Technical Paper #2

Comparing Evaporative and Air-Cooled Condensing in Ammonia and HFC-507 Refrigeration Systems

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Abstract

Ammonia is the ideal industrial refrigerant, with high efficiency and broad utilization in industry and attractive environmental properties. Use of air-cooled ammonia systems is uncommon, though; almost all ammonia systems use evaporative condensers based on past practice and assumptions concerning efficiency and system performance. However, efficient use of air-cooled condensing could allow the benefits of ammonia to be realized more widely. As a follow-on to a previous IIAR paper, "Comparing Evaporative and Air Cooled Condensing for Ammonia Systems" (Scott 2014), this paper compares energy and operating costs of ammonia and hydrofluorocarbon HFC-507 refrigerants using evaporative and air-cooled condensing applied to similar refrigeration systems for refrigerated warehouses in 11 U.S. cities.

Introduction

This paper expands on an earlier paper comparing evaporative (evap) and air-cooled condensing in ammonia refrigeration systems, with additional evaluation of HFC-507 refrigerant. Ammonia refrigeration systems in the United States use evaporative cooled condensers almost exclusively. Due to the large size of most ammonia systems, historical context, and industry perceptions regarding performance and efficiency, air-cooled condensing is seldom considered for ammonia. When air-cooled systems are necessary or preferable, the assumption has generally been that halocarbon systems were the only choice.

Increased water costs and reduced water availability in many areas make water conservation an important component of many companies' sustainability efforts. This paper compares evap-cooled condensing and air-cooled condensing in ammonia and R-507 systems in refrigerated warehouses in 11 U.S. cities. The comparison uses detailed hourly simulation of the refrigeration plant and local electric and water rates. The analysis primarily focuses on energy usage and electric costs, because these are the greatest "unknowns" in considering air-cooled ammonia systems.

The most intriguing and environmentally beneficial opportunity for air-cooled ammonia systems may be as an alternative to halocarbon refrigerants, historically hydrochlorofluorocarbon HCFC-22, which is being phased out, and more recently HFC refrigerants, which are now being phased down. This study compares four system design cases: with ammonia and R-507 refrigerants and with air and evaporative-cooled condensing. In all four cases the refrigeration system is a central industrial plant, with consistent design assumptions. The intent of this study is to provide a focused "apples-to-apples" comparison of refrigerants and condensing means, emphasizing energy and water costs.

Numerous large, built-up halocarbon systems currently exist and are being built. The largest of these tend to use evap condensers. Typically, halocarbon systems, most of which are air cooled, comprise smaller parallel "rack" systems and one or two compressor split systems. Potentially addressing both large and smaller industrial applications, new ammonia system designs are emerging, offering smaller low-charge evap- and air-cooled packages. Comparing smaller ammonia and halocarbon systems in detail is beyond the scope of this paper, but the hope is that these results will prompt consideration of air-cooled condensing for these new smaller ammonia systems as they are developed, thus increasing the reach and benefits of ammonia refrigeration.

Background

Ammonia is the dominant refrigerant in industrial refrigeration systems due to its low cost, availability, and attractive thermodynamic and physical properties, resulting in high system efficiency. Evaporative condensing has been the standard for ammonia systems in the United States, with almost no use of air-cooled condensers until recent years. However, air-cooled ammonia condensers have been used more commonly in Europe over the last 15 years, especially in areas with high water costs. The higher design pressures required for air-cooled systems, affecting compressors, piping, valves, and vessels, have historically limited equipment availability. In compression, ammonia produces high actual discharge temperatures, which are exacerbated by the higher discharge pressures in air-cooled systems. While not a concern with screw compressors, which use oil or liquid cooling in compression, this characteristic is more difficult to address with reciprocating compressors. This, in addition to higher operating pressures, may explain why air-cooled condensing has had little past use in ammonia systems.

Historically, commercial and medium-sized industrial applications and a small portion of large industrial facilities have used halocarbon systems. R-22 was a reliable alternative for decades, and although it is no longer used in new construction, many R-22 systems still exist. In the last 10–15 years, HFC refrigerants, typically R-507 or R-404A, have been used in industrial-scale applications, with both air-cooled and evaporative-cooled condensers. HFC refrigerants are now being phased down in the United States, especially R-507 and R-404A because of their high global warming potential (GWP). The new HFC and HFO (hydrofluoro-olefin) synthetic refrigerants are often problematic in flooded or recirculated industrial applications due to the boiling point transition or “glide” characteristic common in many of the new emerging alternatives. And while ammonia systems have stringent safety requirements and regulations—with attendant costs—HFC refrigerants are increasingly subject to regulations, fees, and taxes globally. In this context, the environmental benefits of ammonia (with zero GWP) along with low cost per pound and high energy efficiency are attractive as an alternative to existing and new synthetic refrigerants, particularly if the ammonia systems can be applied cost-effectively with air-cooled condensers.

Study design

The study design follows that of the previous paper, with improvements in simulation assumptions and methods. A medium-sized refrigerated distribution warehouse was employed, as shown in Figure 1, with freezer, cooler, and dock spaces.

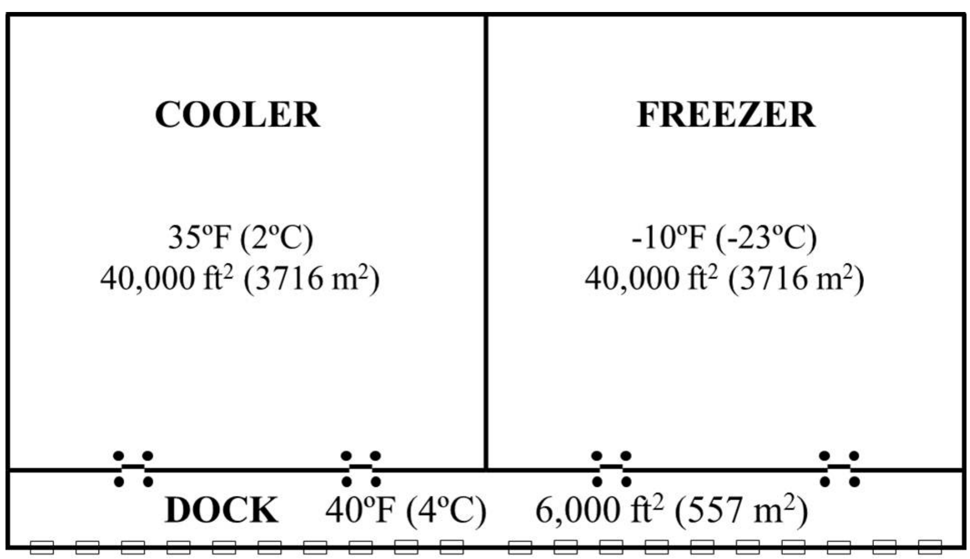

Figure 1. Refrigerated warehouse space layout.

The refrigeration systems were designed with two suction temperature levels, each with two equal-size single-stage noneconomized screw compressors. Recirculated liquid supply was assumed for all evaporator coils, with the high-temperature recirculator feeding the liquid supply to the low-temperature recirculator. Appendix A shows the design assumptions and equipment selections.

The study used 11 cities to obtain a range of weather conditions. Because no particular correlation exists between weather conditions and electric rates across the country, annual energy consumption is the most relevant variable in understanding the effect of climate on the two means of condensing. In addition, the local electric and water rates for the subject cities were used to provide examples of actual operating costs. Table 1 shows the 11 cities along with the ASHRAE (2013b) design dry bulb temperature (DBT) and wet bulb temperature (WBT) conditions. Note that these temperatures reflect more realistic design conditions for refrigeration equipment selection compared with those used in the first paper, which were based on the design temperatures in ASHRAE Standard 90.1 (ASHRAE 2013a).

City	**ASHRAE 0.4% DBT °F (°C)**	**ASHRAE 0.4% WBT °F (°C)**
Dallas, Texas	100 (38)	79 (26)
Chicago, Illinois	92 (33)	78 (26)
Denver, Colorado	94 (34)	65 (18)
Miami, Florida	92 (33)	80 (27)
Salinas, California	83 (28)	65 (18)
Portland, Oregon	92 (33)	71 (22)
Atlanta, Georgia	94 (34)	77 (25)
Charlotte, North Carolina	94 (34)	77 (25)
Fresno, California	104 (40)	74 (23)
Phoenix, Arizona	110 (43)	76 (24)
Minneapolis, Minnesota	91 (33)	77 (25)

Table 1. Study cities and design weather conditions. Source: Data from ASHRAE (2013b).

Local utility costs for electric and water usage were used to provide realistic economic examples but are only examples because electric and water rates can vary greatly within a given climate selection.

Condenser selection

The evap-cooled condenser selections were made using the compressor total heat of rejection (THR), based on compressor capacity, and using the approach (i.e., temperature difference or TD) between saturated condensing temperature (SCT) and entering wet bulb temperature, as shown in Table 2. These condenser approach values are equivalent to the minimum requirements in the California 2013 Title 24 Standards for new refrigerated warehouses (CEC 2013). The closer approach (lower TD) at higher design WBTs does not mean a condenser is necessarily larger as a result; rather the lower TD reflects the physics of moist air and the fact that condensers have greater capacity at the same approach as the WBT increases. This effect can be observed in the heat rejection capacity factor tables all manufacturers of evaporative condensers provide for selection at specific application WBT and SCT conditions (BAC). Thus the operating approach would be lower even for a condenser of the same size with the same THR and a higher WBT. Industry practice often specifies condensing temperature rather than approach temperature, which can result in condensers being over- or undersized, at least from an energy efficiency standpoint. Specifying the condenser approach in this way is more consistent in terms of overall system energy efficiency and the objectives of this study.

Design WBT	**TD**
≤ 76°F (24°C)	20°F (11.1°K)
76–78°F (24–26°C)	19°F (10.6°K)
≥ 78°F (26°C)	18°F (10.0°K)

Table 2. Evap condenser design approach.

The air-cooled condenser selections were based on a 15°F (8.3°K) approach between SCT and ambient entering DBT. The air-cooled design approach is the same for all ambient conditions.

The assumed approach temperatures directly determine the size of the condenser and thus affect the results of the study. These sizes are considered to be a reasonable balance of energy efficiency and cost-effective sizing that could be applied across numerous climates. However, this condenser sizing is not intended to be a comprehensive design recommendation. In actual system design for a particular facility, the optimum condenser is best determined with site-specific modeling of hourly load shape and heat rejection, weather, condenser size characteristics, and—perhaps most important—the control methods and setpoints.

Load calculations

Cooling design loads were calculated for each location, including envelope, infiltration, and internal loads, with refinements made after the first paper in facility operating schedules and doorway traffic assumptions. The design loads, in Btu/h, were used to select compressors and condensers. Table 3 summarizes the loads for one location, Dallas, Texas.

		Freezer		Cooler		Dock	
Transmission		298,611	27%	160,953	17%	61,435	9%
Infiltration		306,270	27%	26,409	3%	468,552	66%
Internal	People	17,400	2%	29,000	3%	5,800	1%
	Equipment	180,000	16%	240,000	26%	60,000	8%
	Fans	134,512	12%	111,424	12%	85,827	12%
	Lights	95,564	9%	95,564	10%	28,669	4%
	Product	41,667	4%	226,042	24%	0	0%
	Defrost	41,875	4%	34,688	4%	0	0%
Total Peak Load		1,115,899	100%	924,080	100%	710,283	100%
Load with Safety Factor		1,283,284	115%	1,062,692	115%	816,826	115%
SF per Ton		374		452		176	
Load for Coil Selection		1,339,079	120%	1,108,896	120%	852,340	120%

Table 3. Design load calculations for Dallas, Texas, location.

The design load is used for equipment sizing, whereas the hourly cooling loads calculated by hourly system modeling are based on weather files, the building envelope response, facility operating schedules, and other modeling assumptions, and not directly determined by any of the design load components.

Compressor selection

To minimize unintended part-load effects, the compressors for each analysis case (location and refrigerant and condenser type) were size-adjusted from a single representative base compressor model for the low- and high-temperature suction levels. In other words, the compressor size was made to match the desired design capacity exactly to avoid unintended part-load effects that limiting selections to actual compressor models would cause. Part-load operation assumed slide valve control and utilized representative compressor part-load performance curves.

Condenser-specific efficiency

Both evap- and air-cooled condensers are available with a very wide range of fan power for a given capacity. In a given cabinet size, for example, evap condensers are available with fan motor power ranging from 10 hp (7.5 kW) to 40 hp (30 kW). Historically, air-cooled condensers have had an even larger range, such as from 2 hp (1.5 kW) to 10 hp (7.5 kW) for the same size belt-drive fan blade on certain condensers. Today, air-cooled condensers tend to utilize direct drive motors and have smaller motors, but they still have a substantial range in power for a given capacity.

Specific efficiency is the term used to define condenser fan power vs. capacity. Specific efficiency is the heat rejection capacity at an assumed specific efficiency rating point divided by the input electrical power for the condenser fans, as well as the spray pump power for evap condensers. Specific efficiency rating conditions are unrelated to application conditions. The rating conditions for evap-cooled condensers and air-cooled condensers are necessarily different, because one is based on WBT and one is based on DBT. For the same reason, numerical comparison of specific efficiencies can only be made between like condenser types, not between air- and evap-cooled condensers. Table 4 shows the rating assumptions and assumed specific efficiencies used in this study.

		Evap	Evap (T24)	Air
Specific Efficiency Rating Basis	SCT °F (°C)	100 (38)	100 (38)	105 (41)
	WBT °F (°C)	70 (21)	70 (21)	
	DBT °F (°C)			95 (35)
Specific Efficiency (BTUh/W)		275	350	90

Table 4. Specific efficiency assumptions. (T24 refers to California Title 24.)

The condenser specific efficiency rating conditions are taken from the values that the California new construction utility incentive programs utilize, where this parameter first came into use, and were more recently published in CEC (2013).

Note that the specific efficiency rating conditions are not special; other rating condition assumptions could evolve in the future, which would result in different specific efficiency numbers for each condenser and for minimum standards. The two California study locations used a specific efficiency of 350 Btu/h/W, consistent with the minimum value required by CEC (2013). The California value was determined to be cost effective for new refrigerated warehouses given California climates, utility rates, and energy efficiency policies. A value of 275 Btu/h/W (e.g., somewhat higher fan horsepower) was assumed for the other study locations. This is the average efficiency estimated by the author to be generally cost effective on a national basis for large axial fan condensers used in refrigerated warehouses.

Note also that the cost-effective specific efficiency assumptions are based on a design with all condenser fans running in unison and using variable speed fan control, as will be discussed subsequently. An alternative design approach could be considered with physically larger condensers using smaller fan motors to obviate the need for variable speed drives, albeit at higher condenser cost. With this alternate design approach, the condensers would have higher specific efficiency with no variable speed control, but with low enough fan power to deliver equivalent efficiencies.

Air-cooled condenser specific efficiency is based on motor sizes available from manufacturers that are either standard or have nominal adaptation to standard products. The study used a specific efficiency of 90 Btu/h/W. No adjustments were made to the specific efficiency assumption for altitude, although, in the case of the Denver location, the air-cooled condenser size would certainly need adjustment for altitude. Air-cooled condenser manufacturers publish capacity adjustments for altitude, but provide no information on motor power at altitude. The typical air-cooled capacity adjustment for 5,000 ft (1,500 m) altitude is approximately 12%, which is roughly similar to the air density change from sea level. Because fan power and density are nominally proportional (i.e., based on affinity laws), the same specific efficiency basis was assumed to be reasonable at higher altitude.

As noted previously, air-cooled specific efficiencies and evap-cooled specific efficiencies cannot be compared directly. Air-cooled condensers require far greater air volume than evap condensers do and thus generally have higher fan power. Table 5 shows the input power for evap-cooled and air-cooled condensers for the 11 study locations.

NH3		Dallas	Chicago	Denver	Miami	Salinas	Portland	Atlanta	Charlotte	Fresno	Phoenix	Mnpolis
Evap Cooled	Fan, kW	28.7	28.7	27.0	28.1	19.9	26.1	28.5	28.5	19.6	26.3	28.3
	Pump, kW	4.2	4.2	4.2	4.2	4.2	4.2	4.2	4.2	4.2	4.2	4.2
	Total	32.9	32.9	31.2	32.3	24.1	30.3	32.7	32.7	23.8	30.5	32.5
Air Cooled	Fan, kW	46.7	44.2	41.7	44.6	38.9	42.5	44.4	44.4	46.7	49.2	43.5

R-507		Dallas	Chicago	Denver	Miami	Salinas	Portland	Atlanta	Charlotte	Fresno	Phoenix	Mnpolis
Evap Cooled	Fan, kW	31.2	30.8	28.8	30.7	21.2	28.1	29.3	29.3	21.3	28.4	29.1
	Pump, kW	4.2	4.2	4.2	4.2	4.2	4.2	4.2	4.2	4.2	4.2	4.2
	Total	35.4	35.0	33.0	34.9	25.4	32.3	33.5	33.5	25.5	32.6	33.3
Air Cooled	Fan, kW	52.3	48.5	46.1	49.0	42.1	46.7	49.0	49.0	52.9	56.9	47.7

Table 5. Condenser power by location.

Hourly modeling

Building and system modeling was performed using the DOE2.2R simulation program (Hirsch 2016). This program includes hourly calculation of loads, refrigeration system performance, and utility costs. The heat load calculations include transmission with consideration of hourly weather and solar effects; infiltration, which utilizes American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) formulas for interzonal (doorway) mass exchange and considers wind velocity; and internal loads, which may be calculated automatically (e.g., evaporator fan speed and thus power and heat) or scheduled as part of input instructions (e.g., product and defrost loads). The refrigeration system portion of the program is mass-flow based and calculated at a component level. Refrigerant mass flow is determined from the cooling loads, with compressor operation developed to meet the required mass flow, and balanced against the available condenser capacity and ambient conditions. Compressor performance is determined from regressions based on saturated suction temperature (SST) and saturated discharge temperature (SDT), with a separate relationship for part-load (e.g., slide valve) efficiency. Control strategies for evaporator fans, supervisory compressor sequencing and part-load control, and condenser setpoint and fan control are all explicitly modeled (within the limits of an hourly simulation model) in a manner consistent with actual control operation. TMY3 weather files were used for hourly ambient temperatures, solar values, and wind velocity.

Adjustments to catalog ratings, primarily equipment derating, are essential to effective modeling and in particular to refrigeration modeling and this study for several reasons, including

- Equipment catalog ratings are based on steady-state operation for new equipment and generally at design (peak) conditions, whereas most hours of operation are not at steady state and the system is operating under off-design conditions (which may not be within the catalog ratings) and at part load.
- Condenser performance values in catalogs have historically not referenced a rating standard, and the ratings are not certified. Recently, some evap condenser manufacturers have started to, or will soon, use the Cooling Technology Institute's standards (CTI 2011) and/or ASHRAE standards (2005) to test evaporative condensers and are moving toward certification of their evap condenser ratings. Manufacturers of air-cooled refrigeration condensers in the United States have not typically referenced rating standards in their catalog ratings. The Air-Conditioning, Heating, and Refrigeration Institute's standard for air-cooled condensers, AHRI (2005), uses test rating conditions that are more suitable for air-conditioning applications than refrigeration, e.g., the 30°F (17°K) approach. Beyond considerations of actual vs. catalog performance at full capacity, factors for performance at part load are less certain, in most cases not published, and, given the many variables, very difficult to test.
- Transient operation, e.g., fan cycling and cyclical pressure variations may have a large effect on condenser operation.

- Field effects including multiple adjacent condensers, building configuration, and effect of prevailing wind result in recirculation of air from the condenser outlet and reduced condenser capacity. Piping pressure drop and flow imbalance would be part of this factor.
- Scale, corrosion, and bio-fouling in evaporative condensers often comprise a large factor, reducing condenser capacity and sometimes condenser longevity.

Derating the catalog capacity values is necessary to address these factors and simulate real-world condenser performance under average hourly conditions. Table 6 summarizes individual factors, largely based on the author's judgment and opinion.

	Evap	Air	
Catalog Capacity	100%	100%	Notes
Applied vs. Catalog Adjustment	0%	10%	Authors opinion there is less certainty with air-cooled
Scale, Fouling and Dirt	20%	10%	Evap fouling is higher on average due to ubiquitous scale
Non-steady State Factors	5%	5%	Small factor, considering large system with variable speed
Field Installaton Effects	5%	10%	More likely air-cooled is more compromised by recirculation
Part Load Effects	5%	5%	Equal assumption
Net De-rate vs. Catalog	**69%**	**66%**	

Table 6. Condenser derating factors for hourly analysis.

These derating factors undoubtedly seem high initially, indicating the realized average capacity is approximately a third less than the catalog ratings. However, based on the author's experience in evaluating expected vs. actual hourly performance at a limited number of facilities, this is not an unreasonable conclusion—particularly when noting that the purpose of these factors is to develop an accurate hourly simulation through the course of a year, including off-design and part-load effects and not just peak design conditions. Individual derating components may be more or less manageable through system design and ongoing system maintenance; for example, a somewhat small amount of scale on evaporative condensers can have a very large effect on capacity.

In terms of this paper, an important question is whether the derating assumptions for evaporative condensers and air-cooled condensers are themselves a strong determinant of the study results. Because the cumulative derating factors shown in Table 6 only differ by 3%, the comparative effect on total system energy simulation results is somewhat small. However, the factors are only estimates. Each derating component justifies more detailed study, first to understand and quantify the effects on condenser capacity and then to define how this knowledge could be used to model system performance and energy use. Rather than applying a single all-encompassing adjustment factor, additional knowledge of each factor would result in better system design, allow more sophisticated modeling, and lead to more effective condenser performance measurement and monitoring.

Head pressure control

Control of head pressure or condensing temperature, which are interchangeable terms in this context, is the essential consideration in comparing evap-cooled and air-cooled condensers. Without a balanced and consistent assumption the results would be skewed. Head pressure control elements include how condenser fans are controlled (cycling or speed modulation), the control strategies used to control fans, and how low head pressure is allowed to drop, as cooler weather permits.

Floating head pressure

Aside from the few (if any) hours in a year that the compressors and condensers run near maximum capacity, a constant opportunity exists to employ controls to optimize the total power the compressors and condenser fans use. For lack of a better description, this is called floating head pressure. Floating head pressure is somewhat vague and can have multiple meanings, but here the term describes the overall effort to maintain the lowest total energy use of compressors and condensers throughout the year. Three elements are involved:

- How low can the head pressure (or condensing temperature) go, weather permitting?
- How are the condenser fans controlled?
- How is the condenser fan control setpoint determined?

Minimum condensing temperature

The lowest possible steady-state condensing temperature is a function of compressor oil separator sizing, other compressor limitations, and system design pertaining to liquid supply to evaporators. Generally all modern systems can operate to 70°F (21°C) SCT or lower, i.e., 114 psig (7.9 bar) for ammonia. Some existing systems need higher pressure during defrost periods; however, newer systems typically need no more than 95 psig (6.6 bar) for defrosting and are equipped with regulators to limit defrost pressure, thereby allowing head pressure reduction to near 95 psig (6.6 bar) pressure with no effect on defrost.

The financial value of designing for a minimum condensing temperature lower than 70°F (21°C) may be small in a warm climate but could yield large incremental savings in a colder climate. This also becomes an important difference between evap- and air-cooled systems in many climates. As noted previously, evap condensers "lose" capacity as the wet bulb temperature drops, in terms of the approach the condenser can achieve for a given heat rejection, whereas an air-cooled condenser maintains the same approach temperature at lower dry bulb temperatures. Coupled with this fact, the difference between DBT and WBT varies through the day and the year, favoring evap-cooled condensers in the hottest weather periods, but favoring air-cooled condensers during the moderate and cool temperatures that typically comprise most of the year.

The minimum condensing temperature setpoint used in this study for all 11 locations and all systems was 60°F (15.6°C).

Fan control

For both evap- and air-cooled condensers the study assumes all fans are controlled in unison with variable speed, rather than fan cycling. The use of all surface, all of the time, is generally the most efficient means of condenser capacity utilization. The affinity laws define physical principles of flow, pressure drop, and power, and specifically the "third-power" relationship between airflow and fan power. Figure 2 shows this relationship as a curve that is applied to compare an example condenser with fan cycling with the same condenser with variable speed fan control, with both condensers at 50% capacity. Condenser capacity is nominally proportional to airflow and fan speed, whereas power varies with the cube of fan speed, thus increasing the part-load condenser efficiency at 50% capacity from 90 to 360 Btu/h/W.

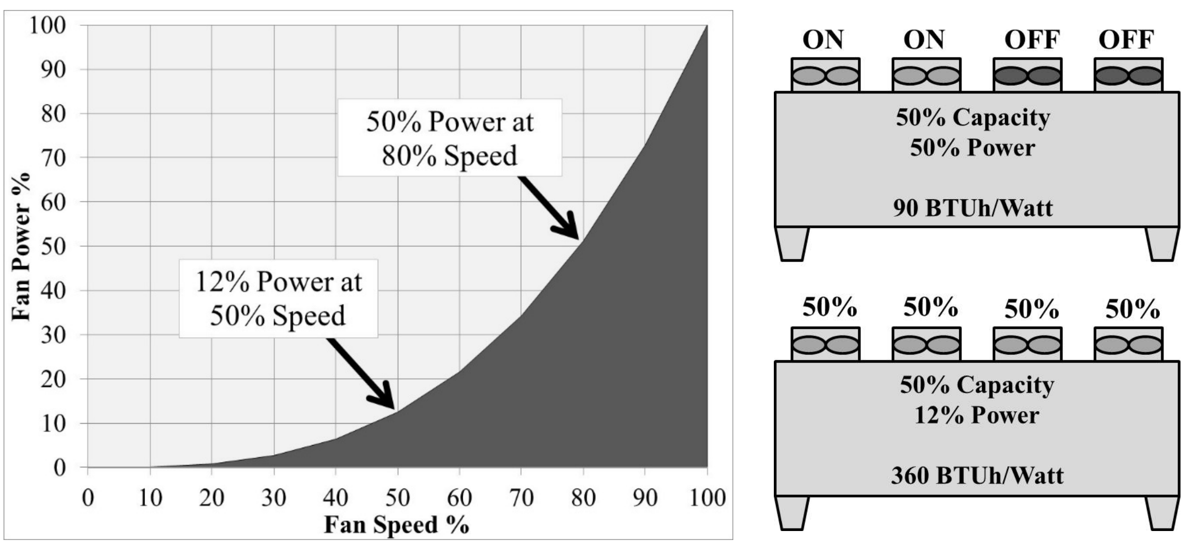

Figure 2. Condenser capacity vs. power for fan cycling and variable speed.

The nonlinear relationship of fan power to airflow, and thus to condensing temperature and compressor power, is important and points to an important aspect of control optimization as described in the following.

Setpoint determination

The final aspect of condenser control and system energy optimization is setpoint determination. The essential objective is balancing the compressor and condenser power to obtain the lowest total power. Based on the sole objective of reducing compressor power, the condenser would simply run at 100% capacity to balance at the lowest head pressure possible at the ambient temperature. However, the condenser uses energy as well, which creates the tradeoff between compressor power and condenser power. As shown previously, fan power vs. condenser capacity is nonlinear, following a third-power relationship. In addition, like all heat exchangers, increased condensing capacity has diminishing returns in terms of the heat exchanger approach. For example, if doubling condenser capacity (and power) reduces the approach (TD) by 20°F to 10°F (11.1°K to 5.6°C), a reduction of 10°F (5.6°C) in condensing temperature, an additional doubling would only reduce the approach and condensing temperature by 5°F (2.8°C), producing only half the benefit at the compressor. Both of these nonlinear relationships complicate the goal of balancing condenser fan control vs. compressor power. Simply put, the goal is to use as much condenser capacity as possible, without increasing condenser power more than the gain achieved in compressor power.

The most common control strategy used to manage floating head pressure for optimum power use is ambient-following logic, where the condenser control setpoint is determined by adding an "offset" value to the current ambient temperature to determine the target saturated condensing temperature setpoint. This offset is typically called the control TD. For evap-cooled condensers WBT is used, and for air-cooled condensers DBT is used. Figure 3 shows a simplified example of ambient-following control. The condensing temperature setpoint follows ambient temperature, bounded by a minimum setpoint limit defined by the system design minimum pressure capability [e.g., 70°F (21.1°C) in this example figure] and typically a maximum setpoint limit as well [e.g., 95°F (35.0°C)] at which running the fans at 100% is desirable to limit maximum system pressures, regardless of energy optimization.

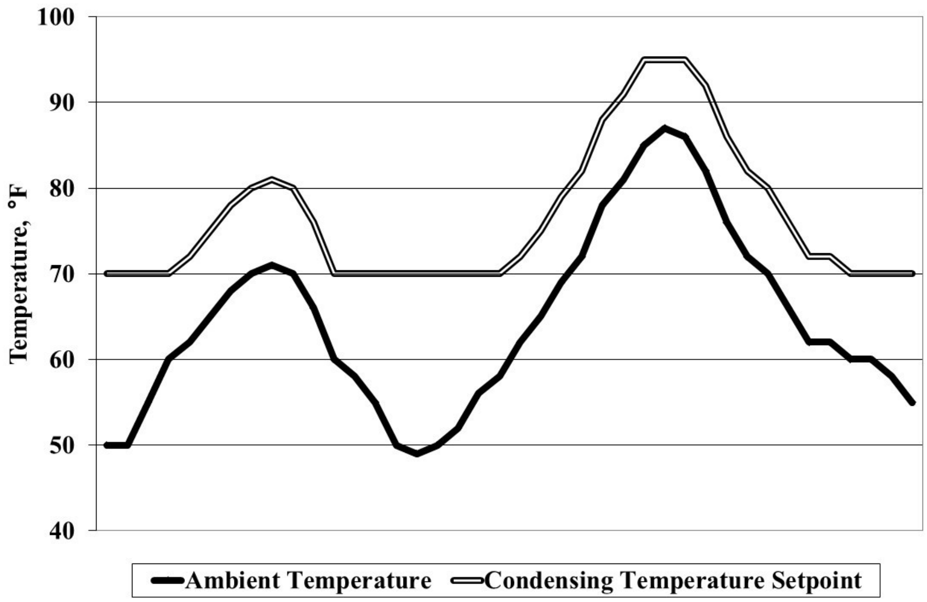

Figure 3. Ambient-following condensing temperature setpoint.

When using an energy simulation, as with this study, the optimum control TD value is determined by iterating the simulation control TD value to obtain the lowest total combined power. To allow for real-world control variations, the control TD is then raised slightly. In actual plant operations, which typically lack detailed guidance from energy analysis, the control TD setpoint is commonly optimized using a condenser fan speed "sweet spot" of 60–80% of target, when not at minimum SCT. An average speed of 60–80% is normally close to the ideal operating point, utilizing a large fraction of the condenser capacity and still providing a sizable reduction in condenser fan power.

Other floating head pressure control and optimization methods are possible and may provide greater savings, but ambient following control is the most common method and for the purpose of this study provides a consistent comparison between evap-cooled and air-cooled condensing.

Water costs

Water, sewer, and water treatment costs are often the impetus for considering air-cooled condensing, in addition to concern regarding future water availability. Table 7 shows the results of this study's investigation of water and sewer rates in all study cities.

	Dallas	Chicago	Denver	Miami	Salinas	Portland	Atlanta	Charlotte	Fresno	Phoenix	Mnpolis
Supply Water, $/CCF	$ 2.60	$ 2.90	$ 2.70	$ 1.60	$ 2.60	$ 3.90	$ 3.70	$ 2.70	$ 1.10	$ 4.10	$ 3.50
Sewer Cost, $/CCF	$ 2.80	$ 2.90	$ 2.80	$ 5.30	$ 1.60	$10.30	$15.80	$ 4.50	$ 1.20	$ 2.50	$ 3.40
Sewer Fraction of Supply	40%	100%	40%	40%	40%	40%	100%	100%	100%	80%	100%
Effective Rate, $/CCF Supply	$ 3.70	$ 5.80	$ 3.80	$ 3.70	$ 3.20	$ 8.00	$19.50	$ 7.20	$ 2.30	$ 6.10	$ 6.90

Table 7. Water and sewer costs.

For cities that adjust the sewer rate based on measured flow or submetering credits to account for evaporated water, the sewer cost was factored to 40% of the supply water cost, and the two costs added to obtain the effective rate for both supply water and sewer costs expressed in $/CCF (100 ft^3) of supply water usage.

Water usage and water treatment

Water consumption from evaporation was estimated using the actual hourly heat of rejection from the simulation model and 1,000 Btu for each pound of water evaporated, which resulted in consumption very close to the industry rule of thumb of 2 GPM (gallons per minute) per 1,000 MBtu/h. In addition, drift was assumed to be 0.001% of the circulation rate and bleed was calculated based on 3.0 cycles of concentration, through use of a good-quality water treatment system with attentive monitoring and maintenance, with an assumed monthly cost of $1,200.

Results

Tables and figures in the following sections show simulation results for the 11 cities.

Energy usage

Table 8 shows the annual energy usage for each location and for both condensing means and both refrigerants. In addition to compressor and condenser energy, "other" energy consists of evaporator coil fan motors, lighting in the refrigerated spaces, and refrigerant recirculation pumps.

NH3 Results	Evaporative Cooled					Air Cooled				Air Cooled Increase (Decrease)	
	Compr (kWh)	Cond Fan (kWh)	Cond Pump (kWh)	Other (kWh)	Total (kWh)	Compr (kWh)	Cond Fan (kWh)	Other (kWh)	Total (kWh)		
Chicago	1,096,160	39,511	36,517	727,782	1,899,970	1,138,850	40,620	727,782	1,907,252	7,282	0.4%
Denver	963,021	32,790	36,513	724,965	1,757,289	1,079,794	38,994	724,965	1,843,753	86,464	4.9%
Portland	1,053,316	49,047	36,518	727,170	1,866,051	1,086,327	42,458	727,170	1,855,955	(10,096)	-0.5%
Dallas	1,364,121	51,412	36,521	736,435	2,188,489	1,460,078	67,651	736,435	2,264,164	75,675	3.5%
Miami	1,653,376	75,158	36,551	744,655	2,509,740	1,782,242	76,781	744,655	2,603,678	93,938	3.7%
Salinas	1,074,840	39,372	36,531	726,428	1,877,171	1,101,877	44,497	726,428	1,872,802	(4,369)	-0.2%
Atlanta	1,252,810	59,983	36,527	732,894	2,082,214	1,320,314	58,969	732,894	2,112,177	29,963	1.4%
Charlotte	1,219,920	56,428	36,524	732,095	2,044,967	1,277,854	54,370	732,095	2,064,319	19,352	0.9%
Fresno	1,185,728	41,541	36,528	735,220	1,999,017	1,342,724	54,950	735,220	2,132,894	133,877	6.7%
Phoenix	1,270,333	63,045	36,525	742,382	2,112,285	1,628,862	68,838	742,382	2,440,082	327,797	15.5%
Minneapolis	1,051,725	35,104	36,519	726,970	1,850,318	1,044,638	35,127	726,970	1,806,735	(43,583)	-2.4%
									Average:	65,118	3.1%

507 Results	Evaporative Cooled					Air Cooled				Air Cooled Increase (Decrease)	
	Compr (kWh)	Cond Fan (kWh)	Cond Pump (kWh)	Other (kWh)	Total (kWh)	Compr (kWh)	Cond Fan (kWh)	Other (kWh)	Total (kWh)		
Chicago	1,147,610	42,301	36,533	837,808	2,064,252	1,203,858	52,814	837,808	2,094,480	30,228	1.5%
Denver	989,933	36,862	36,531	834,991	1,898,317	1,154,452	50,843	834,991	2,040,286	141,969	7.5%
Portland	1,100,813	53,723	36,536	837,196	2,028,268	1,149,773	54,941	837,196	2,041,910	13,642	0.7%
Dallas	1,432,852	71,867	36,534	846,461	2,387,714	1,629,724	67,148	846,461	2,543,333	155,619	6.5%
Miami	1,771,401	78,902	36,533	854,681	2,741,517	1,924,226	100,193	854,681	2,879,100	137,583	5.0%
Salinas	1,108,103	56,521	36,535	836,454	2,037,613	1,149,946	58,148	836,454	2,044,548	6,935	0.3%
Atlanta	1,326,320	61,101	36,534	842,920	2,266,875	1,425,563	77,422	842,920	2,345,905	79,030	3.5%
Charlotte	1,290,701	57,696	36,536	842,121	2,227,054	1,376,966	71,231	842,121	2,290,318	63,264	2.8%
Fresno	1,237,242	59,167	36,535	845,246	2,178,190	1,486,508	71,060	845,246	2,402,814	224,624	10.3%
Phoenix	1,351,915	67,571	36,531	852,408	2,308,425	1,863,977	88,828	852,408	2,805,213	496,788	21.5%
Minneapolis	1,087,145	45,886	36,529	836,996	2,006,556	1,108,287	45,959	836,996	1,991,242	(15,314)	-0.8%
									Average:	121,306	5.4%

Table 8. Annual energy usage.

Figure 4 shows the annual energy usage for all four analysis cases and each location. In nearly all locations, air-cooled condensing uses equal or greater total energy (kWh) than evaporative-cooled condensing. For ammonia systems, this ranges from essentially no difference to a 16% increase in Phoenix, which is a dry climate with very high dry bulb temperatures during much of the year that is obviously attractive for evaporative cooling. For R-507 systems the pattern is similar, with a 22% increase in Phoenix.

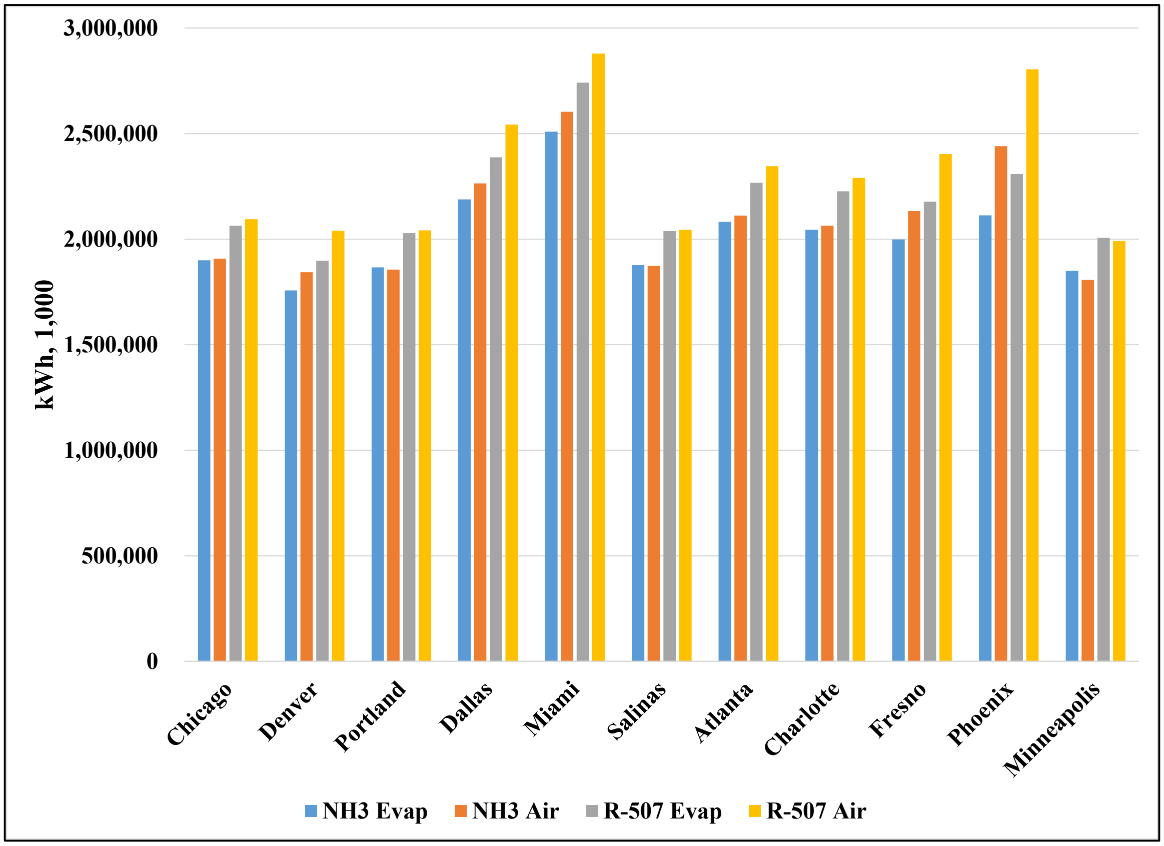

Figure 4. Annual energy usage (kWh).

Ammonia evap-cooled condensing is the reference in Figure 5, against which ammonia air-cooled, R-507 evap-cooled, and R-507 air-cooled condensing are compared, in terms of the percentage difference in annual kWh energy usage.

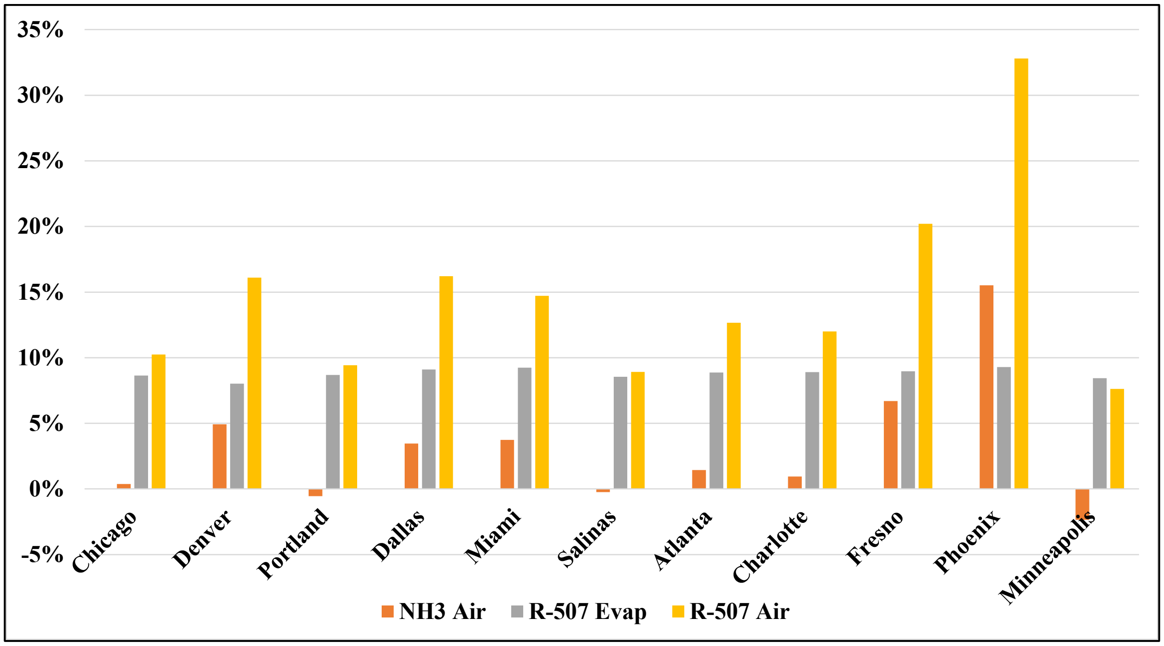

Figure 5. Energy usage vs. ammonia evap cooled (%).

Operating costs

Tables 9 and 10 show the electric utility and water costs for each location and for ammonia and R-507 respectively. Water costs are based on the costs in Table 7 plus water treatment costs as defined previously. The electric costs are separated between energy cost (for kWh usage) and demand charges.

NH3 Results	Evaporative Cooled					Air Cooled		
	Energy Cost ($)	Demand Cost($)	Total Energy Cost ($)	Water Costs	Total Energy and Water Costs	Energy Cost ($)	Demand Cost($)	Total Energy Cost ($)
Chicago	$ 139,458	$ 55,474	$ 194,932	$ 26,410	$ 221,342	$ 139,992	$ 59,554	$ 199,546
Denver	$ 95,610	$ 52,612	$ 148,222	$ 19,270	$ 167,492	$ 101,985	$ 62,456	$ 164,441
Portland	$ 113,738	$ 30,821	$ 144,559	$ 34,354	$ 178,913	$ 113,284	$ 33,169	$ 146,453
Dallas	$ 166,763	$ 15,116	$ 181,879	$ 21,625	$ 203,504	$ 172,529	$ 17,116	$ 189,645
Miami	$ 143,800	$ 86,266	$ 230,066	$ 23,327	$ 253,393	$ 149,091	$ 92,566	$ 241,657
Salinas	$ 172,560	$ 118,986	$ 291,546	$ 18,223	$ 309,769	$ 172,679	$ 132,766	$ 305,445
Atlanta	$ 212,969	$ 49,232	$ 262,201	$ 79,245	$ 341,446	$ 215,292	$ 53,870	$ 269,162
Charlotte	$ 84,574	$ 66,796	$ 151,370	$ 33,384	$ 184,754	$ 85,728	$ 72,860	$ 158,588
Fresno	$ 184,396	$ 126,041	$ 310,437	$ 12,836	$ 323,273	$ 199,430	$ 155,786	$ 355,216
Phoenix	$ 146,889	$ 26,199	$ 173,088	$ 30,922	$ 204,010	$ 174,873	$ 33,988	$ 208,861
Minneapolis	$ 79,456	$ 58,832	$ 138,288	$ 29,471	$ 167,759	$ 77,761	$ 61,770	$ 139,531

NH3 Results	Air Cooled Increase (Decrease)			
	Electric Only ($)	Electric Only (%)	with Water Cost ($)	with Water Cost (%)
Chicago	$ 4,614	2.4%	$ (21,796)	-9.8%
Denver	$ 16,219	10.9%	$ (3,051)	-1.8%
Portland	$ 1,894	1.3%	$ (32,460)	-18.1%
Dallas	$ 7,766	4.3%	$ (13,859)	-6.8%
Miami	$ 11,591	5.0%	$ (11,736)	-4.6%
Salinas	$ 13,899	4.8%	$ (4,324)	-1.4%
Atlanta	$ 6,961	2.7%	$ (72,284)	-21.2%
Charlotte	$ 7,218	4.8%	$ (26,166)	-14.2%
Fresno	$ 44,779	14.4%	$ 31,943	9.9%
Phoenix	$ 35,773	20.7%	$ 4,851	2.4%
Minneapolis	$ 1,243	0.9%	$ (28,228)	-16.8%
Average:	13,814	6.6%	(16,101)	-7.5%

Table 9. Evap- and air-cooled operating costs for ammonia systems.

507 Results	Evaporative Cooled					Air Cooled		
	Energy Cost ($)	Demand Cost($)	Total Energy Cost ($)	Water Costs	Total Energy and Water Costs	Energy Cost ($)	Demand Cost($)	Total Energy Cost ($)
Chicago	$ 151,516	$ 59,141	$ 210,657	$ 26,653	$ 237,310	$ 153,735	$ 64,301	$ 218,036
Denver	$ 103,348	$ 55,485	$ 158,833	$ 19,353	$ 178,186	$ 113,169	$ 69,076	$ 182,245
Portland	$ 123,487	$ 32,729	$ 156,216	$ 34,675	$ 190,891	$ 124,509	$ 35,955	$ 160,464
Dallas	$ 181,944	$ 16,415	$ 198,359	$ 21,836	$ 220,195	$ 193,802	$ 18,717	$ 212,519
Miami	$ 156,650	$ 92,626	$ 249,276	$ 23,690	$ 272,966	$ 164,394	$ 101,254	$ 265,648
Salinas	$ 187,010	$ 127,082	$ 314,092	$ 18,243	$ 332,335	$ 188,148	$ 143,901	$ 332,049
Atlanta	$ 230,157	$ 52,002	$ 282,159	$ 80,447	$ 362,606	$ 236,566	$ 58,706	$ 295,272
Charlotte	$ 91,938	$ 70,835	$ 162,773	$ 33,813	$ 196,586	$ 94,969	$ 79,728	$ 174,697
Fresno	$ 200,526	$ 134,414	$ 334,940	$ 12,904	$ 347,844	$ 224,538	$ 176,620	$ 401,158
Phoenix	$ 160,086	$ 27,741	$ 187,827	$ 31,342	$ 219,169	$ 201,203	$ 38,948	$ 240,151
Minneapolis	$ 86,041	$ 62,791	$ 148,832	$ 29,673	$ 178,505	$ 85,584	$ 67,675	$ 153,259

507 Results	Air Cooled Increase (Decrease)			
	Electric Only ($)	Electric Only (%)	with Water ($)	with Water (%)
Chicago	$ 7,379	3.5%	$ (19,274)	-8.1%
Denver	$ 23,412	14.7%	$ 4,059	2.3%
Portland	$ 4,248	2.7%	$ (30,427)	-15.9%
Dallas	$ 14,160	7.1%	$ (7,676)	-3.5%
Miami	$ 16,372	6.6%	$ (7,318)	-2.7%
Salinas	$ 17,957	5.7%	$ (286)	-0.1%
Atlanta	$ 13,113	4.6%	$ (67,334)	-18.6%
Charlotte	$ 11,924	7.3%	$ (21,889)	-11.1%
Fresno	$ 66,218	19.8%	$ 53,314	15.3%
Phoenix	$ 52,324	27.9%	$ 20,982	9.6%
Minneapolis	$ 4,427	3.0%	$ (25,246)	-14.1%
Average:	21,049	9.4%	(9,191)	-4.3%

Table 10. Evap- and air-cooled operating costs for R-507 systems.

For ammonia systems, the electric operating cost increase for air-cooled systems is 0–5% for eight locations. Denver is 11% higher due to low humidity throughout the year, which is highly advantageous for evap condensing. For Fresno and Phoenix respectively, the electric cost is 14% and 21% higher, due to high ambient DBT in both locations and high on-peak demand charges in Fresno. When water cost is considered, the annual operating cost for air-cooled systems is lower in all but Fresno and Phoenix, by as much as 21%. The cost in Fresno is 10% higher, with

Fresno water costs being the lowest of all 11 locations. Phoenix, with moderately high water costs, has only a small increase in annual cost for air-cooled condensing, an interesting finding considering the high peak DBT in Phoenix.

For R-507, the city-to-city trend is similar, but air-cooled condensing has a greater energy penalty than ammonia and lower savings when considering water.

Figure 6 shows the operating costs including water for all cities and system types.

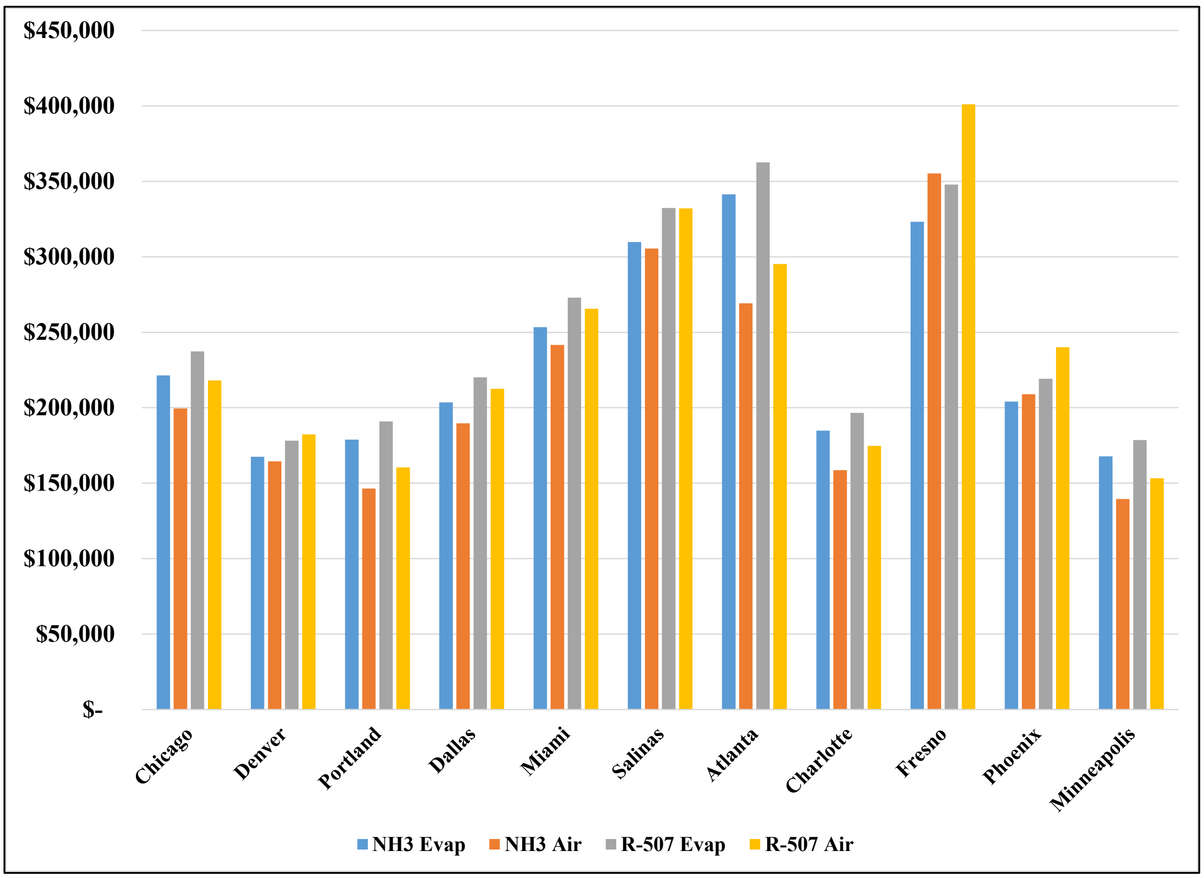

Figure 6. Comparison of operating costs ($).

Table 11 and Figure 7 show the cost savings for ammonia air-cooled, compared with R-507 air-cooled and evap-cooled condensing.

	Ammonia Air Savings vs. R507 Air		Ammonia Air Savings vs. R507 Evap	
Chicago	$ 18,490	8.5%	$ 37,764	15.9%
Denver	$ 17,804	9.8%	$ 13,745	7.7%
Portland	$ 14,011	8.7%	$ 44,438	23.3%
Dallas	$ 22,874	10.8%	$ 30,550	13.9%
Miami	$ 23,991	9.0%	$ 31,309	11.5%
Salinas	$ 26,604	8.0%	$ 26,890	8.1%
Atlanta	$ 26,110	8.8%	$ 93,444	25.8%
Charlotte	$ 16,109	9.2%	$ 37,998	19.3%
Fresno	$ 45,942	11.5%	$ (7,372)	-2.1%
Phoenix	$ 31,290	13.0%	$ 10,308	4.7%
Minneapolis	$ 13,728	9.0%	$ 38,974	21.8%
Average:	$ 23,359	9.7%	$ 32,550	13.6%

Table 11. Ammonia air-cooled savings vs. R-507 air and evap cooled.

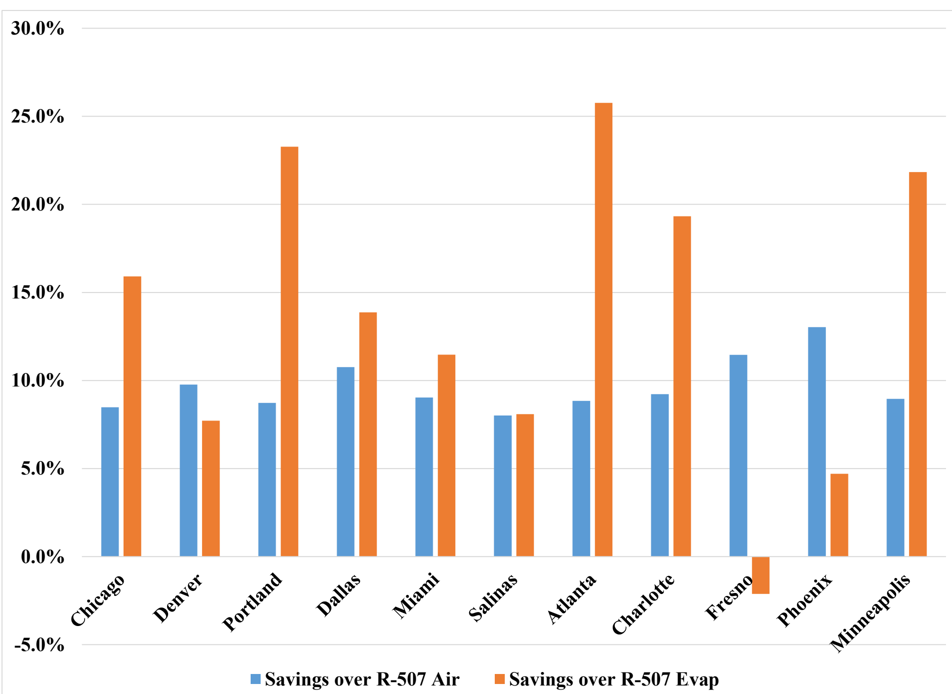

Figure 7. Ammonia air-cooled savings (%).

Ammonia air-cooled saves 8–13% in costs compared with R-507 air cooled and 5–26% compared with R-507 evap cooled, with the exception of a small increase for Fresno.

No load-shifting control was included in the analysis. Load shifting on high-efficiency systems should be undertaken cautiously to avoid increasing total energy usage, but to the extent load shifting is cost effective in all other respects, it would yield comparatively greater benefits on air-cooled systems than on evap-cooled systems. This is due to the higher day-to-night range in dry bulb temperatures than wet bulb temperatures, particularly during peak ambient periods when electric rates are typically highest.

The assumptions of this paper, naturally, affect the results. The assumptions were intended to accurately assess both condensing options with the condenser-related control methods that a modern facility would employ. The sensitivity to changes in various assumptions was not studied. Most assumptions likely have a small comparative difference, whereas others (e.g., minimum condensing temperature setpoint) would be expected to have a larger comparative difference. Also, the condenser derating assumptions are definitely important, and either through error in these assumptions, or actions taken to minimize the factors in a particular design or application, the comparative outcome in energy usage could be materially different. A learning curve could also be expected in applying large air-cooled ammonia condensers (e.g., field effects), although the ammonia plants in most refrigerated warehouses are moderately sized and not significantly beyond the scale of other air-cooled refrigeration and chiller applications.

One very important factor is the assumption of subcooling or flash cooling of the liquid that feeds the low-temperature loads by the high-temperature system, which benefits R-507 far more than it does ammonia. For halocarbon systems without liquid cooling (by a higher temperature system or economizer), energy use and peak demand would be much higher.

Water consumption cited in the study may be somewhat overstated for a facility with excellent water conditions and/or very well managed water treatment. However, in the author's opinion, the water consumption assumptions are more likely to understate the usage for a typical refrigerated warehouse system, because condenser water usage is often not metered or managed vs. expected usage for the actual heat rejection.

Capital cost and payback

The additional costs for air-cooled condensing include

- Air-cooled condenser cost premium over evaporative condensers,
- Cost of increased design pressures for vessels and piping,
- Increased compressor motor cost for higher peak operating pressures,
- Additional condenser piping, and
- Structural support for condensers (potentially lighter weight but larger area).

Detailed equipment selection and installation pricing was not undertaken as part of this paper, because costs vary greatly based on design conditions and site-specific factors. Also, the relevant opportunities for ammonia in lieu of R-507 are often not central systems. Table 12 presents a comparison of condenser costs only, for air-cooled ammonia vs. evap-cooled ammonia, including simple payback based on the estimated operating cost savings.

	Added Cost	Annual Savings	Payback, Years
Chicago	$160,877	$ 21,796	7.4
Denver	$146,214	$ 3,051	47.9
Portland	$137,567	$ 32,460	4.2
Dallas	$151,235	$ 13,859	10.9
Miami	$122,241	$ 11,736	10.4
Salinas	$145,993	$ 4,324	33.8
Atlanta	$153,663	$ 72,284	2.1
Charlotte	$153,663	$ 26,166	5.9
Fresno	$171,722	$(31,943)	N/A
Phoenix	$186,881	$ (4,851)	N/A
Minneapolis	$149,059	$ 28,228	5.3

Table 12. Annual savings for ammonia air cooled vs. ammonia evap cooled

Several locations have reasonable simple paybacks based solely on energy and water savings. The actual first cost difference for a particular facility can, of course, be determined by designing and pricing both evap- and air-cooled systems. Life cycle financial analysis may often be necessary and justified to assess future costs, such as differences in evap- vs. air-cooled condenser replacement cycles, in addition to water conservation objectives. Financial incentives may also be available in some cities for water-saving technologies.

Conclusions

The use of air-cooled condensers for ammonia systems is potentially attractive in many areas of the country, both in lieu of evap-cooled ammonia systems and as a cost-effective and more environmentally friendly option to both evap- and air-cooled HFC systems. Electric energy cost is equal or greater for air-cooled condensing in all areas evaluated, but when water costs are considered, the net operating cost is lower in all but two U.S. locations considered in this paper.

Higher electric operating costs with air-cooled condensing reflect the higher electric rates concurrent with high dry bulb temperatures, when the comparative advantage of evaporative condensing is greatest. No refrigeration load shifting was included in the analysis and may comprise a potential advantage for air-cooled condensing due to the higher daily range of dry bulb temperature compared with wet bulb temperature.

Water usage was calculated based on heat rejection from the hourly simulation and typical water bleed rates. Actual water usage may be lower with better water treatment or could be substantially higher if not carefully controlled.

Given the wide range of water costs and utility rates (and rate shapes in peak periods), site-specific analysis is necessary to identify operating costs of evap-cooled and air-cooled condenser options accurately. For both air-cooled and evap-cooled condensers, the catalog capacity ratings were derated by more than 30% to develop the average capacities for the hourly simulation. This is a significant assumption for which limited field testing exists. Future work is needed for both evap-cooled and air-cooled condensers to evaluate installed average performance to achieve more accurate annualized analysis and establish performance expectations.

The study employed similar system designs and assumptions for all analysis cases to obtain the most accurate comparison of refrigerants and condensing means. However, most HFC systems are smaller built-up systems, skid-mounted multiplex systems, and individual split-system condensing units, with smaller systems mostly using air-cooled condensers. These HFC systems are generally far less efficient than the HFC systems in this study due to smaller compressors, lack of subcooling, and minimal energy efficiency features. Accordingly, the savings for an energy-efficient air-cooled ammonia system would be significantly higher.

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Appendix A. Simulation assumptions

<table>
<tr><td colspan="2">Weather</td></tr>
<tr><td>Design WBT, DBT</td><td>
<table>
<tr><th>City</th><th>ASHRAE 0.4% WBT °F</th><th>ASHRAE 0.4% DBT °F</th></tr>
<tr><td>Dallas</td><td>79</td><td>100</td></tr>
<tr><td>Chicago</td><td>78</td><td>92</td></tr>
<tr><td>Denver</td><td>65</td><td>94</td></tr>
<tr><td>Miami</td><td>80</td><td>92</td></tr>
<tr><td>Salinas</td><td>65</td><td>83</td></tr>
<tr><td>Portland</td><td>71</td><td>92</td></tr>
<tr><td>Atlanta</td><td>77</td><td>94</td></tr>
<tr><td>Charlotte</td><td>77</td><td>94</td></tr>
<tr><td>Fresno</td><td>74</td><td>104</td></tr>
<tr><td>Phoenix</td><td>76</td><td>110</td></tr>
<tr><td>Minneapolis</td><td>77</td><td>91</td></tr>
</table>
Source: Data from ASHRAE (2013b).</td></tr>
<tr><td colspan="2">Compressor Information</td></tr>
<tr><td>Refrigerant</td><td>Ammonia and R-507</td></tr>
<tr><td>Suction group design SST</td><td>Low Temp (LT) System: -23°F (-31°C)
High Temp (HT) System: 22°F (-6°C)</td></tr>
<tr><td>Design SCT</td><td>
<table>
<tr><th>City</th><th>Evap Condenser</th><th>Air Cooled Condenser</th></tr>
<tr><td>Dallas</td><td>97</td><td>115</td></tr>
<tr><td>Chicago</td><td>96</td><td>107</td></tr>
<tr><td>Denver</td><td>85</td><td>109</td></tr>
<tr><td>Miami</td><td>98</td><td>107</td></tr>
<tr><td>Salinas</td><td>85</td><td>98</td></tr>
<tr><td>Portland</td><td>91</td><td>107</td></tr>
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<tr><td>Fresno</td><td>94</td><td>119</td></tr>
<tr><td>Phoenix</td><td>96</td><td>125</td></tr>
<tr><td>Minneapolis</td><td>96</td><td>106</td></tr>
</table>
1°F (0.56°C) loss is assumed between the SCT at the condenser and the saturated discharge temperature at the compressors.</td></tr>
<tr><td>Compressor description</td><td><u>LT System</u>
Serves freezer area. Ammonia screw compressors (2) with slide-valve unloading.</td></tr>
</table>

	HT System Serves cooler and dock areas. Ammonia screw compressors (2) with slide-valve unloading
Compressor capacity, power, nominal motor HP, and motor efficiency at design conditions	LT System Frick RXF-101: 72.2 TR (tons of refrigeration), 208.8 BHP (brake horsepower) at -23°F (-31°C) SST and 100°F (38°C) SCT, 250 nominal hp, 94.5% efficient motor HT System Frick RXF-50: 105.8 TR, 137.3 BHP at 22°F (-6°C) SST and 100°F (38°C) SCT, 150 nominal hp, 93.6% efficient motor The actual compressor capacities were scaled for each city so that the compressors meet the design cooling load.
Suction group SST control strategy	LT System: -23°F (-31°C) fixed SST setpoint HT System: 22°F (-6°C) fixed SST setpoint 1°F (0.56°C) throttling range
Lead compressor unloading strategy	Slide valve unloading
Oil cooling type	Thermosyphon
Superheat	10°F (5.6°C) nonproductive superheat for compressor mass flow
Liquid subcooling for compressor ratings	No condenser subcooling (The low temp liquid is flash cooled by the high-temp system, but compressor performance modeling uses mass flow not cooling capacity.)
Recirculator pumps	Ammonia: LT System: 2hp, 86.5% efficient, assumed 90% loaded HT System: 3hp, 89.5% efficient, assumed 90% loaded R-507: LT System: 7.5 hp, 91% efficient, assumed 90% loaded HT System: 15 hp, 93% efficient, assumed 90% loaded
Evaporator Coil Information	
Air unit fan operation	All zones Fans run 100% of the time, except for defrost. Variable speed control, 65% minimum speed, 2 hours/day forced at 100% speed to reflect typical control response variations vs. hourly simulation.
Defrost assumptions	Cooler: 2 30-minute hot-gas defrosts/day Dock: 2 30-minute off-cycle defrosts/day Freezer: 2 30-minute hot-gas defrosts/day
Air unit quantity	Cooler: 6 Dock: 6 Freezer: 6

<table>
<tr><td>Air unit capacity (per unit)</td><td>Cooler: 190 MBtu/hr at 10°F (5.6°C) TD
Dock: 145 MBtu/hr at 10°F (5.6°C) TD
Freezer: 227 MBtu/hr at 10°F (5.6°C) TD
For energy analysis these capacities are derated by 10% from catalog ratings to account for field effects and other factors.</td></tr>
<tr><td>Design saturated evaporator temperature:</td><td>Cooler: 25°F (-4°C)
Dock: 30°F (-1°C)
Freezer: -20°F (-29°C)</td></tr>
<tr><td>Air flow rate (per unit)</td><td>Cooler: 36,200 CFM (cubic feet per minute)
Dock: 27,700 CFM
Freezer: 43,200 CFM</td></tr>
<tr><td>Fan power</td><td>Cooler: 5.59 kW
Dock: 4.27 kW
Freezer: 6.67 kW
Based on specific efficiency of 34.0 Btu/h/kW at 10°F (5.6°C) TD between saturated evaporator temperature and space temperature</td></tr>
<tr><td colspan="2">Condenser Information</td></tr>
<tr><td>Condenser type</td><td>Evaporative/air cooled</td></tr>
<tr><td>Design temperature difference</td><td>
<table>
<tr><th>City</th><th>Evap Condenser</th><th>Air Cooled Condenser</th></tr>
<tr><td>Dallas</td><td>18</td><td>15</td></tr>
<tr><td>Chicago</td><td>18</td><td>15</td></tr>
<tr><td>Denver</td><td>20</td><td>15</td></tr>
<tr><td>Miami</td><td>18</td><td>15</td></tr>
<tr><td>Salinas</td><td>20</td><td>15</td></tr>
<tr><td>Portland</td><td>20</td><td>15</td></tr>
<tr><td>Atlanta</td><td>19</td><td>15</td></tr>
<tr><td>Charlotte</td><td>19</td><td>15</td></tr>
<tr><td>Fresno</td><td>20</td><td>15</td></tr>
<tr><td>Phoenix</td><td>20</td><td>15</td></tr>
<tr><td>Minneapolis</td><td>19</td><td>15</td></tr>
</table>
Fixed TD of 15°F (8.3°C) was used for air-cooled condensers.
TD for evaporative condensers was determined as follows:
Design WBT ≤ 76°F (24°C), TD = 20°F (11.1°C)
76°F (24°C) < design WBT < 78°F (26°C), TD = 19°F (10.6°C)
Design WBT ≥ 78°F (26°C), TD = 18°F (10.0°C)</td></tr>
</table>

<table>
<tr>
<td>THR capacity at design conditions</td>
<td>
<table>
<tr><th>City</th><th>NH3 Evap Cooled</th><th>NH3 Air Cooled</th><th>507 Evap Cooled</th><th>507 Air Cooled</th></tr>
<tr><td>Dallas</td><td>5,800</td><td>6,300</td><td>6,250</td><td>6,300</td></tr>
<tr><td>Chicago</td><td>5,675</td><td>5,962</td><td>6,106</td><td>5,962</td></tr>
<tr><td>Denver</td><td>5,064</td><td>5,634</td><td>5,377</td><td>5,634</td></tr>
<tr><td>Miami</td><td>5,783</td><td>6,022</td><td>6,242</td><td>6,022</td></tr>
<tr><td>Salinas</td><td>4,966</td><td>5,251</td><td>5,273</td><td>5,251</td></tr>
<tr><td>Portland</td><td>5,341</td><td>5,736</td><td>5,710</td><td>5,736</td></tr>
<tr><td>Atlanta</td><td>5,647</td><td>5,990</td><td>6,077</td><td>5,990</td></tr>
<tr><td>Charlotte</td><td>5,647</td><td>5,990</td><td>6,077</td><td>5,990</td></tr>
<tr><td>Fresno</td><td>5,599</td><td>6,299</td><td>6,009</td><td>6,299</td></tr>
<tr><td>Phoenix</td><td>5,759</td><td>6,640</td><td>6,198</td><td>6,640</td></tr>
<tr><td>Minneapolis</td><td>5,619</td><td>5,877</td><td>6,047</td><td>5,877</td></tr>
</table>
For energy analysis these capacities were derated by 31% for evap-cooled condensers and by 34% for air-cooled condensers to account for fouling, non-steady-state operation, field installation effects, and other factors as noted in the main body of the paper.
</td>
</tr>
<tr>
<td>Pump power and efficiency (for evaporative condenser)</td>
<td>5 hp, assumed 89.5% efficient, 4.17 kW—for all cities
Pump runs continuously</td>
</tr>
<tr>
<td>Fan power</td>
<td>
<table>
<tr><th colspan="2">NH3</th><th>Dallas</th><th>Chicago</th><th>Denver</th><th>Miami</th><th>Salinas</th></tr>
<tr><td rowspan="3">Evap Cooled</td><td>Fan, kW</td><td>28.7</td><td>28.7</td><td>27.0</td><td>28.1</td><td>19.9</td></tr>
<tr><td>Pump, kW</td><td>4.2</td><td>4.2</td><td>4.2</td><td>4.2</td><td>4.2</td></tr>
<tr><td>Total</td><td>32.9</td><td>32.9</td><td>31.2</td><td>32.3</td><td>24.1</td></tr>
<tr><td>Air Cooled</td><td>Fan, kW</td><td>46.7</td><td>44.2</td><td>41.7</td><td>44.6</td><td>38.9</td></tr>
</table>
<table>
<tr><th colspan="2">NH3</th><th>Portland</th><th>Atlanta</th><th>Charlotte</th><th>Fresno</th><th>Phoenix</th><th>Mnpolis</th></tr>
<tr><td rowspan="3">Evap Cooled</td><td>Fan, kW</td><td>26.1</td><td>28.5</td><td>28.5</td><td>19.6</td><td>26.3</td><td>28.3</td></tr>
<tr><td>Pump, kW</td><td>4.2</td><td>4.2</td><td>4.2</td><td>4.2</td><td>4.2</td><td>4.2</td></tr>
<tr><td>Total</td><td>30.3</td><td>32.7</td><td>32.7</td><td>23.8</td><td>30.5</td><td>32.5</td></tr>
<tr><td>Air Cooled</td><td>Fan, kW</td><td>42.5</td><td>44.4</td><td>44.4</td><td>46.7</td><td>49.2</td><td>43.5</td></tr>
</table>
</td>
</tr>
</table>

<table>
<tr><td></td><td>
<table>
<tr><th colspan="2">R-507</th><th>Dallas</th><th>Chicago</th><th>Denver</th><th>Miami</th><th>Salinas</th></tr>
<tr><td rowspan="3">Evap Cooled</td><td>Fan, kW</td><td>31.2</td><td>30.8</td><td>28.8</td><td>30.7</td><td>21.2</td></tr>
<tr><td>Pump, kW</td><td>4.2</td><td>4.2</td><td>4.2</td><td>4.2</td><td>4.2</td></tr>
<tr><td>Total</td><td>35.4</td><td>35.0</td><td>33.0</td><td>34.9</td><td>25.4</td></tr>
<tr><td>Air Cooled</td><td>Fan, kW</td><td>52.3</td><td>48.5</td><td>46.1</td><td>49.0</td><td>42.1</td></tr>
</table>
<table>
<tr><th colspan="2">R-507</th><th>Portland</th><th>Atlanta</th><th>Charlotte</th><th>Fresno</th><th>Phoenix</th><th>Mnpolis</th></tr>
<tr><td rowspan="3">Evap Cooled</td><td>Fan, kW</td><td>28.1</td><td>29.3</td><td>29.3</td><td>21.3</td><td>28.4</td><td>29.1</td></tr>
<tr><td>Pump, kW</td><td>4.2</td><td>4.2</td><td>4.2</td><td>4.2</td><td>4.2</td><td>4.2</td></tr>
<tr><td>Total</td><td>32.3</td><td>33.5</td><td>33.5</td><td>25.5</td><td>32.6</td><td>33.3</td></tr>
<tr><td>Air Cooled</td><td>Fan, kW</td><td>46.7</td><td>49.0</td><td>49.0</td><td>52.9</td><td>56.9</td><td>47.7</td></tr>
</table>
Evaporative condenser power is based on a specific efficiency of 275 Btu/h/W at 100°F (38°C) SCT, 70°F (21°C) WBT, for all locations except Salinas and Fresno, which use the CEC (2013) requirement of 350 Btu/h/W, and assumes 5 hp spray pumps.
Air-cooled condenser power is based on specific efficiency of 90 Btu/h/W at the specific efficiency rating point of 10°F (5.6°C) TD.
</td></tr>
<tr><td>Condenser fan control</td><td>
60°F (16°C) minimum SCT setpoint
Ambient temperature following SCT control (wet bulb reset for evaporative condenser, dry bulb reset for air-cooled condenser)
Variable-speed fan control
1°F (0.56°C) throttling range
Optimum control TD:
<table>
<tr><th>City</th><th>NH3 Evap Cooled</th><th>NH3 Air Cooled</th><th>507 Evap Cooled</th><th>507 Air Cooled</th></tr>
<tr><td>Dallas</td><td>16</td><td>12</td><td>16</td><td>13</td></tr>
<tr><td>Chicago</td><td>16</td><td>12</td><td>17</td><td>12</td></tr>
<tr><td>Denver</td><td>17</td><td>12</td><td>18</td><td>12</td></tr>
<tr><td>Miami</td><td>15</td><td>13</td><td>16</td><td>13</td></tr>
<tr><td>Salinas</td><td>17</td><td>13</td><td>17</td><td>13</td></tr>
<tr><td>Portland</td><td>17</td><td>12</td><td>18</td><td>12</td></tr>
<tr><td>Atlanta</td><td>16</td><td>12</td><td>17</td><td>12</td></tr>
<tr><td>Charlotte</td><td>16</td><td>12</td><td>17</td><td>12</td></tr>
<tr><td>Fresno</td><td>17</td><td>12</td><td>17</td><td>12</td></tr>
<tr><td>Phoenix</td><td>17</td><td>12</td><td>18</td><td>12</td></tr>
<tr><td>Minneapolis</td><td>17</td><td>12</td><td>17</td><td>12</td></tr>
</table>
The control TD is assumed to stay fixed, i.e., no enhanced logic to change the TD with WBT in the case of evap condensing.
</td></tr>
</table>

Load Information	
Facility size	Freezer area: 40,000 ft^2 Cooler area: 40,000 ft^2 Dock area: 12,000 ft^2 Total area: 92,000 ft^2
Ceiling heights	All areas: 30 ft
Temperature setpoints	Freezer: -10°F (-23°C) Cooler: 35°F (2°C) Dock: 40°F (4°C)
Load profiles	Internal loads are product load, lights, infiltration, people, forklifts/pallet lifts, and equipment.
Infiltration, leakage open, closed, etc.	Cooler: 2 10 ft × 10 ft doors from cooler to dock. Freezer: 2 10 ft × 10 ft doors from freezer to dock. Dock: 20 10 ft × 10 ft dock doors. Assumed 200 CFM design infiltration per dock door, subject to infiltration schedule. Interzonal doors assumed to open 20 times per hour, 5 seconds per opening. Freezer doors are assumed to have air curtains or strip curtains with 50% effectiveness. Subject to hourly operations schedule with normal operations for two shifts and reduced operation for third shift and weekends.
Product loads	Freezer: 41.7 MBH (400,000 lb/day product load, from -5°F (-20.6°C) to -10°F (-23.3°C), with specific heat of 0.50) Cooler: 226.0 MBH (400,000 lb/day product load, from 45°F (7.2°C) to 40°F (4.4°C), with specific heat of 0.65, plus 750 tons of respiring product; heat of respiration: 5,500 Btu/h/ton of product per 24 hours) Dock: 0 Btu/h
People loads	36 people maximum Assumed people heat gain: 580 Btu/h sensible, 870 Btu/h latent, 1,450 Btu/h total Subject to hourly operations schedule
Forklifts	16 forklifts, 16 pallet lifts 20 MBtu/h/forklift, 10 MBtu/h/pallet lift Subject to hourly operations schedule
Facility Envelope Insulation	
Climate data	TMY3 weather files for each city
Azimuth	0°
Roof construction	Freezer Construction: Built-up roof, R-36 insulation Inside film resistance: 0.90 hr-ft^2-°F/Btu

	Absorptance: 0.45 (thermal emittance of 0.55) Cooler Construction: Built-up roof, R-28 insulation Inside film resistance: 0.90 hr-ft^2-°F/Btu Absorptance: 0.45 (thermal emittance of 0.55) Dock Construction: Built-up roof, R-28 insulation Inside film resistance: 0.90 hr-ft^2-°F/Btu Absorptance: 0.45 (thermal emittance of 0.55)
Wall construction	Freezer R-36 insulation Cooler R-28 insulation Dock R-28 insulation Interzonal wall R-26 insulation
Floor construction	Freezer 8 in. concrete slab, R-36 insulation Cooler 8 in. concrete slab (no insulation) Dock 8 in. concrete slab (no insulation)
Hours of operation	7 days/week, 24 hours per day with normal activity during two shifts and reduced activity during one shift and weekends
Lighting	
Lighting power density	All areas: 0.7 W/ft^2

Notes: