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DEVELOPMENT OF 20 IEER ROOFTOP UNITS – A SIMULATION STUDY

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ABSTRACT

Based on detailed steady-state system and component modeling, we developed a rooftop unit system design, which is able to achieve IEER (Integrated Energy Efficiency Ratio) higher than 20. We modeled fin-&-tube and micro-channel heat exchangers using segment-to-segment approach, and use ARI 10-coefficient compressor map to simulate compressor performance. The system modeling is based on a component-based modeling approach, which facilitates flexible simulation of complicated system configurations. Starting with a baseline system having IEER of 16.6, we extensively investigated numerous technical options, i.e. varying compressor sizes, heat exchanger fin densities, fin-&-tube or micro-channel heat exchanger, suction line heat exchanger, desiccant wheel, tandem compressor, variable-speed compressor, and condenser evaporative pre-cooling; and developed an innovative system configuration combining a tandem compression system with a variable-speed compression system. The combined system can achieve high IEER as well as process the outdoor ventilation air over an extensive range. We successfully evaluated the design concept for a 20-ton (70.4 kW) unit as well as a 10-ton (35.2 kW) unit. All the selected components are readily accessible on the market, and we validated the performance predictions against existing Rooftop Unit (RTU) products at the rating condition. This paper illustrates a potentially cost-effective high IEER RTU design.

1. INTRODUCTION

Recently, the US Department of Energy released a new high-efficiency design specification for commercial air conditioners with 10-ton (35.2 kW) to 20-ton (70.4 kW) capacity. It targets a high-performance Integrated Energy Efficiency Rating (IEER) of 18.0 (Reference: High Performance Rooftop Unit Challenge). ANSI/AHRI 340/360 gives the standard for rating IEER. It puts significant weighting on part-load performances, i.e. 61.7% weight for EER at 75% capacity, 23.8% weight for EER at 50% capacity. This requires smooth capacity modulation with varying air and refrigerant flow rates, and sufficient utilization of all the heat exchanger surface areas at part-load conditions. Advanced components need to be used to meet the design targets, like micro-channel heat exchangers, high efficiency fans and compressors, etc. Other means to enhance the efficiency are to use accessories like condenser evaporative precooling, desiccant wheel, etc.

The IEER rating can't tell the whole story about energy saving, since it doesn't consider the outdoor air free cooling effect. Rooftop units in commercial buildings operate long hours for space cooling when the ambient temperature is lower than the indoor. Outdoor air economizing or dedicated outdoor air system can lead to significant energy savings. The main challenge of utilizing outdoor air is to remove the extra moisture. The outdoor air condition can vary over a large range, which requires flexible dehumidification capability.

Based on the above considerations, we developed an innovative system configuration, which combines a tandem compression system and variable speed compression system, as shown in Figure 1.

The major considerations for the system design are summarized below,

1. The tandem system provides majority of the capacity by running at 50% or 100% capacity and the variable speed system provides capacity fine tuning. Thus, it provides continuous capacity modulation capability.
2. This should result in a cost-effective product. It provides smooth capacity modulation while avoiding the need for large size, variable-refrigerant-flow compressors. The choices of commercial variable-refrigerant-flow compressors are limited, and so they can be rather expensive. We intend to use small-size, variable-speed compressors, having capacities of 3-ton (10.6 kW) to 5-ton (17.6 kW), for which the choices are plentiful. We

can not only choose variable-speed scroll compressors, but also rotary compressors having fairly good efficiency.

3. The variable speed compressor can control evaporating pressure. It will be used for dehumidification, for example, processing humid outdoor air. Dampers will be used to regulate air flow path at different capacity levels, so that the indoor air flow will only pass through active coils. The damper position can be adjusted to make the variable-speed compression system become a dedicated outdoor air system. Furthermore, altering the compressor speed and the blower speed can match the sensible and latent loads simultaneously, and the outdoor air flow fraction can be optimized to achieve maximum efficiency at varied outdoor air temperature and humidity conditions.
4. This system is easily modifiable. To further improve the performance for special applications, we can add a desiccant wheel for humid zones, and use condenser evaporative pre-cooling at desert conditions. Further, design engineers could replace the fixed-speed tandem compressors with digital scroll tandem compressors, or a larger size variable-speed compressor, for better part-load performance, but at higher cost.
5. This system has superior reliability, since it has two parallel vapor compression systems, and the two compressors in the tandem system rarely fail together.

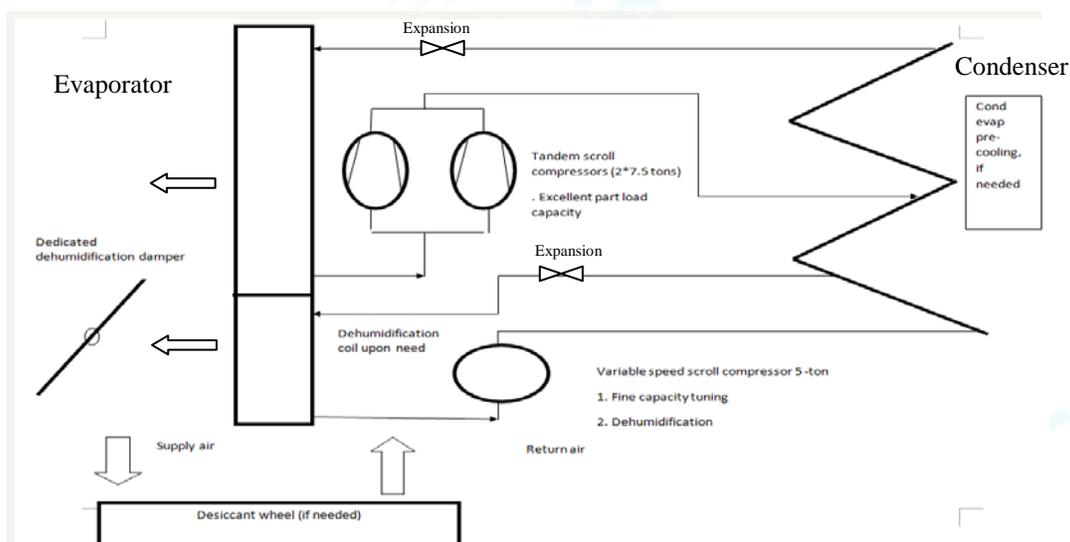


Figure 1: System Configuration of Next Generation Rooftop Unit

2. MODEL DESCRIPTIONS

We have developed steady-state simulation models covering most categories of residential and light commercial space cooling, space heating and water heating components, like compressors, heat exchangers, pumps, fans, etc. Our models are fundamentally based, can simulate detailed geometry and circuitry, and accept real air-side and refrigerant-side boundary conditions.

Our in-house component models have different complexity levels, which fall to three categories, i.e. bulk models, phase-to-phase models, and discretized models. The bulk models are usually based on Effectiveness-NTU or UA-LMTD approach, to simulate the component as a whole. The phase-to-phase models separate the refrigerant side to vapor, two-phase and liquid regions, and each region has individual air side and refrigerant side entering states. The discretized models use segment-to-segment approach, which divide a heat exchanger into to numerous mini-segments; each segment has individual refrigerant and air entering parameters, and considers possible phase separation; the mini segments are basic building blocks, which are used to build up heat exchangers having arbitrary circuitry, geometry, and represent any boundary conditions. All our phase-to-phase and segment-to-segment heat exchanger models are able to calculate refrigerant charge inventory. For the system modeling, we have been developing a component-based modeling framework, which allows connecting steady-state component models in any manner, for example, simulating the vapor compression system with coupled desiccant wheel and indoor coil; simulating parallel air-to-refrigerant condenser and water heater, etc. Some component models used in this study are introduced as below.

Compressors:

- Single-speed Compressor: We use ARI 10-coefficient compressor maps to calculate mass flow rate, power consumption; simulate energy balance from inlet to outlet using the calculated power and given heat loss ratio; we also consider the actual suction state to correct the map mass flow and power predictions.
- Variable-speed compressor: The model accepts multiple sets of mass flow and power curves, and does linear or quadratic interpolation between speed levels.

Heat Exchangers:

- Segment-to-segment fin-&-tube condenser: It uses a segment-to-segment modeling approach; Each tube segment has individual air side and refrigerant side entering states, and considers possible phase transition; the coil model can simulate arbitrary tube and fin geometries and circuitries, any refrigerant side entering and exit states, misdistribution, and accept two-dimensional air side temperature, humidity and velocity local inputs; The tube circuitry and 2-D boundary conditions are provided by an input file.
- Segment-to-segment fin-&-tube evaporator: In addition to the functionalities of the segment-to-segment fin-tube condenser, the evaporator model is capable of simulating the dehumidification process.
- Segment-to-segment micro-channel condenser: The model uses a segment-to-segment modeling approach; Each micro-channel port segment has individual air-side and refrigerant-side entering states, and considers possible phase transition; the coil model can simulate arbitrary port (round, triangle, etc.), fin geometries and circuitries (serpentine, slab, etc.), any refrigerant side entering and exit states, misdistribution, and accept two-dimensional air side temperature, humidity and velocity local inputs.
- Segment-to-segment micro-channel evaporator: In addition to the functionalities of the segment-to-segment micro-channel condenser, the evaporator model is capable of simulating the dehumidification process.

Expansion Devices:

- Idealized TXV: The compressor suction superheat degree is explicitly specified as well as the condenser subcooling degree.

Fans and Blowers:

- Single-speed fan: Given airflow rate, the model uses a fan curve to simulate static head, power consumption, and calculate air-side temperature increment from inlet to outlet.
- Variable-speed fan: The model accepts multiple sets of fan curves, and does linear interpolation between speed levels.

Accessories:

- Desiccant wheel: Analogy between heat and mass transfer is used to simulate process side and regeneration side energy transfer based on given effectiveness; the effectiveness shall be determined from manufacturer's data; the model can simulate any entering air temperature and humidity levels, and predict temperature and humidity change at the process side and the regeneration side; this is the same model used in TRNSYS.

3. 20-TON ROOFTOP UNIT SYSTEM DESIGN

3.1 Baseline Unit – Heat Exchanger Sizing, Blower and Fan

We chose a 20-ton (70.4 kW) baseline unit, which represents the highest efficiency RTUs on the market (IEER = 16.6). The baseline unit is an interlaced system using four scroll compressors feeding parallel refrigeration systems; variable indoor air flow; thermostatic expansion valve; and staged condenser fans (six fans). It uses larger indoor and outdoor fin-&-tube (FT) coils, high efficiency condenser fans and indoor blower, in comparison to the other 20-ton units on the market. Here, we use the same heat exchanger sizing, indoor blower and outdoor fans as our baseline configuration. Some general information on the heat exchangers is given in Table 1.

Table 1: Condenser and Evaporator of 20-Ton Unit

Parameters	Fin-&-Tube Condenser Coil	Fin-&-Tube Evaporator Coil
Face area [sq. ft]	70.6 (6.56 m ²)	33.3 (3.09 m ²)
Tube diameter [in]	3/8 (0.0095 m)	3/8 (0.0095 m)
Number of rows	3 (cross counter-flow)	3 (cross counter-flow)
Fins per inch	14 (551 fins/m)	14 (551 fins/m)

We have the tandem system use 3/4 of the surface area for both the condenser and the evaporator, and the variable speed system uses the remaining. Furthermore, we optimized the number of circuits to achieve proper refrigerant-side pressure drops at nominal refrigerant mass flow rates. In addition, we directly adopted the staged condenser fans and the multi-speed indoor blower used in the baseline unit. Assuming a nominal external static pressure of 0.40 inch of H₂O (99.64 Pa) (minimum value, according to ANSI/AHRI 340/360), the blower consumes 3.6 HP (2685 W) at 8000 cfm (3.78 m³/s) 1.57 HP (1171 W) at 5000 cfm (2.36 m³/s), 0.94 HP (701 W) at 3000 cfm (1.416 m³/s), and 0.74 HP (552 W) at 2000 cfm (0.9439 m³/s). The six parallel condenser fans use 1900 Watts to provide 22,500 cfm (10.62 m³/s) of air flow at full capacity. We assume conservatively that the condenser fan power drops proportionally with the modulated outdoor air flow rate in the following simulations.

3.2 Capacity Modulation and General System Setting

We use a typical 400 cfm/ton indoor air flow rate, i.e. 8000 cfm for the 100% capacity for the 20-tons unit, and use the same 22,500 cfm for the condenser air flow. The general modulation levels, the corresponding air flow rates and the fan powers are listed in Table 2.

Table 2: Modulation Parameters of 20-Ton Unit

Capacity Levels	100%	75%	50%	25%
Number of running compressors in tandem system	2	1	1	0
Number of running compressors in variable-speed system	1	1	0	1
Indoor air flow rate of tandem system [CFM] (1 cfm = 4.719E-4 m ³ /s)	6000	3000	3000	0
Indoor air flow rate of variable-speed system [CFM]	2000	2000	0	2000
Outdoor air flow rate of tandem system [CFM]	16875	8438	8438	0
Outdoor air flow rate of variable-speed system [CFM]	5625	5625	0	5625
Total indoor blower power @ static head of 0.40 in H ₂ O [W]	2662	1171	701	552
Total outdoor fan power [W]	1900	1188	713	475

At all the modulation levels, we set the superheat degree at the exit of each individual evaporator at 10 R (5.6 K), and subcooling degree at each condenser exit at 10 R (5.6 K). This system design aims to use the tandem system to provide the major capacity and the variable-speed system is modulated to supplement the remaining capacity at each rated capacity level.

3.3 Choices of Compressors

Manufacturer A has variable-speed scroll compressors, ranging from 3-ton to 5-ton (1 ton = 3.517 kW). The manufacturer claims that the variable-speed scroll compressors can achieve +/-60% variations from the rated capacity. We chose a 5-ton compressor for the variable-speed system. The compressor map has coefficients at five speed levels.

With respect to the tandem compressors, we evaluated two options, Manufacturer A's 14- and 15-ton scrolls. Both of them can provide 20-ton capacity, coupling with different speeds of the 5-ton variable-speed compressor. We list the two options below.

Option 1: Fin-&-tube evaporator, Fin-&-tube condenser, 14-ton tandem compressor (TD), 5-ton variable speed compressor (VS).

This is to use the three-row, fin-&-tube evaporator and condenser, the 14-ton tandem compressor coupled with the 5-ton variable-speed compressor in two parallel, refrigeration systems.

Option 2: Fin-&-tube evaporator, Fin-&-tube condenser, 15-ton TD, 5-ton VS.

In comparison to Option 1, Option 2 uses the 15-ton TD in the place of the 14-ton TD, with other components the same.

Conclusion:

By comparing the two options in Table 5, we can see that the 15-ton tandem is a more efficient pair than the 14-ton tandem, especially with one compressor running. When coupling with the 15-ton tandem, smaller speeds are required for the VS system, and then the VS system gets higher efficiency. In addition, the 15-ton tandem system has larger dehumidification capability than the 14-ton tandem system, due to its larger refrigerant mass flow rate in the same evaporator. Consequently, the 15-ton tandem is an apparent choice over the 14-ton tandem.

We had a cross-comparison validation point between our designs and the baseline unit in Table 3, at 100% capacity, 95 F outdoor air temperature, and indoor DB/WB of 80 °F/67 °F (26.7 °C/19.6 °C).

Table 3: Comparisons between Our Designs and Baseline Unit at 100% Capacity

Products	Net Capacity [kBtu/h]	EER [Btu/h/W]	SHR
14 ton TD and 5-ton VS	24.4 (7.15 kW)	12.5 (COP = 3.66)	77%
15 ton TD and 5-ton VS	25.5 (7.47 kW)	12.6 (COP = 3.69)	76%
Baseline Unit	23.6 (6.92 kW)	12.6 (COP = 3.69)	75%

As indicated in Table 3, our system performances match very closely to the baseline unit, since we are using the same combination of indoor blower, condenser fans, heat exchanger dimensions, and we all use scroll compressors at regular efficiencies. The consistency indicates that our system simulations are in the ballpark of the real product performance. Consequently, we can extend the system simulations to other conditions and capacity ranges with good credibility.

3.4 Choices of Heat Exchangers

A convenient solution for enhancing heat exchanger effectiveness is to have denser fins; however, the added fin density also increases the fan power as a penalty. A micro-channel heat exchanger doubles the heat exchanger effectiveness without increasing the fan power, as compared to a fin-&-tube heat exchanger having the same volume. On the other hand, micro-channel heat exchangers have disadvantages in water drainage and distribution with two-phase entrance. Hence, micro-channel heat exchangers are good choices to replace fin-&-tube heat exchangers as condensers, while not so promising yet as evaporators.

We studied two micro-channel (MHX) heat exchangers having the same frontal area as the fin-&-tube condenser coil. These two MHX coils use the same type of tube and fin. The tube has width of 1 inch (0.0254 m) and height of 0.072 inch; it has 11 rectangular inner ports, whose hydraulic diameter is 0.05 inch. The fin has a depth of 1 inch, height of 0.34 inch, thickness of 0.0062 inch, and the fin density is 20 fins per inch. Both the fin and the tube are made of aluminum. With the same frontal area as the fin-&-tube condenser, we pack 100 tubes in one slab (row). To minimize the refrigerant pressure drop in the two-phase and vapor sections, we designed a step-circuited pattern having four passes with 40 tubes, 30 tubes, 20 tubes and 10 tubes in the passes along the refrigerant flow direction. For the two MHX coils, one option is one slab (row) and has almost the same heat transfer effectiveness as the three-row FT condenser; the other option has two identical slabs. The two-row MHX coil has the same air side pressure drop as the three-row FT coil and the one-row MHX coil has half of the air side pressure drop. We calibrated our MHX models against a manufacturer's design software.

To estimate the fan power reduction due to the reduced air-side pressure drop, we compared the selected baseline to another 20-ton unit which uses a 1-row MHX condenser. The alternative has similar condenser air flow rate and smaller condenser frontal area than the baseline unit, and its 1-row MHX condenser has 20 fins per inch. By comparing the condenser fan powers, we assumed that using a 1-row MHX condenser to replace the 3-row FT condenser can lead to fan power saving of 36% at the same air flow rate. We investigated three options as below for choosing heat exchangers.

Option 3: Fin-&-tube evaporator, Fin-&-tube condenser, 15-ton TD, 5-ton VS, 20 fins per inch.

This option starts from Option 2, by adding denser fins to both the condenser and evaporator (from 14 to 20 fins per inch). We ignored the fan power increase for the related simulations. And so, the evaluation is the best case scenario for the FT case.

Option 4: Fin-&-tube evaporator, 2-row MHX condenser, 15-ton TD, 5-ton VS

This option starts from Option 2, by using a 2-row MHX to replace the 3-row FT condenser. We assume that the condenser fan power would be the same due to the similar air side pressure drops.

Option 5: Fin-&-tube evaporator, 1-row MHX condenser, 15-ton TD, 5-ton VS

This option starts from Option 2, by using a 1-row MHX to replace the 3-row FT condenser. We assume 36% reduction in the condenser fan power.

Conclusion:

By comparing the above three options in Table 5, we can see minor efficiency improvements for Option 3 and Option 4 with the most gain from option 5. If we consider the fan power increase regarding to Option 3, the benefit

from increasing the fin densities would be even smaller. It can be concluded that in the highly modulated rooftop system, increasing heat transfer effectiveness actually leads to minimum efficiency enhancement, since the original coils already have sufficient effectiveness at the part load conditions with reduced refrigerant mass flow rate. Consequently, the real benefit using the MHXs is the reduction in the condenser fan power. That is why Option 5 is the most advantageous choice, which uses a 1-row MHX condenser to accomplish the same heat transfer duty but reducing the condenser fan power consumption.

3.5 Choices of Accessories

Once we select the compressor and the heat exchangers, the basic RTU configuration is set. Manufacturers tend to provide choices of accessories for customers with special requests. Certainly, these accessories will add product cost. We evaluated five options of accessories as below, for pursuing higher IEER.

Option 6: Fin-&-tube evaporator, 1-row MHX condenser, 15-ton TD, 5-ton VS, suction line accumulator.

This option starts from Option 5, by adding suction line accumulators upstream of the compressors. A suction line accumulator can minimize the suction superheat degree, and thus enlarge two-phase area in the evaporator. In the case, saturated vapor exits the evaporator and enters the suction line accumulator.

Option 7: Fin-&-tube evaporator, 1-row MHX condenser, 15-ton TD, 5-ton VS, liquid line accumulator.

This option starts from Option 5, by adding liquid line accumulators upstream of the expansion. A liquid line accumulator can minimize the condenser subcooling degree, and thus enlarge two-phase area in the condenser. However, small subcooling degrees reduce the cooling capacity as a penalty. For the related simulations in the case, we specified the condenser exit subcooling degree as 1.0 (0.56 K).

Option 8: Fin-&-tube evaporator, 1-row MHX condenser, 15-ton TD, 5-ton VS, suction line heat exchanger (SLHX).

This option starts from Option 5, by adding a refrigerant-to-refrigerant heat exchanger between the suction line and the liquid line. This option can enhance cooling capacity while minimize the superheat degree. We only investigated the SLHX addition to the tandem system. For the related simulations in the case, we specified the condenser exit subcooling degree at 5.0 R (2.78 K) (allowing for additional subcooling in the SLHX), and the compressor suction superheat degree at 20.0 R (11.1 K). The heat transfer effectiveness of the SLHX is chosen as 0.3, which represents a practical choice, since it is mainly a single-phase heat exchanger and we want to limit the expense.

Option 9: Fin-&-tube evaporator, 1-row MHX condenser, 15-ton TD, 5-ton VS, desiccant wheel.

This option starts from Option 5, by adding a desiccant wheel to the indoor. The process side of the desiccant wheel is at the supply air path, and the regeneration side of the desiccant wheel is at the return air path. There is no external heat source used for the regeneration. This technique is called the Cromer cycle. The intention is to enhance the dehumidification capability of the evaporating coil by recycling moisture from the supply side to the return side. For the related system simulations, we applied the desiccant wheel in both the tandem and the variable-speed system, assuming the same indoor air flow rate and no added blower power for the initial analysis, which represents the best case scenario. In addition, the process effectiveness of the desiccant wheel is fixed at 0.1, and the regeneration effectiveness is 0.7, these effectiveness values represent a low performance wheel. We use the same desiccant wheel modeling method as TRNSYS, and effectiveness values were obtained from Panaras et al. (2010).

Option 10: Fin-&-tube evaporator, 1-row MHX condenser, 15-ton TD, 5-ton VS, condenser evaporative pre-cooling

This option starts from Option 5, by adding condenser evaporative pre-cooling. The parameter for gauging efficiency of the evaporative pre-cooling process is the wet bulb (WB) efficiency, which is a measure of the drop in the dry bulb temperature due to the evaporative process. The wet bulb efficiency, E_{wb} , is defined in Equation (1) as

$$E_{wb} = \frac{T_{db,i} - T_{db,o}}{T_{db,i} - T_{wb,i}} \quad (1)$$

where E_{wb} is the wet bulb efficiency; $T_{db,i}$ and $T_{db,o}$ are the entering and leaving air dry bulb temperatures, respectively; and $T_{wb,i}$ is the entering air wet bulb temperature. A manufacturer's 8-inch pad was selected as the evaporative pre-cooling pad. The 8 inch pad was recommended by the manufacturer for commercial application. The wet bulb efficiency is 0.84 at the frontal velocity of our design. ANSI/AHRI 340/360 defines the ambient wet bulb temperature for evaluating the evaporative pre-cooling effect at individual capacity levels, which are given in Table 4. In addition, the reduced air dry bulb temperatures entering the condenser coil are listed in Table 4, which are calculated using Equation 1.

It has to be mentioned that the evaporative pre-cooling pad adds air side pressure drop and increase the condenser fan power consumption. We obtained the pad pressure drop from the manufacturer's performance data and add it to the pressure drop of the 1-row MHX condenser, and the sum is about 100% relative to the pressure drop across the original baseline 3-row condenser coil. And thus, we use the same condenser fan power consumption as the baseline, when adding the evaporative pre-cooling pad.

Table 4: Condenser Entering Dry Bulb Temperatures by Applying 8 inch Evaporative Pre-cooling Pad

Capacity Level	100%	75%	50%	25%
Ambient Air Dry Bulb [°F]	95.0 (35°C)	81.5 (27.5°C)	68.0 (20°C)	65.0 (18.3°C)
Ambient Air Wet Bulb [°F]	75.0 (23.9°C)	66.3 (19.1°C)	57.5 (14.2°C)	52.8 (11.6°C)
Condenser Entering Air Dry Bulb [°F]	78.2 (25.7°C)	68.7 (20.4°C)	59.2 (15.1°C)	54.8 (12.7°C)

Conclusion:

By comparing the above five options in Table 5, Option 6 to Option 8 mainly serve to enhance the heat transfer situations by minimizing the superheat or subcooling degree. However, since the indoor and outdoor heat exchangers have already been oversized at part-load conditions for the modulated systems, the benefits by enhancing the heat transfer are minor. The option of adding liquid line accumulators even degrades the performance, because it reduces the total cooling capacity while having minor influence on power consumption. In a modulated refrigeration system, a suction line accumulator usually is needed for protecting the compressor, but is not necessary for increasing the system efficiency.

Adding a desiccant wheel, upstream and downstream of the evaporating coil, can significantly increase the dehumidification ability, which approximately changes the SHR from 70% to 50%. However, this doesn't necessarily enhance the efficiency, since the desiccant wheel also lowers the air temperature entering the evaporator coil, and results in lower suction pressure. Considering the added power due to the extra air side pressure drop and rotating the wheel, the performance degradation may be even larger. Certainly, using gas heat to regenerate the desiccant wheel can not only enhance the dehumidification capability but also would lift the evaporating pressure. This option is not considered here since it adds too much complexity to the system. It is recommended to add the desiccant wheel option where there are specific dehumidification needs, for instance, in hospitals.

Condenser evaporative pre-cooling is an easy add-on or retrofit choice for an existing rooftop unit. It is an ideal solution to upgrade the system efficiency for highly modulated systems, in cases where enhancing heat transfer only leads to minor benefit. The evaporative pre-cooling effect lowers the sink temperature, which narrows the Carnot cycle source-to-sink temperature difference. Another way of changing the sink temperature is to use a ground-source heat exchanger for condensing. However, ground-source-coupled RTUs are not popular on the market so far, and they should not be used for a cooling only application. Present evaporative pre-cooling technologies can boost the system efficiency largely but they come at the expense of water consumption. In addition, it has penalties of scaling, bacterial growth that can cause infections and higher maintaining cost for cleaning. An economic analysis should be conducted to see if the addition is really worthwhile according to individual climate zones. In Table 5, items 2, 5 and 10 are highlighted, since they are recommended options.

Table 5: Performance Predictions of Multiple Design Options for 20-ton Unit (Options as listed above)

Capacity Level	100%	75%	50%	25%	IEER
Ambient Temperature [°F]	95.0	81.5	68.0	65.0	
Baseline System:					16.6
Option 1: Fin-&tube evaporator, Fin-&tube condenser, 14-ton TD, 5-ton VS					17.4
RPM of VS compressor	3000	4500	0	3600	
EER of VS system	13.2	16.2	0.0	20.1	
EER of tandem system	12.4	16.4	19.1	0.0	
EER of combined system	12.5	16.3	19.1	20.1	
SHR of combined system	77%	73%	71%	76%	
Net cooling capacity (combined) [Btu/h (0.293 W)]	244316	181097	116425	63011	
Option 2: Fin-&tube evaporator, Fin-&tube condenser, 15-ton TD, 5-ton VS					18.0

Capacity Level	100%	75%	50%	25%	IEER
RPM of VS compressor	2700	3600	0	3600	
EER of VS system	13.0	16.9	0.0	20.1	
EER of tandem system	12.5	16.9	20.2	0.0	
EER of combined system	12.6	16.9	20.2	20.1	
SHR of combined system	76%	73%	68%	76%	
Net cooling capacity (combined) [Btu/h]	254553	179697	129171	63011	
Option 3: Fin-&tube evaporator, Fin-&tube condenser, 15-ton TD, 5-ton VS, 20 fins per inch					18.4
RPM of VS compressor	2700	3600	0	3600	
EER of combined system	13.1	17.2	20.5	20.8	
SHR of combined system	76%	73%	68%	77%	
Net cooling capacity (combined) [Btu/h]	261791	181704	129790	64560	
Option 4: Fin-&tube evaporator, 2-row MHX condenser, 15-ton TD, 5-ton VS					18.4
RPM of VS compressor	2700	3600	0	3600	
EER of combined system	13.1	17.3	20.6	20.7	
SHR of combined system	75%	73%	68%	76%	
Net cooling capacity (combined) [Btu/h]	258357	181260	130228	63621	
Option 5: Fin-&tube evaporator, 1-row MHX condenser, 15-ton TD, 5-ton VS (36% condenser fan power reduction)					18.8
RPM of VS compressor	2700	3600	0	3600	
EER of combined system	13.1	17.6	21.0	21.3	
SHR of combined system	75%	73%	71%	76%	
Net cooling capacity (combined) [Btu/h]	255825	179976	129253	63242	
Option 6: Fin-&tube evaporator, 1-row MHX condenser, 15-ton TD, 5-ton VS, suction line accumulator					18.7
RPM of VS compressor	2700	3600	0	3600	
EER of combined system	13.0	17.5	20.9	21.3	
SHR of combined system	76%	73%	68%	76%	
Net cooling capacity (combined) [Btu/h]	255132	179771	129208	63362	
Option 7: Fin-&tube evaporator, 1-row MHX condenser, 15-ton TD, 5-ton VS, liquid line accumulator					18.3
RPM of VS compressor	2700	3600	0	3600	
EER of combined system	12.8	17.2	20.5	20.9	
SHR of combined system	77%	74%	69%	77%	
Net cooling capacity (combined) [Btu/h]	246954	174174	125666	61549	
Option 8: FT evaporator, 1-row MHX condenser, 15-ton TD, 5-ton VS, suction line heat exchanger in tandem system					18.7
RPM of VS compressor	2700	3600	0	3600	
EER of combined system	13.1	17.5	20.9	21.3	
SHR of combined system	76%	73%	69%	76%	
Net cooling capacity (combined) [Btu/h]	254266	178520	127538	63242	
Option 9: Fin-&tube evaporator, 1-row MHX condenser, 15-ton TD, 5-ton VS, desiccant wheel					18.5
RPM of VS compressor	2700	3600	0	3600	
EER of combined system	12.8	17.3	20.7	21.0	
SHR of combined system	54%	52%	51%	52%	
Net cooling capacity (combined) [Btu/h]	248114	176579	126710	62121	
Option 10: Fin-&tube evaporator, 1-row MHX condenser, 15-ton TD, 5-ton VS, condenser evaporative precooling (assuming wet bulb efficiency of 0.84, no condenser fan power saving)					20.9
RPM of VS compressor	2700	3600	0	3600	
EER of combined system	15.8	20.0	22.6	22.6	
SHR of combined system	73%	71%	67%	75%	
Net cooling capacity (combined) [Btu/h]	278845	191498	135365	65524	

Figure 1 illustrates the relative performance gains with various technical options, in terms of IEER compared to the baseline system (IEER = 16.6).

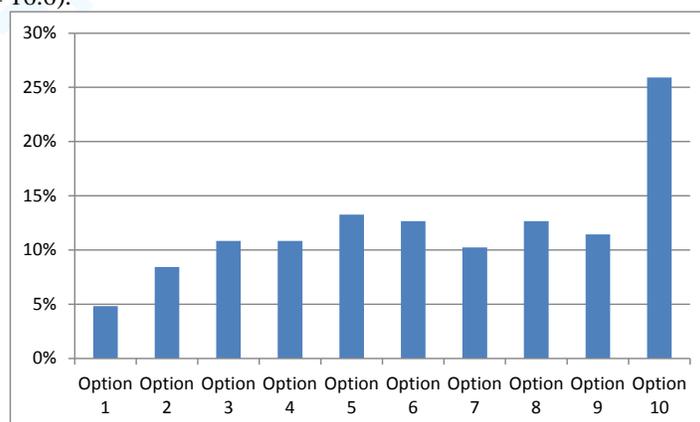


Figure 2: Relative Performance Gains with Various Technical Options

4. 10-TON ROOFTOP UNIT SYSTEM DESIGN

We applied the same system configuration to design a 10-ton unit, i.e. using combined tandem and variable-speed systems. The 10-ton baseline unit, we started with, has an IEER of 14.7 and the rated EER of 12.3 at 95 °F ambient temperature and 100% capacity. The 10-ton baseline unit has two parallel refrigeration circuits. Similar to the 20-ton system, we adopted the same heat exchanger frontal area, the indoor blower and outdoor fan data from the baseline system. The heat exchanger parameters and the modulation levels can be seen in Table 6 and Table 7.

Table 6: Condenser and Evaporator of 10-Ton Unit

Parameters	Fin-&-Tube Condenser Coil	Fin-&-Tube Evaporator Coil
Face area [sq. ft]	47.1 (4.38 m ²)	15.6 (1.45 m ²)
Tube diameter [in]	3/8 (0.0095 m)	3/8 (0.0095 m)
Number of rows	3 (cross counter-flow)	3 (cross counter-flow)
Fins per inch	14 (551 fins/m)	14 (551 fins/m)

Table 7: Modulation Parameters of 10-Ton Unit

Capacity Levels	100%	75%	50%	25%
Number of Running Compressors in Tandem system	2	1	1	0
Number of Running Compressors in Variable-Speed System	1	1	0	1
Indoor Air Flow Rate of Tandem System [CFM] (1 cfm = 4.719E-4 m ³ /s)	3000	1500	1500	0
Indoor Air Flow Rate of Variable-Speed System [CFM]	1000	1000	0	1000
Outdoor Air Flow Rate of Tandem System [CFM]	8850	4425	4425	0
Outdoor Air Flow Rate of Variable Speed System [CFM]	2950	2950	0	2950
Total Indoor Blower Power @ Static Head of 0.30 inH ₂ O [W] (note: 0.30 inH ₂ O is the minimum external static pressure required for 10-ton units, ANSI/AHRI 340/360)	1357	656	509	339
Total Outdoor Fan Power [W]	940	588	353	235

We selected the Manufacturer A's 3-ton variable speed scroll for the variable speed system, and its 7-ton tandem compressor for the tandem system. Similar to the methodology used for the 20-ton unit, the step-by-step design improvements can be seen in Table 8.

Table 8: Performance Predictions of Multiple Design Options for 10-ton Unit (Options as listed above)

Capacity Level	100%	75%	50%	25%	IEER
Ambient Temperature [°F]	95.0	81.5	68.0	65.0	
Baseline System:					14.7
Option 1: Fin-&-tube evaporator, Fin-&-tube condenser, 7-ton TD, 3-ton VS					18.4
RPM of VS compressor	1800	2700	0	3600	
EER of combined system	12.8	17.3	20.3	20.8	
SHR of combined system	76%	75%	69%	74%	

Capacity Level	100%	75%	50%	25%	IEER
Net cooling capacity (combined) [Btu/h(0.293 W)]	121763	83261	61407	33388	
Option 2: FT evaporator, 1-row MHX condenser, 7-ton TD, 3-ton VS (assuming 36% condenser fan power reduction)					19.0
RPM of VS compressor	1800	2700	0	3600	
EER of combined system	13.2	17.9	21.0	21.8	
SHR of combined system	75%	75%	69%	74%	
Net cooling capacity (combined) [Btu/h]	121732	83035	61373	33386	
Option 3: Fin-&tube evaporator, 1-Row MHX condenser, 7-ton TD, 3-ton VS, condenser evaporative precooling (assuming wet bulb efficiency of 0.84, no condenser fan power reduction)					21.6
RPM of VS compressor	1800	2700	0	3600	
EER of combined system	16.3	20.8	23.0	24.0	
SHR of combined system	74%	73%	68%	72%	
Net cooling capacity (combined) [Btu/h]	131308	87540	63286	34602	

It is noted that, for the 1-row micro-channel condenser, we used the same type of tube-and-fin heat exchanger as the 20-ton unit, except that we arranged with 30 tubes, 20 tubes, 14 tubes and 8 tubes in the passes along the refrigerant flow direction.

5. SUMMARY

From the investigations above, we can draw conclusions as below,

1. Combining a tandem system and a variable-speed system can easily achieve IEERs higher than 18.0 for both 20-ton and 10-ton rooftop units. If higher efficiency is desired, condenser evaporative precooling can be applied to achieve IEERs above 20.
2. For a highly modulated system, enhancing heat transfer would only lead to minor benefits. Design engineers should pay more attention to reducing power consumption of the motion movers like fans and compressors. A single-row micro-channel heat exchanger is applied in this study, which can accomplish the same heat transfer load with shorter air flow path and smaller heat exchanger volume, and thus reduce the condenser fan power.
3. All the components we selected are readily accessible on the market. We use small size, variable-speed compressors coupled with fixed-speed tandem compressors, aiming to have a cost-effective design.
4. Our component-based equipment modeling toolkit is used to investigate all the options above. For all the component modeling, we calibrated our model against manufacturers' performance data; for equipment modeling, we matched the system efficiencies to the baseline units. Thus, the system performance predictions should have good credibility.

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REFERENCES

- ANSI/ARI Standard 540-99, 1999, "Positive Displacement Refrigerant Compressors and Compressor Units", Air-Conditioning and Refrigeration Institute, Arlington, VA
- ANSI/AHRI Standard 340/360, 2007, "Performance Rating of Commercial and Industrial Unitary Air-Conditioning and Heat Pump Equipment", Air-Conditioning, Heating, and Refrigeration Institute, Arlington, VA
- Charles, J., Cromer, "Cromer Cycle Air Conditioner: A Unique Air Conditioner Desiccant Cycle to Enhance Dehumidification and Save Energy", *Proceedings of 12th Symposium on Improving Building Systems in Hot and Humid Climates*, San Antonio, TX, May 15-17, 2000
- High Performance Rooftop Unit Challenge,
http://apps1.eere.energy.gov/buildings/publications/pdfs/alliances/cbea_rtu_spec_long.pdf
- Panaras, S., E. Mathioulakis, V. Belessiotis, N. Kyriakis, 2010, "Experimental validation of a simplified approach for a desiccant wheel model", *Energy and Building*, Vol. 42, pp. 1719-1725.
- S. A. Klein. 2010, "TRNSYS 16.0 User Manual".