Hydronics Industry Alliance

Significant Concerns: AHRI/ANSI Standard 1230 Performance Rating of Variable Refrigerant Flow (VRF) Multi-Split Air-Conditioning and Heat Pump Equipment

Executive Summary

The Hydronics Industry Alliance – Commercial (HIA-C)[1] provides information in this white paper to identify areas of concern related to the existing, and proposed amended, provisions of AHRI/ANSI Standard 1230. This standard is currently in a negotiated revision process administered by the US Department of Energy.[2] It is the HIA-C's contention, supported by the information presented herein, that the test methodologies set forth in the current version of AHRI 1230 produce cooling energy efficiency ratings for VRF equipment which exceed actual operating efficiencies in the field by 25% to 60%, with an average of approximately 53% (see Appendix A). AHRI 1230 also produces heating energy efficiency ratings for VRF equipment which exceed actual operating efficiencies in the field by 30% to 60%, with an average of approximately 48% (Appendix H).

This is a critical area of concern given that ratings derived from the Standard 1230 test methodology are used by building owners, architects, electric utility energy efficiency program administrators, and HVAC design professionals to select amongst various HVAC system types. Currently there are no reliable data sources other than the AHRI ratings available to these key interest groups to assess equipment efficiencies. As a result, the AHRI ratings are used to compare efficiencies of various *systems* based on the rated IEER of various *equipment*, despite the original intent of AHRI to level the playing field between *different manufacturers of the same equipment*. This was not the original intention, however it is now the norm.

Because no other reliable data exists it is incumbent on AHRI to acknowledge that their ratings are used in this way in order to craft a standard which results in ratings that can be used to compare equipment across system types. Without such acknowledgement and appropriate remedial action, the perception that certain manufacturers have been allowed to manipulate the system will become more prevalent.

Discussion

Standard 1230 is essentially a standard for residential equipment, not a standard for VRF systems installed in commercial buildings. As a result, the standard allows VRF manufacturers to take advantage of the allowable test conditions to inflate the resulting IEER ratings of a system operating in a commercial building.

Cooling Ratings

Entering Refrigerant Temperature

The primary example of favorable test conditions is the allowable supply fluid (refrigerant) suction temperature to which the equipment may be tested. VRF is an outgrowth of direct

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[1] The Hydronics Industry Alliance – Commercial (HIA-C) is a non-profit alliance of hydronic equipment manufacturers and partners operating in North America. A list of our member companies can be found at http://www.hia-c.org . HIA-C member companies are involved in the sale of commercial-size hydronic-based heating and cooling products.

[2] Docket Number EERE-2018-BT-STD-0003: Energy Conservation Standards for Variable Refrigerant Flow Multi-Split Air Conditioners and Heat Pumps: https://www.regulations.gov/docketBrowser?rpp=25&so=DESC&sb=commentDueDate&po=0&D=EERE-2018-BT-STD-0003

expansion (DX) split systems, where a single condensing unit is matched to a single evaporator. AHRI Standard 210/240 governs the testing of this equipment. 210/240 does not define the supply, or suction temperature, of the fluid (refrigerant) to the evaporator, only the entering air temperature for the (indoor) evaporator or (outdoor) condensing unit.

Similarly, Standard 1230 for VRF equipment does not define the supply fluid (refrigerant) suction temperature to the evaporator, only the supply air temperature to the (indoor) evaporator or (outdoor) condensing unit. This despite that fact that VRF systems are now configured as a central system with multiple evaporators. For central systems the industry's conventional control sequence is to control the supply fluid temperature to the terminal units, not just the entering air temperatures. VRF systems are no exception.

As an example, AHRI Standard 550/590 requires chillers to be tested at a constant supply fluid (chilled water) temperature of 44°F. Since Standard 1230 allows manufacturers to choose their own supply or suction temperature of the fluid (refrigerant) for the rating test, manufacturers test to a suction temperature higher than what occurs in actual operating conditions. This is done by locking the compressor speed and maintaining full airflow. This results in a higher tested efficiency than what is experienced in the field.

Specifically, tests are started at a suction temperature of 55°F at 100% load, and increase to 70°F at 25% load[3]. In the real world, refrigerant systems are typically controlled to a constant suction temperature between 43°F to 45°F by an electronic expansion valve. There is approximately a 1.7% increase in IEER for every 1°F increase in suction temperature (see Appendix B). This results in an inflated increase in rated efficiency of approximately 27%. This accounts for about half of the 50% difference between real world efficiencies and IEER efficiencies as published by AHRI.

AHRI's standards already do this for other test parameters, e.g. the entering air conditions for almost all terminal units are specified at 80°F DB and 67°F WB. One argument by the VRF manufacturers against testing to a constant suction temperature is that the manufacturers have (proprietary) sequences to reset the suction temperature higher under low load or outdoor air conditions to "optimize" the energy performance of the system. This can also be done by chiller manufacturers, yet Standard 550/590 does not permit any such resetting for testing. In addition, different VRF manufacturers have different reset sequences. The only way to fairly compare different manufacturers' efficiencies is to test with the same constant suction temperature, the same as chiller chilled water temperatures.

Again, to level the playing field and provide fair comparisons between different *types of equipment or systems,* Standard 1230 should require testing to be done at a constant suction temperature of 44°F, the same as Standard 550/590 for chillers. This is within the ability of the VRF manufacturers; if the suction temperature can currently be manipulated to higher temperatures than the "native," or "as shipped" controls during testing, then the suction temperature can also be maintained at a constant temperature similar to the "native" controls.

HIA-C's recommendation is to specify a constant supply or suction fluid (refrigerant) temperature of 44°F for Standard 1230 test conditions. This will level the playing field with other AHRI central HVAC system test standards.

Refrigerant Piping Length

Standard 1230 does not address the loss in efficiency of a VRF system from the pumping energy lost to pumping refrigerant in the piping system.

In particular, the difference between residential and commercial installations is the test provision requiring only 25 feet of refrigerant line for the test setup. This pipe length might be adequate

[3] See slides 7 and 12: Differences Between Test Conditions and Real-World Operation for VRF Systems https://www.regulations.gov/document?D=EERE-2018-BT-STD-0003-0016

for some single family residential applications, but for commercial applications it is clearly inadequate, resulting in less compressor energy consumption than is experienced in the real world (see Appendix C).

1230 attempts to correct the cooling *capacity* lost to the energy used to pump refrigerant around the piping system rather than provide cooling capacity. No attempt is made to correct the *efficiency* or IEER for this lost pumping energy. This 25' of pipe is addressed in 1230, but is shown as a correction factor *increase* in capacity for test installation line length above 25 feet rather than a *decrease* in efficiency for actual line length installed in the field in a user's installation.

HIA-C's recommendation is to include a correction factor decrease in efficiency for actual line length above 25 feet for a real world field installation. This would include both cooling and heating operation.

Entering Air Temperature
Also at issue with Standard 1230 are the test conditions for air entering the evaporator. Standard entering air test conditions are 80°F DB/67°F WB. This is one area where AHRI is consistent between equipment or system types, requiring the same entering air conditions. However, this gives refrigerant systems an advantage over air and water systems. Increasing the entering air temperature increases the temperature difference between the air and the refrigerant, thereby increasing the heat transfer and efficiency. A water-based terminal unit such as chilled beam or fan coil cannot do this.

Almost no one in North America maintains a setpoint of 80°F during cooling. This is typically 73°F DB/61°F WB at the same 50% RH. The increase in IEER from 80°F/67°F to 73°F/61°F is approximately 8% (see Appendix D). Therefore a water based terminal unit is at an 8% disadvantage out of the box.

HIA-C's recommendation is that AHRI 1230 require testing and publishing for at least both of these two entering air conditions, rather than the higher temperature alone.

Evaporator/Condenser Combinations
Standard 1230 does not define which indoor evaporator is to be tested with a given outdoor condensing unit. As a result, testing of indoor units is limited to one style each for ducted and un-ducted testing. Other styles of indoor units that require lower suction temperatures and lower efficiencies are ignored. A similar opportunity to test configurations that are not a "basic" model exists for packaged rooftop ratings in AHRI Standard 340/360. However, Standard 340/360 requires that all "Individual models that contain/use different or alternate version of the same component shall either be represented separately as a unique Basic Model or certified with the same Basic Model based on testing of the least efficient configuration."[4]

In other words, according to AHRI Standard 340/360 a manufacturer can certify and list the *most* efficient combination, but must also certify and list the *least* efficient combination of indoor and outdoor components/units. On the other hand, Standard 1230 allows a VRF manufacturer to certify and list the *most* efficient combination only. This difference can be up to approximately 4.5% (see Appendix E).

HIA-C's recommendation is to require the listing and certification of both the least and most efficient combination of indoor and outdoor units in the same way as Standard 340/360.

Dehumidification
Because of the ability to operate VRF systems at elevated refrigerant supply/suction temperatures the equipment typically does not adequately dehumidify. Test results for a typical 1230 VRF test show that the discharge air temperature of the evaporator fan coil varies from

[4] AHRI STANDARD 340/360-2019, Appendix F, Section (D1) (D1.1.2)

approximately 59F at 100% load point, to 73F at 25% load point. These elevated discharge air temperatures will not dehumidify.

The draft 1230 currently being proposed attempts to address this shortcoming. The draft 1230 is proposing that an evaporator fan coil produce a sensible heat ratio (SHR) of .82 at 100% load and .85 at 75% load. This is not much different than the current test results. Typical discharge air temperatures for an SHR of .82 will be in the mid 50F range. For an SHR of .85 this will be a discharge air temperature in the upper 50F range. This compares to the current test results of 59F for the 100% and 61F for the 75% load point.

There is currently no requirement being proposed for SHR's at the 50% and 25% load point despite the fact that this represents almost 40% of the part load conditions. The test results could still use the previous elevated discharge air temperatures that provide no dehumidification.

The requirement of the .82 SHR at 100% is for only 2% of the part load conditions. This leaves the one SHR of .85 at the 75% load point as virtually the only requirement to demonstrate dehumidification.

For humid climates this means almost no dehumidification. The SHR for the load in humid climates in the US varies from .70 in the Southeast (Jacksonville, FL) to .86 in the Southwest (Phoenix, AZ) (Appendix G). Thus for humid climates in the US at an SHR of the load of .70 there is virtually no dehumidification. For dry climates in the US an SHR of .85 is barely below the SHR of even the driest climates let alone humid climates.

A typical hydronic fan coil with a supply fluid (chilled water) temperature of 45F, as required in AHRI Standard 440 for fan coil units, has an SHR of at least .78 (Appendix G). With custom circuiting and airflow this can be lowered to at least 0.63 (Appendix G). Thus a hydronic fan coil can dehumidify in even the most humid climates of the US while the proposed SHR's for the draft 1230 can barely dehumidify even in dry climates.

HIA-C's recommendation is to require an SHR of 0.78 at all load conditions similar to what a standard fan coil tested to Standard 440 would produce.

Cumulative Effect
Combining these rerates of VRF ratings allowed by Standard 1230 can explain the nearly 50% difference between real world IEER's and AHRI's published ratings as follows:

Suction temperature @ constant 44F (see Appendix B)	= 26.8%
Entering air temperature at 73°F (see Appendix D)	= 7.9%
Refrigerant line length @ 300'	
300' x 6% of Compressor HP / 100' x 300'	
X 24% of Compressor HP at minimum speed (Appendix C)	= 4.1%
Different indoor unit (see Appendix E)	= 3.4%
Oil return cycle (see Appendix F)	= 2.0%
Total	= 47.2%

Heating Ratings

It is the HIA-C's contention, supported by the information presented herein, that the test methodologies set forth in the current version of AHRI 1230 produce heating energy efficiency ratings for VRF equipment which exceed actual operating efficiencies in the field by 30% to 60%, with an average of approximately 48% (Appendix H).

Again, this is a critical area of concern given that ratings derived from the Standard 1230 test methodology are used by building owners, architects, electric utility energy efficiency program administrators, and HVAC design professionals to select amongst various HVAC system types. Currently there are no reliable data sources other than the AHRI ratings available to these key interest groups to assess equipment efficiencies. As a result, the AHRI ratings are used to compare efficiencies of various *system* types based on the rated IEER, despite the original intent of AHRI to level the playing field between *different manufacturers of the same equipment*. This was not the original intention, however it is now the norm.

Hyper Heat Operation
The outdoor unit in a VRF system is an air source heat pump. As a result it loses both capacity and efficiency with a drop in ambient or outdoor temperature in the heating mode.

To compensate for this drop in capacity the variable speed compressor is speeded up to increase the mass flow rate and the heat transfer or heating capacity. This enables a Hyper Heat over speed unit to produce a constant capacity as the outdoor temperature drops. At approximately 10F to 15F the Hyper Heat unit reaches its' maximum over speed of approximately 50% and then remains at this speed. As the outdoor temperature drops the capacity then drops proportionately to what it would have been for a constant speed compressor (Appendix I).

However, this increase in capacity at lower outdoor air temperatures is not free and comes at a cost of reduced efficiency. The First Law of Thermodynamics is still the First Law, you can't get something for nothing. The COP of a constant speed heat pump at 47F is approximately 3.5 vs. 3.2 for the variable speed heat pump, not a large difference. However, at 17F this difference is 2.6 vs 1.1 an almost 60% difference. In addition, a COP of 1.1 is almost straight electric heat.

1230 currently does not require the testing and publication of hyper-heat or over speed capacities or COP's.

Some VRF manufacturers are also claiming their units can heat to -25F. The amount of heat available at -25F from an air source heat pump is substantially reduced. To heat a building at this low ambient temperature the unit will have to have backup heat. Some manufacturers are wrapping the condenser coils with electric strip heat as the backup. This is a COP of 1.0.

The constant speed COP's are the published COP's. However, the units are operated in the variable speed mode, but there is no published data on the lower COP's. This is misleading because only the constant speed COP's are published yet the unit is operated at the lower variable speed COP's.

This difference in the seasonal HSPF of an over speed Hyper Heat unit vs. a constant speed outdoor unit is approximately 7% (Appendix J) based on the AHRI Standard 210/240 climate zone 4 rating.

HIA-C's recommendation is to require the listing and certification of both the constant and hyper-heat or over speed heating capacities and COP's at 47F and 17F as well as low temperature operation of -25F.

Refrigerant Piping Length
As noted previously Standard 1230 does not address the loss in efficiency of a VRF system from the compressor pumping energy lost to pumping refrigerant throughout the piping system.

1230 attempts to correct the cooling capacity to this lost energy used to pump refrigerant around the piping system rather than provide cooling. No attempt is made to correct the IEER or COP efficiency for this lost pumping energy in either cooling or heating.

For Hyper Heat operation this is a substantial loss. The savings in pumping energy for a variable speed compressor, fan or hydronic pump is substantial since pumping horsepower is a function of the flow cubed. For cooling this can be as much as 75% savings in pumping energy (Appendix E) of a variable speed pump vs. a constant speed in a typical HVAC system.

Conversely, in heating, the pumping energy for a Hyper Heat unit increases by the cube of the flow due to speeding up the compressor and mass flow to achieve higher heating capacities. This can be as much as a 75% increase in pumping energy (Appendix D).

HIA-C's recommendation is to include a correction factor decrease in efficiency for actual line length above 25 feet for a real world field installation. This would include both cooling and heating operation.

Defrost Operation
Any air source heat pump in almost any climate zone in North America will develop frost on the coil of the outdoor unit at some point.

While details of defrost control algorithms are not public, manufacturers, typically initiate a defrost cycle by the presence of a sufficiently cold coil temperature for a specific length of time. Defrost terminates once the coil warms to a certain temperature (which occurs very quickly if no frost is present on the coil). The number and duration of defrost cycles, therefore, is generally a function of outdoor air temperature, humidity, and heat pump output.

During defrost the unit is operated in a reverse cycle by drawing heat from inside the building through the indoor unit to use to defrost the coil of the outdoor unit. In other words the unit operates in the air conditioning mode during the heating season. Not only does this create a comfort problem, but the unit is operating at zero efficiency. It is actually less than zero, because once the unit switches back to heating the unit has to reheat the building from the heat removed during the defrost cycle.

This inefficiency can be up to 3% (Appendix K).

Cumulative Effect
Combining these rerates of VRF ratings allowed by Standard 1230 the difference between real world HSPF's and AHRI's published ratings are as follows:

Hyper Heat Operation (Appendix J)	= 7.1%
Refrigerant line length @ 300'	
300' x 6% of Compressor HP / 100' x 300'	
X 75% of Compressor HP at minimum speed (Appendix C)	= 13.5%
Defrost cycle (see Appendix K)	= 2.9%
Total	= 23.5%

Conclusions
In summary, stakeholders are using AHRI ratings derived from the AHRI Standard 1230 testing procedures to compare the performance of *different types of systems,* not just different manufacturers within the same equipment category. AHRI's stated original intention for the standard was to compare the performance of different manufacturer's equipment within the same category. However, in order to do a reasonable job of allowing the comparison of different types of equipment and systems across different product categories, a standard must require the equipment to be tested much closer to real world operating conditions than the current version of Standard 1230 allows.

AHRI's other standards test equipment to more stringent conditions than 1230 allowing VRF equipment to publish cooling IEER and heating COP ratings that are artificially higher than other equipment types.

Standard 1230 does not require testing of central VRF refrigerant systems to real world operating conditions of a constant fluid (refrigerant) supply (suction) temperature, unlike Standard 550/590 for central water system chillers, which does.

Standard 1230 does not require testing of both the *least and most* efficient combinations of indoor and outdoor units, unlike Standard 340/360 for unitary equipment, which does.

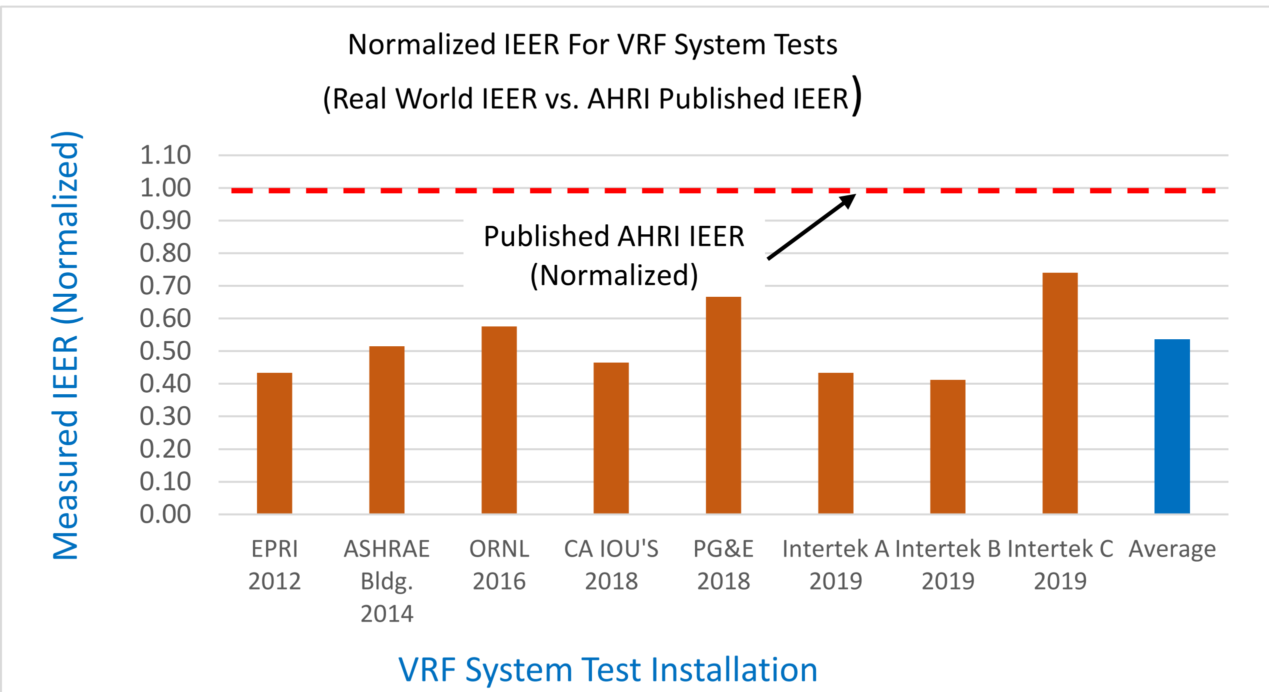
Standard 1230 does not require testing at Sensible Heat Ratios low enough to produce dehumidification in climate zones in the US similar to what a hydronic fan coil can produce when tested to Standard 440.

Finally, it should be noted that seldom are VRF systems installed with filters with the correct Minimum Efficiency Reporting Value (MERV) rating – while this is more of an issue with installing contractors or building owner/operators rather than with VRF equipment manufacturers, it nevertheless is a reality in the field. Using filters that comply with ASHRAE Standard 52.2 further impacts system efficiencies by increasing static pressure. Owner/operators of VRF systems frequently note that even when using lower MERV filters, they must be cleaned frequently or problems with freezing condenser coils occur.

AHRI should look on this as an opportunity to correct the earlier oversight and make Standard 1230 a viable standard for commercial VRF systems and. In addition, AHRI can also level the playing field between 1230 and other AHRI test standards to make the comparison between VRF and other *systems* fair. The question is are DOE, AHRI and the Advocates up to the task?
It was the intent of the Working Group to narrow the discrepancy of VRF field performance to the published efficiencies of 1230 of approximately 50% to within 10% to 20%. However, it appears the discrepancy will only be lowered by 10% to 20%, not to <u>within</u> 10% to 20%.

Appendix A – Real-World Operating Data for VRF Systems

VRF System Installation Real World Operating Data	Abbreviation
Electric Power Research Institute	EPRI 2012
ASHRAE Headquarters	ASHRAE Bldg. 2014
Oak Ridge National Laboratory	ORNL 2016
California Investor Owned Utilities (Pacific Gas & Electric, Southern California Edison, San Diego Gas & Electric)	CA IOUs 2018
Pacific Gas and Electric - Davis, CA	PG&E 2018
Intertek DOE Appliance Standards and Rulemaking Federal Advisory Committee (ASRAC) VRF Working Group Manufacturer A	Intertek A 2019
Intertek DOE Appliance Standards and Rulemaking Federal Advisory Committee (ASRAC) VRF Working Group Manufacturer B	Intertek B 2019
Intertek DOE Appliance Standards and Rulemaking Federal Advisory Committee (ASRAC) VRF Working Group Manufacturer C	Intertek C 2019



Appendix B – IEER Increase for Suction Temperature Increase

Refrigeration Equipment

	EER increase per degree F in suction temperature
SWEP https://www.swep.net/refrigerant-handbook/3.-compressors/3.4-compressor-performance/	1.94%
Wiki Books https://en.wikibooks.org/wiki/Energy_Efficiency_Reference/Refrigeration/Walkthrough_Checklist	2.00%
Process Engineering https://www.processengineer.info/refrigeration/effect-of-evaporating-condensing-temperatures-on-system-efficiency.html	2.08%
Average	**2.01%**

Chiller Equipment

	EER increase per degree F increase in LWT
York YCAL0019	1.3%
York YCAL0066	1.5%
Daikin AGZ010	1.6%
Daikin AGZ070	1.4%
Average	1.5%

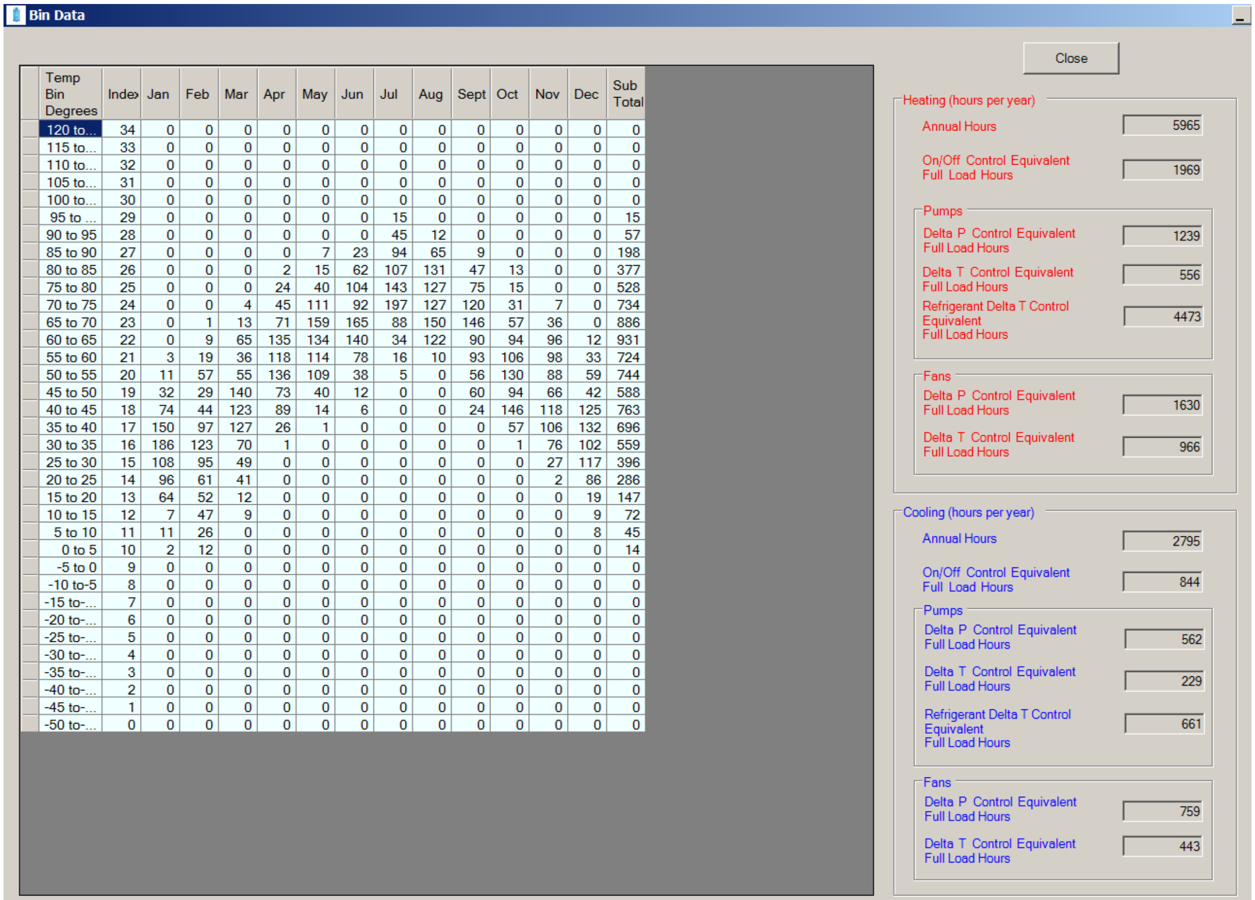
Average for Refrigeration and Chiller Equipment	**1.7%**

IEER Efficiency Increase

	Part Load				**IEER**
	100%	75%	50%	25%	
Suction Temperature for Test (F)	54	57	62	70	
Suction Temperature for Native Control Operation (F)	44	44	44	44	
Suction Temperature Difference	10	13	18	26	
Efficiency Difference per 1F Decrease in Suction Temperature	1.7%	1.7%	1.7%	1.7%	
Efficiency Difference for Suction Temperature Difference between AHRI test and Native Control operation conditions	17%	22%	31%	44%	
IEER Weighting Factor	0.02	0.617	0.238	0.125	
Weighted Efficiency Difference	0.34%	13.64%	7.28%	5.53%	26.8%

Appendix C – Pumping Energy

ASHRAE Climate Zone 4 – Cincinnati



Temp Bin Degrees	Index	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sept	Oct	Nov	Dec	Sub Total
120 to...	34	0	0	0	0	0	0	0	0	0	0	0	0	0
115 to...	33	0	0	0	0	0	0	0	0	0	0	0	0	0
110 to...	32	0	0	0	0	0	0	0	0	0	0	0	0	0
105 to...	31	0	0	0	0	0	0	0	0	0	0	0	0	0
100 to...	30	0	0	0	0	0	0	0	0	0	0	0	0	0
95 to ...	29	0	0	0	0	0	0	15	0	0	0	0	0	15
90 to 95	28	0	0	0	0	0	0	45	12	0	0	0	0	57
85 to 90	27	0	0	0	0	7	23	94	65	9	0	0	0	198
80 to 85	26	0	0	0	2	15	62	107	131	47	13	0	0	377
75 to 80	25	0	0	0	24	40	104	143	127	75	15	0	0	528
70 to 75	24	0	0	4	45	111	92	197	127	120	31	7	0	734
65 to 70	23	0	1	13	71	159	165	88	150	146	57	36	0	886
60 to 65	22	0	9	65	135	134	140	34	122	90	94	96	12	931
55 to 60	21	3	19	36	118	114	78	16	10	93	106	98	33	724
50 to 55	20	11	57	55	136	109	38	5	0	56	130	88	59	744
45 to 50	19	32	29	140	73	40	12	0	0	60	94	66	42	588
40 to 45	18	74	44	123	89	14	6	0	0	24	146	118	125	763
35 to 40	17	150	97	127	26	1	0	0	0	0	57	106	132	696
30 to 35	16	186	123	70	1	0	0	0	0	0	1	76	102	559
25 to 30	15	108	95	49	0	0	0	0	0	0	0	27	117	396
20 to 25	14	96	61	41	0	0	0	0	0	0	0	2	86	286
15 to 20	13	64	52	12	0	0	0	0	0	0	0	0	19	147
10 to 15	12	7	47	9	0	0	0	0	0	0	0	0	9	72
5 to 10	11	11	26	0	0	0	0	0	0	0	0	0	8	45
0 to 5	10	2	12	0	0	0	0	0	0	0	0	0	0	14
-5 to 0	9	0	0	0	0	0	0	0	0	0	0	0	0	0
-10 to-5	8	0	0	0	0	0	0	0	0	0	0	0	0	0
-15 to-...	7	0	0	0	0	0	0	0	0	0	0	0	0	0
-20 to-...	6	0	0	0	0	0	0	0	0	0	0	0	0	0
-25 to-...	5	0	0	0	0	0	0	0	0	0	0	0	0	0
-30 to-...	4	0	0	0	0	0	0	0	0	0	0	0	0	0
-35 to-...	3	0	0	0	0	0	0	0	0	0	0	0	0	0
-40 to-...	2	0	0	0	0	0	0	0	0	0	0	0	0	0
-45 to-...	1	0	0	0	0	0	0	0	0	0	0	0	0	0
-50 to-...	0	0	0	0	0	0	0	0	0	0	0	0	0	0

Cooling pumping energy at 60% minimum speed
= 661 EFLH / 2795 cooling hours
= 24% of constant speed pump

Heating pumping energy at 150% over speed
= 4473 EFLH / 5965 heating hours
= 75% of constant speed pump

Appendix D – IEER Increase for Entering Air Temperature Decrease

WaterFurnace Versatec[5]

Cooling Capacity Corrections

Entering Air WB °F	Total Clg Cap	Sensible Cooling Capacity Multipliers - Entering DB °F										Power Input	Heat of Rejection
		60	65	70	75	80	80.6	85	90	95	100		
55	0.898	0.723	0.866	1.048	1.185	*	*	*	*	*	*	0.985	0.913
60	0.912		0.632	0.880	1.078	1.244	1.260	*	*	*	*	0.994	0.927
63	0.945			0.768	0.960	1.150	1.175	*	*	*	*	0.996	0.954
65	0.976			0.694	0.881	1.079	1.085	1.270	*	*	*	0.997	0.972
66.2	0.983			0.655	0.842	1.040	1.060	1.232	*	*	*	0.999	0.986
67	**1.000**			0.616	0.806	**1.000**	1.023	1.193	1.330	1.480	*	**1.000**	**1.000**
70	1.053				0.693	0.879	0.900	1.075	1.205	1.404	*	1.003	1.044
75	1.168					0.687	0.715	0.875	1.040	1.261	1.476	1.007	1.141

NOTE: *Sensible capacity equals total capacity at conditions shown. 4/22/12

Capacity Correction

Correction Factor at 80/67 =	1.000
Correction Factor at 75/60 =	1.078
Correction Factor at 70/60 =	0.880
Correction Factor at 73/60 =	0.959
Correction Factor at 75/63 =	0.960
Correction Factor at 70/63 =	0.768
Correction Factor at 73/63 =	0.845
Correction Factor at 73/61 =	0.921

Power Input Correction

Correction Factor at 80/67 =	1.000
Correction Factor at 73/60 =	0.994
Correction Factor at 73/63 =	0.996
Correction Factor at 73/61 =	0.995

Adjustment = Capacity Change / Power Input Change = (1 - 0.921)/0.995 = 0.079

[5] https://www.waterfurnace.com/literature/versatec/sc2750au.pdf

Appendix E – IEER Increase for different indoor unit

See: AHRI Program Name: Air Conditioners and Air Conditioner Coils
AHRI Certified Reference Numbers 3818083 and 4175084[6]
American Standard Condenser and Evaporator combinations

Most Efficient Indoor/Outdoor Unit Configuration, IEER = 11.50
Lease Efficient Indoor/Outdoor Unit Configuration, IEER = 11.00
IEER Increase for different indoor unit = (11.00 – 11.50)/ 11.00 = -0.045

Use adjustment of 4.5%

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[6] https://www.ahridirectory.org/NewSearch?programId=3&searchTypeId=3&productTypeId=3401

Appendix F – IEER Decrease for oil return cycle

VRF heat pump cycles approximately every 8 hours for 10 minutes for oil return.

Assume that efficiency of VRF system is zero during oil return cycle.

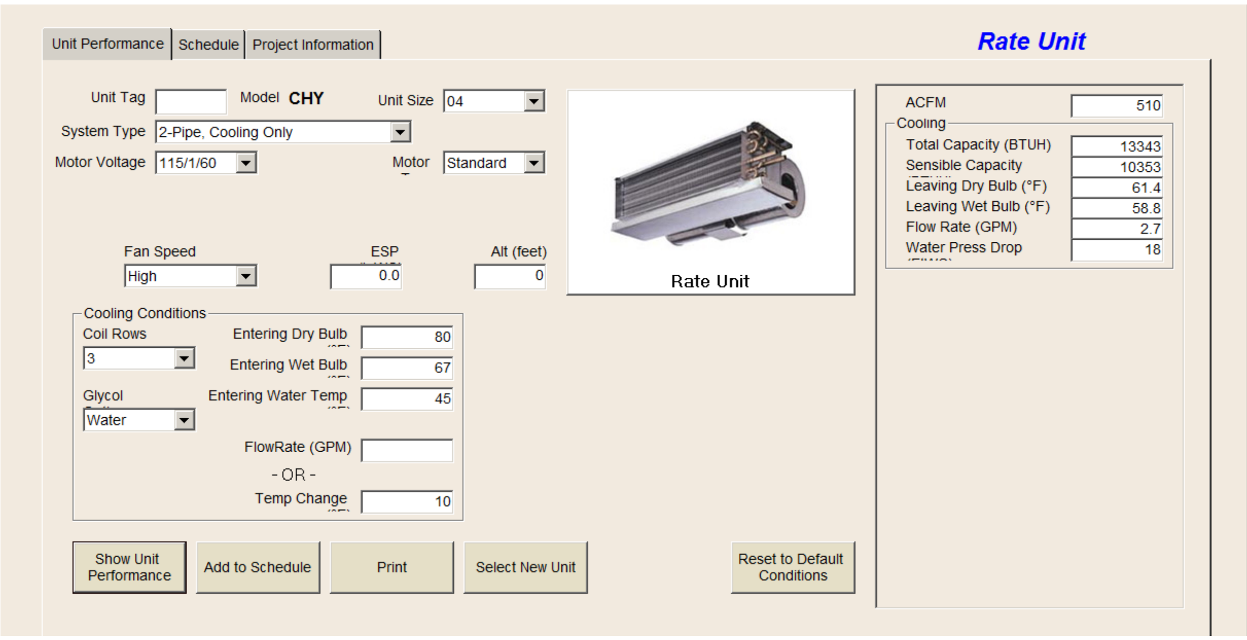
De-rate for oil return cycle
= 10 min / 60 min/hr x 8 hr)
= 2.1%

Appendix G – Sensible Heat Ratios of Cooling Loads in the US

Sensible Heat Ratios of typical office occupancy cooling loads in the US

City	Sensible Heat (MBH)	Latent Heat (MBH)	Sensible Heat Ratio
• Boston	3.6	1.1	0.77
• Jacksonville	3.7	1.6	0.70
• St. Louis	4.0	1.2	0.77
• New Orleans	3.8	1.6	0.70
• Phoenix	4.3	0.7	0.86
• Seattle	3.5	0.8	0.81
• San Diego	3.7	0.8	0.82

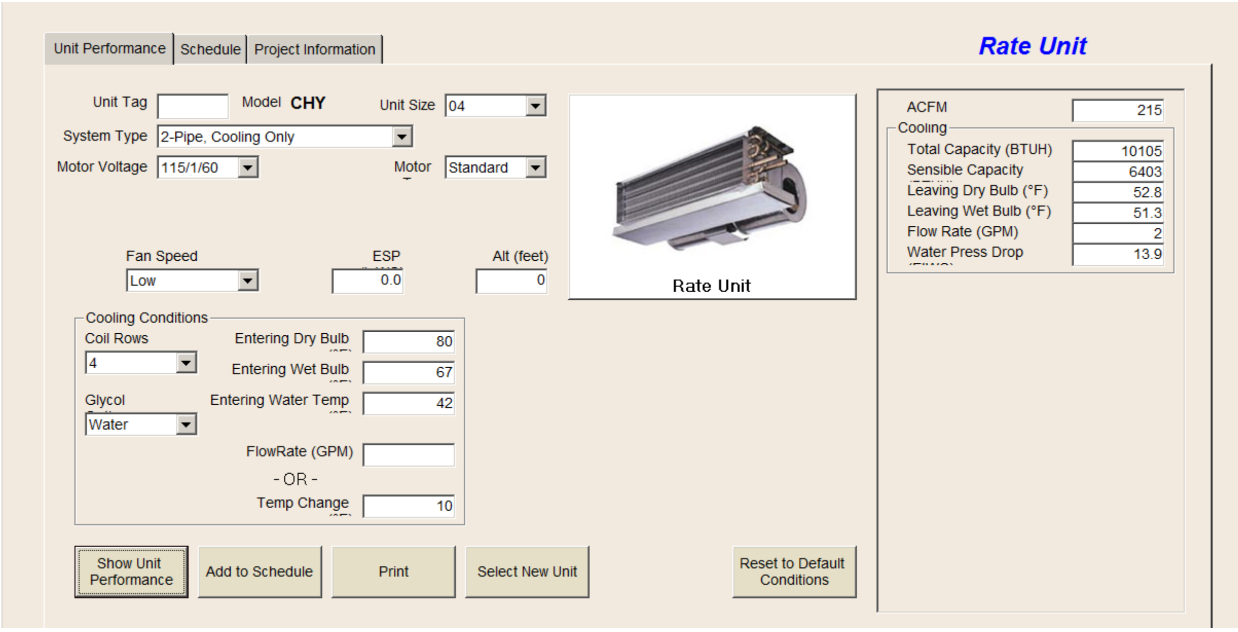
Sensible Heat Ratio of Standard Hydronic Fan Coil Unit – IEC CHY Size 04



Standard 400 cfm 3 row unit at 45F entering water temperature and High fan speed

Sensible Heat Ratio = 10353 / 13343
= .78

Sensible Heat Ratio of Optimized Fan Coil – IEC CHY Size 04

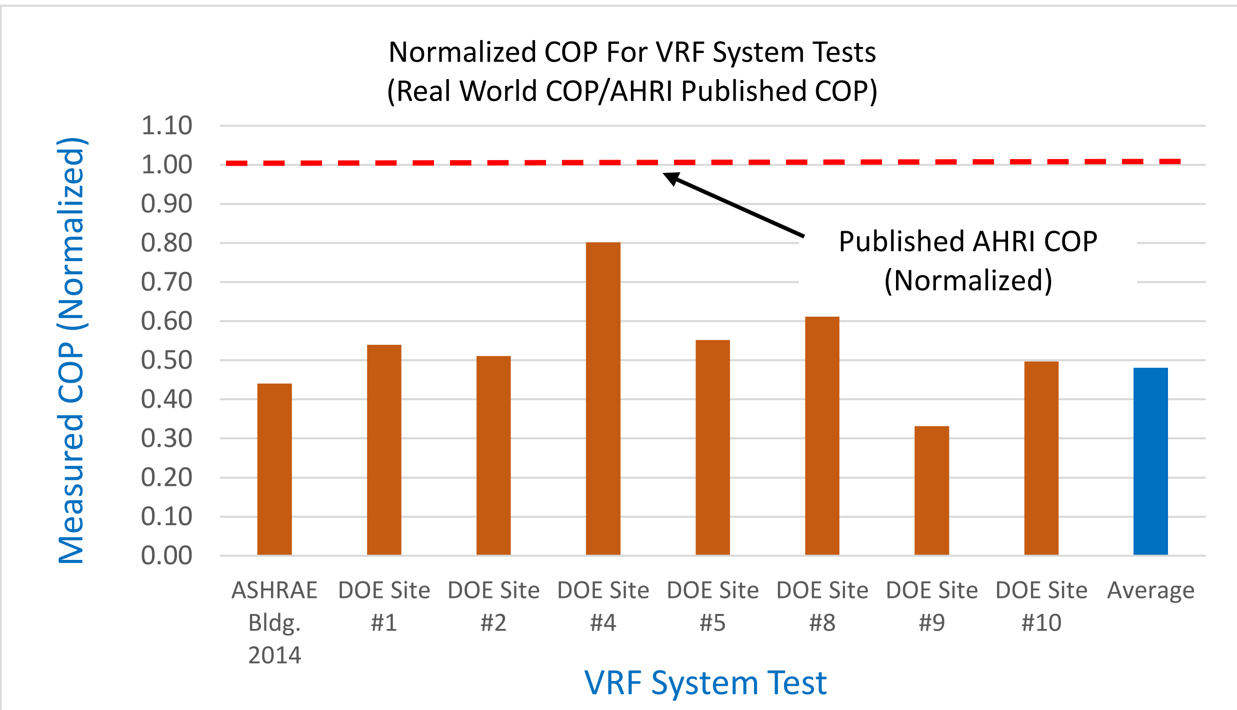


Optimized 400 cfm 4 row unit at 45F entering water temperature and Low fan speed

Sensible Heat Ratio = 6403 / 10105
= .63

Appendix H – Real-World Operating Data for VRF Systems

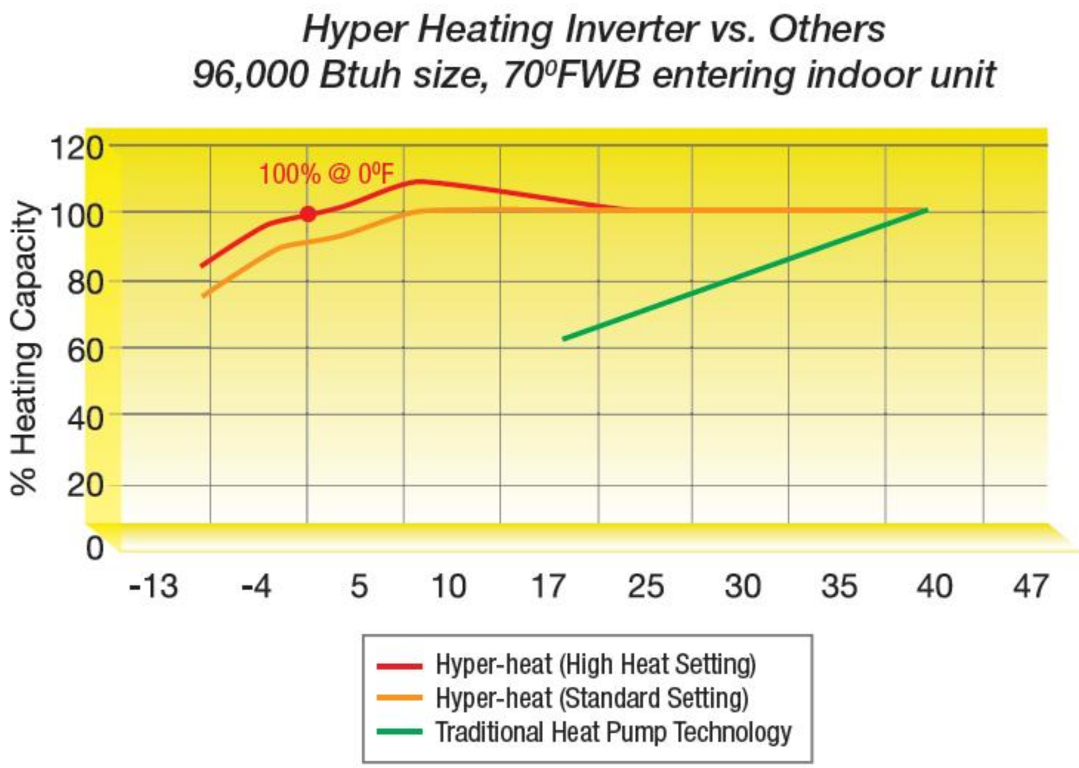
VRF System Installation Real World Operating Data	Abbreviation
ASHRAE Headquarters Building[7]	ASHRAE Bldg. 2014
DOE Field Performance of Inverter-Driven Heat Pumps in Cold Climates	DOE Site #1
DOE Field Performance of Inverter-Driven Heat Pumps in Cold Climates	DOE Site #2
DOE Field Performance of Inverter-Driven Heat Pumps in Cold Climates	DOE Site #4
DOE Field Performance of Inverter-Driven Heat Pumps in Cold Climates	DOE Site #5
DOE Field Performance of Inverter-Driven Heat Pumps in Cold Climates	DOE Site #8
DOE Field Performance of Inverter-Driven Heat Pumps in Cold Climates	DOE Site #9
DOE Field Performance of Inverter-Driven Heat Pumps in Cold Climates	DOE Site #10



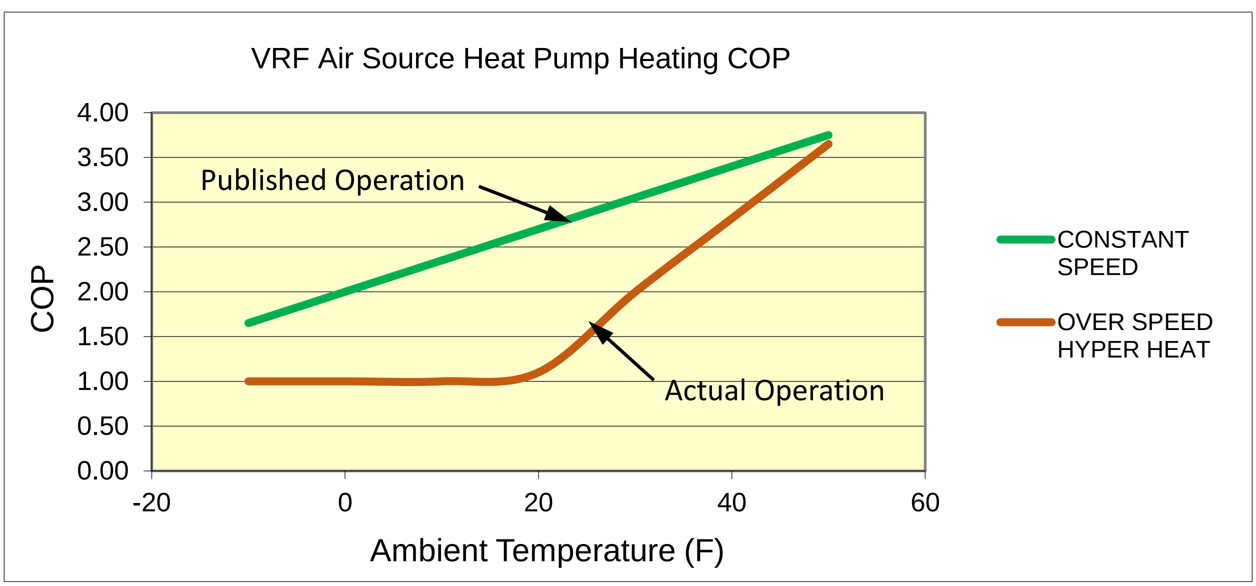
[7] https://shareok.org/handle/11244/25719

Appendix I – Hyper Heat Operation

Capacity of Outdoor Unit



Efficiency of Outdoor Unit



Appendix J – IEER Decrease for Hyper Heat Operations

From ANSI/AHRI Standard 210/240-2017 Performance Rating of Unitary Air-conditioning & Air-source Heat Pump Equipment[8]

Table 18. Distribution of Fractional Heating Hours in Temperature Bins, Heating Load Hours, and Outdoor Design Temperature for Different Climatic Regions, Region IV

Constant Speed VRF Unit	Variable Speed Hyper Heat VRF Unit
COP_{47} = 3.43	COP_{47} = 3.43
COP = 0.035 * Ambient Air °F + (COP_{47} -1.65)	COP = 0.08 * Ambient Air °F + (COP_{47} -3.76)

Heating Seasonal Weighted Average COP 3.22	Heating Seasonal Weighted Average COP 2.99
Heating Seasonal HSPF = 11.0	Heating Seasonal HSPF = 10.2
Hyper Heat vs. Constant Speed HSPF Decrease = 7.1%	

[8] http://www.ahrinet.org/App_Content/ahri/files/STANDARDS/AHRI/AHRI_Standard_210-240_2017.pdf

Appendix K - COP Decrease for Defrost Operation

Field Performance of Inverter-Driven Heat Pumps in Cold Climates[9]

Table 8. Overall COP With and Without Defrost Cycles Included

Site	Overall COP	COP w/o Defrost	Days Monitored
1	1.61	1.69	204
2	1.99	2.01	141
4	2.31	2.44	142
5	1.71	1.73	28
8	2.33	2.41	44
9	1.11	1.12	57
10	2.06	2.11	51

Site # from DOE Study	Overall COP with Defrost	Overall COP w/o Defrost	Defrost Penalty
1	1.61	1.69	4.97%
2	1.99	2.02	1.51%
4	2.31	2.44	5.63%
5	1.71	1.73	1.17%
8	2.33	2.41	3.43%
9	1.11	1.12	0.90%
10	2.06	2.11	2.43%
Average Defrost Penalty			2.86%

[9] https://www1.eere.energy.gov/buildings/publications/pdfs/building_america/inverter-driven-heat-pumps-cold.pdf