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Development of a Hydronic Rooftop Unit – HyPak-MA

Final Report

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ACRONYMS & ABBREVIATIONS

(Alphabetical)

- 3D Three-dimensional
- AEP Advanced Energy Products
- APT Automated Performance Testing System
- ARI Air Conditioning and Refrigeration Institute
- ASHRAE American Society of Heating Refrigeration and Air-Conditioning Engineers, Inc.
- BHP Brake Horsepower
- BTU British thermal units
- BTU/HR British thermal units per hour
- CEWC Counter-flow Evaporative Water Cooler
- CFD Computational Fluid Dynamics
- CFM Cubic feet per minute
- CO_2 Carbon dioxide
- DB Dry bulb temperature
- DDC Direct digital controller
- DEG Davis Energy Group, Inc.
- DCT Des Champs Technologies, Inc.
- DOE Department of Energy
- EA Exhaust air
- EER Energy efficiency rating
- FPI Fins per inch
- GPM Gallons per minute
- GPPS General purpose polystyrene
- GWH Gas water heater
- HIPS High-impact polystyrene
- HP Horsepower
- HR Hour
- HVAC Heating, ventilation, & air-conditioning
- HX Heat exchanger
- HYPAK Hydronic Rooftop Packaged Unit
- HZ Hertz
- IAQ Indoor air quality
- IN Inch
- InH_20 Inches of water column
- InWC Inches of water column
- kW Kilowatts
- LBNL Lawrence Berkeley National Laboratory
- LEED Leadership in Energy and Environmental Design
- MERV Minimum efficiency reporting value
- MOP Maximum operating pressure

- NETL National Energy Technology Laboratory
- OA Outdoor air
- PC Personal computer
- PG&E Pacific Gas & Electric Company
- PVC Polyvinyl chloride
- RA Return air
- R&D Research and development
- RH Relative humidity
- RPM Rotations per minute
- RTD Resistance temperature device
- RTU Conventional roof-top cooling/heating unit
- SA Supply air
- SCFM Standard cubic feet per minute
- SMUD Sacramento Municipal Utility District
- TEFC Total enclosed fan cooled
- TXV Thermostatic expansion valve
- USGBC U.S. Green Building Council
- VA Ventilation air
- VAC Volts alternating current
- VCEC Vertical Counter-flow Evaporative Cooler
- VFD Variable frequency drive
- W Watts
- WB Wet bulb temperature
- WC Water column
- Wg Water gauge

1. Executive Summary

Project Background

The majority of U.S. commercial floor space is cooled by rooftop HVAC units (RTUs). RTU popularity derives chiefly from their low initial cost and relative ease of service access without disturbing building occupants. Unfortunately, current RTUs are inherently inefficient due to a combination of characteristics that unnecessarily increase cooling loads and energy use. 36% percent of annual U.S. energy, and two-thirds of electricity, is consumed in and by buildings[1]. Commercial buildings consume approximately 4.2 quads of energy each year at a cost of $230 billion per year, with HVAC equipment consuming 1.2 quads of electricity[1,2]. More than half of all U.S. commercial floor space is cooled by packaged HVAC units, most of which are rooftop units (RTUs)[3,4]. Inefficient RTUs create an estimated 3.5% of U.S. CO_2 emissions, thus contributing significantly to global warming[5]. Also, RTUs often fail to maintain adequate ventilation air and air filtration, reducing indoor air quality.

This is the second HyPak project to be supported by DOE through NETL. The prior project, referred to as HyPak-1 in this report, had two rounds of prototype fabrication and testing as well as computer modeling and market research. The HyPak-1 prototypes demonstrated the high performance capabilities of the HyPak concept, but made it clear that further development was required to reduce heat exchanger cost and improve system reliability before HyPak commercialization can commence. The HyPak-1 prototypes were limited to about 25% ventilation air fraction, limiting performance and marketability. The current project is intended to develop a “mixed-air” product that is capable of full 0-100% modulation in ventilation air fraction, hence it was referred to as HyPak-MA in the proposal. (For simplicity, the -MA has been dropped when referencing the current project.)

Project Objective

The objective of the HyPak Project is to design, develop and test a hydronic RTU that provides a quantum improvement over conventional RTU performance. Our proposal targeted 60% and 50% reduction in electrical energy use by the HyPak RTU for dry and humid climates, respectively, when compared with a conventional unit, and reduction in peak energy consumption of 50% and 33% respectively. In addition to performance targets, our goal is to develop a production-ready design with durability, reliability and maintainability similar to air-cooled packaged equipment, and that can be commercialized immediately following the conclusion of this project.

HyPak Concept

The proprietary HyPak concept uses many innovative features including evaporative condensing, ventilation air pre-cooling, high-efficiency heating, heat recovery, and efficient filtration. HyPak’s name derives from the evaporative cooling of the condenser and ventilation air and is short for “Hydronic Packaged Unit.” The heart of the HyPak concept is the Vertical Counter-flow Evaporative Cooler (VCEC) that combines indirect evaporative pre-cooling of building ventilation air with direct evaporative condenser water cooling into a single reliable and low-cost component. This plate-type air-to-air heat exchanger is wetted in every other passage for evaporative cooling of condenser water, with the remaining passages dry for indirect evaporative cooling. The VCEC is made from thermoformed thin-sheet plastic in a fan-fold configuration with heat-sealed edges.

Project Team

The HyPak team includes three organizations:

- Davis Energy Group (DEG) – overall management and design, test planning, instrumentation selection and setup;
- Des Champs Technologies, Inc. (DCT), which was recently acquired by Munters – detailed design and manufacturing of prototypes, laboratory testing; and
- Pride Polymers – VCEC thermoforming and fabrication.

Under the anticipated commercialization scenario, DCT will manufacture and market HyPak products. Pride Polymers will manufacture the VCEC modules for Advanced Energy Products (AEP), a recent spin-off of DEG, who will supply them to DCT for HyPak production. AEP will also market HyPak in the West and work to secure utility incentive and demonstration programs. AEP will also make the VCEC heat exchanger available to other HVAC manufacturers as well.

Phase 1 Tasks and Accomplishments

The four Phase 1 tasks were structured to result in design, fabrication, and laboratory testing of a HyPak prototype, with particular emphasis placed on development of the low-cost VCEC module. The team began the current project by re-evaluating the draft configuration supplied in our proposal. In an effort to eliminate the HyPak-1 problem of water entering the indoor air passages, the proposal configuration specified two separate evaporative modules. Early in Phase 1 we realized that the ventilation air pre-cooling module would have to be sized to accommodate full airflow, yet would only condition 10-40% ventilation rates for the majority of operating periods, resulting in poor volumetric efficiency. DEG and DCT evaluated several different HyPak and VCEC configurations, settling on the basic layouts that were developed in Tasks 1 and 2. Results and accomplishments by task are summarized below:

Task 1: Develop VCEC for Laboratory Testing

With the return to a single evaporative module, DEG focused on creating a design that would reliably segregate moist and dry air streams. The resulting VCEC design was simulated using computation fluid dynamics software to optimize air flow distribution and to minimize pressure drop. Pressure gradients from the CFD software were used to optimize the location of spacers to maintain passage openings without excessive pressure drop.

The HyPak-MA proposal included Fagerdala as the thermoforming partner. Early in Phase 1, Fagerdala dropped out of the HyPak team as their parent company focused on all its efforts on their molded foam business. After an extensive search to find a suitable custom thermoforming partner with a machine large enough to produce VCEC plates, Pride Polymers was invited to join the team with their extremely large 30”x60” Lyle machine, giving us greater flexibility than with Fagerdala. Pride Polymers coordinated tooling fabrication and installation. Early VCEC prototypes suffered from cracks, but a change in material and tooling modifications resulted in crack-free VCEC modules. DEG designed and procured stainless steel casings to hold each 24” wide VCEC module and to simplify seal design and integration into the Phase 1 prototype.

Task 2: Design and Fabricate Mixed-Air HyPak Prototype

After evaluating a variety of prototype designs, the team reached a consensus on the prototype configuration. DCT then selected the major components with input from DEG, including design of the refrigerant circuit using a tandem compressor arrangement, a unique modulating evaporator coil, and a wet condenser with integral subcooler. DCT prepared a complete submittal (Appendix A) including a layout drawing, which was revised several times after input from stakeholders at DCT and DEG. Once the major component selection and layout were finalized, DCT designed the detailed sheet metal parts, ordered components, and assembled the Phase 1 prototype. The VCEC and the water feed system were the final components installed.

Task 3: Laboratory Test and Refine Prototype

Once the prototype configuration was established, DEG prepared a detailed Test Plan (Appendix B). The Test Plan specified all instrumentation, sensor locations, test conditions, and performance parameters to be evaluated. DCT installed the prototype into their test equipment, some of which was fabricated specially for this test. Engineers from DEG traveled to DCT for the start of testing. During initial testing, we found that the tower fan was not delivering the required flow rate. A larger fan motor solved this problem. We determined that the VCEC was successfully segregating moist and dry air, but water was entering the return duct during certain operating conditions. Water was entering through a damper at the sump and was running through the cabinet walls. The test protocol was modified to keep this damper closed for all test runs. DCT completed a full battery of tests on the Phase 1 prototype. With the exception of the water leakage, we found no other reliability issues. DCT forwarded the test data to DEG for analysis.

Task 4: Evaluate and Report

DEG analyzed the test data from Task 3 and prepared this Phase 1 Report to summarize project activities and report HyPak results and future potential.

Phase 1 Test Results

At 104°F dry-bulb and 72°F wet-bulb outdoor air temperatures, and with 100% ventilation air, the Phase 1 prototype delivered 28.6 tons of cooling capacity and a system EER of 19.3. The ARI 340/360 standard rating “nameplate” capacity and EER were 17.3 tons and 9.1, respectively. However, the ARI rating condition is a poor match for actual design conditions in what are expected to be the primary HyPak markets: buildings in dry climates with average to above-average ventilation air requirements. ARI rating condition has outdoor air conditions of 95°F dry-bulb and 75°F wet-bulb temperatures (about 40% relative humidity), and no ventilation air. Because it uses an evaporative condenser and indirect evaporative ventilation air cooling, HyPak performance is closely tied to ambient wet-bulb temperature, unlike air-cooled equipment, whose performance is linked to ambient dry-bulb temperature. This unfairly handicaps HyPak performance because HyPak is intended for applications with design conditions of less than 20% RH and wet-bulb temperatures that range from 59°F to 71°F. (See table 4-2 for examples of intended design conditions.) This means that HyPak will have much higher performance in real-world applications with 20-100% ventilation air requirements and hot, dry design conditions, than is indicated by the ARI rating condition. Furthermore, air-cooled equipment deliver 5-20% less performance (capacity and EER) at these real-world conditions than their "nameplate" performance would indicate.

The Phase 1 tests results are summarized in the table below. To compare the performance of the HyPak unit to an air-cooled unit, we have included estimated performance for a Trane high-efficiency 20 ton air-cooled RTU in the grey cells. (For more information on how we estimated the Trane performance, see pages 4-5 to 4-7.) Although the HyPak unit trails the air-cooled Trane unit at ARI rating condition, its performance is clearly superior in real-world applications with the design conditions and ventilation airflow rates shown in the table.

		Test	ARI 340 Rating Condition - 0% VA		Western Design Conditions					
	Parameter	Units			20% VA		40% VA		100% VA	
Outdoor Air	Dry-bulb temperature	ºF	**98.1**		**106.5**		**105.3**		**103.5**	
	Wet-bulb temperature	ºF	**78.74**		**71.46**		**73.19**		**71.87**	
	Flow rate (to condenser)	SCFM	4662		4974		4523		5112	
Vent Air (after VCEC)	Dry-bulb temperature	ºF	**85.9**		**78.3**		**81.7**		**72.7**	
	Wet-bulb temperature	ºF	78.2		62.5		65.2		66.3	
	Flow rate	SCFM	197		1578		3120		5200	
Unit			HyPak	Trane	HyPak	Trane	HyPak	Trane	HyPak	Trane
Total power		kW	24.6	25.1	20.9	28.6	20.8	28.3	19.6	26.6
Compressor Cooling		**BTU/hr**	**207,500**	251,200	**205,527**	243,400	**200,812**	248,500	**211,741**	245,800
		tons	**17.3**	20.9	**17.1**	20.3	**16.7**	20.7	**17.6**	20.5
VCEC Ventilation Air Pre-cooling		**BTU/hr**	**0**		**42,721**		**77,711**		**131,932**	
		tons	**0**		**3.6**		**6.5**		**11.0**	
Total Cooling		**BTU/hr**	**207,500**	251,200	**248,248**	243,400	**278,523**	248,500	**343,673**	245,800
		tons	**17.3**	20.9	**20.7**	20.3	**23.2**	20.7	**28.6**	20.5
System EER			**9.1**	10.0	**13.0**	8.5	**14.7**	8.8	**19.3**	9.3

Estimated performance for Trane YCF241C high-efficiency 20 ton air-cooled RTU shown in grey cells.

Tests at 20% and 40% ventilation air and at Western design conditions are much better for understanding HyPak performance in real-world design conditions. In these tests, the VCEC delivered 3.6 and 6.5 tons respectively of indirect ventilation pre-cooling. The refrigeration system produced 17.1 and 16.7 tons respectively, barely reduced from the 17.3 tons at ARI rating condition despite 5-8°F higher OA dry-bulb and the additional of the VCEC dry passage pre-cooling heat. System EERs were 13.0 and 14.7 respectively.

Across all tests with compressor operation, the Phase 1 prototype delivered a wide range of 17.3 to 31.5 tons of total cooling due to the large indirect evaporative cooling contribution made by the VCEC at tests with higher ventilation rates. The VCEC delivered up to 13.9 tons of indirect evaporative cooling. In all tests, the VCEC was able to cool ventilation air to within 2°F of the return air dry-bulb temperature, essentially eliminating the penalty of increased ventilation rates. In most cases, the pre-cooled ventilation air dry-bulb temperature was below that of the return air.

EER and capacity values for conventional air-cooled RTUs are typically lower at Western design conditions because the high ambient dry-bulb conditions increase condensing temperatures. HyPak condensing temperatures are tied to ambient wet-bulb temperatures, rather than dry-bulb temperatures, so capacity and EER of the HyPak prototype *increases* at Western design conditions because the wet-bulb temperatures are lower than in the ARI rating condition.

The VCEC performance far exceeded test results of prior HyPak evaporative heat exchangers. But higher condensing and lower evaporating temperatures resulted in lower compressor efficiency than prior HyPak work. Similarly, fan energy was about higher (per SCFM). In Phase 2 we should be able to reduce total power levels by 15-25%, increasing EERs at all operating conditions.

Looking more closely at VCEC performance without compressor operation shown in the table below, the VCEC shows great potential for use in stand-alone ventilation air coolers. At high ventilation rates, indirect effectiveness remains high and cooling capacity grows nearly linearly.

				Western Maximum		VCEC Evaluation		
		Test Condition						
		DRY/WET fraction		68%	102%	40%	67%	89%
DRY PASSAGES		Flow Rate	SCFM	3077	5224	1945	3184	3991
	IN	**Dry-bulb temperature**	**ºF**	**103.2**	**105.5**	**94.8**	**95.0**	**96.0**
		Wet-bulb temperature	ºF	73.4	74.1	75.7	76.5	76.4
	OUT	**Dry-bulb temperature**	**ºF**	**76.3**	**68.1**	**75.1**	**76.3**	**78.1**
		Wet-bulb temperature	ºF	63.5	63.3	67.6	68.5	69.1
WET PASSAGES		Flow Rate	SCFM	4516	5112	4915	4782	4507
	IN	Dry-bulb temperature	ºF	105.8	84.6	98.9	96.0	98.1
		Wet-bulb temperature	**ºF**	**73.9**	**68.9**	**76.5**	**76.5**	**76.7**
	OUT	Dry-bulb temperature	ºF	82.6	78.0	82.0	82.4	83.2
VCEC ventilation air pre-cooling			**BTU/hr**	**112,439**	**209,068**	**62,288**	**102,248**	**118,741**
			tons	**9.4**	**17.4**	**5.2**	**8.5**	**9.9**
VCEC indirect cooling effectiveness			**%**	**92%**	**102%**	**88%**	**96%**	**84%**

Conclusions and Recommendations

Based on the work completed in Phase 1 to develop HyPak, a high-efficiency rooftop packaged unit, we offer the following conclusions:

- There is a significant opportunity to improve the efficiency of widely-used “mid-sized” (10-50 ton) packaged HVAC units, particularly regarding cooling and ventilation performance, where energy consumption can be cut by more than 50%.

- The major opportunity for improving cooling performance applies evaporative cooling to both the condensing function of the refrigeration cycle and indirectly to ventilation air without moisture addition. This evaporative cooling strategy significantly reduces annual energy consumption, and is even more effective at reducing peak cooling demand.
- The HyPak configuration is stable and production-ready.
- The Vertical Counter-flow Evaporative Cooler (VCEC) delivers most of the energy savings of the HyPak system. Secondary advantages are an innovative two-stage refrigeration system, variable speed supply blower motor, and intelligent controls with real-time feedback of ventilation airflow rate.
- The fan-fold design of the VCEC allows it to be cost-effectively produced with a minimal labor component. Computer simulation resulted in good performance out-of-the-box.
- The highest EERs (18.4-21.3) were recorded at Western design conditions with 100% ventilation air delivery. This is in stark contract to conventional air-cooled RTUs which operate at their lowest efficiencies at design conditions and peak demand periods. The Phase 2 HyPak prototypes will have EERs 2-4 points higher than in Phase 1 due to reduced blower motor and compressor power consumption.
- The performance of the VCEC is excellent, both with and without simultaneous compressor operation. Results without compressor operation included indirect evaporative effectiveness ranged from 84% to 96%. Indirect cooling capacity ranged from 48,660 BTU/hr with 1560 SCFM of dry passage flow, to 209,100 BTU/hr with 5224 SCFM. In both cases, ventilation air was entering the VCEC at over 100°F dry-bulb temperature and leaving at between 63°F and 78.1°F.
- Even with the compressors on, the temperature of the ventilation air was between 67°F and 82°F for all dry passage flow rates. This means that there is nearly no penalty for higher ventilation rates during peak conditions. Furthermore, at most off-peak conditions pre-cooled ventilation air has a lower dew point than return air.
- VCEC heat-sealing must be automated and integrated into the inline thermoforming line to reach cost targets. Wider passages will reduce pressure drop supply fan energy with a minimal reduction in cooling performance, but will require replacing most tooling.
- The innovative refrigeration system needs more testing and development to reduce compressor energy and assess system stability at part-loads. Fan energy levels can be reduced by 25%, increasing EERs.
- Minimizing maintenance costs and demonstrating a substantial reduction in energy costs are essential to success for a new evaporatively-cooled RTU.

Based on these conclusions, we strongly recommend Phase 2 follow-on funding support. In addition, we request that DOE and NETL increase the budget and scope for Phase 2 to support the following additional development efforts:

- *Continued evaluation of the Phase 1 prototype in early Phase 2 to evaluate improved components and to test controls. Fans and motors will be swapped in an effort to reduce fan energy consumption, while maintaining adequate flow rates. This will allow the team to continue development of the "balance of system" at DCT, when it would otherwise be put on hold as VCEC development in Task 2.1 is expected to take considerably longer* *than prototype design and fabrication in Task 2.2. This also increases the likelihood that the Phase 2 prototypes will have high performance and reliability out of-the-box, which will in turn increase the likelihood of commercialization immediately following the conclusion of Phase 2.*

- *Further VCEC development. Although the Phase 1 VCEC prototypes worked well, it is clear that developing a commercialization-ready VCEC module will require greater resources than anticipated. At least half of the tooling used in Task 1.1 will require replacement, and the custom fabrication work needed to automate the heat sealing process will be substantial. The VCEC is clearly the make-or-break component of the HyPak system, with low-cost production the strongest factor in its success.*
- *Controls development for intelligent operating mode selection, and to deliver the maximum ventilation airflow rate possible without reducing efficiency). The controller will provide real-time feedback of ventilation airflow rate. A web-based interface will be developed for maximum flexibility, user-friendliness, and remote monitoring and fault diagnosis.*

2. Introduction

2.1. Background

More than half of all U.S. commercial floor space is cooled by packaged units, most of which are rooftop units (RTUs).[3] Pacific Gas & Electric Company (PG&E) reports that of 3.5 million tons of commercial cooling capacity in its service territory, two-thirds is provided by packaged units.[4] There are strong reasons for the popularity of these units. RTUs are inexpensive, they provide a measure of zonal control, are easy to install, can be serviced without disrupting occupants, and are familiar to contractors, engineers and operators. Central systems with chillers and boilers are generally more efficient than RTUs, but have been losing market share due to higher first costs and greater complexity.

2.1.1. The Problem

Today's packaged units are inefficient. Their rooftop location exposes them to elevated temperatures that increase ventilation air cooling loads and reduce refrigeration efficiency. Conventional RTUs have a single speed blower motor that uses the same amount of power whether it's satisfying part loads or peak loads. "Direct expansion" evaporator coils with uncontrollable surface temperatures often cause unnecessary latent cooling, and thin coils with closely-spaced fins collect dust and bacterial growth and produce downstream moisture in plenums and ducts due to "pull-off" of condensate droplets that bridge the narrow gap between the fins.

Inefficient RTUs have major consequences. Buildings account for 36% percent of annual U.S. energy use and two-thirds of electricity use[1]. Commercial buildings consume approximately 4.2 quads of energy each year at a cost of $230 billion per year, with commercial HVAC equipment consuming 1.2 quads of electricity[1,2]. Energy consumption for buildings generates 35% of all CO_2 emissions; an estimated 10% of this amount is due to RTUs[5]. This output contributes significantly to global warming. Thus, improvements in equipment efficiency can have major impact on overall energy consumption and global warming.

Another problem with rooftop units is that they often fail to maintain adequate indoor air quality. Inadequate ventilation is one cause of poor air quality with RTUs, which have no system for monitoring the ventilation air rate. Many RTUs are equipped with economizers to deliver additional outdoor air in favorable conditions, but economizer operation is often faulty due to poor design, lack of maintenance, or insufficient exhaust air. A California field survey found that only 16% of RTUs met minimum required ventilation rates[6]. Lower ventilation rates are associated with increased respiratory illnesses and a worsening in perceived air quality[7]. Inadequate filtration is another cause of poor air quality with RTUs, whose filters typically have low efficiencies for particles smaller than ~2 micrometers. Studies have found increased particle concentrations to be associated with respiratory and cardiovascular deaths, hospital admissions, asthma, respiratory symptoms, and diminished lung function[8]. With better HVAC filters, indoor small particle concentrations could be maintained 75% lower than outdoor air[9].

2.1.2. The HyPak-1 Project

This is the second HyPak project to be supported by DOE through NETL. The prior project, referred to as HyPak-1 in this report, had two rounds of prototype fabrication and testing as well as computer modeling and market research. The HyPak-1 prototypes demonstrated the high performance capabilities of the HyPak concept, but made it clear that further development was required to reduce heat exchanger cost and improve system reliability before HyPak commercialization can commence. The HyPak-1 prototypes were limited to about 25% ventilation air fraction, limiting performance and marketability. The current project is intended to develop a "mixed-air" product that is capable of full 0-100% modulation in ventilation air fraction; hence it was referred to as HyPak-MA in the proposal.

2.1.3. The HyPak-MA Project

The overriding objective of the HyPak-MA Project is to design, develop and field test a high-efficiency rooftop unit that provides a quantum improvement over conventional RTU performance. Our proposal targeted 60% and 50% reduction in electrical annual energy use by the HyPak RTU for dry and humid climates, respectively, when compared with a conventional unit, and reduction in peak energy consumption of 50% and 33% respectively. An additional objective is the development of an evaporative heat exchanger (Vertical Counter-flow Evaporative Cooler or VCEC) that efficiently cools water and ventilation air in a single module. We aim to accomplish these goals while improving indoor air quality. (For simplicity, the -MA has been dropped in this report when referring to the current project.)

2.1.4. The HyPak Team

The HyPak team includes three organizations:

- Davis Energy Group (DEG) – overall management and design, test planning, instrumentation selection and setup;
- Des Champs Technologies, Inc. (DCT), recently acquired by Munters – detailed design and manufacturing of prototypes, laboratory testing; and
- Pride Polymers – VCEC thermoforming and fabrication.

Under the anticipated commercialization scenario, DCT will manufacture and market HyPak prototypes. Pride Polymers will manufacture the VCEC modules for Advanced Energy Products (AEP), a recent spin-off of DEG, who will supply them to DCT for HyPak production. AEP will also market HyPak in the West and work to secure utility incentive and demonstration programs. AEP will also make the VCEC heat exchanger available to other HVAC manufacturers as well.

2.1.5. Phase 1 Goals

The three Phase 1 tasks were structured to result in design, fabrication, and laboratory testing of an improved HyPak prototype capable of delivering 0-100% ventilation air, with full modulation in between. A separate task ensured particular emphasis was placed on development of the low-cost thermoformed plastic VCEC module.

2.2. Phase 1 Accomplishments

In Phase 1 we:

- Investigated several possible configurations for the HyPak prototype
- Reached consensus on the selected configuration
- Designed a VCEC fan-folded evaporative heat exchanger using CFD software
- Developed inline thermoforming tooling to produce the VCEC prototype modules
- Fabricated four VCEC modules with manually heat-sealed edges
- Completed detailed design work and component selection for a 20-ton prototype
- Fabricated the prototype using the VCEC modules
- Laboratory tested the prototype under a variety of cooling conditions
- Completed this report

2.3. HyPak Design Concept

The HyPak design concept consists of:

- a unique VCEC evaporative heat exchanger that combines direct evaporative cooling of condenser water with indirectly evaporative cooling of building ventilation air into a single low-cost module;
- a novel modulating (two-stage) R-410A refrigeration system with an evaporatively cooled condenser and tandem scroll compressors;
- variable speed supply blower;
- innovative controls that:
    - optimize operating mode selection to select the most efficient operating mode by taking into account the dry-bulb, wet-bulb enthalpy and/or dew point of return air and outdoor air,
    - provide real-time adjustment and feedback of ventilation airflow rate via web-based interface,
    - coordinate operation of subsystems to avoid failures such as evaporator freeze-up, and
    - include basic automated fault diagnosis;
- hydronic heat delivery combined with a high-efficiency, variable-capacity tankless water heater;
- high efficiency two-stage air filtration; and
- true economizer operation.

Conceptually, HyPak works as shown in Figure 2-1.

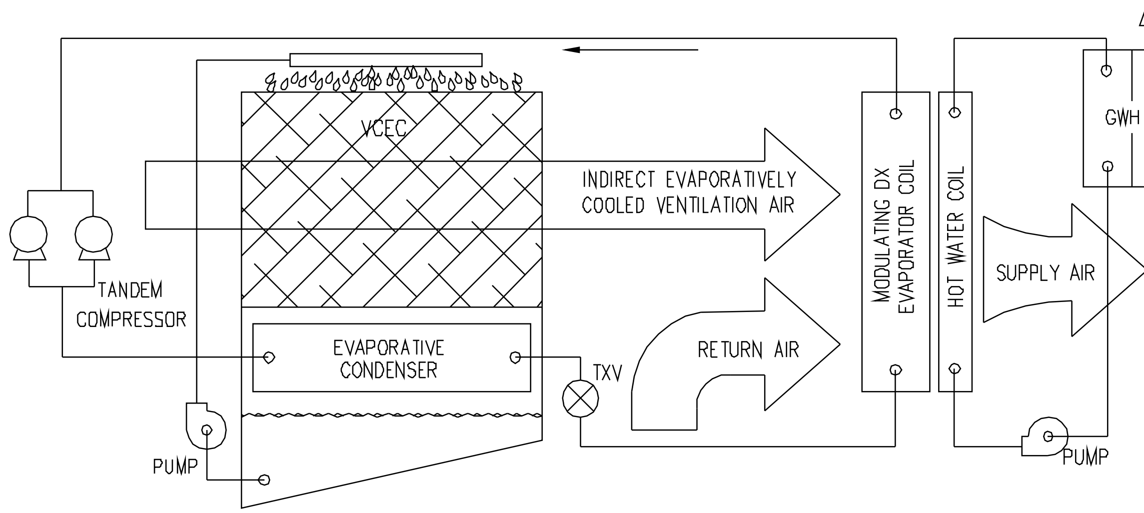

Figure 2-1: HyPak Phase 1 Prototype Schematic

2.4. HyPak Design Features

The HyPak design developed in this project has several specific advantages over conventional RTUs that allow it to meet equivalent building cooling loads with reductions in annual electrical energy consumption up to 60% and reduce peak energy consumption by up to 50%.

- The water-cooled condenser combined with the VCEC effectively de-couples the condensing temperature from the ambient dry bulb temperature and allows the refrigerant system to operate much more efficiently than air-cooled condensers, especially in dry climates. Evaporative condensers increase capacity and reduce compressor power.
- Indirect evaporative pre-cooling of the ventilation air in the VCEC reduces compressor load significantly during peak demand hours, allowing for smaller compressors. This is particularly effective in applications that require significant amounts of ventilation air, such as schools, retail outlets and restaurants.
- The VCEC is a plate-type air-to-air heat exchanger consisting of alternating dry and wet passages. Because ventilation air passes only through the sealed dry passages, *no moisture is added to the conditioned space*, hence the term *indirect* evaporative cooling.
- During periods with low ambient temperatures and modest cooling loads, the HyPak controls operate in normal full-bypass economizer mode.
- During part-load conditions with higher cooling loads and warmer ambient temperatures, the HyPak controls energize the exhaust blower and pump. As in full-bypass mode, return air is not recirculated, but damper positions are changed to use the dry passages of the VCEC for indirect evaporative cooling. The excellent performance of the VCEC means that indirect evaporative cooling is often sufficient to meet substantial cooling loads without compressor operation, extending semi-economizer operation to reduce energy consumption.
- A variable speed supply blower maximizes efficiency in part-load conditions.
- A weir-type water distribution system instead of high pressure drop spray nozzles.
- The refrigeration system:

  - Uses pre-packaged tandem compressors to reduce cost;
  - Uses a single circuit to take full advantage of the large condenser and evaporator heat exchangers during single-compressor operation (conventional RTUs use multiple circuits in parallel to reduce complexity at the expense of part-load performance);
  - Uses an oversized evaporator aircoil to:
    - increase evaporative temperature to increase compressor efficiency, and
    - decrease latent fraction to eliminate unnecessary dehumidification in dry climates.
- A web-based control interface to improve user-friendliness when adjusting setpoint and occupancy schedules, provide real-time ventilation airflow rate feedback, and allow remote diagnostics.

At the same time, HyPak delivers higher indoor air quality with a two-stage air filtration system for minimal pressure drops and long replacement intervals. A 25% efficient pre-filter cleans outdoor air before it enters the VCEC dry passages, reducing main filter loading by removing larger particles, followed by a 65% efficient main filter that cleans all supply air. Furthermore, the VCEC nearly eliminates the energy penalty normally associated with cooling ventilation air, encouraging operators to supply more fresh air to the building. The intelligent control algorithm will deliver more ventilation air than the specified minimum whenever possible without increasing energy consumption or reducing system efficiency.

4. Results and Discussion

4.1. Test Results

We collected laboratory test data in April and May 2007. DCT staff ran the tests, collected data, and forwarded data to DEG for all analysis. All data included in this report is after the minor modifications made to the prototype, as described in Section 3.4.4. The laboratory test results are broken down into three areas of emphasis.

- *Overall system performance* discusses the capacity and efficiency of the entire HyPak Phase 1 prototype.
- *VCEC performance* discusses the indirect cooling capacity and effectiveness of this critical component.
- *Refrigeration performance* discusses the capacity and efficiency of the refrigeration sub-system.

Selected representative test data is presented and discussed in this section. For complete Phase 1 test data, refer to Appendix C.

4.1.1. Overall System Performance

Table 4-1 summarizes test data for seven test runs with both compressors operating.

Lines 6 and 7 show the condition of outdoor air entering the unit for each test. Dry-bulb temperatures were within 3°F of the targets shown in Table 3-1. Wet-bulb temperatures, which have a greater impact on the relevance of evaporative performance testing, were off by even more, up to 6°F too high for test 14A. In all cases, the wet-bulb temperature was at least 1.5°F warmer than the target, reducing performance.

In the test setup shown in Figures 3-30, outdoor air was conditioned and then split into two branches, one for the VCEC dry passage ventilation inlet and one for the condenser and VCEC wet passage inlet. Because of lower ambient shop temperatures and non-insulated ducts and transition boxes, the VCEC dry passage ventilation inlet air was usually warmer than the condenser inlet. Lower VA flowrates exacerbated this behavior, with good agreement in 40% VA tests (6A, 9A) and there was even some reversal for 100% VA tests (6E, 14A). Because this behavior was caused by sensible heat addition, we assumed that both airstreams had identical dew points, regardless of dry-bulb temperature. Data shown in line 6 and 7 are for the condenser inlet. (Both data points are shown in Appendix C.)

Condenser and VCEC wet passage flow rates (line 8) were slightly less than our target of 6000 SCFM due to the high pressure drop of the VCEC wet passages.

Table 4-1: Laboratory Testing Results

1			Test Condition	1	5A	6A	6E	8A	9A	14A
2				ARI 340	Western Maximum			Western Summer		West. Shoulder
3	Group	Parameter	Units	Standard Rating	20% VA	40% VA	100% VA	20% VA	40% VA	100% VA
4		Date/Time		4/27/2007 9:23	4/20/2007 7:44	4/25/2007 10:20	4/26/2007 10:31	4/19/2007 9:16	4/25/2007 14:27	4/26/2007 14:20
5		Compressors		both ON	both ON	both ON	both ON	both ON	both ON	both ON
6	Outdoor Air	Dry-bulb temperature	**ºF**	**98.1**	**106.5**	**105.3**	**103.5**	**95.8**	**96.7**	**85.5**
7		Wet-bulb temperature	**ºF**	**78.7**	**71.5**	**73.2**	**71.9**	**69.4**	**70.5**	**67.9**
8		Flow rate (to condenser)	SCFM	4662	4974	4523	5112	4199	2922	5112
9	Vent Air (after VCEC)	Dry-bulb temperature	**ºF**	**85.9**	**78.3**	**81.7**	**72.7**	**77.4**	**77.9**	**67.6**
10		Wet-bulb temperature	ºF	78.2	62.5	65.2	66.3	63.0	64.3	63.8
11		Flow rate	SCFM	197	1578	3120	5200	1572	2862	5322
12	Return Air	Dry-bulb temperature	ºF	82.2	79.9	79.1	74.9	76.3	76.7	77.7
13		Wet-bulb temperature	ºF	67.9	67.5	65.7	68.7	63.2	64.3	64.4
14		Flow rate	SCFM	7448	4735	3375	5531	4697	6109	5522
15	Supply Air	Dry-bulb temperature	ºF	64.9	59.7	60.5	61.0	58.0	60.4	58.1
16		Wet-bulb temperature	ºF	61.9	58.3	58.3	59.0	56.1	57.6	56.5
17		Flow rate	SCFM	7645	6313	6495	5200	6269	8971	5322
18	Refrigeration	Evaporating temp	ºF	49.0	46.1	45.9	45.2	43.7	42.2	42.5
19		Superheat	ºF	25.9	29.3	28.0	27.0	25.1	25.9	25.7
20		Condensing temp	ºF	112.9	106.4	109.5	99.7	106.2	106.6	96.5
21		Subcooling	ºF	2.2	1.2	1.6	0.7	-0.3	8.2	2.3
22		Compressor EER		14.2	15.1	14.4	16.8	14.5	14.8	17.2
23		**Compressor cooling**	btu/hr	207,500	205,527	200,812	211,741	194,934	199,790	208,802
24			**tons**	**17.3**	**17.1**	**16.7**	**17.6**	**16.2**	**16.6**	**17.4**
25	Power	Compressor A power	kW	7.30	6.81	6.99	6.33	6.78	6.80	6.08
26		Compressor B power	kW	7.27	6.78	6.95	6.27	6.70	6.71	6.05
27		Supply fan power	kW	6.93	4.17	3.86	3.41	4.73	4.23	3.46
28		Exhaust fan power	kW	2.46	2.62	2.49	3.14	1.86	2.63	3.18
29		Total power	kW	24.57	20.89	20.80	19.60	20.59	20.86	19.26
30	VCEC/Condenser water flow rate		GPM	10.0	10.0	10.0	10.0	10.0	10.0	10.0
31	**VCEC Ventilation Air Pre-cooling**		BTU/hr	0	42,721	77,711	131,932	33,467	78,869	111,843
32			**tons**	**0**	**3.6**	**6.5**	**11.0**	**2.8**	**6.6**	**9.3**
33	**Total Cooling Capacity**		BTU/hr	207,500	248,248	278,523	343,673	228,400	278,659	320,645
34			**tons**	**17.3**	**20.7**	**23.2**	**28.6**	**19.0**	**23.2**	**26.7**
35	**System EER**			**9.1**	**13.0**	**14.7**	**19.3**	**12.2**	**14.6**	**18.4**

The dry-bulb and wet-bulb temperatures of the VA (lines 9 and 10) were close to or below those for RA (lines 12 and 13) for all tests, indicating that most, or all, of the ventilation air cooling load was handled by the VCEC.

RA dry-bulb temperature (line 12) was within 2.2°F of our targets for all tests except 6E, where it was 5.1°F too cool. RA wet-bulb temperature (line 13) was within 2°F for all tests.

The VA flow rate (line 11) was close to our targets for the 20% and 40% VA tests (5A, 6A, 8A, 9A). The 100% VA tests only reached 5200 SCFM and 5300 SCFM, about 13% below our target, due to the high pressure drop of the VCEC dry passages at high flow rate.

The mass flow meter and refrigeration pressure and temperature sensors performed flawlessly to deliver accurate measurement of the capacity of the refrigeration system. Combining them with the real-time refrigerant enthalpy calculations in the Labview interface made short work of charging, troubleshooting and assessing the performance of the refrigeration system. The refrigeration system capacity and other parameters shown in lines 18 to 24 were taken directly from the Labview output without the need for post-processing (with some spot-checking for accuracy). See Section 4.1.4 for more information about the refrigeration system performance.

Power measurements in lines 25 through 29 were taken directly from watt transducers. Not shown are ancillary power values for controls, dampers and other smaller components, which were between 440 and 610 watts for all tests.

At the start of the Phase 1 testing, we tried varying the water flow rate through the condenser and VCEC. As in earlier HyPak testing, this made no appreciable difference in refrigerant condensing temperature. This is in contrast to cooling towers, which typically operate with 3 GPM per ton of capacity. Excessive water is used to reduce scale in cooling towers and to improve distribution, but the amount of water required for evaporation is much less. For HyPak we are able to get away with only 0.5 GPM per ton (or less) because of the efficient water distribution system. In addition, a thick layer of water on the wet passage walls has been shown to reduce performance in earlier HyPak testing and other indirect evaporative heat exchanger testing. This is because the thick layer resists heat transfer between the evaporating water and the dry air on the other side of the plastic plate.

VCEC ventilation air pre-cooling (lines 31, 32) was calculated from the VA flow rate and change in enthalpy across the VCEC dry passages. Because the only OA RH sensor was located at the condenser inlet, to determine the wet-bulb and enthalpy at the dry passage inlet, where there was only a thermocouple grid, we had no choice other than to assume that both the OA and VA inlets had identical dew point temperatures. This is a reasonable assumption because there is no addition or removal of moisture between the point where these two air streams split and where they entered the prototype. (This assumes no duct leakage.)

Total cooling capacity (lines 33, 34) is the mechanical cooling (lines 23, 24) combined with the VCEC ventilation air pre-cooling (lines 31, 32). System EER is the total cooling capacity (line 33) divided by the total power (line 29).

The ARI 340 "nameplate" capacity and EER were 207,500 BTU/hr (17.3 tons) and 9.1, respectively. (One ton of cooling is equivalent to 12,000 BTU/hr.) Although this EER is roughly equivalent to the nameplate EER of standard efficiency air-cooled RTUs, and about 1.5 points below that of a high-efficiency air-cooled RTU, the ARI 340/360 rating condition is a poor match for actual design conditions in what are expected to be the primary HyPak markets: buildings in dry climates with higher-than-average ventilation air requirements.

ARI 340/360 rating conditions are 95°F dry-bulb and 75°F wet-bulb temperatures. This is equivalent to 40% humidity at 95°F, which is much more humid than design conditions in the Central Valley of California, Las Vegas, Phoenix, Salt Lake City, and most of

Southern California. In addition, the dry-bulb temperature is well below design conditions on those climates. Selected 1% design conditions are shown in Table 4-2.

Table 4-2: Primary HyPak Market Design Conditions

Location	1% Dry-Bulb Design Condition	Coincident Wet-Bulb	Relative Humidity
ARI 340/360 Standard Rating Condition	95	75	40%
Sacramento	101	70	21%
Bakersfield	104	70	18%
Redding	105	68	14%
Riverside	100	68	19%
Palm Springs	112	71	12%
Las Vegas	108	66	9%
Reno	96	61	10%
Phoenix	109	71	15%
Salt Lake City	97	62	11%
Denver	93	59	10%

As can be seen from Table 4-2, design conditions in target HyPak markets range from 9% to 21% relative humidity, well below the 40% RH in ARI 340/360 standard rating condition. Although the ARI 340/360 "nameplate" rating capacity and EER are a convenient way to compare various types of HVAC systems irrespective of climate, this "one size fits all" rating condition is of limited utility when understanding how well these systems will perform in real-world applications. As the other test runs show, *the performance of the HyPak unit increases (sometimes as much as 112%) from ARI rating conditions to the Western design applications shown in tests 5A, 6A, 6E, both in terms of capacity and EER.* This contrasts with air-cooled equipment, which typically loses 5-10% in capacity, and 10-20% in EER. Performance of air-cooled equipment is tied to dry-bulb temperatures, unlike HyPak, which is driven by wet-bulb temperatures. The wet-bulb temperatures at design conditions (which typically coincide with peak regional annual energy consumption) in Table 4-2 average 8.4°F lower than the ARI rating condition wet-bulb temperature of 75°F.

Tests 5A and 6A (Western Maximum + 20% and 40% VA) are much better for understanding HyPak performance in real-world design conditions. In these tests, the VCEC delivered 42,700 BTU/hr and 77,700 BTU/hr respectively of indirect ventilation pre-cooling. The refrigeration system produced 205,500 BTU/hr and 200,800 BTU/hr respectively, barely reduced from the 207,500 BTU/hr at ARI despite 5-8°F higher OA dry-bulb and the addition of the VCEC dry passage pre-cooling heat. System EERs were 13.0 and 14.7 respectively.

Test 6E demonstrates the ability of the HyPak prototype to deliver very high levels of indirect evaporative cooling and high system EER. *At 104°F dry-bulb and 72°F wet-bulb and with 100% ventilation air, the Phase 1 prototype delivered 343,800 BTU/hr (28.6 tons) of cooling capacity and a system EER of 19.3.*

Across all tests with compressor operation, the Phase 1 prototype delivered between 17.3 and 28.6 tons of total cooling (207,500-343,800 BTU/hr). Compressor cooling capacity was relatively level between 16.2 and 17.6 tons (194,900-211,700 BTU/hr). The cause of the wide range in total cooling capacity is the large indirect evaporative cooling contribution made by the VCEC at tests with higher ventilation rates. The VCEC delivered up to 11.0 tons (131,900 BTU/hr) of indirect evaporative cooling. In all tests, the VCEC was able to cool ventilation air to within 2°F of the return air dry-bulb temperature, essentially eliminating the penalty of increased ventilation rates. In most cases, the pre-cooled ventilation air dry-bulb temperature was below that of the return air.

EER values ranged from 9.1 to 19.3. EER values for conventional air-cooled RTUs are typically lower at Western design conditions because the high ambient dry-bulb conditions increase condensing temperatures. Because Western dry-bulb design conditions are typically 100-110°F, the capacity of air-cooled RTUs is typically de-rated by 5-15% as a result of higher condensing temperatures than at the 95°F dry-bulb temperature of the ARI standard rating condition. Because condensing temperatures in evaporative condensers are tied to ambient wet-bulb temperatures, rather than dry-bulb temperatures, capacity and EER of the HyPak prototype increases at Western design conditions because the wet-bulb temperatures (66-72°F) are lower than in the ARI standard rating condition (75°F). *Improvements to the HyPak design in Phase 2 will focus on lowering power consumption, and should result in EERs about 2-3 points higher than those presented in this report.*

For conventional RTU applications with substantial ventilation air fraction, the air passing over the evaporator coil is warmer, increasing evaporating temperature slightly, with a small increase in EER and capacity. However, this is a “phantom” benefit because the warmer air also increases supply air temperature, reducing the compressor capacity available to satisfy the cooling load of the conditioned space. *For HyPak, the VCEC is able to handle the entire ventilation air cooling load, leaving all of the compressor capacity available to satisfy the conditioned space cooling load.*

4.1.2. Comparison to Air-Cooled RTU Performance

In Phase 2 we hope to field test a HyPak prototype side-by-side with a high-efficiency air-cooled RTU for baseline comparisons. Until then, the only way to evaluate the advantage of the HyPak system is to estimate the performance of an air-cooled unit at the HyPak test conditions.

Although they are #2 in the marketplace behind Carrier, we believe that Trane Company makes the highest quality air-cooled packaged RTUs on the market. We selected a high-efficiency 20 ton R-22 Trane Voyager unit, which will be one of the primary competitors to HyPak. Trane supplies detailed fan power and compressor capacity information for this unit at a variety of operating conditions. Using a psychrometric analysis, we calculated what the mixed-air conditions entering the coil of a high efficiency Trane unit would be using the flow rates and conditions of return air and entering ventilation air. Together with the outdoor air dry-bulb entering the condenser, we interpolated the compressor capacity from tables in the Trane specification manual for this RTU. Supply fan energy was also interpolated from tables in the specification manual. We assumed condenser fan energy to be 1.5 kW, on the low side for the two 1.0 BHP motors. Trane does not supply detailed power consumption tables for the compressor power, so we had to estimate this. However, Trane does provide total power consumption at ARI rating condition, making it possible to accurately estimate the compressor power at ARI rating condition. Trane also supplies basic information about the condenser and evaporator coils. From this relatively complete data set at ARI rating conditions, we were able to accurately estimate the condensing and evaporating temperatures using a compressor spreadsheet for a Copeland scroll compressor. (Trane makes their own scroll compressors, but Copeland are considered the industry standard for performance and reliability.) We could then estimate the condensing and evaporating temperatures at HyPak test conditions with a reasonably high degree of certainty. Plugging these into the Copeland spreadsheet, we could estimate the compressor power at each test condition, checking each test condition against the capacity interpolated from the Trane data. The results of this analysis are shown in Table 4-3, with estimated Trane RTU performance shown in yellow.

Table 4-3: Estimated Trane RTU Performance at HyPak Test Conditions

1			Test Condition	1	TRANE		5A	TRANE	6A	TRANE	8A	TRANE	9A	TRANE
2				ARI 340/360 Standard Rating Condition			Western Maximum				Western Summer			
3	Group	Parameter	Units				20% VA		40% VA		20% VA		40% VA	
4		Date/Time/Notes		4/27/2007	From YCD241C Spec[1]		4/20/2007	YCD241C	4/25/2007	YCD241C	4/19/2007	YCD241C	4/25/2007	YCD241C
5				AM	@ T1 cond.[2]	At SRC[3]	7:44 AM	Spec[1]	10:20 AM	Spec[1]	9:16 AM	Spec[1]	2:27 PM	Spec[1]
6	Outdoor Air	Dry-bulb temperature	°F	98.1		95.0	106.5		105.3		95.8		96.7	
7		Wet-bulb temperature	°F	78.7		75.0	71.5		73.2		69.4		70.5	
8		Flow rate (to condenser)	SCFM	4662	13,700	13,700	4974	13,700	4523	13,700	4199	13,700	2922	13,700
9	Vent Air (after VCEC)	Dry-bulb temperature	°F	85.9			78.3		81.7		77.4		77.9	
10		Wet-bulb temperature	°F	78.2			62.5		65.2		63.0		64.3	
11		Flow rate	SCFM	197			1578		3120		1572		2862	
12	Return Air	Dry-bulb temperature	°F	82.2		80.0	79.9		79.1		76.3		76.7	
13		Wet-bulb temperature	°F	67.9		67.0	67.5		65.7		63.2		64.3	
14		Flow rate	SCFM	7448		7000	4735		3375		4697		6109	
15	Mixed Air Before Coil	Dry-bulb temperature	°F	82.2	82.2	80.0	79.5	84.1	80.4	89.7	76.5	79.8	77.1	82.6
16		Wet-bulb temperature	°F	67.9	67.9	67.0	66.3	68.5	65.4	69.3	63.1	64.8	64.3	66.3
17		Flow rate	SCFM	7645		7000	6313		6495		6269		8971	
18	Refrigera-tion	Evaporating temp	°F	49.0	48.0	47.0	46.1	49.0	45.9	50.0	43.7	47.0	42.2	48.0
20		Condensing temp	°F	112.9	132.0	130.0	106.4	137.0	109.5	136.0	106.2	133.5	106.6	131.0
22		Compressor EER		14.2	12.3	13.2	15.1	10.1	14.4	10.5	14.5	10.5	14.8	11.3
23		**Mechanical cooling**	**btu/hr**	**207,500**	**251,200**	**250,000**	**205,527**	**243,400**	**200,812**	**248,500**	**194,934**	**241,500**	**199,790**	**252,000**
24	Power	Compressor A power	kW	7.30	10.25	9.50	6.81	12.00	6.99	11.85	6.78	11.50	6.80	11.15
25		Compressor B power	kW	7.27	10.25	9.50	6.78	12.00	6.95	11.85	6.70	11.50	6.71	11.15
26		Supply fan power	kW	6.93	3.06	3.06	4.17	3.06	3.86	3.06	4.73	3.06	4.23	3.06
27		Exhaust fan power	kW	2.46	1.50	1.50	2.62	1.50	2.49	1.50	1.86	1.50	2.63	1.50
28		Total power	kW	24.57	25.06	23.56	20.89	28.56	20.80	28.26	20.59	27.56	20.86	26.86
29	VCEC/Condenser water flow rate		GPM	10.0			10.0		10.0		10.0		10.0	
30	VCEC Ventilation Air Pre-cooling		BTU/hr	0			42,721		77,711		33,467		78,869	
31	**Total Cooling**		**BTU/hr**	**207,500**	**251,200**	**250,000**	**248,248**	**243,400**	**278,523**	**248,500**	**228,400**	**241,500**	**278,659**	**252,000**
32	**System EER**			**9.1**	**10.0**	**10.6**	**13.0**	**8.5**	**14.7**	**8.8**	**12.2**	**8.8**	**14.6**	**9.4**

[1] Trane gas/electric high-efficiency RTU 20 ton nominal capacity, downflow, with TXV and 2" pleated filters, data from specification manual, directly or estimated/interpolated

[2] Estimated results if Trane RTU was tested at exactly the same conditions at the HyPak Test 1, which attempted to match ARI 340 Standard Rating Conditions

[3] Values at ARI 340 Standard Rating Conditions

As Table 4-3 shows, the performance of the Trane RTU is driven by two factors: condensing temperature (which is closely related to outdoor air dry-bulb temperature) and evaporating temperature (which is closely related to the mixed air dry-bulb and wet-bulb temperatures). With the cooler return air temperatures of tests 8A and 9A, the evaporating temperature drops, taking compressor capacity and EER with it. At the high outdoor air temperatures of Test 5A and 6A, condensing temperature goes way up, driving down the Trane's compressor capacity and EER. The Trane RTU compressor capacity remains within a 10,000 BTU/hr band for all conditions, but the HyPak unit benefits from the indirect evaporative pre-cooling capacity of the VCEC, and its capacity goes up with outdoor air and/or ventilation airflow rates, which conveniently corresponds to the load placed upon it.

The critical comparison is shown in red. These are the total cooling capacities for both the HyPak and Trane unit at typical Western design conditions in real-world zones with moderate ventilation airflow rates. (The Trane specification manual uses a 40% VA application in one of their sizing examples.) Engineers specifying an RTU for real-world applications in the markets shown in Table 4-2 will immediately recognize that the HyPak unit has greater capacity than the Trane unit at their design conditions, regardless of the ARI nameplate capacity. Of less importance to engineers, but of great importance to utilities and regulatory agencies, is the roughly ***50% higher EER values for the HyPak unit at design conditions.*** (Furthermore, EERs for production HyPak units are expected to be 2-3 points higher than those shown in this report as fan power will be less.)

This comparison clearly highlights the shortcomings of the ARI rating condition for real-world applications in dry Western climates. It also shows the main strength of HyPak: that its capacity is at or near it maximum during design conditions, when it is needed most. For utilities and agencies seeking to reduce peak demand, the similar behavior of HyPak EER will be highly attractive. This is in stark contrast to air-cooled equipment, whose capacity and EER are at their lowest when they are needed most, leaving them unable to satisfy cooling loads at design conditions, and resulting in oversized equipment and short-cycling at part-load conditions.

4.1.3. Comparison to 2004 Testing of HyPak-1 Prototype

During the laboratory testing in Phase 3 of the HyPak-1 project in 2004, EER values were much more consistent across all test runs. There are two explanations for this.

- Power levels were much lower in 2004 than in 2007, even when accounting for the larger size of the 2007 prototype. Total power increased from about 5.5 kW to about 21 kW.
    - Supply fan energy increased from about 750 W in 2004 to about 5 kW. (Blower motor size was increased from 1 BHP to 10 BHP. The blower type was changed from a plug fan to a centrifugal blower to reduce cabinet size and work better with high pressure drop duct systems.)
    - Exhaust fan energy increased from about 650 W in 2004 to about 2.5 kW. (Blower motor size was increased from ¾ BHP to 5 BHP. The blower type was changed from a propeller fan to a centrifugal blower to overcome wet passage pressure drop.)
    - Higher compressor power levels are discussed in detail in Section 4.1.4.
- The VCEC provided much higher levels of indirect evaporative cooling in 2007 than the CEWC used in the 2004 prototype. For tests with high levels of ventilation air (6E and 14A), the 2007 Phase 1 HyPak prototype was able to overcome the high power levels and generate higher system EERs than during any test run of 2004.

One of the major problems with the 2004 prototype was insufficient fan power. This generated high EERs, but would have been a problem with most duct systems. The 2007 Phase 1 prototype has enough fan power to satisfy all but the largest 20 ton duct systems. A comparison of 2004 and 2007 HyPak test data is shown in Table 4-4.

Table 4-4: Comparison of Phase 1 Data with 2004 HyPak-1 Data

1			Test Condition	2004	2007	2004	2007	2004	2007
2				ARI 340 Standard Rating		Western Maximum		Western Summer	
3	Group	Parameter	Units		1		5A		8A
6	Outdoor Air	Dry-bulb temperature	**ºF**	**96.4**	**98.1**	**104.6**	**106.5**	**2188.9**	**95.8**
7		Wet-bulb temperature	**ºF**	74.4	**78.7**	71.1	**71.5**	**95.6**	**69.4**
8		Flow rate (to condenser)	SCFM	**2183**	4662	**2243**	4974	75	4199
9	Vent Air (after VCEC)	Dry-bulb temperature	**ºF**	79.6	**85.9**	79.5	**78.3**	78.9	**77.4**
10		Wet-bulb temperature	ºF	70.6	78.2	66.2	62.5	71.3	63.0
11		Flow rate	SCFM	**0**	197	**363**	1578	**430**	1572
12	Return Air	Dry-bulb temperature	ºF	80.3	82.2	78.7	79.9	75.8	76.3
13		Wet-bulb temperature	ºF	65.7	67.9	64.1	67.5	63.8	63.2
15	Supply Air	Dry-bulb temperature	ºF	60.7	64.9	58.9	59.7	59.6	58.0
16		Wet-bulb temperature	ºF	58.4	61.9	56.5	58.3	56.3	56.1
17		Flow rate	SCFM	2987	7645	2997	6313	3005	6269
18	Refrigeration	Evaporating temperature	ºF	51.0	49.0	52.0	46.1	49.0	43.7
19		Superheat	ºF	6.8	25.9	4.7	29.3	5.7	25.1
20		Condensing temperature	ºF	112.0	112.9	102.0	106.4	100.0	106.2
21		Subcooling	ºF	26.3	2.2	18.7	1.2	16.9	-0.3
22		Compressor EER		23.3	14.2	22.6	15.1	22.3	14.5
23		**Mechanical cooling**	**btu/hr**	**87,762**	**207,500**	**83,392**	**205,527**	**85,010**	**194,934**
24	Power	Compressor A power	kW	3.76	7.30	3.69	6.81	3.82	6.78
25		Compressor B power	kW		7.27		6.78		6.70
26		Supply fan power	kW	0.73	6.93	0.80	4.17	0.82	4.73
27		Exhaust fan power	kW	0.68	2.46	0.65	2.62	0.66	1.86
28		Total power	kW	5.48	24.57	5.47	20.89	5.62	20.59
29	Sump temperature		ºF	79.5	91.6	77.9	87.1	75.5	88.3
30	Sump approach		ºF	5.1	12.8	3.4	15.7	4.4	18.9
31	VCEC/Condenser water flow rate		GPM	18.0	10.0	18.0	10.0	18.0	10.0
32	**VCEC Ventilation Air Pre-cooling**		**BTU/hr**	**0**	**0**	**6,369**	**42,721**	**5,405**	**33,467**
33	**Total Cooling**		**BTU/hr**	**87,762**	**207,500**	**89,762**	**248,248**	**90,415**	**228,400**
34	**System EER**			**16.0**	**9.1**	**16.4**	**13.0**	**16.1**	**12.2**

4.1.4. Evaporative Heat Exchanger Performance

In additional to the test runs shown in Table 4-1, we conducted tests with the compressors off to better isolate the VCEC performance. These test runs are shown in Table 4-5.

Table 4-5: VCEC Test Data with Compressors OFF

#					5B	6B	6D	14B	19A	19B	19C	19D
1			Test Condition		5B	6B	6D	14B	19A	19B	19C	19D
2			DRY/WET fraction		31%	68%	102%	105%	40%	67%	89%	106%
3	DRY PASSAGES		Flow Rate	SCFM	1559	3077	5224	5351	1945	3184	3991	4806
4	DRY PASSAGES	INLET	**Dry-bulb temperature**	**ºF**	**94.7**	**103.2**	**105.5**	**84.9**	**94.8**	**95.0**	**96.0**	**89.0**
5	DRY PASSAGES	INLET	Wet-bulb temperature	ºF	70.0	73.4	73.3	66.2	75.7	76.5	76.4	76.4
6	DRY PASSAGES	INLET	Dew point	ºF	58.0	60.1	58.6	56.1	68.4	69.6	69.2	72.0
7	DRY PASSAGES	OUTLET	**Dry-bulb temperature**	**ºF**	**71.5**	**76.3**	**69.1**	**62.8**	**75.1**	**76.3**	**78.1**	**74.3**
8	DRY PASSAGES	OUTLET	Wet-bulb temperature	ºF	60.9	63.5	64.3	59.7	67.6	68.5	69.1	71.3
9	DRY PASSAGES	OUTLET	Dew point	ºF	54.8	56.5	62.2	58.1	64.5	65.3	65.5	70.5
10	DRY PASSAGES		Pressure drop	inWC	0.16	0.34	1.32	1.41	0.16	0.33	0.48	1.06
11	WET PASSAGES		Flow Rate	SCFM	5074	4971	5433	5506	5095	5082	5014	4898
12	WET PASSAGES	INLET	Dry-bulb temperature	ºF	102.5	105.8	84.9	77.0	98.9	96.0	98.1	89.8
13	WET PASSAGES	INLET	**Wet-bulb temperature**	**ºF**	**72.0**	**73.9**	**68.8**	**64.8**	**76.5**	**76.5**	**76.7**	**76.3**
14	WET PASSAGES	INLET	Dew point	ºF	58.0	60.1	61.0	58.2	68.4	69.6	69.2	72.0
15	WET PASSAGES	OUTLET	Dry-bulb temperature	ºF	79.0	82.6	78.0	71.7	82.0	82.4	83.2	81.3
16	WET PASSAGES		Pressure drop	inWC	0.70	0.49	0.69	0.65	0.76	0.81	1.05	0.48
17	VCEC/condenser water flow rate			GPM	10.0	10.0	10.0	10.0	10.0	10.0	10.0	10.0
18	Sump water temp			ºF	69.7	76.1	70.2	66.7	77.7	77.7	78.1	75.9
19	Sump water approach			ºF	-2.3	2.2	1.4	1.9	1.2	1.2	1.3	-0.5
20	Wet bulb depression (wet passages)			ºF	30.5	31.8	16.1	12.2	22.4	19.5	21.4	13.5
21	Dew point change (in dry passages)			ºF	-3.2	-3.5	3.5	2.0	-4.0	-4.3	-3.7	-1.4
22	**VCEC ventilation air pre-cooling**			**BTU/hr**	**48,659**	**112,439**	**174,317**	**112,517**	**62,288**	**102,248**	**118,741**	**102,154**
23				**tons**	**4.1**	**9.4**	**14.5**	**9.4**	**5.2**	**8.5**	**9.9**	**8.5**
24	**VCEC indirect cooling effectiveness**			**%**	**102%**	**92%**	**99%**	**110%**	**88%**	**96%**	**84%**	**109%**

For all four of the early 100% VA tests (6E, 14A, 6D, 14B, but not 19D) the Return-to-Sump Damper (location K in Figure 3-3) was accidentally opened. This was not detected until closer inspection of the test data after testing was completed, and there as not enough time to repeat the testing. This allowed return air to enter the sump and mix with the outdoor air before entering the wet passages, resulting in much lower dry-bulb and wet-bulb temperatures than for the same tests with lower VA rates. With both return air dampers open, return air was going toward both the sump *and* the main filter/coil section. Without knowing how much was going in each direction, we could only estimate what the mixed-air conditions in the sump section had been in each test. We assumed that the exhaust fan, which was set at the same speed for all tests, was moving the same amount of air as the average for all other tests. By subtracting out the component for the condenser OA inlet (which was measured via an orifice plate) we estimated the RA-to-sump flow rate and the mixed air conditions.

Line 2 shows the ratio of dry passage flow rate to wet passage flow rate. Indirect evaporative air coolers are usually tested with equal flow in both passages. In the case of the VCEC, this is somewhat artificial because of the requirement that it accommodate the full range of 0-100% ventilation airflow rates in the dry passages. On the other hand, we kept the flowrate in the wet passages within a narrow band to simulate the single-speed exhaust blower in the anticipated production HyPak unit specification.

Sump water temperature (line 18) was measured in the water line between the sump and the pump inlet. Sump water approach (line 19) is the difference between the wet-bulb of the air entering the wet passages and the sump water temperature. High performance evaporative coolers are able to attain water temperatures 1-2°F above the entering air wet-bulb temperature (approach) when there is no other heat addition, as from an evaporative condenser. The VCEC generally complies with this rule for all tests, except for test 5B, which has sump water 2.3°F *cooler* than the wet-bulb of the air entering the wet passages. This may have been caused by an instrumentation error, or perhaps from a surge of fresh water entering the sump from the float valve during the test period.

Line 20 shows the wet-bulb depression of the air entering the wet passages. Wet-bulb depression is the difference between the dry-bulb and wet-bulb of air at a given location, and is considered to be the driver in evaporative capacity. In general, the larger the depression, the more cooling capacity. This does not appear to be the case for the VCEC when comparing tests 6B and 6D. The wet-bulb depression in 6D is about half that of 6B due to the incorrect RA damper setting, which lowers the wet passage entering dry-bulb temperature substantially. However, this RA addition also lowers the wet-bulb temperature. A larger wet-bulb depression improves evaporative performance, but only for a fixed dry-bulb temperature. As expected, wet-bulb temperature itself appears to have a much stronger influence on evaporative performance than wet-bulb depression.

<u>The four primary indicators of VCEC performance are shown in lines 7 and 21-24.</u>

- *Dry passage leaving air dry-bulb temperature* indicates whether any additional mechanical cooling is required to cool ventilation air to the setpoint.
- *Dew point change* shows if moisture is infiltrating the dry passages.
- *Indirect evaporative capacity* shows how much compressor capacity can be offset by using a VCEC to handle ventilation air cooling loads.
- *Indirect evaporative effectiveness* is the ultimate measure of performance for indirect evaporative air coolers.

The air leaving the dry passages had a dry-bulb temperature (line 7) that was at, or below, a typical thermostat setpoint for all test runs. This means that additional compressor cooling would *not* be required before the air can be delivered to the space, unless there was a cooling load from the conditioned space. This will generally be the case at the hotter conditions, but probably not for test 14B. This confirms that the VCEC will substantially reduce the number of hours when compressor operation is required. This pseudo-economizer mode, with both blowers and the pump running but not the compressors, is exemplified by test 14B. With outdoor conditions of 85°F WB and 66°F DB and the compressors off, the supply air temperature was 61.8°F, not much above the typical supply air temperature of 55-60°F for a DX system.

Line 21 shows the change in dew point as air moves through the dry passages of the VCEC. Because there is no latent cooling or addition in the dry passages, this should be zero for all tests. A positive value would indicate moisture is reaching the dry passages, and would be a serious concern for the VCEC. Instead, the negative values point to discrepancies in the test setup, instrumentation or analysis.

Lines 22 and 23 shows the indirect evaporative capacity of the VCEC as determined by the dry passage flow rate (in lbs/hr) multiplied by the change in enthalpy across the dry passage. Without compressor heat added to the system, the VCEC delivered substantial amounts of indirect evaporative cooling. This indicates how well the VCEC would perform if it was used in a dedicated ventilation air pre-cooling stage, either as part of a larger system such as those made by DCT, or in a stand-alone ventilation air cooler, such as the Indirect Evaporative Heat Recovery Ventilator concept currently being promoted by DEG to various funding sources for future development. For conventional air-cooled equipment in the same conditions, this entire load would need to be cooled by mechanical compressor cooling, in addition to the cooling load of the conditioned space itself.

For those buildings that require 100% VA for all or many of the occupied hours, the VCEC can deliver between 11.0 and 17.4 tons of indirect evaporative cooling at design conditions. For more conventional retail, school, and restaurant applications requiring 20-40% VA, the VCEC can deliver between 4.1 and 9.4 tons of indirect evaporative cooling at design conditions.

Indirect evaporative effectiveness (line 24) is defined as:

$$\frac{\text{dry passage entering dry-bulb} - \text{dry passage leaving dry-bulb}}{\text{dry passage entering dry-bulb} - \text{wet passage entering wet-bulb}}$$

It is generally assumed to have a maximum of 100%, but that is only for cases with identical conditions entering both passages. Because we conditioned the outside air entering the HyPak prototype, as well as the RA damper position issue, we had situations where the airstreams were quite different. In addition, we were not measuring the airflow entering the sump through the RA damper; we could only estimate the flow. That limited the accuracy of the dry-bulb and wet-bulb data entering the wet passages (lines 12, 13). We believe that explains the effectiveness values greater than 100% in line 24 for tests 6D and 6E. In tests 6B, 19A, 19B, 19C, and 19D, the dry-bulb and wet-bulb of the air entering both passages was very similar, and we calculated indirect evaporative effectiveness values of 84-92% for these tests. We consider this to be the appropriate range of values for publication. With different entering conditions, we measured a range of indirect evaporative effectiveness for the VCEC of 102-109%.

We expected the indirect evaporative capacity to increase with the dry passage flowrate. However, we were much more surprised to see the indirect evaporative effectiveness stay in a range of 84-109% for all tests. The best comparisons can be drawn between tests 19A, 19B and 19C. We conducted the 19-series tests after analyzing data from the prior runs to get a better idea of VCEC performance. They were intended to minimize the discrepancy between the two OA inlet locations so that both passages had nearly identical entering air conditions. This required the OA DB to be relatively close to the ambient shop temperature, and the tests were run in as quick succession as possible (with the RA damper closed). Although these tests are a poor match for Western design conditions (too humid) they gave us an excellent insight into the VCEC behavior, as shown in Figure 4-1.

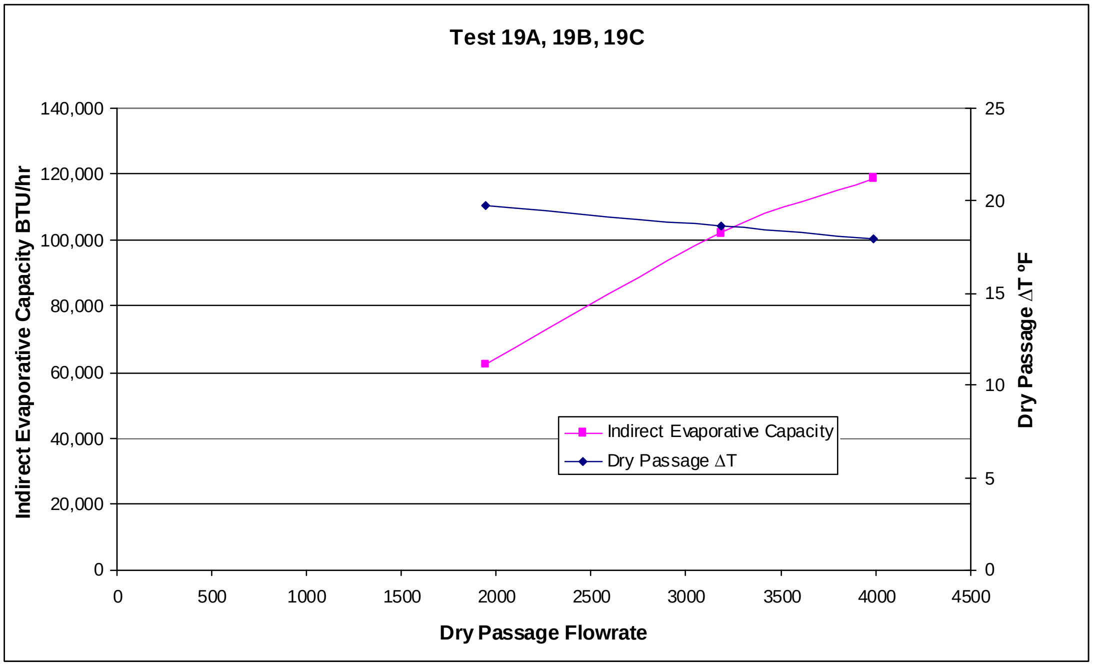

Figure 4-1: VCEC Data for 19-Series Tests

As dry passage flow doubled from 40% to 90% of the wet passage flow rate, the ΔT across the dry passage was reduced from 19.7 to 17.9°F, and capacity grew at a rate nearly proportional to the dry passage flow rate. This indicates that the air in the dry passage at low flow rates is getting close to its lower limit soon after entering the dry passage, and spends a good portion of the dwell time without being cooled further. (The lower limit is driven by the wet-bulb of the air entering the wet passages.) As the amount of air in the dry passage increases, the dwell time is reduced, but the air is still getting very close to its lower limit. This indicates that the VCEC is likely capable of cooling even larger amounts of dry passage air without much of an increase in dry passage leaving air dry-bulb temperature; it also appears to confirm the otherwise hard-to-believe temperature distribution from the CFD software and shown in Figure 3-11.

Although the VCEC demonstrated high evaporative performance in the Phase 1 testing, the pressure drop in both the dry and wet passages was higher than we would have liked. The dry passage pressure drop for all test runs (with and without compressor operation) is shown in Figure 4-2, and for the wet passages in Figure 4-3. Dry passage flow appears to follow cube-law flow curve behavior, but the wet passage flow has a scatter-shot appearance that defies conventional wisdom. The single outlier was probably caused by instrumentation or operator error rather than a legitimate data point. But at this time, we have no explanation for the other data points and further testing will be required to learn more. Before testing, we were concerned that higher flows in the dry passage are sucking the dry passages closed slightly, opening the wet passages to allow more flow. But this does not appear to match the data. It is possible that plate vibration and moisture are making it impossible to get a good measure of wet passage pressure drop. Unlike the dry passage, we did not dedicate a channel to the measuring pressure drop, but had to subtract the static pressure measurements from two channels; this added uncertainty to the data.

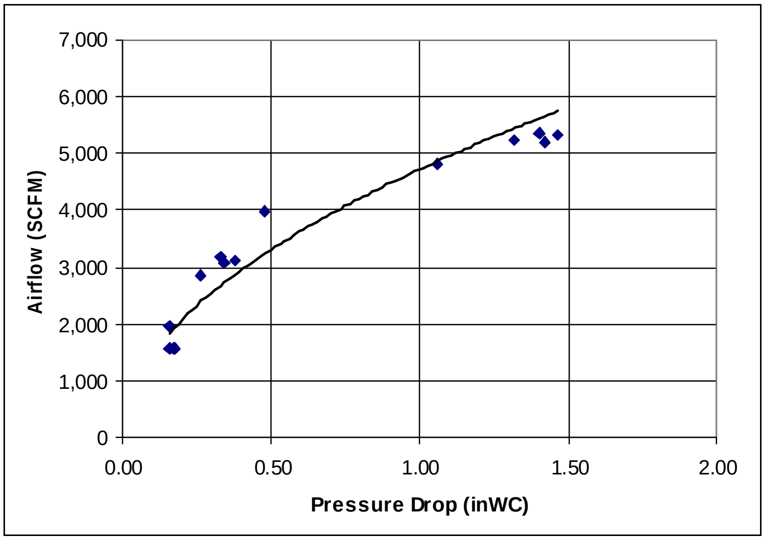

Figure 4-2: Flow in VCEC Dry Passages for All Tests

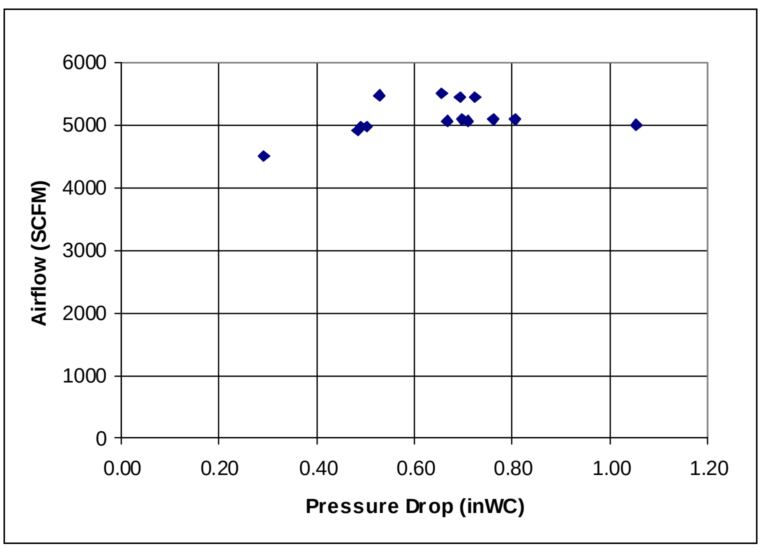

Figure 4-3: Flow in VCEC Wet Passages for All Tests

VCEC data for the test runs with the compressors on is shown in Table 4-6. These are the same runs reported in Table 4-1, but with the VCEC data points from Table 4-5. We attempted to match the test conditions between runs with and without compressor operation as closely as possible. The matching test conditions are 5A-5B, 6A-6B, 6D-6E, 14A-14B. The discrepancies varied for each test pair, with 6A-6B the closest test pair. With the compressors on, the wet passages of the VCEC must reject vastly more heat than when cooling only VA. (There is also some evaporation occurring at the condenser itself, but this increases the moisture content of the air entering the wet passages, making the VCEC work harder.) Line 24 shows the cooling capacity from Table 4-1. The condenser of a refrigeration system must reject not only the heat extracted from the evaporator, but also the power consumed by the compressors. (1 kW = 3412 BTU/hr) The heat rejected at the condenser is shown in line 25. Lines 26 and 27 combine line 25 with the indirect evaporative cooling of VA (line 22) to get the total heat rejected by the air passing through the condenser and VCEC wet passages.

Table 4-6: VCEC Test Data with Compressors ON

1			Test Condition		5A	6A	6E	8A	9A	14A
2			DRY/WET fraction		32%	69%	102%	37%	98%	104%
3	DRY PASSAGES		Flow Rate	SCFM	1578	3120	5200	1572	2862	5322
4	DRY PASSAGES	INLET	**Dry-bulb temperature**	**ºF**	**96.5**	**101.1**	**107.1**	**90.2**	**95.1**	**87.1**
5	DRY PASSAGES	INLET	Wet-bulb temperature	ºF	69.5	72.0	73.0	68.3	71.1	68.8
6	DRY PASSAGES	INLET	Dew point	ºF	55.9	58.2	57.0	56.9	59.7	59.8
7	DRY PASSAGES	OUTLET	**Dry-bulb temperature**	**ºF**	**77.3**	**81.7**	**72.7**	**76.4**	**76.9**	**66.5**
8	DRY PASSAGES	OUTLET	Wet-bulb temperature	ºF	61.6	65.2	66.3	62.0	63.4	62.7
9	DRY PASSAGES	OUTLET	Dew point	ºF	52.3	56.4	63.4	53.5	55.8	60.9
10	DRY PASSAGES		Pressure drop	inWC	0.17	0.38	1.42	0.18	0.26	1.47
11	WET PASSAGES		Flow Rate	SCFM	5057	4966	5437	4507	5066	5463
12	WET PASSAGES	INLET	Dry-bulb temperature	ºF	104.1	106.2	77.1	95.8	95.9	78.4
13	WET PASSAGES	INLET	**Wet-bulb temperature**	**ºF**	**71.5**	**73.2**	**65.2**	**69.8**	**71.1**	**64.5**
14	WET PASSAGES	INLET	Dew point	ºF	55.9	58.2	58.9	56.9	59.7	56.9
15	WET PASSAGES	OUTLET	Dry-bulb temperature	ºF	88.6	90.5	84.4	87.0	85.5	79.5
16	WET PASSAGES		Pressure drop	inWC	0.67	0.50	0.73	0.29	0.71	0.53
17	VCEC/condenser water flow rate			GPM	10.0	10.0	10.0	10.0	10.0	10.0
18	Sump water temp			ºF	87.1	90.4	84.2	88.8	88.3	79.8
19	Sump water approach			ºF	15.6	17.2	19.0	19.0	17.2	15.3
20	Wet bulb depression (wet passages)			ºF	32.6	33.0	11.9	26.0	24.8	13.9
21	Dew point change (in dry passages)			ºF	-3.6	-1.8	6.5	-3.4	-3.9	1.1
22	**VCEC ventilation air pre-cooling**			**BTU/hr**	**42,721**	**77,711**	**131,932**	**33,467**	**78,869**	**111,843**
23	**VCEC indirect cooling effectiveness**			**%**	**77%**	**69%**	**82%**	**68%**	**76%**	**91%**
24	Compressor cooling capacity			BTU/hr	205,527	200,812	211,741	194,934	199,790	208,802
25	Heat rejected at condenser			BTU/hr	251,893	248,376	254,733	240,905	245,891	250,164
26	**Total heat rejected in VCEC**			**BTU/hr**	**294,614**	**326,087**	**421,337**	**274,372**	**324,761**	**362,006**
27	**Total heat rejected in VCEC**			**tons**	**24.6**	**27.2**	**35.1**	**22.9**	**27.1**	**30.2**

Even with an additional 250,000 BTU/hr of condenser heat entering the wet passage air, the VCEC continued to provide a large amount of indirect evaporative cooling to the ventilation air. Comparing the matching test pairs, *the VA pre-cooling capacity dropped by only 11-31% with the compressors on.* For typical design conditions (tests 5A, 6A), the VCEC was still able to deliver between 3.6 and 6.5 tons of indirect evaporative cooling at 100°F DB / 70°F WB outdoor air, with 20-40% ventilation air.

4.1.5. Refrigeration System Performance

Against the outstanding performance of the VCEC, the performance of the refrigeration system in the Phase 1 HyPak prototype was somewhat below our expectations.

Compressor EER is a good measure of the performance of the entire refrigeration system. In the Phase 1 testing, the compressor EER was between 14.2 and 17.2, while in the final Phase 3 laboratory testing of the prior Hypak-1 project it ranged between 21.1 and 23.6.

(In Table 4.1, compressor EER (line 21) is the ratio of mechanical cooling (line 23) over total compressor power (lines 25 and 26). Because it does not include fan power, compressor EER is always higher than overall system EER in conventional packaged HVAC products. Because HyPak includes substantial indirect evaporative cooling capacity, it is possible for the system EER to exceed the compressor EER.)

To explain this performance shortfall we examined several parameters of the refrigeration system that combine to influence compressor EER and compared them to the data recorded in the 2004 Phase 3 testing in the HyPak-1 project. Comparisons between the 2004 and 2007 tests runs with the closest operating conditions are shown in Table 4-4.

The primary driver of the lower compressor EER appears to be much higher compressor power consumption than in 2004 testing. 2007 power consumption for the tandem compressors ranged from 12.6 to 14.0 kW at Western design conditions, while in 2004 it was just 3.8 kW. Although the 2007 prototype had about 140% more capacity than the 2004 prototype, the compressors used about 300% more power.

Reducing condensing temperature (line 20, Table 4-4) is generally the easiest way to improve refrigeration system performance. Compressors are typically rated at 130°F condensing temperature, because that is the average condensing temperature for an air-cooled system at 95°F ARI 340/360 rating condition. In actual design conditions for Western applications, condensing temperatures are usually even higher. Against this backdrop, the condensing temperatures of 96.5-112.9°F recorded in the Phase 1 testing seem reasonable. However, evaporative condensers are usually able to attain condensing temperatures below 100°F at Western design conditions.

The easiest way to reduce condensing temperature in an evaporative system is to increase the condenser size. Because the evaporative condenser is made from copper for corrosion resistance, this has obvious cost consequences. The 2004 condenser was only 4 rows compared to 7 rows in the 2007 condenser, and the face area in the 2004 condenser was slightly more than half that of the 2007 condenser. Based on 10 and 20 ton nominal capacities, the 2007 condenser is 33% larger per ton. However, the 2007 prototype delivered 207,500 BTU/hr (17.3 tons) at ARI rating condition (no VA), while the 2004 prototype delivered only 87,762 BTU/hr (7.3 tons) at the same conditions (line 33). When equalized over these compressor capacities, the two condensers were much closer in size per ton, with the 2004 condenser actually 13% larger per ton than the 2007 condenser. In addition, the 2004 condenser had rifled tubes, while the 2007 condenser did not, giving it further advantage. This partially explains why the 2004 prototype had lower condenser temperatures than the 2007 prototype by 4.4 and 6.2°F for realistic design conditions (tests 5A and 8A).

Evaporating temperature can have an even stronger impact on refrigerant system efficiency. A 1°F increase in evaporating temperature has about 1.5 to 2 times the increase in efficiency of a 1°F decrease in condensing temperature. However, increasing evaporating temperature by 1°F is harder to achieve than a similar decrease in condensing temperature, so the costs to achieve the increase in efficiency are probably similar, because they come mostly from the size of the evaporator and condenser heat exchangers (in this case aircoils, finned for the evaporator and unfinned for the condenser). Comparing 2004 and 2007 testing, we found an even greater discrepancy in evaporating temperature between the two tests than for condensing temperature, indicating that this is a better place to target improvements. Evaporating temperatures were 5.3 and 6.4°F higher in 2004 than in 2007 at realistic design conditions.

Both prototypes used 6 row evaporator coils. Because the face area of the 2007 evaporator was 2.4 times that of the 2004 evaporator, they were effectively identical when adjusted for the actual compressor capacities measured in the ARI 340/360 standard rating test condition.

Superheating refrigerant vapor and subcooling liquid refrigerant are necessary to ensure stable system operation. Superheating ensures that liquid does not enter the compressor, which can damage the compressor. (This is called “slugging.”) Subcooling ensures that the refrigerant is entirely in the liquid state as it passes through the thermostatic expansion valve (TXV). Vapor in the TXV (called “flashing”) will cause the expansion valve to flutter as it tries to adjust to what it detects is an increase in the volumetric flow rate due to the lower density of refrigerant vapor.

However, both superheat and subcooling come with a penalty in terms of system efficiency. This is because a portion of the evaporator coil is used for superheating vapor via sensible heat transfer, reducing the surface area of the evaporator available for latent heat transfer, which is much more effective at cooling the air stream. Similarly, by devoting a section of the condenser to liquid subcooling (via sensible heat transfer), less of the condenser is available for latent heat transfer. Typical HVAC refrigerant systems are set with 15-20°F of superheat and subcooling.

Because it can be controlled by the setting and configuration of the TXV, superheat is somewhat easier to adjust. Despite this, the 2007 prototype had an excess of superheat, between 25-30°F for all test runs, while the 2004 prototype had 4-7°F of superheat. On the other hand, the 2004 prototype had plenty of subcooling (16-26°F), while the 2007 prototype had less than 3°F of subcooling for all tests except one. This is surprising because the 2007 condenser had a dedicated subcooling circuit, while the 2004 condenser did not (but it did have a receiver, which was not necessary for the 2007 prototype because of the simpler control circuit without a pumpdown cycle).

Another factor appears to have been compressor selection. The nominal capacity of the Carlyle tandem scroll compressors used in the Phase 1 prototype was 13.9 tons (combined, at 130°F/45°F condensing/evaporating temperatures). The single Copeland scroll used in the 2004 prototype had a nominal capacity of only 4.8 tons. By stretching the capacity of the 4.8 ton compressor to match a 10 ton nominal system with 3000 SCFM of supply air, the 2004 prototype may have been better able to take advantage of the oversized evaporator and condenser heat exchangers, but had only 7.3 tons at the ARI standard rating condition. In the 2007 prototype, we sought to achieve a nominal capacity closer to our 20 ton target, and we did with 17.3 tons. However, this gave the compressor less evaporator and condenser heat exchanger size to "grow into."

When combined with the much higher indirect evaporative capacity of the VCEC, the 2007 prototype delivered about 280,000 BTU/hr (23.3 tons) of total cooling at 40% VA design conditions, making this unit equivalent to a 25 ton conventional air-cooled RTU. However, a 25 ton unit would require the capacity to move about 10,000 SCFM, which may be possible with the Phase 1 prototype, but only by increasing the supply fan speed. A better fit may be to use a smaller tandem compressor with a nominal capacity of 10-12 tons (the next size smaller from Carlyle is 11.3 tons) and allow it to stretch to about 20 tons of equivalent capacity with the heat exchangers (VCEC and aircoils) of this prototype. For a 20% VA application, the current tandem compressor pair is probably a good choice, delivering 20.6 tons of cooling at design conditions, which should slightly exceed that of a conventional 20 ton air-cooled RTU.

Brand selection may have had an impact, although Carlyle claims to have higher efficiency than the Copeland units.

Refrigeration circuits are complex systems, and all of the factors described here are closely interrelated. Using a very large evaporator with a small condenser will generally not result in a high compressor EER. Similarly, dialing-out superheat or increasing subcooling is not as easy as turning a setscrew. For these reasons, improving the efficiency of the HyPak system will require a dedicated budget and test period in Phase 2.

4.2. Next Steps

With Phase 1 completed, we have a good understanding of where future efforts must be concentrated to complete development the HyPak high-efficiency RTU to the point where it is ready for production, field trials, and market introduction. For the most part, these focus areas are consistent with the tasklist in the Statement of Project Objectives (SOPO) from the June 1, 2005 HyPak-MA contract. However, we believe that additional budget and schedule focus should be committed to some new areas, in addition to those anticipated 36 months ago at the time that we drafted the proposal.

4.2.1. Phase 2 Tasks from Statement of Project Objectives

The Phase 2 task descriptions from the June 1, 2005 HyPak-MA contract SOPO are shown below.

> *Task 2.1: Prepare VCEC for Volume Production.* The recipient shall refine the plate design to improve performance, minimize leakage and allow a rapid "Z-fold" VCEC assembly. An automated assembly line will be developed at Pride Polymers to thermoform and assemble VCECs in one continuous operation.

Task 2.2: Design and Fabricate Production Prototypes. The recipient shall refine the Phase 1 cabinet design and component selections will be refined in anticipation of volume production. Control algorithms will be optimized to maximize ventilation and minimize energy use. Two pre-production prototypes will be fabricated for use in laboratory and field testing.

Task 2.3: Laboratory and Field Test Prototypes. The recipient shall develop test plans will be developed for both laboratory and field testing. The first pre-production prototype will be tested at DCT. The second pre-production prototype will be installed on a field site in the Sacramento area. DEG will select and install instrumentation and monitor the unit for capacity and efficiency. DEG will assess reliability and maintenance concerns. Additional demonstration sites will be identified for pilot production units to be built using team and third party funds.

Task 2.4: Evaluate and Report. The recipient shall compile design information and test results into a report. Commercialization efforts and production planning will be documented.

4.2.2. Evaporative Heat Exchanger Development

Development of the VCEC evaporative heat exchanger will continue to be a primary area of emphasis for the HyPak project as described in the SOPO under Task 2.1. With the outstanding performance of the VCEC in Phase 1, Phase 2 work will focus on lowering manufacturing costs and preparing the production process for commercialization.

- As described in the SOPO, we will concentrate on integration of the heat sealing process into the Lyle thermoforming line at Pride Polymers. Fabrication and design costs are expected to be substantial, but implementation of this system is critical to reach internal VCEC price targets.
- Good evaporative performance indicates that the airflow distribution in the dry and wet passages was most likely good. However, we would like to confirm this before any modifications are made to the thermoforming tooling. That will require producing a VCEC module from clear plastic and testing it with smoke to physically confirm the airflow distribution in the Phase 1 design. We may also evaluate water distribution with colored water solution.
- Some suppliers of indirect evaporative heat exchangers claim that flocking is not necessary because scale buildup from hardness minerals is sufficient to get good water distribution. This requires an initial break-in period to build up enough scale (which flakes off due to plate vibration before becoming a problem). We would like to test this with some simple long-term testing at DEG of a flocked VCEC module alongside a clear, un-flocked VCEC test module.
- Fire code requirements and flame-resistant material additives will be investigated.
- At least 50% of the VCEC thermoforming tooling will need to be replaced, including the entire primary tooling (shown in Figure 3-17) to address the following concerns with the Phase 1 tooling.
    - The hinge geometry did not work, requiring hand-creasing.

  - Any airflow or water distribution problems found in later VCEC evaluation.
  - Evaporative performance of the VCEC was good, but it came with a larger than acceptable fan energy cost. Because ΔT and cooling effectiveness (Figure 4-1) were maintained even at the highest dry passage flow rates, we will most likely be able to widen the dry passage plate spacing without losing much evaporative performance, while reducing supply fan energy by about half (at the highest flow rates). (See Figures 3-10 and 3-11 for additional information.) CFD software will be used to determine the optimum dry passage plate spacing. A similar analysis will be performed on the wet passage spacing to reduce exhaust blower energy.
- Also important is development of a low-cost casing for VCEC modules. The stainless steel casings used in Phase 1 worked well, but their costs are prohibitive for volume production. In Phase 2, we will develop a suitable production casing made from corrugated plastic.

Based on a Task 1.1 budget review, we spent $184,282 to develop the Phase 1 VCEC. With only $139,350 allocated to Task 1.1, VCEC development in Phase 1 was $45,922 over budget. *We estimate that it will require about $300,000 to complete the VCEC development outlined above. Task 2.1 is allocated $214,560, about $30,000 more than Task 1.1, but this leaves an anticipated budget shortfall of about $85,000 in Task 2.1.*

4.2.3. Phase 2 Prototype Development

We anticipate evaluating and incorporating some or all of the following design changes in the Phase 2 HyPak prototype during Task 2.2:

- Optimize fan and motor selection. Fan energy consumption was higher than necessary in Phase 1 testing. We will most likely keep the supply fan as-is, with efforts to reduce supply fan energy consumption focused on reducing the pressure drop of the VCEC dry passages in Task 2.1. If pressure drop can be reduced substantially, we may be able to use a smaller supply fan motor. For the exhaust fan, we would like to evaluate other blower types, such as propeller fans or motorized impellers. Such fans offer higher efficiency than squirrel-cage centrifugal blowers, but are not able to meet high static pressure requirements.
- We believe that the condenser design can be improved to reduce condensing temperatures. Design improvements will most likely be more surface area and/or rifled tubing, which will require optimization of cost versus performance.
- The evaporator will require a similar evaluation and optimization of heat exchanger size and cost.
- We would like to reduce superheat levels, and possibly increase subcooling levels. This may require evaporator and condenser circuit changes, or use of different TXVs.
- We were not able to evaluate the part-load performance of the system. HyPak has two means to modulate cooling capacity: (a) reduced supply fan speed, and (b) single compressor operation. We must evaluate system capacity and stability for all possible combinations of these two measures before prototypes can be installed in the field. Lower fan speed risks coil-freeze up, and single compressor operation must

continue to be stable with reasonable levels of superheat and subcooling. Proper operation of the liquid line solenoid valve for evaporator modulation should be investigated, along with the possibility of changing the evaporator split circuit ratio. (See Figure 3-5 for more information.) Part-load performance should be evaluated before and after any changes are made to the refrigeration system.

- We will eliminate the Return-to-Sump Damper (location K in Figure 3-3), which will also allow us to eliminate the condenser OA inlet damper (location N in Figure 3-3). Unfortunately, this will eliminate the possibility of heat recovery in heating mode or enthalpy recovery in cooling mode, but both of those are expected to be modest for the initial HyPak target market in California. After production begins, we believe that we will be able to solve the problem of moisture in the return plenum and duct with the Return-to-Sump Damper, bringing back the heat and enthalpy recovery that will enhance cost-effectiveness in secondary HyPak markets. A compromise solution may be to retain both dampers, but use a seasonal blank-off plate that can be removed to allow heat recovery in the winter. (A sensor will be included to prevent compressor operation without the plate installed.)
- Without the Return-to-Sump Damper, HyPak will require a separate exhaust air blower similar to those used on most conventional RTUs to avoid pressurizing the building. This will need to be incorporated into the Phase 2 prototype design.
- The spray nozzle manifold underneath the evaporative condenser in the Phase 1 prototype was expensive to fabricate and offered no improvement over using only the VCEC water feed system. We will not use it in future HyPak prototypes, but we would like to evaluate possible changes to the VCEC water feed system.
- We will most likely need a damper between the VA pre-filter and the VCEC to enable pure economizer function bypassing the VCEC. The controls will use this mode only when the additional energy of the exhaust fan and sump pump is not offset by additional cooling capacity. In addition, economizers are required in California, our primary target for initial HyPak sales.
- Circulator pumps offer higher efficiencies than submersible pumps, but their net positive suction requirements mean that they must be installed outside of the HyPak cabinet. For prior HyPak prototypes this added cost to the field installation. Submersible pumps can be installed inside the sump in the factory. We will most likely use submersible pumps in Phase 2 prototypes and the first production units.
- The Phase 1 prototype used a Johnson Metasys DX9100 DDC controller, but the controller was driven manually from a laptop, rather than using an algorithm to select operating modes and adjust component operating levels and states. We developed a basic Sequence of Operations for the Phase 1 prototype, which was a further development of the SOO used for the HyPak-1 field test unit control algorithm. That algorithm was designed to work with standard 3 stage thermostats, but we have chosen to use an indoor sensor in the zone and move all operational decisions to the DDC controller on-board the HyPak unit. This saves thermostat cost and allows for full modulation, but it will most likely require a more powerful controller (at a higher cost). A web-based interface will be used on field test units and early production units to provide automated fault diagnostics and remote monitoring, logging and troubleshooting. (This will probably become optional for later production units to reduce cost.) Automated fault diagnostics may monitor important system functions

such as damper actuation, supply airflow (to avoid coil freeze-up), filter pressure drop, and refrigerant head pressure. Development of this control system will require substantial resources and time. (For more description of the controller logic, refer to page 3-8.)

We completed design and fabrication of a single prototype in Task 1.2 for approximately the budgeted sum of $191,860. *Task 2.2 has a budget of $164,360. We estimate that an additional $100,000 will be required in order to evaluate these potential design changes and fabricate two Phase 2 prototypes as per the SOPO.*

4.2.4. Continued Testing of Phase 1 Prototype

We believe that we can learn substantially more about the Phase 1 prototype. It would be an excellent test mule to evaluate the benefits of modifications discussed in Section 4.2.3. Continued testing and evaluation of the Phase 1 prototype would also allow DCT to continue development and durability assessment of the prototype at the same time as DEG and Pride Polymers are working on VCEC development. (Phase 2 prototypes are intended to use VCECs produced using the production-spec tooling developed in Task 2.1.) In particular, this would allow DCT to develop the sophisticated controls for intelligent selection of operating mode to maximize HyPak year-round efficiency.

Evaluating as many of the design changes as possible using the Phase 1 prototype will increase the likelihood that the Phase 2 prototypes will be true pre-production specification units, and will minimize the time and effort required to prepare for commercialization.

The cost for this additional testing is included in the $100,000 request in Section 4.2.3.

4.2.5. Phase 2 Testing

Task 2.3 in the SOPO calls for one unit to be laboratory tested at DCT, and second prototype to be field tested in the Sacramento area. We would like to change this slightly so that the Phase 1 prototype (updated to Phase 2 specification as described in Sections 4.2.3 and 4.2.4) becomes the Task 2.3 laboratory test unit and remains at DCT. We would still build two prototypes in Task 2.2, with one for field testing at the DEG warehouse and the second reserved for a future field test, most likely in a retail application, to be installed a few months before or after the end of Phase 2. Several utilities and agencies have expressed interest in supporting a field test.

Field testing at DEG would have several advantages.

- Because the facility is not currently air conditioned, there is no consequence of system failure or shutdown for modifications or repairs.
- Ground level installation for easy access.
- DEG can evaluate changes and make repairs easier than if the unit is installed on a commercial building some distance away.
- If necessary, we can combine neighboring warehouse units to create as large of a zone and cooling load as required to simulate design conditions.

- The Davis, California climate has about 120 days with highs over 90°F, and about 30 days with highs over 100°F.
- To simplify performance comparisons, a baseline air-cooled RTU can serve the same large zone. This will ensure identical return air and outside air temperatures, and identical return and supply static pressures.
- Any reliability problems encountered at DEG can be addressed in the second Phase 2 unit before it is installed on a commercial space.

In Task 1.2, we exceeded the budget amount of $202,410 by about $30,000. Savings in other tasks offset some of this shortfall, with the rest resulting in cost match significantly higher than required by the contract. ***We estimate that the $202,270 allocated to Task 2.3 will be sufficient.***

4.2.6. Commercialization Planning

The market for commercial heating, ventilation and air conditioning equipment is undergoing significant change. According to an article in the *Air Conditioning Heating Refrigeration News* of May 14, 2007, construction spending is strong and consequently demand for commercial HVAC equipment is healthy, “However, … building managers and owners are not necessarily looking for the same old, same old. They are interested in new and innovative products and services that will help them save energy costs and improve comfort while reducing their impact on the environment”[10] A number of factors are driving the market. Primary among them are the rising price of energy, particularly electricity, and enhanced awareness and concern about climate change. In addition, electric utilities are having increasing problems meeting peak demand. These factors are leading to utility and governmental incentives for low energy and low emissions HVAC equipment and designs. *The News* article sums it up by stating “… it is abundantly clear that the trend toward greener buildings is here to stay.”

A consequence of this trend is the growing adoption of the U.S. Green Building Council’s Leadership in Energy and Environmental Design (LEED) green building rating system. LEED offers credit for energy efficient cooling and heating systems and for improved indoor air quality (IAQ). *The News* article describes the indoor air quality factor as follows: “Another positive trend is the focus on improving comfort and IAQ in the commercial building. Just like sustainability, most building owners realize that great comfort and IAQ are not only good for their employees, customers and tenants, they are also good for the bottom line. For example, keeping employees healthy helps eliminate sick days and increases productivity.”[11] It is safe to say that HyPak is poised to enter a very receptive market.

Davis Energy Group has begun laying plans for the commercialization of HyPak to take advantage of this growing demand for green HVAC products. Recently, DEG spun off a

[10] *“Commercial Market is Red Hot and Green,”* by Joanna R. Turpin. Air Conditioning Heating Refrigeration News, May 14, 2007; page 1.
[11] Michael C. Walker, product manager, commercial rooftop products, Lennox Industries. In *“Commercial Market is Red Hot and Green,”* by Joanna R. Turpin. Air Conditioning Heating Refrigeration News, May 14, 2007; page 1.

company whose purpose is the commercialization of low energy, low emissions cooling and heating equipment for buildings. This company, Advanced Energy Products Corp. (AEP), recently launched its first product line – NightBreeze – an automated ventilation cooling system for new residential construction. That product is being well received and AEP is now seeking capitalization so that it can expand rapidly. The company presented at a recent CleanTech conference in Santa Clara, California at which a number of venture capital firms and other financial companies were in attendance. As a consequence, discussions about capitalization have commenced with two of those companies. The outlook is for AEP to be well capitalized and prepared to commercialize HyPak immediately upon completion of its development and testing.

The plan is to first offer HyPak in the dry climate West, where its energy savings and performance will be best. As its reliability and payback are demonstrated, marketing efforts will expand eastward. To increase the pace of adoption, AEP will work with utilities to get focused demonstration programs and incentives in place for the technology. The company has already been successful at this for its NightBreeze product line.

Regarding manufacturing, it is our intent to continue to work with Pride Polymers and have them supply the vertical counter-flow evaporative cooler heat exchanger. Likewise, we will continue to work with Munters/Des Champs to have them manufacture and supply HyPak units. It is also quite possible that the HyPak technology will also be sold through the Munters/Des Champs sales organization. In this way, we intend to achieve our objective of proving HyPak's reliability and economic effectiveness in the field and then rapidly growing the market for the technology.

5. Conclusions and Recommendations

Based on the work completed in Phase 1 to develop HyPak, a high-efficiency rooftop packaged unit, we offer the following conclusions:

- Our review of conventional rooftop units (RTUs) in comparison with our test results indicates there is a significant opportunity to improve the efficiency of widely-used "mid-sized" (10-50 ton) packaged HVAC units, particularly regarding cooling and ventilation performance. In cooling mode, the best available units deliver only 12 BTU/watt, while our results indicate that 15-21 BTU/watt can be cost-effectively delivered with an improved unit.
- The major opportunity for improving cooling performance applies evaporative cooling to both the condensing function of the refrigeration cycle and indirectly to ventilation air. While this evaporative cooling strategy significantly reduces annual energy consumption, it is even more effective at reducing peak cooling demand, since evaporative benefits are greatest at high outdoor temperatures.
- After four project phases including the prior HyPak-1 project, the HyPak concept has reached a stable configuration that is production-ready. This configuration offers excellent cooling performance and can be manufactured using current DCT techniques and processes.
- The Vertical Counter-flow Evaporative Cooler (VCEC) delivers most of the energy savings of the HyPak system. Secondary advantages are an innovative modulating refrigerant system, variable speed supply blower motor, and intelligent controls with real-time feedback of ventilation airflow rate.
- The fan-fold design of the VCEC allows it to be cost-effectively produced through an automated in-line thermoforming and heat-sealing process with a minimal labor component. Computational fluid dynamics software was critical to simulating various spacer patterns to optimize airflow distribution, and resulted in good performance out-of-the-box.
- System EERs ranged from 9.1 to 21.3 at a variety of operating conditions. The highest EERs were recorded at design conditions matching typical HyPak applications (commercial buildings in Western population centers). This is in stark contract to conventional air-cooled RTUs which operate at their lowest efficiencies at design conditions and peak demand periods. The Phase 2 HyPak prototypes will have higher EERs due to reduced blower motor and compressor power consumption.
- The performance of the VCEC is excellent, both with and without simultaneous compressor operation. Results without compressor operation included indirect evaporative effectiveness ranged from 84% to 96% for tests with similar air entering both the wet and dry passages. Indirect cooling capacity ranged from 48,660 BTU/hr with 1560 SCFM of dry passage flow, to 209,100 BTU/hr with 5224 SCFM. In both cases, air was entering the VCEC at over 100°F dry-bulb temperature, and left at between 63°F and 78.1°F, meaning that compressor operation is required only to satisfy building loads. (Wet passage flow was constant at about 5000 SCFM. No moisture is added to the ventilation air as it passes through the VCEC.)

- Even with the compressors on, the temperature of the ventilation air was between 67°F and 82°F for all dry passage flow rates. This means that there is nearly no penalty for higher ventilation rates during peak conditions, and that at most off-peak conditions pre-cooled ventilation air has a lower enthalpy than return air. Eliminating the energy penalty associated with higher ventilation rates will make HyPak attractive for LEED projects and other buildings that value high indoor air quality.
- The innovative refrigeration system needs more testing and development to reduce compressor energy and assess system stability at part-loads. Fan energy levels can be reduced by 25%, increasing EERs.
- Most HVAC manufacturers are reluctant to introduce evaporative cooling to these relatively small rooftop units because of maintenance concerns. Minimizing maintenance costs and demonstrating a substantial reduction in energy costs are essential to success for a new evaporatively-based rooftop unit.

Based on these conclusions, we strongly recommend Phase 2 follow-on funding support. In addition, we request that DOE and NETL increase the budget and scope for Phase 2 to support the following additional development efforts:

- *Continued evaluation of the Phase 1 prototype in early Phase 2 to evaluate improved components and to test controls. Fans and motors will be swapped in an effort to reduce fan energy consumption, while maintaining adequate flow rates. This will allow the team to continue development of the "balance of system" at DCT, when it would otherwise be put on hold as VCEC development in Task 2.1 is expected to take considerably longer than prototype design and fabrication in Task 2.2. This also increases the likelihood that the Phase 2 prototypes will have high performance and reliability out of-the-box, which will in turn increase the likelihood of commercialization immediately following the conclusion of Phase 2.*
- *Further VCEC development. Initial VCEC development efforts in Task 1.1 consumed $184,282 against a proposal budget of $139,350. Although the Phase 1 VCEC prototypes worked well, it is clear that developing a commercialization-ready VCEC module will require greater resources than anticipated. Task 2.1 (Prepare VCEC for Volume Production) will require replacing at least half of the tooling used in Task 1.1, and the custom fabrication work needed to automate the heat sealing process will be substantial. The VCEC is clearly the make-or-break component of the HyPak system, with low-cost production the strongest factor in its success.*
- *Controls development for intelligent operating mode selection, and to deliver the maximum ventilation airflow rate possible without reducing efficiency (always meeting the minimum ventilation airflow rate required during occupied periods). The controller will monitor the pressure drop across the dry passages of the VCEC to provide real-time feedback of ventilation airflow rate. Sensor selection must be optimized to keep costs as low as possible without compromising controls performance. A web-based interface will be developed for maximum flexibility, user-friendliness, and remote monitoring and fault diagnosis.*