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Pressure Relief Capacity Determination for Equipment

Previously in The Cold Front (2006), we provided an introduction to the purpose, function, and principles of operation for pressure relief valves. In this issue, we discuss vapor relief valve capacity determination for industrial refrigeration equipment. We begin by reviewing the relevant requirements in codes and standards. Then we introduce fundamental considerations for relief valve sizing and corresponding principles that provide the foundation for relief valve capacity determination. We conclude by providing an example of capacity determination for a scraped-surface heat exchanger. In a future edition of the Cold Front, we will cover liquid-service or hydrostatic over-pressure protection.

Introduction

What components or portions of a refrigeration system require pressure relief protection? At a minimum, the pressure vessels used in refrigeration systems are required to carry pressure relief protection per Section

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In This Issue

- Pressure Relief Capacity Determination for Equipment 1-8
- Upcoming Ammonia Classes 2
- Noteworthy 2
- Engineering Safety Relief Systems Webcourse 9

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UG-125 of the ASME Boiler and Pressure Vessel Code (Section VIII Division 1, ASME 2010). Other equipment, such as heat exchangers, can be built to the requirements of the B&PV Code and if they are stamped as meeting those requirements (i.e. they carry an ASME stamp), those components are also required to be equipped with appropriately sized and certified pressure relief valves for over-pressure protection.

Section 9.4.1 of ASHRAE Standard 15 (2010) establishes a broad requirement for refrigeration systems to be protected by pressure relief devices.

9.4.1 *Refrigerating systems shall be protected by a pressure relief device or other approved means to safely relieve pressure due to fire or other abnormal conditions.*

The "fire" condition referred to in 9.4.1 provides the design basis for relief valve capacity determination most commonly applied to the pressure vessels used in refrigeration systems. In this case, a pressure relief valve is sized with sufficient vapor flow capacity such that the heat load on the vessel due to the external source of heat input (a fire radiating heat to the vessel) will not over-pressure the vessel. Section 9.7.5 of ASHRAE 15 (2010) provides a standardized straightforward approach for relief capacity determination for pressure vessels due to external heat addition (Fenton and Richards, 2001; Reindl and Jekel, 2006).

9.7.5 *The minimum required discharge capacity of the pressure-relief device or fusible plug for each pressure vessel shall be determined by the following:*

$$C = f \cdot D \cdot L$$

where

C = *minimum required discharge capacity of the relief device expressed as mass flow of air, [lb_m/min]*

D = *outside diameter of vessel, [ft]*

L = *length of vessel, [ft]*

f = *factor dependent upon type of refrigerant (see Table 2 of ASHRAE 15)*

Upcoming Ammonia Courses

Engineering Safety Relief Systems
June 13-17, 2009 via the Web!

Process Hazard Analysis for Ammonia Refrigeration Systems
September 13-15, 2011 Madison, WI

Design of NH_3 Refrigeration Systems for Peak Performance and Efficiency
September 19-23, 2011 Madison, WI

Introduction to Ammonia Refrigeration Systems
October 19-21, 2011 Madison, WI

Principles & Practices of Mechanical Integrity
November 2-4, 2011 Madison, WI

Intermediate Ammonia Refrigeration Systems
December 7-9, 2011 Madison, WI

Process Safety Management Audits for Compliance and Continuous Safety Improvement
January 25-27, 2012 Madison, WI

Energy Efficiency Improvement Strategies for Ammonia Refrigeration Systems
February 8-10, 2012 Madison, WI

Noteworthy

- Check the IRC website to access presentations made at the **2011 IRC Research and Technology Forum.**
- Mark your calendars now for the **2012 IRC Research and Technology Forum** – **May 2-3, 2012** at the Pyle Center in Madison, WI.
- Send items of note for next newsletter to **Todd Jekel**, **tbjekel@wisc.edu**.

Notes:

1. *When combustible materials are used within 20 ft (6.1 m) of a pressure vessel, multiply the value of f by 2.5.*
2. *When a single pressure-relief device or fusible plug is used to protect more than one pressure vessel, the required capacity shall be the sum of the capacities required for each pressure vessel.*
3. *The formula is based on fire conditions. Other heat sources shall be calculated separately.*

The added requirement of 9.4.1 in ASHRAE 15 to guard against "other abnormal conditions" is vague and necessitates the application of sound engineering judgment to anticipate other sources that could lead to over pressuring refrigeration systems and equipment. In this issue of the **Cold Front**, we will consider examples of "other sources of heat load" that can lead to over pressurization of equipment where pressure relief valves are being applied for compliance with codes (ASME 2010) or for enhanced safety.

Facilities that are subject to OSHA's Process Safety Management Standard are required to establish and document a design and design basis for the safety relief systems protecting equipment as part of their covered process [29 CFR 1910.119(d)(3)(i)(D)]. The details provided in this issue of the **Cold Front** are intended to be supportive of this requirement.

Basic Principles of Relief Capacity Determination

For refrigeration systems, there are three sources of work or thermal energy input that can potentially lead to over pressuring equipment:

1. Positive displacement compressors
2. External heat addition
3. Internal heat addition

Section 9.8 of ASHRAE 15 (2010) prescribes the conditions for the relief capacity determination applied to positive displacement compressors and Appendix D (informative) of Standard 15 provides one method for calculating the capacity of pressure relief device(s) protecting a positive displacement compressor.

The fire condition discussed in the previous section is the most common *external heat addition* scenario for relief valve sizing. Over-pressure caused by an *internal heat addition* can occur when secondary fluids, food products, or other liquids at high temperatures flow through heat exchangers giving up heat to the refrigerant within the exchanger. The following are examples of equipment subject to internal heat loads:

- Shell-and-tube heat exchangers
- Plate-type heat exchangers
- Fluid product bulk tanks
- Scraped-surface heat exchangers
- Other types of heat transfer equipment requiring over-pressure protection

Section 8.2.1.4 of IIAR 2 (2008) requires that heat loads from cleaning operations be considered during the process of sizing relief valve's capacity for process heat exchangers.

8.2.1.4 *Heat loads from cleaning operations shall be considered when designing the relief capacity and control of process heat exchangers.*

The first step in the process of relief valve sizing for internal heat load scenarios is to assess whether or not the internal heat load is capable of over-pressuring a heat exchanger. A necessary condition to create an over-

pressure condition due to an internal heat load is that the temperature of the heat input source must be higher than the saturation temperature corresponding to the maximum refrigerant pressure - assumed to be equal to the component's maximum allowable working pressure (MAWP). Table 1 provides key property data for anhydrous ammonia with the first two columns showing a range of MAWPs typical for ammonia refrigeration system components and the corresponding refrigerant saturation temperature.

Consider a heat exchanger with an MAWP of 300 psig. The refrigerant-side of this heat exchanger will not experience over-pressure until fluid temperatures on the secondary side are greater than 126.5°F. In a situation where the secondary fluid temperature will always be below 126.5°F, this heat exchanger would be sized considering an external load scenario only.

Table 1: Saturation temperature for anhydrous ammonia at various maximum allowable working pressures.

MAWP [psig]	T_{sat} [°F]	Accumulated Pressure [psig]	Enthalpy of Vaporization [Btu/lb$_m$]
150	84.3	179.7	488
175	93.0	207.2	479
200	100.8	234.7	470
225	108.0	262.2	461
250	114.6	289.7	453
275	120.8	317.2	446
300	126.5	344.7	438
325	132.0	372.2	431
350	137.1	399.7	424
375	142.0	427.2	417
400	146.7	454.7	410

Once it has been established than an internal heat source has the potential of creating over-pressure, the next step is to determine the corresponding rate of refrigerant evaporation attributable to the heat source. To avoid the risk of component failure due to over-pressure, the relief valve specified & installed must have equal or more flow carrying capability than the maximum anticipated rate of refrigerant evaporation based on the internal or external source of heat addition being considered.

Relief Capacity Determination for Internal Heat Load Scenarios

Let's now look at what is required to determine the required pressure relief capacity for over-pressure situations that originate from internal heat load sources. The basic steps in the process include:

1. Verify that the internal heat load will have a temperature in excess of the saturation temperature corresponding to a refrigerant pressure equal to the relief device set pressure or component MAWP – whichever is less.
2. Determine the maximum heat load on the equipment, $\dot{Q}_{max}$.
3. Determine the rate of refrigerant evaporation at the maximum heat load condition, $\dot{m}_{ref}$.
4. Convert the refrigerant mass flow rate to an air mass flow rate basis.

5. Select candidate relief valves that have equal or greater air mass flow capacities than required by the equipment as-determined in Step 4.
6. Verify that the inlet pressure losses due to piping and three-way valves does not diminish the relief valve's capacity below the minimum required as-determined in Step 4.

In the previous section, we discussed what is required in Step 1. Step 2 involves determining the maximum heat load on the equipment. The maximum heat load will depend on a number of factors but the following are typical heat loads that could result in over-pressuring equipment.

- High temperature product supplied to the heat exchanger
- Clean-In-Place (CIP) fluid supplied to the heat exchanger
- Abnormal conditions on the secondary fluid circuit
- Other conditions relevant to the installation

Each situation needs to consider the nature of the internal heat load and appropriate analysis techniques applied to quantify the magnitude of the internal heat load. It is important to point out that the heat load needs to be evaluated at conditions consistent with the relieving condition. In other words for a constant secondary fluid condition, the thermal load on a heat exchanger will diminish as the refrigerant pressure increases due to the rising refrigerant saturation temperature within the heat exchanger.

Once the heat load has been quantified, we proceed to Step 3 to calculate the refrigerant vapor production rate at the pressure relief device relieving condition. The refrigerant mass flow rate is calculated by Eq. 1.

$$\dot{m}_{ref} = \frac{\dot{Q}_{max}}{(h_{vapor} - h_{liquid})} \qquad \text{Eq. 1}$$

where

$\dot{m}_{ref}$ is the refrigerant vapor production rate due to the applied internal heat load [lb_m/min].

$\dot{Q}_{max}$ is the maximum expected internal heat load, [Btu/min].

h_{vapor} is the refrigerant saturated <u>vapor</u> enthalpy evaluated at the fully accumulated relief device set pressure ($P = P_{set} \cdot 1.1 + 14.696$), [Btu/$lb_m$].

h_{liquid} is the refrigerant saturated <u>liquid</u> enthalpy evaluated at the fully accumulated relief device set pressure ($P = P_{set} \cdot 1.1 + 14.696$), [Btu/$lb_m$].

With Step 3 complete, we can now convert the refrigerant mass flow to an equivalent air mass flow basis. The mass conversion factor from refrigerant to air, r_w, is accomplished by the following (ASHRAE 2010):

$$\dot{m}_{air} = r_w \cdot \dot{m}_{ref} \qquad \text{Eq. 2}$$

where r_w is the refrigerant to standard air mass flow conversion factor given by:

$$r_w = \frac{C_a}{C_r}\sqrt{\frac{T_r}{T_a}}\sqrt{\frac{M_a}{M_r}} \qquad \text{Eq. 3}$$

with

$$C_r = 520\sqrt{k\left(\frac{2}{k+1}\right)^{\frac{k+1}{k-1}}} \qquad \text{Eq. 4}$$

where

k is the ratio of specific heats for refrigerant vapor evaluated at the fully accumulated relief device set pressure ($P = P_{set} \cdot 1.1 + 14.696$).

C_r is a constant for the refrigerant vapor (as given in Eq. 4)

C_a is a constant for air = 356 (per ASHRAE 15-2010 Appendix D)

M_a is the molar mass of air = 28.97

M_r is the molar mass of the refrigerant

T_a is the absolute temperature of the air (ASHRAE 15-2010 Appendix D T_a = 520°R)

T_r is the absolute temperature of the refrigerant at the relieving condition

Once the minimum air-equivalent mass flow is determined, Step 5 involves finding candidate relief valves that have at least this minimum required capacity. It is important to avoid significantly oversizing relief valves due to its effect on vent piping sizing as well as the potential for an abnormal relief device operation known as "chatter" (Reindl and Jekel 2006).

With one or more candidate relief valves identified, the calculation of inlet pressure losses follows. Pressure losses will occur when refrigerant flows from the protected component to the relief piping connection and through the three-way manifold (if-equipped). These pressure losses, effectively, lower the inlet pressure to the relief device; thereby, decreasing the valve's capacity. Inlet pressure losses shall not diminish the relief valve's capacity below the minimum capacity as-determined in Step 4. We will discuss inlet pressure losses more fully in a future edition of the **Cold Front**.

This completes the process of relief valve capacity determination for internal heat loads applied to equipment. At a minimum, the relief valve capacity required to satisfy an external heat load due to a fire condition should be compared with the relief valve capacity for the internal heat load. The larger of the two capacities must then be used as the basis for relief valve selection.

Example: Scraped-Surface Heat Exchanger

Consider a Scraped-Surface Heat eXchanger (SSHX) that normally operates at low temperatures to freeze product (such as ice cream mix). The SSHX consists of a cylindrical barrel that product flows through during operation. Dasher blades rotate within the barrel to prevent product from adhering (freezing) to the inside surfaces of the barrel. Surrounding the barrel is a jacket containing evaporating anhydrous ammonia refrigerant.

The scenario we will consider is an over-pressure condition potentially created by hot CIP fluid being supplied to the barrel while the refrigerant-side of the SSHX is isolated from the refrigeration system. The maximum expected CIP fluid supply temperature in this example is 160°F. The following data obtained were obtained from the manufacturer of the SSHX:

- U = 300 Btu/hr-ft^2-°F
- A = 14.5 ft^2
- $MAWP$ = 150 psig [refrigerant-side of the unit]

Based on this information, determine the minimum required relief valve capacity needed to protect the SSHX from over-pressure due to the internal load from CIP. We assume here that the set pressure of the relief valve that will be applied to the SSHX is 150 psig - equal to the heat exchanger's MAWP.

Step 1: Verify the CIP fluid temperature is high enough to over-pressure the heat exchanger

Referring to Table 1, the saturation temperature of ammonia at the 150 psig MAWP of the SSHX is

84.3°F. The supply temperature of the CIP fluid is considerably higher than the refrigerant's saturation temperature at the heat exchanger's MAWP so the potential for over-pressuring the heat exchanger does exist.

Step 2: Determine the maximum heat load

The heat load can be determined from the following rate equation:

$$\dot{Q}_{max} = U \cdot A(LMTD) \qquad \text{Eq. 5}$$

where

U is the overall heat transfer coefficient for the heat exchanger, Btu/hr-ft^2-°F
A is the area of the heat exchanger upon which the overall transfer coefficient is based, ft^2
$LMTD$ is the log-mean temperature difference, °F

Because the refrigerant in the jacket of the heat exchanger will be evaporating, the log-mean temperature difference is approximated as the difference in temperature between the entering CIP fluid and the refrigerant's saturation temperature (89.6°F) evaluated at the fully accumulated relieving pressure (150·1.1+14.696=179.7 psia). The maximum expected heat load for the over-pressure condition then becomes:

$$\dot{Q}_{max} = 300\frac{Btu}{hr-ft^2-°F} \cdot 14.5\, ft^2(160-89.6)°\text{F} \cdot \frac{1\, hr}{60\, min} = 5{,}104\frac{Btu}{min} \qquad \text{Eq. 6}$$

Step 3: Determine the refrigerant mass flow rate at the relieving condition

The refrigerant mass flow is calculated using Eq. 1. Table 1 includes the enthalpy of vaporization at the fully accumulated relieving condition for a range of relief device set pressures. For the 150 psig condition, the enthalpy of vaporization is 488 Btu/lb$_m$. The mass flow rate of refrigerant at the relieving condition is:

$$\dot{\boldsymbol{m}}_{\boldsymbol{ref}} = \frac{5{,}104\, Btu/min}{488\, Btu/lbm} = 10.5\, \frac{lbm}{min} \qquad \text{Eq. 7}$$

Step 4: Convert refrigerant mass flow rate to an air basis

The first step in the process of converting the refrigerant mass flow to an air-equivalent basis is to determine the ratio of specific heats, k, for saturated refrigerant vapor at the fully accumulated relieving condition. In this case, k=1.484. We can now calculate the refrigerant constant as given by Eq. 4.

$$C_r = 520\sqrt{k\left(\frac{2}{k+1}\right)^{\frac{k+1}{k-1}}} = 520\sqrt{1.484\left(\frac{2}{1.484+1}\right)^{\frac{1.484+1}{1.484-1}}} = 363.2 \qquad \text{Eq. 8}$$

We now apply Eq. 3 to determine the refrigerant-air mass flow rate conversion factor where the temperature of the refrigerant is assumed to be saturated vapor at the fully accumulated relieving condition and the molar mass of ammonia is 17.03.

$$r_w = \frac{356}{363.2}\sqrt{\frac{549.6}{520}}\sqrt{\frac{28.97}{17.03}} = 1.314 \qquad \text{Eq. 9}$$

The equivalent air mass flow rate is then determined by applying Eq. 2.

$$\dot{m}_{air} = r_w \cdot \dot{m}_{ref} = 1.314 \cdot 10.5 = 13.8 \left(\frac{lbm}{min}\right)_{air} \qquad \text{Eq. 10}$$

The process would then proceed with the selection of candidate relief valves followed by verifying that suitable relief capacity is available when inlet pressure losses have been accounted.

Conclusion

Process equipment including heat exchangers equipped with pressure relief protection devices require a basis of sizing for determining the minimum required pressure relief device capacity. Like pressure vessels, other types of process equipment may experience over-pressure conditions due to external heat loads. Methods of relief capacity determination for equipment due to external heat loads were previously presented by Reindl and Jekel (2009). In this issue of the **Cold Front**, we considered relief device capacity determination due to internal heat loads.

The fundamental step in determining the minimum required relief device capacity for protecting refrigeration equipment from over-pressure due to internal heat loads is quantifying the rate of refrigerant vaporization at the elevated pressure coincident with the relieving condition. Once the rate of vaporization is quantified, the refrigerant mass flow is converted to an air equivalent basis to allow candidate relief devices to be selected. Once a candidate relief device is selected, the inlet pressure losses to the relief valve must be quantified and the effects of inlet pressure losses evaluated to ensure they do not diminish the candidate relief device's capacity below that required to protect the equipment.

References

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REMINDER

Engineering Safety Relief Systems Webcourse

June 13-17, 9-11am Central, Anywhere via the Internet

Overview

This workshop is your opportunity to develop or improve your understanding of engineered safety relief systems. Our primary focus is industrial refrigeration systems using anhydrous ammonia but many of the principles we will discuss apply equally to other applications & refrigerants as well.

Audience

Whether you are an end-user, equipment manufacturer, design engineer, or contractor, this course will help you build your capabilities in the area of the principles and practices of engineering safety relief systems. Participate and develop your understanding of:

- Codes and Standards related to safety relief systems
- Key aspects of engineering code-compliant relief systems
- Capacity determination for non-standard equipment like heat exchangers
- Methods for proper sizing of relief vent piping, including headered vent systems

Get Access to the IRC's Safety Relief Systems Vent Tool

The IRC has developed a web-based safety relief systems analysis tool. This powerful tool has a high degree of flexibility to analyze, engineer, and document safety relief systems for industrial refrigeration applications.

Tool Features

- Graphical user interface to configure relief system to be analyzed
- Ability to handle headered systems & multiple relief scenarios
- Quick and accurate algorithm to solve compressible flow equations
- Relief valve selection wizard
- Equivalent lengths for elbows & fittings included
- Detailed compliance checks for each system component
- One-click reports for easy printing

Access to this tool is provided free of charge to those completing this course.

More Information

The course brochure is available here. If you have any additional questions, please call us toll-free at 1-866-635-4721 or e-mail us at info@irc.wisc.edu.