WHITE PAPER

ANSI/AHRI Standard 420-2008 for Fan and Coil Evaporators - Benefits and Costs

Abstract

Fan and coil evaporators as used in the industrial refrigeration industry can be certified for performance per ANSI/AHRI Standard 420-2008, Performance Rating of Forced-Circulation Free-Delivery Unit Coolers for Refrigeration. This paper will explain how the standard works and illustrate the costs related to evaporators that may not meet the performance criteria of the standard.

The Standard

ANSI/AHRI standards benefit the HVAC and Refrigeration industry by ensuring the performance and efficiency of manufactured equipment is accurate and reliable. These standards are produced by the Air-Conditioning, Heating and Refrigeration Institute and its member companies under ANSI guidelines. There are numerous standards for many different types of HVAC equipment. For instance Standard 410 is for fin and tube air heating or cooling coils only without an attached fan and standard 510 is for positive displacement ammonia compressors. AHRI Standard 490 and Cooling Technology Institute (CTI) Standard 106 both cover aspects of evaporative refrigerant condensers which also can be thermally certified under CTI Standard 201, Thermal Performance Certification of Evaporative Heat Rejection Equipment. The standards typically define testing criteria and the associated instrumentation tolerances for various operating conditions and set the minimum and maximum acceptable capacity and power deviations. When a manufacturer states that a product line is per a certain ANSI/AHRI or CTI standard, then that manufacturer is subject to having a random production unit tested by an independent laboratory to confirm the product performs per its factory rating.

Standard 420-2008 applies to refrigerant fed evaporators, of either the direct expansion or liquid overfeed type. There are five standard rating conditions as listed below:

Condition #1 Wet Coil 50 °F 75% RH air on with a 35 °F refrigerant evaporating temperature
Condition #2 Dry Coil 50 °F <45% RH air on with a 35 °F refrigerant evaporating temperature
Condition #3 Dry Coil 35 °F <50% RH air on with a 25 °F refrigerant evaporating temperature
Condition #4 Dry Coil 10 °F <46% RH air on with a 0 °F refrigerant evaporating temperature
Condition #5 Dry Coil -10 °F <43% RH air on with a -20 °F refrigerant evaporating temperature

When an evaporator is marketed, it will have two published ratings, one of the 5 standard rating conditions above and the application rating, reflecting the actual operating condition, if different than the closest standard rating condition.

To pass the performance test, an evaporator must meet the applicable standard condition rating per section 5.3, which states:

> “Tolerances. To comply with this standard, any representative production unit selected at random, when tested at the Standard Rating Conditions, shall have a Gross Total Cooling Effect not less than 95% of its published Standard Rating and not exceed 105% of its Rated Power”.

Translating Standard 420 to Everyday Use

All evaporators with fans produce a gross cooling or refrigerating effect that is dependent upon many variables such as; fin material, size and shape, tube material, size and shape, fin spacing, tube spacing, refrigerant feed method and LMTD. The aforementioned variables all apply to the fin and tube coil assembly only. Unit coolers or evaporators are usually provided by the manufacturer with a mounted fan assembly(s) to draw or blow air through the coil assembly. The choice of fan and motor will determine the air quantity and velocity through the coil assembly. Higher air velocities produce greater capacities for a given coil assembly. However, this comes at the expense of exponentially greater fan power consumption.

Standard 420 measures gross cooling effect which is equal to all of the heat that the coil assembly passes into the refrigerant. It also measures the power consumed by the fan(s). The gross cooling effect less the fan power is the net cooling effect, which is the portion of that evaporator's capacity available for removing heat from the refrigerated space.

For this freezer study, the calculated design load is 280 Tons which includes a 10% safety factor per the attached load estimate summary. This load is removed from the freezer through eight 35.1 Ton evaporators. They are rated at a -10 °F room (return air) temperature and -20 °F recirculated ammonia feed. (Lines 1-20) The refrigeration system is a single staged economized screw compressor with a low fan power evaporative condenser.

Accurate Sizing and Selection

To properly and accurately select and size an evaporator, the total load of the refrigerated space must be determined. Historically, there are many "rules of thumb" that have been used for this purpose. In most instances, these "rules of thumb" and/or the safety factors that are applied to them are very conservative and result in the purchase and installation of excess evaporator capacity. For instance, a range of 250 ft^2/Ton to 450 ft^2/Ton is representative of -10 F Freezer load "rule of thumb" estimates. If the actual heat load of the freezer was 450 ft^2/Ton, then selecting equipment at 250 ft^2/ton would result in a safety factor of 80% and grossly oversized refrigeration equipment.

The tools to accurately calculate the load in a refrigerated space have been available for over 40 years. During that period many refinements to their accuracy have been developed and made available by research and experimentation. Long hand calculations are time consuming, hence the continued use of "rules of thumb". With the availability of computer programs and spreadsheets, these calculations can be done quickly and a number of "what if" scenarios can be investigated. Greater accuracy in load estimating will result in more appropriately sized refrigeration equipment. **This is not limited to evaporators, but also compressors and condensers.**

When the equipment is sized to more precisely calculated loads, having a certified rating to ensure full performance becomes more critical to deliver the required capacity.

As shown on the attached load summary sheet, the four main components of heat load into a refrigerated space are;

1) Thermal transmission through the walls, roof and floor (27%)
2) Infiltration of warmer moist outside air (25%)
3) Heat from incoming product that is warmer than the refrigerated space (21%)
4) Personnel, electrical and machinery loads within the space (27%)

By considering the certainty of each of these four components, a good judgment for the safety factor can be applied. For instance; Component 1 at 27% ± 7%, Component 2 at 25% ± 20%, Component 3 at 21% ± 10% and Component 4 at 27% ± 3% results in the 10% safety factor used in this study's load estimate.

The fourth component is directly influenced by the efficiency of the evaporator selection, which is discussed in the Evaporator Efficiency section later in this paper.

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Additional heat transfer surface in any refrigeration surface is always a positive thing. However, it is imperative additional capacity in the form of more fan power is not confused with additional surface with the same or less fan power.

Tables A, A1 & B reflect the effect substandard evaporator performance has on a generic cold storage warehouse with a -10 °F design operating temperature. The effect on operating expense from a performance shortfall as shown by these tables will occur regardless of the safety factor applied to the load calculation. Oversizing the equipment may satisfy temperatures at peak loads, but the operating cost of not meeting stated performance remains.

To illustrate the effects of less capacity or increased performance, the system at 100% design load, 280 TR, will operate with each evaporator defrosting once per day, leaving 23.5 hours in a day to remove heat from the refrigerated space (Line 30). The compressor and condenser are sized to match the eight evaporators. When the actual system load is less than the design load, the entire refrigeration system will cycle off once the room temperature is satisfied (Lines 37, 43 & 49). The operating year is divided into various part load segments; 100% load for 10% of the time, 80% load for 35% of the time, 60% load for 40% of the time and 40% load for 15% of the time (Lines 33, 40, 46 & 52).

When the net cooling capacity of the evaporator is less than 100% design, either the room temperature increases to widen the TD and compensate for the capacity shortfall (Line 28) or the required run hours must increase to remove the same amount of heat during part load conditions (Lines 37, 43 & 49). It should be noted that at all times with 100% load, there are not enough hours in a day for an underperforming evaporator to achieve the desired room temperature. The same is true at 80% load when an evaporator is performing at 80% or less than the stated capacity. Consequently the room temperature rises to -9.6 °F, -8.8 °F,or -7.8 °F as shown in lines 28 and 35, respectively.

Effect of Reduced Capacity on Room Temperature or Run Hours to Maintain Room Temp (from Table A)

Line	% Evaporator Rated Capacity	100%	95%	90%	85%	80%	75%	70%
A.19	Capacity, BTU/Hr	421,200	400,140	379,080	358,020	336,960	315,900	294,840
A.23	Fan Motor Heat, Btu/hr	52,068	52,068	52,068	52,068	52,068	52,068	52,068
A.24	Net Capacity, Btu/hr	369,132	348,072	327,012	305,952	284,892	263,832	242,772
A.27	Total Net Capacity, Tons	246.1	232.0	218.0	204.0	189.9	175.9	161.8
A.28	**Revised Room Temp °F**	**-10**	**-9.4**	**-8.7**	**-7.9**	**-7.0**	**-6.0**	**-4.8**
A.30	100% Load Run Hours		23.5	23.5	23.5	23.5	23.5	23.5
	- OR -							
A.37	80% Load Run Hours	18.8	19.9	21.2	22.7	24.4*	26.3*	28.6*
A.36						*23.5 Max		
A.35	**Revised Room Temp °F**					**-9.6**	**-8.8**	**-7.8**
A.43	60% Load Run Hours	14.1	15.0	15.9	17.0	18.3	19.7	21.4
A.49	40% Load Run Hours	9.4	10.0	10.6	11.3	12.2	13.2	14.3

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The converse of this would be lowering the compressor suction to widen the TD and revise the capacity shortfall with a proportionally greater TD to achieve the desired room temperature. Table A1 reflects that and will be discussed later.

Effect of Reduced Capacity on Suction Temp Needed to Maintain Room Temperature (from Table A1)

Line	% Evaporator Rated Capacity	100%	95%	90%	85%	80%	75%	70%
A1.28	**Revised Sat. Evap Temp °F**	**-20**	**-20.6**	**-21.3**	**-22.1**	**-23.0**	**-24.0**	**-25.2**

When the fan power is greater than 100% design (Table B), the net evaporator capacity is reduced with the same impact as described in the above paragraph.

Effect of Fan Power on Room Temperature (from Table B)

Line	% Fan Power	100%	105%	110%	115%	120%	125%	130%	135%	140%
B.18	Total Gross Capacity, Tons	280.8								
B.19	Total Gross Capacity, BTU/Hr	421,200								
B.23	Fan Motor Heat, Btu/hr	52,068	54,671	57,274	59,878	62,481.2	65,085	67,688	70,291	72,895
B.24	Net Capacity, Btu/hr	369,132	366,529	363,926	361,322	358,719	356,115	353,512	350,909	348,305
B.27	Total Net Capacity, Tons	246.1	244.4	242.6	240.9	239.1	237.4	235.7	233.9	232.2
B.28	**Revised Room Temp °F**	**-10**	**-9.9**	**-9.9**	**-9.8**	**-9.7**	**-9.6**	**-9.6**	**-9.5**	**-9.4**

During part load operation it is typical that the saturated condensing temperature will reduce as well. 95 °F is used for 100% load, 90 °F for 80% load, 80 °F for 60% load and 75 °F compressor minimum for 40% load. This is reflected in the kW / ton values, which use a major compressor manufacturer's ratings with a 93% efficiency for the compressor drive motor. The kW/ton values also include a 3 HP recirculator pump, 2-5 HP condenser fans and a 5 HP spray pump, all at 90% efficiency (Lines 32, 39, 45 & 51).

Total net evaporator capacity and the operating hours are combined for a ton-hour value for each load segment (Lines 31, 38, 44 & 50). The kW/ton of the refrigerant pump, compressor and condenser is added to the evaporator kW/ton and multiplied by the ton-hours for a kW-hr value in each load segment (Lines 34, 41, 47 & 53).

These daily kW-hr subtotals are totaled and multiplied by 365 days for an annual kW-hr energy consumption (Line 57). An energy cost of $0.10 /kW-hr monetizes the annual cost penalty (Line 60) of an evaporator choice that does not meet the performance standards of ANSI/AHRI Standard 420.

Table A1 reflects the use of a greater TD to regain the capacity shortfall. When the compressor suction pressure and corresponding evaporating temperature is lowered, the kW/Ton of the compressor increases. This modified table then reflects 23.5 daily run hours at the lower suction pressures whenever the room temperature would have risen above -10 °F. The difference between table A and A1 is reflective of the true cost to maintain a constant -10 °F room temperature while compensating for evaporators with a performance shortfall.

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Operating Cost Penalty to Maintain Room Temperature by Increased Fan Run Hours (from Table A)

Line	% Evaporator Rated Capacity	100%	95%	90%	85%	80%	75%	70%
A.57	Annual kW- hr	3,765,995	3,808,447	3,860,255	3,923,515	3,938,563	3,901,539	3,877,343
A.58	Annual Difference, kW-hr		42,452	94,260	157,520	172,568	135,544	111,348
	Annual Operating $	$376,600						
A.60	**Annual $ Increment**		**$4,245**	**$9,426**	**$15,752**	**$17,257**	**$13,554**	**$11,135**

Operating Cost Penalty to Maintain Room Temperature by Reduced Suction Temperature (from Table A1)

Line	% Evaporator Rated Capacity	100%	95%	90%	85%	80%	75%	70%
A1.57	Annual kW- hr	3,765,995	3,846,336	3,939,257	4,044,616	4,123,197	4,264,938	4,455,846
A1.58	Annual Difference, kW-hr		80,341	173,262	278,621	357,202	498,943	689,851
	Annual Operating $	$376,600						
A1.60	**Annual $ Increment**		**$8,034**	**$17,326**	**$27,862**	**$35,720**	**$49,894**	**$68,985**

The increased annual costs for not performing as published will be ongoing for the life of the facility resulting in cumulative lost profits far in excess of any cost premium for evaporators that meet the criteria of ANSI/AHRI Standard 420. The evaporators used in this study have a contractor cost of less than $200,000.00. A 10% shortfall in performance over 20 Years is almost equal to the original cost of the evaporators. If the suction pressure is lowered to maintain -10 °F at all times, then the resulting higher annual operating cost will be equivalent to the evaporator first cost of $200,000 in 12 years.

Considerations to Performance Shortfall

If there is no margin of safety in the heat load calculation and the evaporators are sized to match the load, they will operate continuously except when in defrost. When the load diminishes due to cooler ambient conditions, less infiltration or a diminished product load; the evaporators can and should cycle off. This stops the flow of fan energy into the room for the most efficient operation. If continuous airflow or precise temperature regulation is required, then VFD's should be considered. If there is a performance shortfall at design load, then the room temperature will rise until the increased TD offsets the missing capacity or the suction pressure has to be lowered accordingly. At part load, more run hours will be required.

Evaporator Efficiency

The other aspect of Standard 420 ratings, as shown and discussed above, is the fan power. Fan energy has a two-fold impact. In addition to reducing the efficiency of the evaporator, extra fan energy must be removed by the compressors and condensers with their additional specific energy use resulting in higher operating costs over the life of the system. Ultimately, refrigeration system designers, contractors, and end-users need to remember, it is capacity that is being purchased and not pounds of steel.

Table C is a 22 square foot face area coil assembly that has a gross rating of 7.3 Tons with a 1.5 HP fan and the same coil equipped with a 5 HP fan will have a gross rating of 9.4 Tons due to a much higher face velocity, 982 ft/min vs. 650 ft/min. Assuming a fan motor efficiency of 85% converts 1 HP to 2,994 Btu/Hr. The first instance is 4,670.8 Btu/hr or approximately 3/8 Ton, the latter is 17,964.7 Btu/hr or approximately 1.5 Tons. This leaves a net capacity, gross cooling effect less fan motor heat of 6.9 Tons for the 1.5 HP selection and 7.9 Tons for the 5 HP selection. The 29% gain in gross capacity is reduced to 14% when net capacity is compared. The efficiency of the low power selection, Btu/hr per BHP is over triple that of the high powered selection. This is not insignificant, in the cold storage study of Tables A & B above, the evaporator fan energy is approximately 25% of the total refrigeration system energy including the compressor and condenser.

Effect of Fan Motor Heat on Net Cooling Effect (from Table C)

	Efficient Selection	Inefficient Selection
Fan HP, Nominal	1.5	5.0
Fan HP, Rated	1.56	6.00
Fan RPM	870.0	1160.0
Face Velocity, ft/min	650	982
Fan Heat, Btu/hr	4,670.8	17,964.7
Gross Capacity, BTU/Hr	87,360.0	112,440.0
Gross capacity, Tons	7.3	9.4
Percent gain	-	29%
Net Capacity, Btu/Hr	82,689.2	94,475.3
Net capacity, Tons	6.9	7.9
Percent gain	-	14%
NET BTUH/BHP	53,005.9	15,745.9
Percent gain	337%	-
Relative $/Net Btuh	1.00	0.90

System Performance Standards

California has enacted system performance standards under Title 24. ASHRAE is also addressing this with Standard 90.1. Both of these standards have noble goals of increasing the system efficiency of HVAC and refrigeration systems. An integral part of these standards is to specify minimum energy efficiency standards for the various components that make up these systems. The bedrock foundation of a system performance standard is accurate equipment ratings. Compliance with the applicable ANSI/AHRI or CTI testing and rating standard ensures that the stated performance is delivered.

Meeting these standards generally requires a cost premium to purchase heat transfer equipment with increased surface area and low fan, pump and or compression power. **A concerned facility owner can ensure that this extra expense will deliver the expected efficiency gains by insisting on certified equipment ratings.**

A Note about Rating Methods

ANSI/AHRI Standard 420 rates the evaporator with the measured air inlet temperature sampled at the entering face of the coil assembly and the saturated refrigerant temperature leaving the coil. The DT-1 rating method used most commonly in the U.S. also uses the air temperature entering the coil face (RA) less the saturated refrigerant temperature (SET) to establish the rating temperature difference (TD).

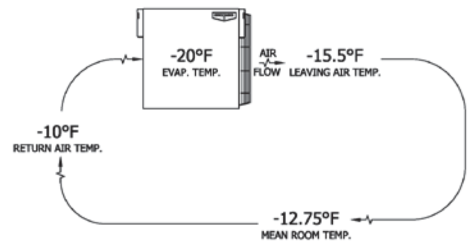
The DT-M rating method used by some European based manufacturers uses the median room temperature (MRA) less the saturated refrigerant temperature (SET) to establish the TD. The median room temperature is by definition the mid-point between the cold leaving air temperature and the warmest air entering the coil face.

If both rating methods use a 10 °F TD, then the air entering the coil is -10 °F for a DT-1 rated coil and -5.8 °F for a DT-M rated coil. **Confusing a DT-M rating with a DT-1 rating will immediately result in a 32% shortfall in expected capacity. Refer to the 75% and 70% capacity columns in table A or A1 to see the financial impact of this confusion.**

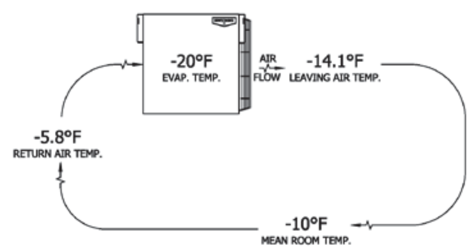
Be sure to compare DT-1 ratings to any ANSI/AHRI Standard 420 certified evaporator.

Refer to Table D (portion shown below). The same evaporator is rated two ways. The first rating column reflects the traditional DT-1 rating which corresponds with ANSI/AHRI 420. The second column demonstrates what happens when a DT-M rated coil is confused with a DT-1 rating, thus overstating the capacity by 47% or resulting in a 32% shortfall depending upon the base point of comparison.

The DT-M rating method overstates coil capacity to an even greater extent as air velocity is reduced or when coil rows or fin count are increased. DT-M rated capacities can be over 60% more than DT-1 rated capacities.



DT-1 - ANSI/AHRI 420



DT-M

Capacity Variation due to Ratings Method (from Table D)

	DT-1 - ANSI/AHRI 420	**DT-M**
Sat. Evap. Temp., F (SET)	-20	-20
Return Air Temp, F (RA)	-10	-5.8
Leaving Air Temp, F	-15.5	-14.1
Mean Room Air Temp., F (MRA)	-12.75	-10.0
DT1 - TD (RA-SET)	**10.0**	**14.2**
DTM - TD (MRA-SET)	7.25	10.0
Capacity, BTU/Hr	421,200	620,400
Gross Capacity, Tons	**35.1**	**51.7**

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Summary

This paper has demonstrated that is in a facility owner's best interest to purchase properly sized, efficient evaporators that produce maximum capacity with minimal fan power. It is also important to understand which evaporator rating method is used when comparing competitive bids.

This paper also clearly shows the significant financial impact that can result from evaporators that are mis-rated or simply do not perform as advertised. The purchase of evaporators with ANSI/AHRI 420 certified ratings per the DT-1 rating method, can prevent a costly mistake with many years of impact to the bottom line.

Author Biography

This technical paper was prepared by Jeff Welch, PE of Welch Engineering Corporation. A consulting firm offering fluid flow and performance analyses, teaching and expert witness services to the Industrial Refrigeration industry. Comments or questions can be sent to WelchEngCorp@gmail.com.