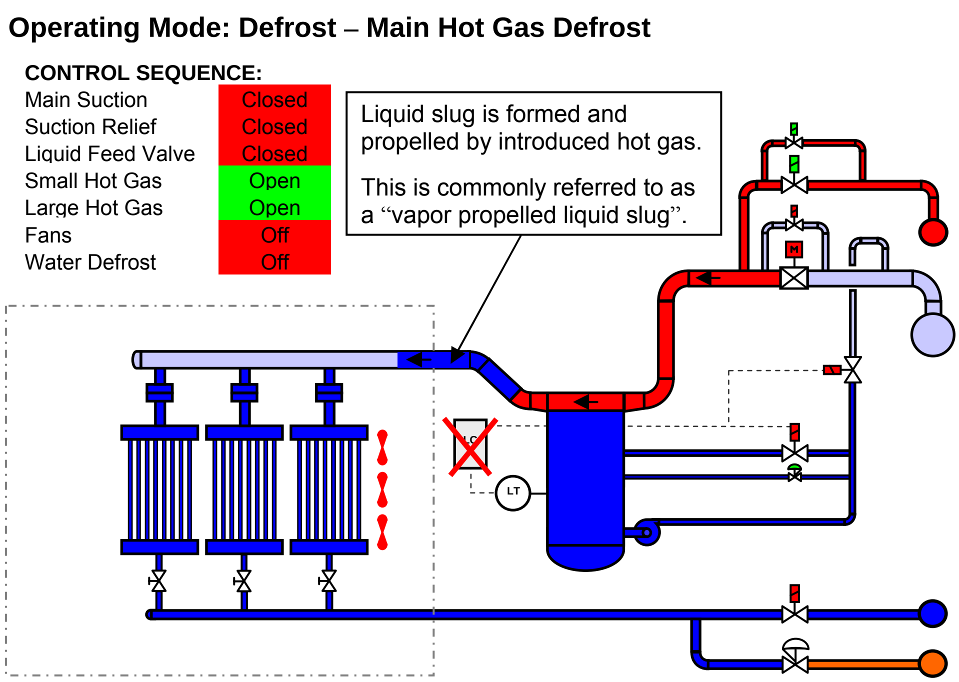


Figure 21: Vapor downstream of the liquid slug begins to collapse generating a "condensation induced shock".

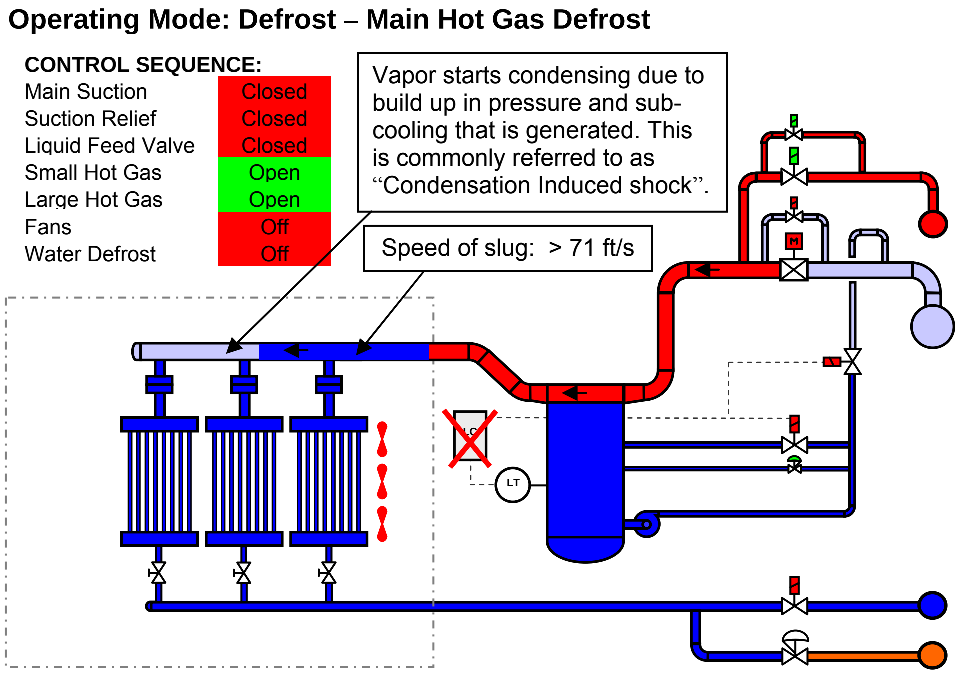


Figure 22: Impact of liquid slug with end cap and resulting hydraulic shock generates a transient pressure between 3,500 psig and 4,530 psig. Model and calculations suggests a bursting pressure of 4,053 psig.

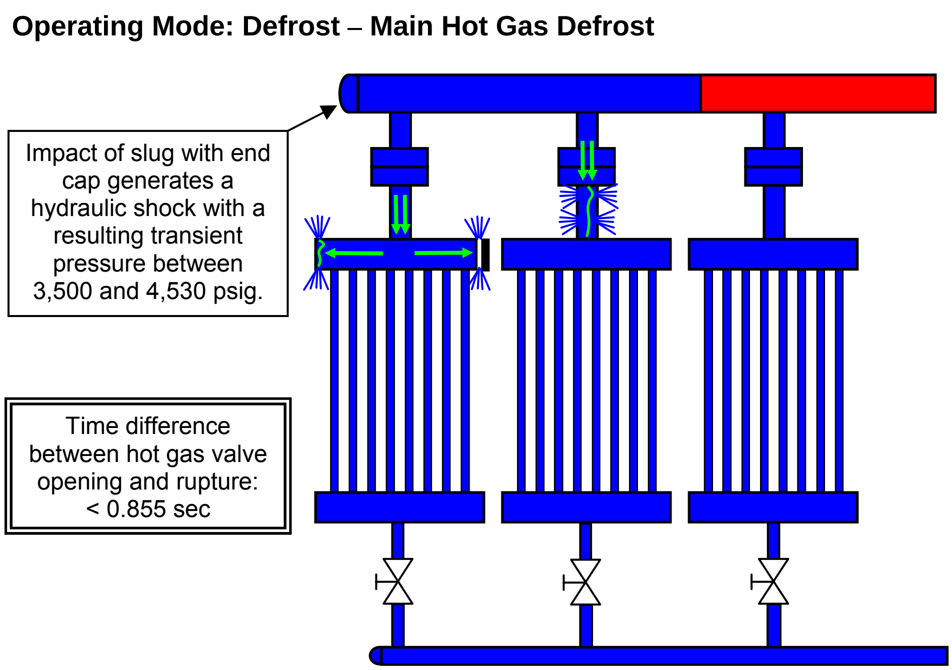
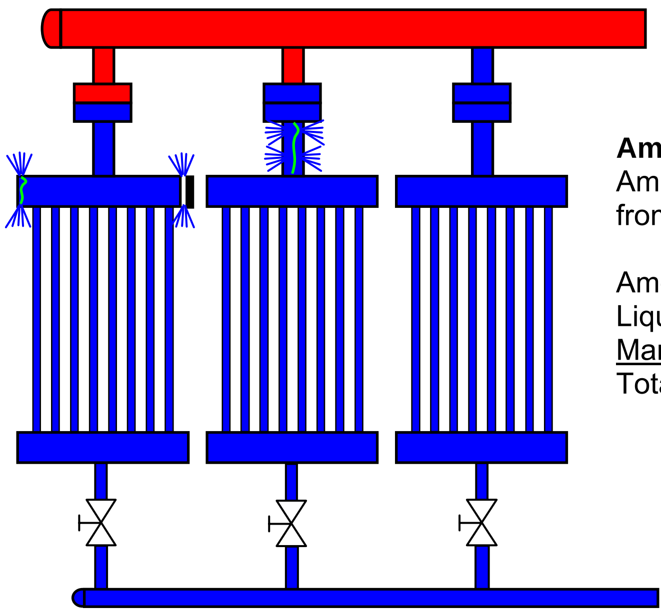


Figure 23: First and initial step of ammonia release.

Operating Mode: Defrost – Main Hot Gas Defrost



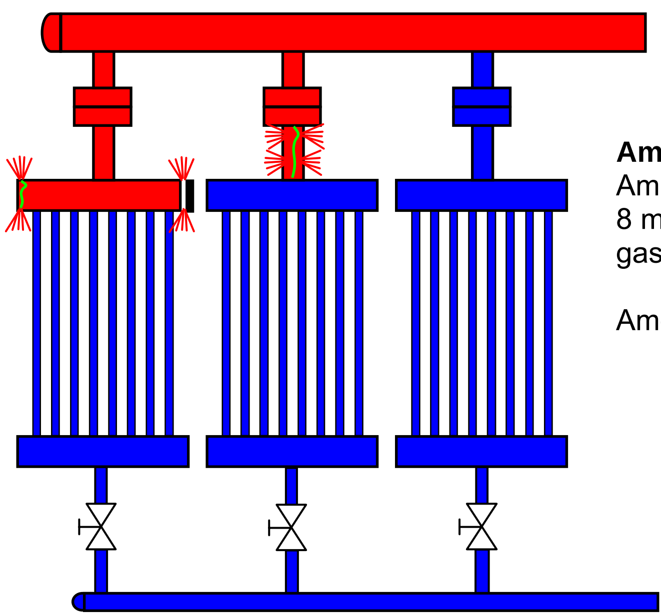
Time: 00 Minutes

Ammonia Release - Step 1
Ammonia liquid is released from liquid slug and manifolds:

Amount Released:
Liquid Slug: ~ 107 lb
Manifolds: ~ 60 lb
Total: ~ 167 lb

Figure 24: Step two of ammonia release.

Operating Mode: Defrost – Main Hot Gas Defrost



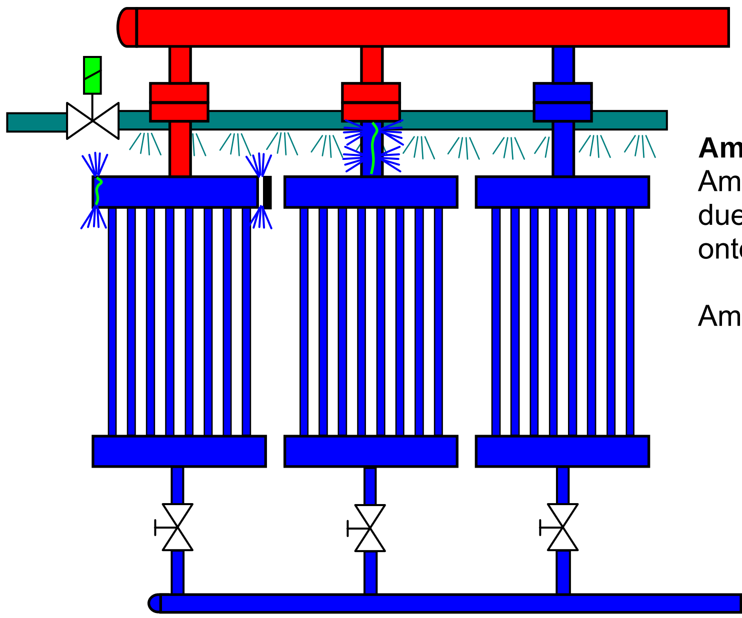
Time: 00 to 08 Minutes

Ammonia Release - Step 2
Ammonia vapor is released for 8 minutes originating from hot gas valve.

Amount released: ~ 2,705 lb

Figure 25: Step three of ammonia release.

Operating Mode: Defrost – Hot Gas and Water



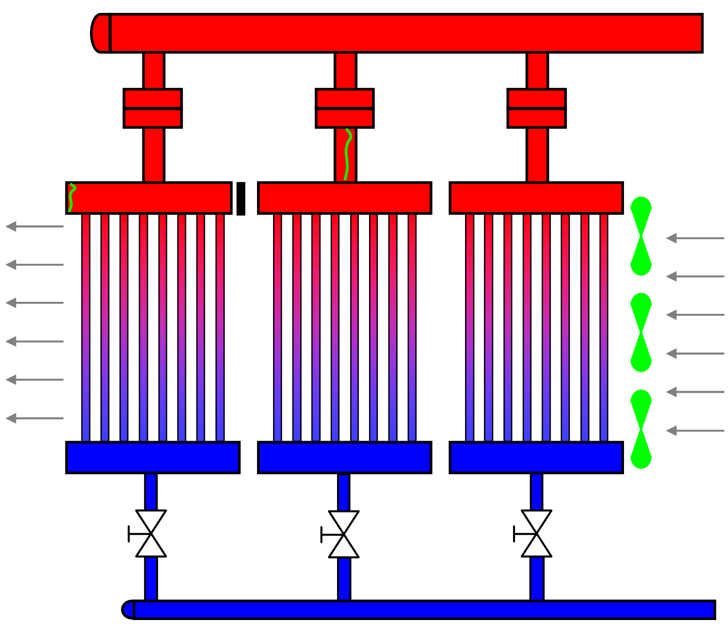
Time: 08 to 13 Minutes

Ammonia Release - Step 3
Ammonia liquid is released due to water being sprayed onto coils.

Amount released: ~ 2,500 lb

Figure 26: Step four of ammonia release.

Operating Mode: Defrost – Fans On to Spin Off Residual Water



Time: 13 to 16 Minutes

Ammonia Release - Step 4
Fans turn on for 3 minutes pushing ammonia vapor into factory through in-feed and out-feed conveyors, and open doors.