

LOW-TEMPERATURE REFRIGERATION OPTIONS

INTRODUCTION

We are on the cusp of a major change in the refrigeration world as the phase-out of R-22 starts to accelerate next year. Beginning in 2010, new equipment using R-22 will be banned in accordance with the Montreal Protocol. Although R-22 refrigerant itself will continue to be manufactured for another decade, its eventual end of production will quickly approach as one considers the future fate of those industrial refrigeration systems designed and currently operating with R-22. To further complicate matters, there is mounting evidence that the next class of halocarbon refrigerants (hydroflourocarbons or HFCs) is also at risk for regulatory phase-out. For example, the Federated States of Micronesia and Mauritius submitted a proposal in May 2009 that would amend the Montreal Protocol to regulate and phase-down the use of HFC refrigerants (UNEP 2009). Although HFC refrigerants are rarely used in large industrial refrigeration systems, the suggestion at removing yet another class of refrigerant working fluids (or refrigerants within that class) is striking fear in the eyes of both manufacturers and end-users of HFCs.

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From the standpoint of refrigerant selection, is there any place for refuge from these changes? Some have suggested that natural refrigerants are a logical choice. *Natural refrigerants* are substances that can be found naturally occurring in nature and include: ammonia (NH₃, R-717), carbon dioxide (CO₂, R-744), hydrocarbons, water and air (ASHRAE 2009). Of course ammonia has been the refrigerant of choice in the industrial sector for more than a sesquicentennial. Currently, there is an increasing interest in CO₂ as a working fluid in thermal systems. At low temperatures, CO₂ can be used either as a cascade refrigerant in a vapor compression system (Page 2002, Homsy 2003, Nielsen 2003, Lee et al. 2006, and Pillis 2009) or as a secondary heat transfer fluid capable of undergoing a phase change (Martin 2006). At high temperatures, CO₂ shows promise as a *supercritical*¹ working fluid in advanced Brayton cycles for power generation (Chapman 2009, Sienicki 2009, and Turchi 2009). In this article, we discuss key properties of CO₂ and the potential benefits of its use as a working fluid in refrigeration systems. We will first look at the thermodynamic properties of CO₂ and contrast its properties with other refrigerants. We will then consider its implementation as a secondary or cascade refrigerant. System alternatives are presented and efficiency characteristics for cascade options estimated. The article concludes with advantages and disadvantages of CO₂ as a low temperature refrigerant.

THERMODYNAMIC PROPERTIES

The thermodynamic and physical properties of individual refrigerant alternatives strongly influence their suitability for use in industrial and other refrigeration applications. The thermo-physical properties of refrigerants will affect a refrigeration system's operating efficiency, capital cost, and physical size of equipment. In addition, some properties, such as saturation pressure, provide an indication of its safety and operability.

UPCOMING AMMONIA COURSES

- Design of NH₃ Refrigeration Systems for Peak Performance and Efficiency*
September 21-25, 2009 Madison, WI
- Introduction to Ammonia Refrigeration*
October 7-9, 2009 Madison, WI
- Principles & Practices of Mechanical Integrity for Industrial Refrigeration Systems*
November 4-6, 2009 Madison, WI
- Intermediate Ammonia Refrigeration*
December 3-5, 2009 Madison, WI
- Engineering Safety Relief Systems*
December 14-18, 2009 via the Web!
- Auditing for Process Safety Management*
January 11-13, 2010 Madison, WI
- Energy Efficiency Improvement Strategies for Ammonia Refrigeration Systems*
February 10-12, 2010 Madison, WI
- Introduction to Ammonia Refrigeration Systems*
March 3-5, 2010 Madison, WI
- Ammonia Refrigeration System Safety*
April 7-9, 2010 Madison, WI

See www.irc.wisc.edu/education/ for more information.

NOTEWORTHY

- The webcourse **ENGINEERING SAFETY RELIEF SYSTEMS** is scheduled for **December 14-18, 2009**. See more information [here](#).
- Congratulations to **PETE MUMANACHIT** with his new position as a refrigeration engineer with **SCHOEP'S ICE CREAM**.
- Send items of note for next newsletter to **TODD JEKEL**, tbjekel@wisc.edu.

¹ The critical pressure and temperature for carbon dioxide is 1,070 psia (72.8 bar) and 87.8°F (31°C), respectively. The term *supercritical* indicates that the working fluid is utilized at conditions above its critical pressure and temperature.

The following are desirable refrigerant thermodynamic properties:

- *Saturation pressures above atmospheric pressure* at the lowest operating temperatures in the system (prevents air infiltration during operation)
- *Saturation pressures that are not extreme* at the highest operating temperature in the system (accommodates reasonable design working pressures for equipment)
- *Low vapor specific volume* (high vapor density) at the lowest operating temperature in the plant (reduces the potential physical size of compression equipment)
- *High heat of vaporization* (reduces mass flow requirements to meet a given refrigeration load and coupled with low vapor specific volume allows for smaller pipe sizes and compression equipment)

Although there are other properties that could be considered, our attention here will be focused on the above property list. First, we will consider how the working pressures of different refrigerant alternatives compare over a range of low-side operating temperatures.

FIGURE 1 shows how the saturation pressure of various refrigerant choices varies over a range operating temperatures consistent with the load requirements for industrial refrigeration systems. Because the range of working pressures is quite wide for the refrigerants shown, **FIGURE 2** shows the same pressure-temperature behavior for those refrigerants that are nearer to atmospheric pressure. Considering the desire to select a refrigerant with its saturation pressure near or slightly above atmospheric pressure, only three (3) of the refrigerants shown meet this goal: R-410a, R-508B, and CO₂ (R-744). Refrigerant R-134a (HFC) has the lowest comparative operating pressure over the temperature range shown. As the operating pressures decrease, the likelihood of air infiltrating the refrigeration system will proportionally increase and the accumulation of air into a refrigeration system will result in decreased operating efficiency. Unfortunately, a characteristic of R-744 and R-508B is their high working pressures. At +20°F (-6.7°C), CO₂ will be at 422 psia (29.1 bar) while the pressure of R-508B is a slightly lower but still significant 341 psia (23.5 bar). At normal room temperatures such as 70°F (21°C) (as would be present during “stand-still” conditions), the saturation pressure for R-744 is 800 psia (55.2 bar) while R-508B would be above its critical temperature of 57°F (13.9°C). Higher design pressures increase the capital and installation costs of refrigeration system components and safety risks.

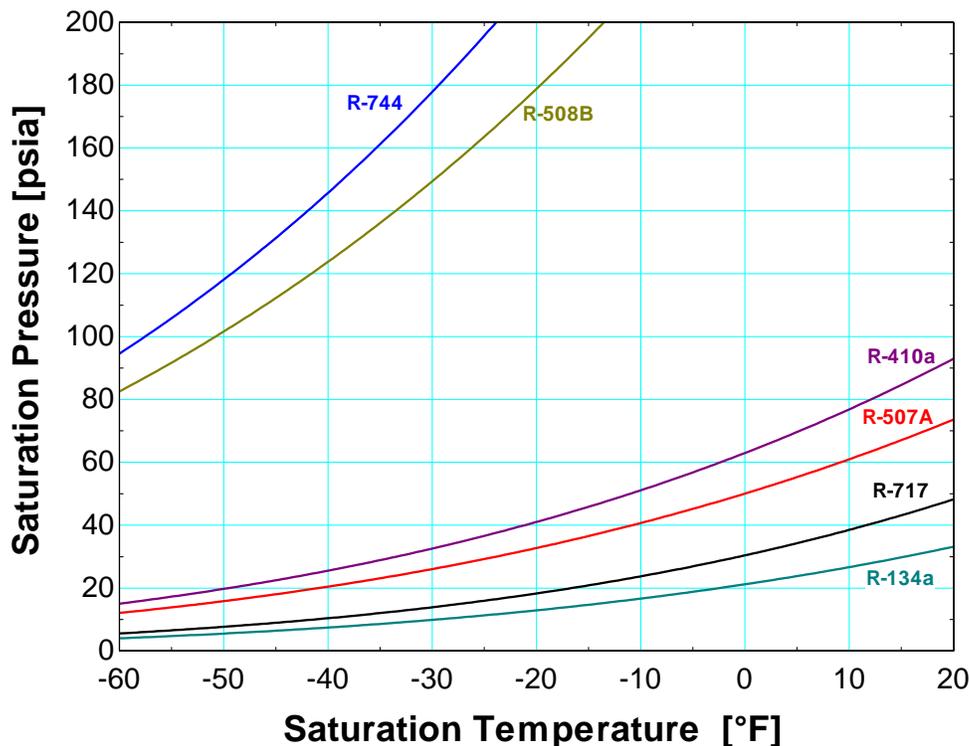


FIGURE 1: Saturation pressure characteristics for a range of refrigerant alternatives.

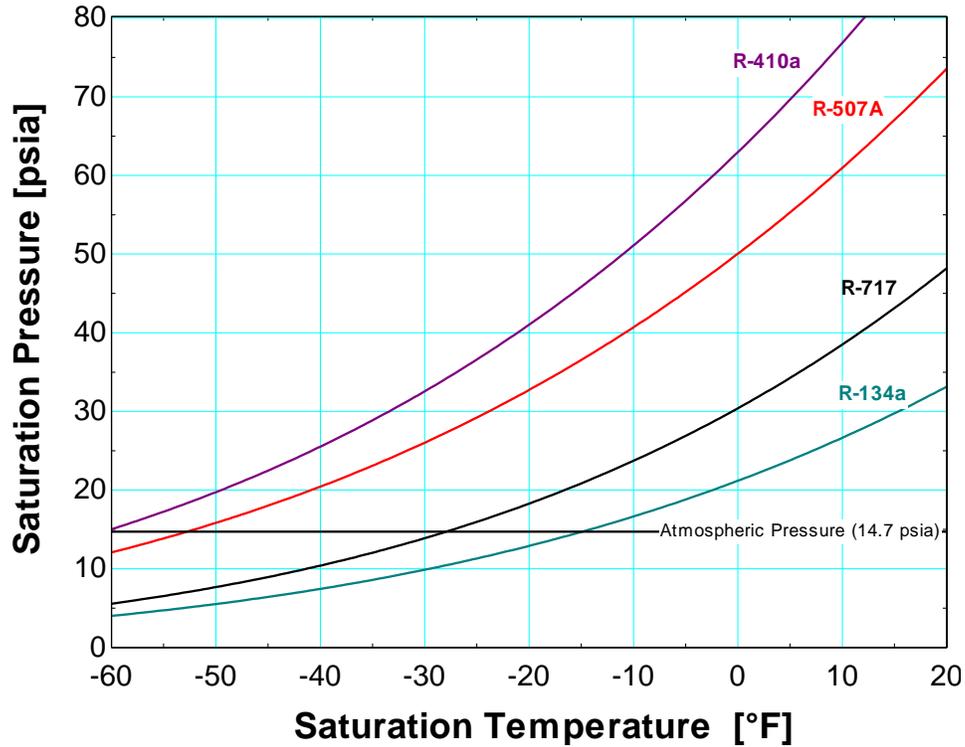


FIGURE 2: Saturation pressure characteristics for near-atmospheric pressure refrigerant alternatives.

Next on the property list is the specific volume of saturated vapor. In this case, we are interested in the specific volume for refrigerant alternatives evaluated over the range of temperatures that correspond to compressor suction conditions typically found in industrial refrigeration systems. **FIGURE 3** shows the specific volume of saturated vapor for each refrigerant alternative at typical low-side saturation temperatures. Note that the specific volume is shown on a logarithmic scale.

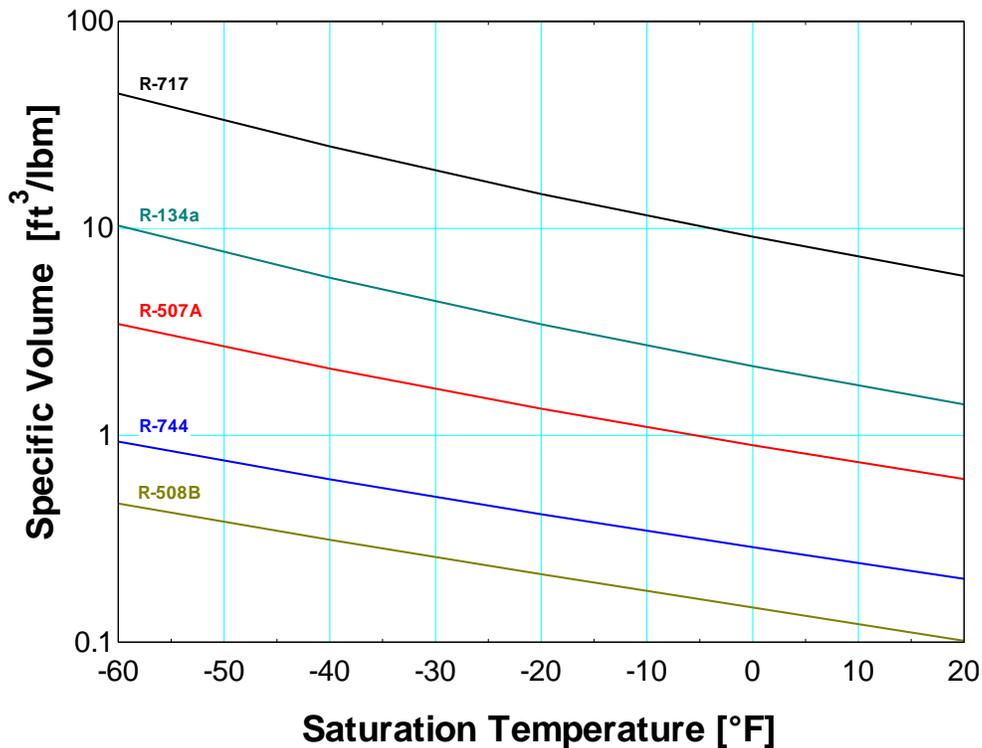


FIGURE 3: Specific volume of saturated vapor refrigerants over a range of typical saturated suction temperatures.

Ammonia (R-717) stands out with its extremely high vapor specific volume while both CO₂ (R-744) and R-508B both retain low vapor specific volume over the entire temperature range. Although not a direct measure of the physical size of compression equipment required, refrigerants that have a low vapor specific volume (high vapor density) at suction conditions have the potential for smaller compression equipment per unit of refrigeration capacity.

A better indicator of the relative physical size of compression equipment required at low temperatures is the compressor's volumetric flow rate at suction conditions to achieve a unit of refrigeration load (tons, kW_t). The volumetric flow rate required to achieve a unit refrigeration capacity (ton, kW_t) includes the effects of vapor specific volume as well as the latent heat of vaporization characteristics for each refrigerant.

FIGURE 4 shows the comparative results for lower temperature operation assuming a compound compressor arrangement with a 0°F (-17.8°C) intermediate temperature. Carbon dioxide (R-744) and R-508B have the lowest volumetric rate requirement which translates into smaller compression equipment as well as smaller suction line sizes for a given pressure loss. R-134a has the highest volume flow requirements of the alternatives shown.

Figure 3 showed ammonia had the highest specific volume – why doesn't ammonia have the highest volumetric flow requirement? The answer lies in the fact that ammonia has the highest heat of vaporization of the refrigerants considered. The high heat of vaporization means that a lower mass (and volumetric) flow is needed.

Analysis of the thermo-physical properties shows that there a number of trade-offs exist when selecting a refrigerant. Not one of the refrigerants included in the above analysis exhibits all of the desirable characteristics over the entire range of typical operation; consequently, refrigeration engineers and designers have to exercise some design creativity in attempts to limit the operation of a refrigeration system to ranges where favorable operating characteristics are attained. This leads to concepts such as volatile secondary loop and cascade refrigeration systems. A volatile secondary loop system uses a refrigerant as a heat transfer fluid that undergoes a phase change as it circulates to meet loads but no compression equipment is utilized. A cascade refrigeration system has separate refrigerants circulating though one or more vapor compression systems with heat being transferred between the systems. In this article, we focus on the cascade refrigeration system option.

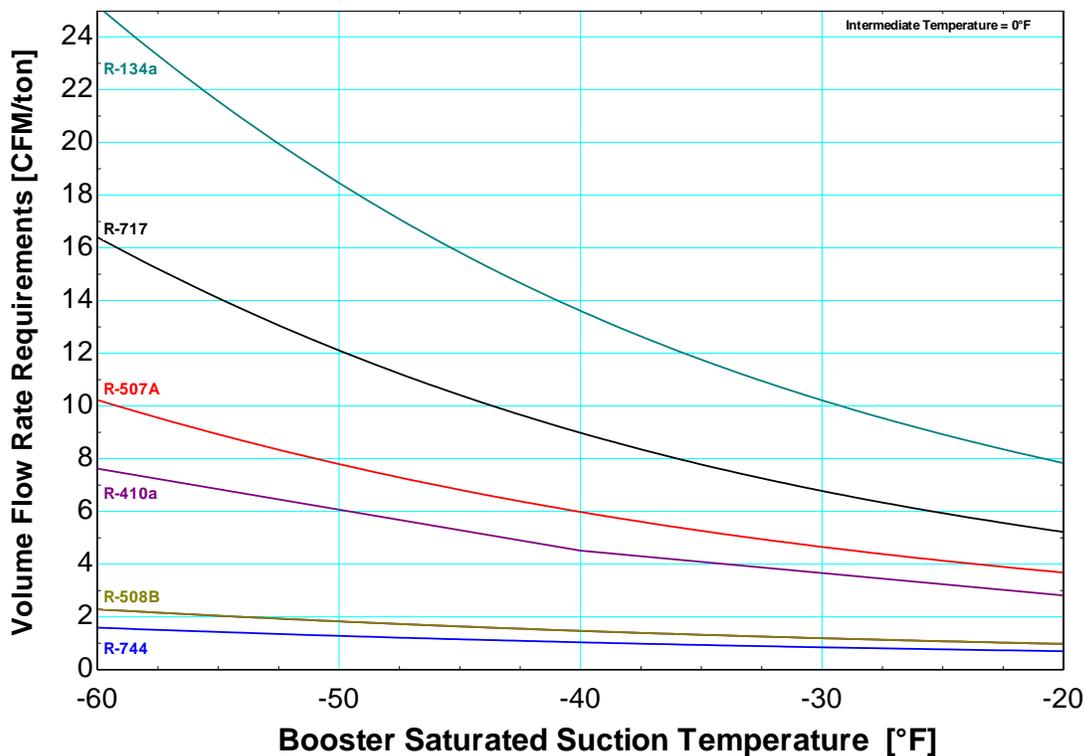


FIGURE 4: Unit refrigeration volumetric flow rate requirements for different refrigerants.

CASCADE REFRIGERATION SYSTEMS

A cascade refrigeration system utilizes separate refrigerating circuits coupled by the exchange of thermal energy only rather than by the exchange of refrigerant flow as would be the case in a compound system. ASHRAE Standard 15-2007 defines a cascade refrigeration system as:

a refrigerating system having two or more refrigerant circuits, each with a pressure-imposing element, a condenser, and an evaporator, where the evaporator of one circuit absorbs the heat rejected by another (lower-temperature) circuit.

By definition, a cascade refrigeration system requires the coupling of at least two (2) separate refrigeration (i.e. vapor compression) circuits; however, an infinite number of thermally-connected systems are possible. Compression equipment, heat exchangers, valves, and a dedicated refrigerant charge are required for each individual circuit. **FIGURE 5** shows a simple cascade refrigeration system utilizing ammonia and CO₂. In this simple arrangement, low temperature loads are met by pumping cold liquid CO₂ to evaporators that transfer heat into the CO₂ causing evaporation. Because the evaporators are overfed, a mixture of saturated vapor and saturated liquid carbon dioxide is returned to the low temperature recirculator where the vapor and liquid phases are separated. Low temperature CO₂ vapor is compressed to a higher pressure/temperature and directed to the cascade condenser. In the cascade condenser, the warmer CO₂ gives up its heat to the lower temperature ammonia which is evaporating on the other side of the heat exchanger. As the CO₂ gives up its heat to the ammonia, it condenses to a liquid at the higher pressure. The high pressure liquid CO₂ is then throttled to a lower pressure in the recirculator. The throttling process creates flash gas as it lowers the temperature of the CO₂ flowing into the recirculator. The flash gas and vapor returning from the load is re-compressed by the CO₂ compressor completing the circuit. In the ammonia circuit, the evaporated low temperature ammonia vapor flows to the high stage suction via the surge drum. The ammonia compressor raises the pressure of the refrigerant so heat can be rejected from the entire system to the ambient environment. Liquefied ammonia leaves the evaporative condenser and is collected in the high pressure receiver before being throttled to a lower pressure in the surge drum as-needed to maintain an adequate level of low temperature ammonia to allow continued heat absorption from the CO₂ circuit.

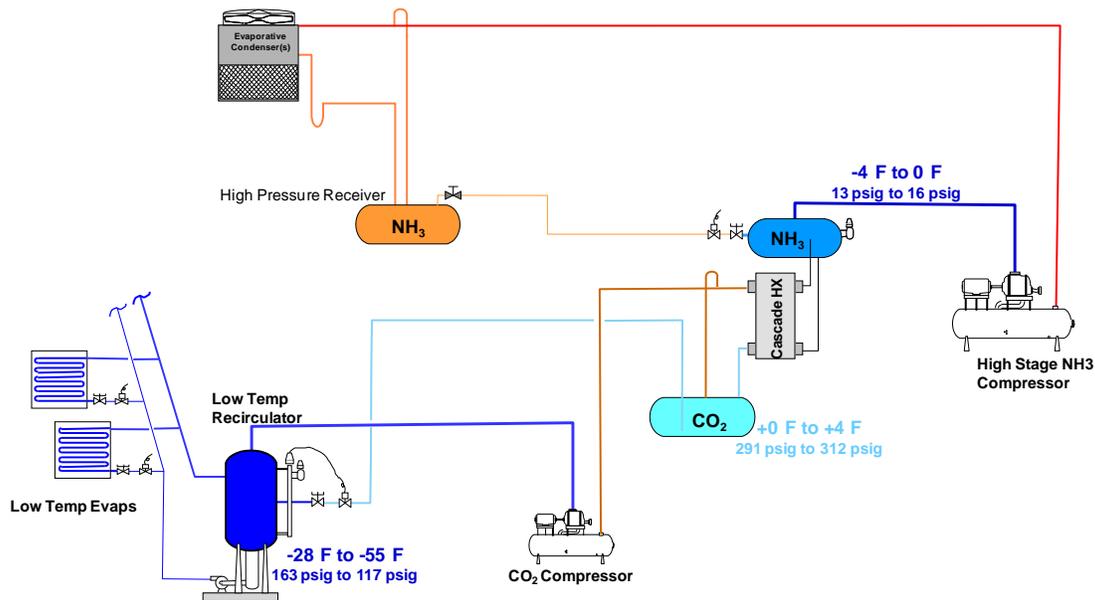


FIGURE 5: Cascade refrigeration system using both ammonia and carbon dioxide.

Page (2002) presented the results of an R-22 system replacement for a freeze-drying operation at a plant in the United Kingdom. The system consisted of 680 tons (2,390 kW_t) of low temperature load operating with

CO₂ evaporating at -63.4°F (-53°C) and condensing at 1.4°F (-17°C) in the cascade condensing heat exchanger. The low-temperature circuit utilized oil-free reciprocating compressors while the high-temperature circuit utilized economized twin screw compressors. Homsy (2003) discussed a number of lessons-learned as part of the installation and start-up of the CO₂/NH₃ cascade system documented by Page (2002). One of the more notable findings was the short operating life of the oil-less CO₂ compressors. The short compressor life expectancy was due to the poor lubricity of the carbon dioxide which required an alternative material selection for the piston rings. Nielsen and Lund (2002) describe the development of an CO₂/NH₃ cascade refrigeration system for low-temperature freezing of fish on an ocean-going vessel. The system featured oil-lubricated reciprocating compressors for the low-temperature circuit and twin screw compressors for the high-temperature circuit. The low-temperature circuit operates at an evaporating temperature of -58°F (-50°C). The authors highlighted the value of a smaller footprint for the cascade system compared to a direct-ammonia system on this ocean-going vessel. The authors also cited reduced risk of product damage due to contamination with the CO₂/NH₃ cascade system since CO₂ is used on the process-side of the refrigerant plant while ammonia is confined to a machinery room.

PERFORMANCE COMPARISONS

Increasingly, end-users are more carefully scrutinizing system designs to ensure they are capable of achieving high operating efficiency. The prospect of achieving greater operating efficiency with a cascade refrigeration system has piqued the interest of manufacturers and contractors as well as the end-users they serve. *Can a cascade refrigeration system be more efficient than a compound system?*

Mumanachit (2009) conducted a detailed analysis of industrial refrigeration system arrangements for a low temperature freezing operation. **FIGURE 6** shows Mumanachit's results expressed in terms of the system coefficient of performance (refrigeration capacity over the power input to system) for both two-stage ammonia (compound) and cascade CO₂/NH₃ system alternatives. As the evaporating temperature increases, the operating efficiency of the compound ammonia system exceeds that of the CO₂/NH₃ cascade system; however, a break-even efficiency is eventually reached and the cascade system efficiency exceeds the compound ammonia system at lower operating temperatures. At full-load operation, the system's condensing pressure/temperature influences the break-even evaporating temperature for a given cascade heat exchanger approach temperature. At a condensing pressure of 160 psia (145 psig), the CO₂/NH₃ cascade system option will be more efficient than the two-stage ammonia option for evaporating temperatures lower than -54°F (-47.8°C). The CO₂/NH₃ cascade option is never more efficient than a compound ammonia system at suction temperatures warmer than -52°F (-46.7°C). The break-even temperatures are within 2°F (1.1°C) of those reported by Pillis (2009). The break-even temperature will depend on the approach temperature in the cascade heat exchanger. This effect will be discussed shortly.

Because the previous performance comparison is for a single operating point, some may not consider them as illustrative of the performance that can be expected for an actual system operating against a varying load in a climate where seasonal ambient temperature fluctuations will influence the hours of operation at various head pressures (to the system's minimum). For a system that operates at part-load conditions, the approach temperature on the cascade heat exchanger will decrease; thereby, lowering the break-even temperature. Apart from the influence of system part-load effects on the performance of the cascade heat exchanger, the part-load operating efficiency of individual pieces of compression equipment/technology (screw compressor vs. reciprocating) will also influence the efficiency for both compound and cascade systems. These part-load effects are not included in the results presented in this article.

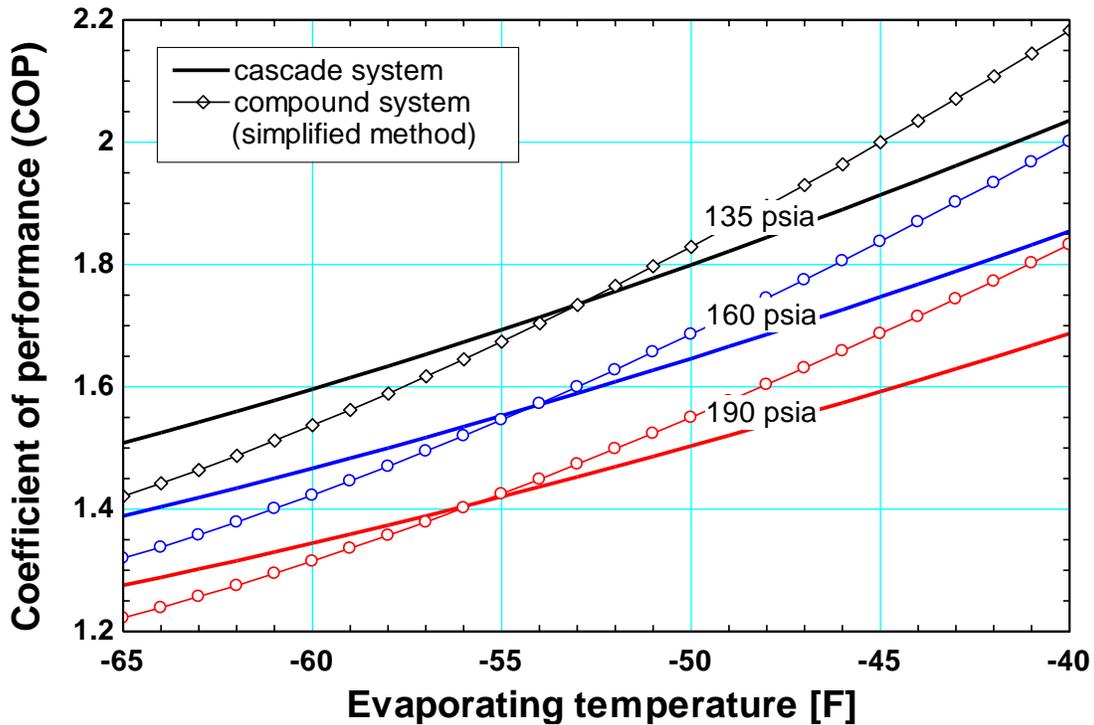


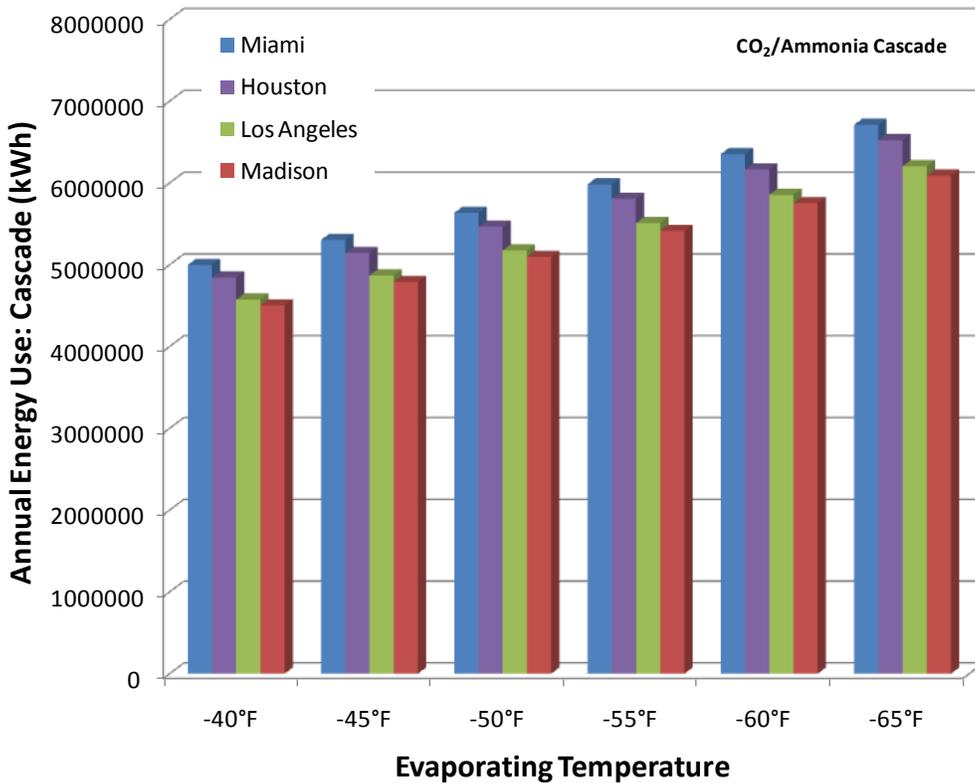
FIGURE 6: Comparative results for compound direct-ammonia and CO₂/NH₃ cascade refrigeration systems with condensing pressure as a parameter (Mumanachit 2009).

To address those concerns, Mumanachit evaluated the efficiency of compound and CO₂/NH₃ cascade options assuming the low temperature load was a freezing system operating at full-load on one of two defined production schedules to meet plant production demands and in climate types that included: Houston, TX, Los Angeles, CA; Madison, WI; and Miami, FL. For all climates, the system’s head pressure floated but was constrained to a minimum of 120 psig (8.3 barg). Other key aspects of the system comparisons are shown in the table below.

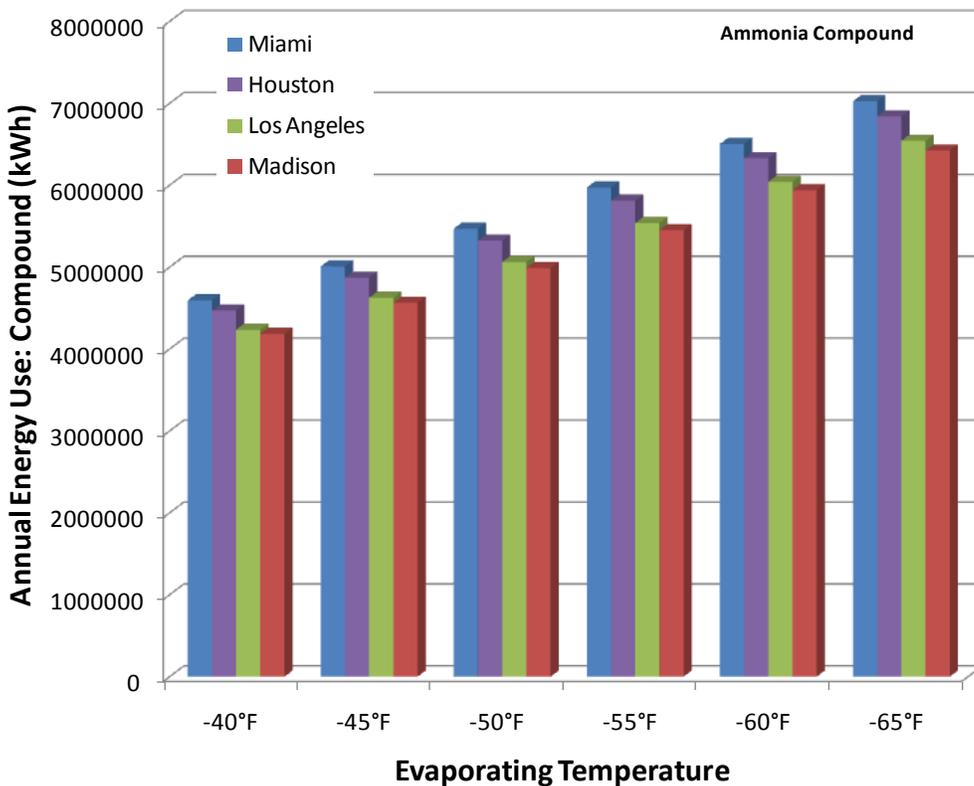
Table 1: Key characteristics of system comparisons.

CHARACTERISTIC	VALUES
Input parameter(s)	Ambient air condition (TMY2 weather data)
Locations	Miami, FL; Madison, WI; Los Angeles, CA; Houston, TX
Mode(s) of operation	<ul style="list-style-type: none"> • 8 hr/day (2,920 hr/yr) and • 10 hr/day (3,650 hr/yr)
Head pressure limit	Variable based on weather with a 120 psig (8.3 barg) minimum
Evaporator heat load	680 tons (2,390 kW _t) (constant)
Evaporating temperatures	Variable: -40 to -65°F (-40 to 53.9°C)
Cascade HX approach temperature	10°F (5.6°C) (constant)

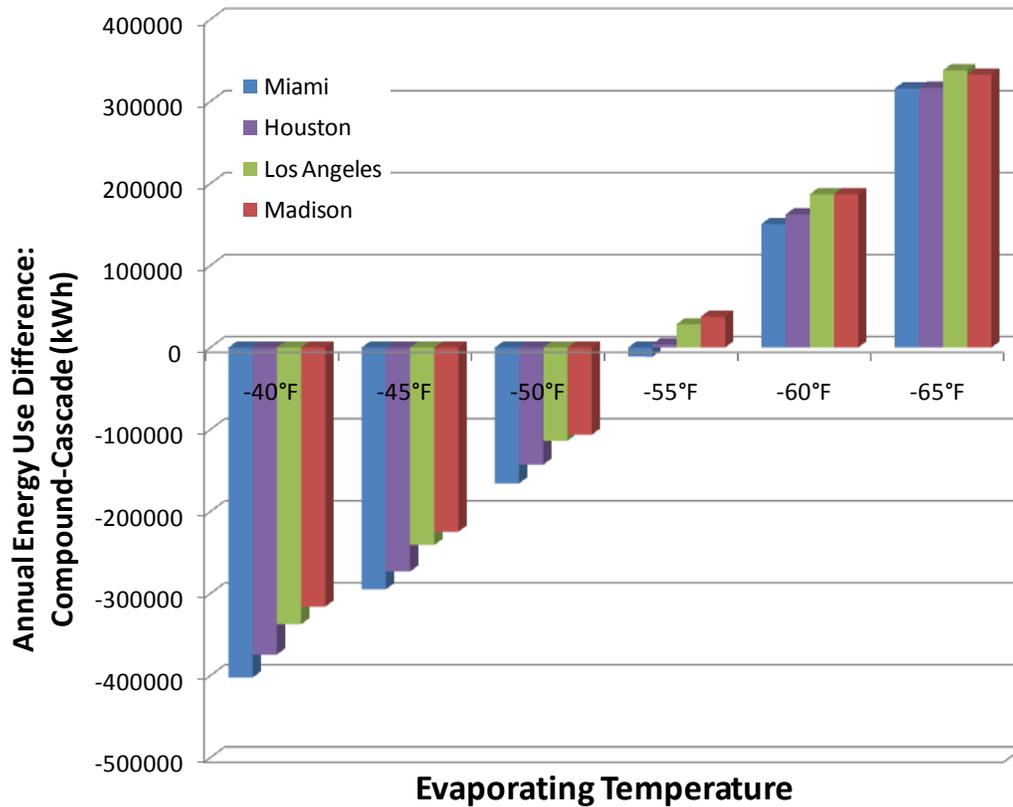
FIGURE 7 shows the annual energy consumption for both direct-ammonia and cascade refrigeration systems over a range of freezing system evaporating temperatures.



(A)



(B)



(c)

FIGURE 7: Location dependent results showing annual energy usage for (a) compound direct-ammonia and (b) CO₂/NH₃ cascade refrigeration and (c) difference: compound – cascade systems (Mumanachit 2009).

The analysis clearly shows that the cascade system option exhibits higher annual energy consumption for warmer evaporating temperatures. The comparative annual energy consumption for the cascade system improves as the evaporating temperature decreases. Mumanachit extended his work to evaluate 20 year operation in a life-cycle analysis. In order to compare both systems over the life-cycle, he defined the *premium difference*. The *premium difference* is the present value of the life-cycle operating cost difference between the two (2) system options. Positive values of premium difference are indicative of the compound system having lower energy costs. The concept of a *premium difference* is to allow the effects of system capital cost differences to be assessed when operating efficiencies of the two options differ. **FIGURE 8** shows the premium difference between the compound and cascade system options for all climates over a range of evaporating temperatures. Positive values of the premium difference represent the present cascade system energy costs in excess of the energy costs for the compound system.

For a system operating at -45°F (-42.8°C) evaporating temperature, the premium difference for Madison is \$100k (US). In other words, this means that if a cascade system alternative could be installed for a capital cost difference of less than \$100k compared to the compound ammonia system, the cascade system would yield life-cycle savings. At evaporating temperatures below -56°F (-48.9°C), the premium difference is negative indicating that the cascade system is the lower energy cost option. Clearly, these results are strongly dependent on the utility rates and other economic parameters in the life-cycle analysis. The assumed blended electric cost for Mumanachit’s analysis was 0.06\$/kWh. Higher electric costs will greatly amplify the premium difference results.

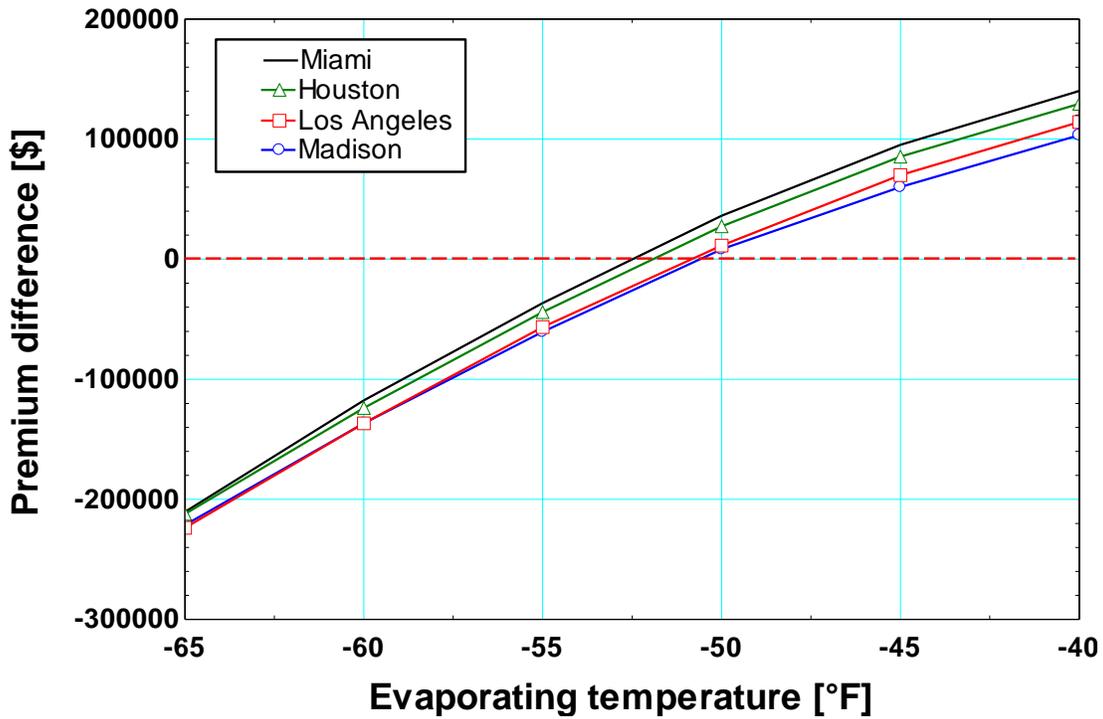


FIGURE 8: Premium differences for compound direct-ammonia and CO₂/NH₃ cascade refrigeration systems (Mumanachit 2009).

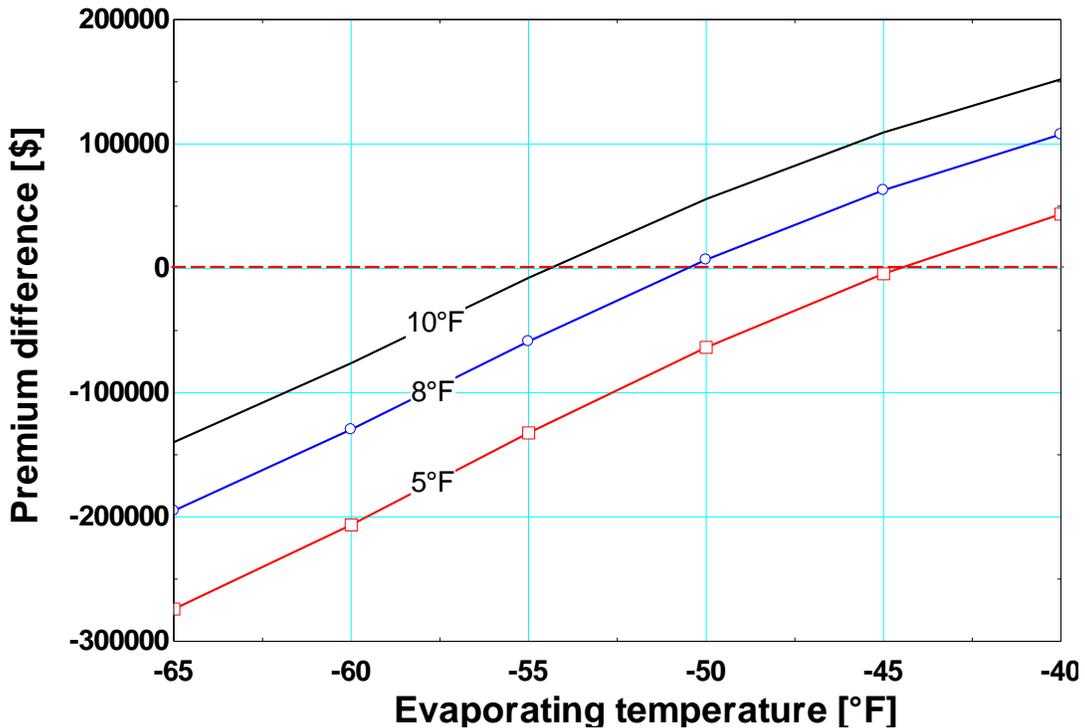


FIGURE 9: Effect of cascade heat exchanger approach temperature on the premium differences for compound direct-ammonia and CO₂/ammonia cascade refrigeration systems (Mumanachit 2009).

An important system design parameter that has not been discussed in detail as of yet is the approach temperature for the cascade heat exchanger. The approach temperature is the difference in temperature between the condensing CO₂ and the evaporating ammonia. The approach temperature has a dramatic effect on the evaporator temperature that yields a break-even (zero premium difference) operating cost. **FIGURE**

9 shows the effect of the cascade heat exchanger design approach temperature for operation at a constant head pressure (145 psig [10 barg]). As the design approach temperature for the cascade heat exchanger decreases (larger heat exchanger), the break-even evaporating temperature drops significantly. With a 10°F (5.6°C) approach, the break-even temperature for the two system options is -55°F (-48.3°C) at full-load. By increasing the size of the cascade heat exchanger to achieve a 5°F (2.8°C) approach at full-load, a higher break-even evaporating temperature of -45°F (-42.8°C) is realized. Given this information, specifying a large cascade heat exchanger to raise the efficiency of the cascade system option to break-even at warmer evaporating temperatures seems like a logical approach. Unfortunately, as the size of the cascade heat exchanger increases, its capital cost rises nonlinearly.

Mumanachit found that the optimum cascade heat exchanger temperature difference is approximately 10°F (5.6°C). At this temperature difference, the range of evaporating temperatures that result in life-cycle savings for the cascade system option is the broadest; thereby, allowing economic operation at higher evaporating temperatures over 20 years of the system operation.

CONCLUSIONS

In this edition of the **COLD FRONT**, we considered two-stage direct ammonia as well as CO₂/NH₃ cascade system options to meet low temperature refrigeration loads. For evaporating temperatures in the range of -52°F (-46.7°C) and warmer, the two-stage ammonia system has a higher operating efficiency compared to a CO₂/NH₃ cascade system. For loads requiring evaporating temperatures below -52°F (-46.7°C), the CO₂/NH₃ cascade system offers an efficiency advantage over the compound ammonia system. The life-cycle operating cost difference between the compound and cascade systems can be quantified and is defined as the *premium difference*. The *premium difference* represents the present value of the life-cycle energy cost for the CO₂/NH₃ cascade system compared with the compound system. It can be viewed as the maximum capital cost difference that can be tolerated for comparable life-cycle system cost.

Although the CO₂/NH₃ cascade system option only becomes more efficient than a two-stage ammonia system at very low evaporating temperatures, it does offer another advantage. Using CO₂ as a refrigerant to meet evaporator loads decreases risks associated with product contamination (such as in low temperature warehouses and freezing systems) and potential risks of ammonia exposure to plant personnel in the event of refrigerant leakage. While there is a reduced risk of exposure to ammonia in CO₂ systems, CO₂ presents other risks when used in confined spaces (e.g. spiral freezers). The primary risk to personnel in the event of exposure to CO₂ is asphyxiation (PEL_{CO2} = 50,000 ppm).

There are advantage and disadvantages to each refrigeration technology and configuration. This is no different for the two (2) technologies we focused on in this article: compound ammonia systems and CO₂/NH₃ cascade. The following table highlights some of the advantages and disadvantages of both.

COMPOUND AMMONIA	CO₂/NH₃ CASCADE
PROS	PROS
<ul style="list-style-type: none"> • High operating efficiency over a wide temperature range • Reasonable design working pressures • Well established technology • Utilizes natural refrigerant • Ammonia is self-alarming 	<ul style="list-style-type: none"> • Achieves high operating efficiencies at very low (below -52°F [-46.7°C]) evaporating temperature requirements • Both carbon dioxide and ammonia are natural refrigerants • Operates above atmospheric pressure (no air and water contamination during operation) • Smaller piping and valves • Field experience indicates equipment is easier and safer to service • When used as a secondary fluid, oil fouling of heat exchangers can be eliminated (no oil draining is required)
CONS	CONS
<ul style="list-style-type: none"> • Requires large compression equipment at low refrigerant operating temperatures • Operation in a vacuum at low temperatures (resulting in air infiltration with purge requirement) • Toxic chemical circulating to evaporators (human risk and product contamination risk) • Requires management of oil on the low-side (the commonly used manual oil draining is a high risk procedure) • Incidents and accidents can have severe consequences 	<ul style="list-style-type: none"> • Very high working pressures at moderately high temperatures • Potential risks of mixing ammonia and carbon dioxide (e.g. during cascade heat exchanger leak and during charging) • Re-emerging technology – not widely used and limited experience with contractors in the North America • Liquid leaks to atmosphere can create a solid (“dry ice”) • Typically requires the use of electric defrost • Carbon dioxide is not self-alarming (asphyxiation risk)

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REMINDER

IRC END-USER SAFETY BEST PRACTICES WORKSHOP

SEPTEMBER 30-OCTOBER 1, MADISON, WI

Overview

The Industrial Refrigeration Consortium (IRC) will host a workshop focused on cultivating best practices relating to refrigeration system safety and operations. In addition to a detailed look at the **operating procedures** and **operator training** elements of the Process Safety Management Standard, this workshop is your opportunity to compare your organization's experiences in machinery room best practices with other end-users participating. We will also take time to explore trends in incidents and accidents.

Format

In a workshop format, we will review the PSM requirements for developing operating procedures and staff training. We will also explore OSHA-issued interpretations for these elements to establish a clear understanding of the standard's requirements. Workshop attendees will then share their experiences and strategies for meeting those requirements based on the PSM content for their organization. Best practices will be identified and shared with participants.

Goals

The goals of this workshop are to

1. Cultivate and diffuse best practices relating to safety and process safety
2. Achieve meaningful improvement in process safety management improvements
3. Review specific requirements for compliance with provisions of the PSM standard
4. Review relevant OSHA-issued interpretations on specific provisions of the PSM standard

Attendees

This workshop is open to employees of IRC member companies only. There is no cost for attending.