Final Technical Report

Recovery Act: Adaptive Environmentally Contained Power and Cooling IT Infrastructure for the Data Center

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List of Acronyms

ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers
BBU	Battery Backup Unit
BER	Basic Encoding Rules
BMU	Battery Management Unit
CMS	Controls and Monitoring Subsystem
COP	Coefficient of Performance
COP _G	Grand COP
CU	Cooling Unit
DAQ	Data Acquisition
DC	Direct Current Dedicated Controller
HEX	Heat Exchanger
HMI	Human Machine Interface
MDC	Modular Data Center
OID	Object ID
PC	Primary controller
PDU	Power Distribution Unit
PID	Proportional-Integral-Derivative
PLC	Programmable Logic Controller
PV-UPS	Photovoltaic Uninterruptible Power Supply
Q	IT Load (kilowatts)
RAT	Return Air Temperature (Celsius)
RWT	Return Water Temperature (Celsius)
SAT	Supply Air Temperature (Celsius)
SNMP	Simple Network Management Protocol
SWT	Supply Water Temperature (Celsius)

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Executive Summary

A 2007 report by the US Environmental Protection Agency (EPA) estimated that in 2006 data centers consumed 61 billion kilowatt-hours (kWh), or about 1.5% of total US electricity consumption costing \$4.5 billion¹. Federal servers and data centers accounted for 10% of this electricity use. Based on compute and storage trends at that time, the report forecasted that data center power consumption could reach more than 100 billion kWh at an annual cost of \$7.4 billion by 2011. This energy demand would require ten additional power plants to meet the peak load on power grids from data centers. The effects of changes in technology trends, such as server consolidation and virtualization, on these estimates are uncertain. However, these estimates demonstrate the magnitude of the problem and the need for energy efficiency strategies by data centers.

As part of the American Recovery and Reinvestment Act of 2009, the U.S. Department of Energy (DOE) awarded HP and the Eaton Corporation a grant to research and develop power and cooling technologies for a fully enclosed Information Technology (IT) rack system with an electrical IT load up to 100 kilowatts (kW). The enclosure system developed under this cost-sharing grant provides internal power distribution and cooling, with high-voltage alternating current and chilled water as the primary inputs. The system also accepts power from solar alternative energy sources.

The largest segment of the US data center market is the small and medium business (SMB) segment. Therefore, this project targets the energy efficiency of SMB data centers to make the greatest impact on US energy consumption and greenhouse gas emissions.

The most common metric used to estimate data center efficiency is power usage effectiveness (PUE). PUE is the ratio of the total power consumed by a data center to the power consumed by the IT equipment.

For the target SMB data center market, the baseline PUE is known to be 2.04². This means that for every watt consumed by the IT equipment, over one watt is used to provide power and cooling.

The purpose of this project was to develop high efficiency power and cooling solutions and to achieve a PUE of 1.25 for a fully enclosed 100-kW IT rack system. These R&D efforts build on power and cooling technologies HP developed for the HP Modular Cooling System and HP Performance Optimized Data Center container solutions, and they build on power technologies developed by Eaton.

¹ U.S. Environmental Protection Agency (EPA), "Report to congress on server and data center energy efficiency," public law 109-431, Aug 2007.

² Sullivan, A., 2009, "Data Center Efficiency with ENERGY STAR," presentation at 7x24 Exchange, Spring 2009, Florida.

We conducted the necessary R&D, testing, and manufacturing of elements for a self-contained enclosure supporting a 100 kW IT load. This Modular Data Center (MDC) uses high voltage AC (480 V) and chilled water inputs. It consists of a highly efficient high-voltage direct current (HVDC) power distribution subsystem, alternative energy interface, HVDC Battery Backup Unit, closed-loop cooling subsystem, and a controls and monitoring subsystem.

Our approach over the two-year program was to build, test, and improve on several iterations of the subsystems, then integrate and test them. Some areas of development matured faster than others. We began development of the mechanical and thermal subsystems first so they were the most mature. Other areas, such as the rectifier development, were more technically challenging and the controls and monitoring system had a later start. But overall the final solution included working and efficient products across the board. We did not have the UPS system with its photovoltaic (PV) simulator shipped to the integration site in Houston due to size and weight limitations of our lab. We did test this UPS thoroughly with both simulated and actual alternative energy inputs at the development site.

The body of the paper provides details of our results. The target goal of a PUE of 1.25 was achieved, but is of course highly dependent on the energy required to deliver the chilled water. We also talk about the partial PUE, which better focuses on the efficiency of our close-coupled enclosure. For test purposes, we used a dedicated 35-ton chiller, but this is not the recommended configuration for the best PUE. Substituting an actual chilled water energy requirement determines the final PUE. Cooler climates, free cooling, and higher cold aisle temperatures improve the PUE.

The following MDC elements are summarized below:

- Modular, deployable mechanical infrastructure
- HVAC and HVDC power distribution subsystems
- DC Battery Backup Unit subsystem
- Alternative energy interface
- High-efficiency, closed-loop cooling subsystem
- Controls and monitoring subsystem

Modular mechanical infrastructure

HP researched and developed the self-contained MDC enclosure that provides four to eight bays for 42U³ IT racks and provides space for cooling and power distribution subsystems (Figure 1). The enclosure includes an independent cooling unit (CU) bay in the center that provides a cooling capacity up to 100 kW of IT load. The enclosure is sealed and partially insulated so that very little heat escapes into the external environment.

³ One rack unit, U, is 1.75 inches (44.45 mm) high.

Figure 1: MDC enclosure

High-voltage AC distribution subsystem

Eaton designed and built an integrated high voltage AC distribution subsystem that feeds redundant 480 V, 3-phase power to Eaton rectifiers (cycloconverters) located in each IT rack. The cycloconverter converts 480 V AC to 370 V DC for input to the HVDC power distribution unit (PDU) for each IT rack. The PDU distributes 360-380 V DC to the IT loads.

Alternative energy interface

Eaton developed a photovoltaic uninterruptible power supply (PV-UPS) system based upon their existing transformer-less technology. The design includes a ground fault protection scheme to disconnect the PV panels from the system in case there is a ground fault anywhere in the PV array. With more development, a MDC equipped with this PV-UPS will be able to accept power from a wind energy source.

HVDC Battery Backup Unit

HP researched and developed an integrated HVDC Battery Backup Unit (BBU). The BBU provides alternate power for the IT loads any time there is a disruption in power from the main power source, including transitioning from the main power source to a backup generator. The BBU features a bi-directional HVDC-DC converter and lithium iron phosphate battery technology, among other innovations.

Cooling subsystem

HP researched and developed a cooling unit that can match the cooling requirements of the 100-kW IT load. The cooling unit, located in the center bay of the MDC, includes a heat exchanger

and three centrifugal system fans from EBM-Pabst Inc. that provide bulk airflow to the IT equipment. We conducted tests to characterize and analyze the thermal performance of the heat exchanger (HEX). The analysis shows that system power consumption can be reduced by increasing the supply air temperature to the ASHRAE-allowed limit of 32°C and maintaining a supply water temperature below 25°C for an IT load of 100 kW. We tested and analyzed the performance of the system fans, but we cannot fully analyze their contribution to the overall system load without direct power measurements from the server fans. Regardless, the MDC design surpassed the proposed PUE objective of 1.25 by efficiently utilizing the cooling hardware to reduce system power while maintaining environmental conditions. However, the energy requirement of chilled water from a dedicated chiller or from a larger campus loop will have a larger effect on the PUE.

Controls and monitoring subsystem

The implementation of the controls and monitoring subsystem serves as a proof of concept of a consolidated thermal controls and optimization framework developed by HP Labs. This framework integrates controls and monitoring components, adaptive control, and optimization techniques that are scalable and adaptable to existing and future product lines. The outcome obtained during testing shows promising results, indicating that the system can adapt to environmental changes while sustaining the most efficient operating parameters.

Market analysis and commercialization

Based on discussions with customers during the course of this program, there is an increasing emphasis on energy efficient power and cooling solutions in data centers. This trend is demonstrated by customers trying to improve the PUE and Total Cost of Ownership (TCO) of their data centers. Further, the accelerated adoption of the self-contained HP Performance Optimized Datacenter (POD) demonstrates that customers are willing to embrace energy efficient modular systems quickly.

HP sees a definite market need for closed-loop IT systems. We believe that these systems have good market potential because solutions such as the MDC can be utilized inside existing data centers, and solutions such as the HP POD can be used externally for larger deployments.

At this time as we continue to develop the market there has not been a final decision to productize this solution.

Introduction

Under a DOE grant awarded as part of the American Recovery and Reinvestment Act of 2009, HP and Eaton researched and developed several energy-efficient technologies required for a fully enclosed IT rack system for 100 kW of IT load. The IT rack system, herein called the Modular Data Center (MDC) provides its own internal power and cooling with HVAC—defined as 400 to 480 V AC—and chilled water as the primary inputs into the system. The MDC can also accept power from an alternative energy source such as solar, and eventually wind. For maximum efficiency, the internal power distribution converts HVAC to HVDC—defined as 360 to 380 V DC—and distributes it to the IT loads.

The management scheme of the MDC aggressively controls energy use to ensure the best use of available power and cooling resources. The MDC has an active controls and monitoring subsystem that optimizes the internal environment for given dynamic IT loads and changing system conditions. These conditions include variations in supply water and air temperatures. The controls and monitoring subsystem can also interface with the data center's Building Management Systems (BMS) to provide a complete end-to-end view of the power and cooling chain.

The goal of developing these energy efficient technologies is to achieve a PUE of 1.25, which is a 76% reduction in infrastructure energy needed to support a 100 kW IT load compared to current data center designs.

The project management plan consisted of two phases (Phase I and Phase II) lasting approximately one year each. Phase I focused on researching and developing critical technology elements, including a reconfigurable power and cooling architecture, control algorithms, and hardware risk mitigations. During Phase I, we completed investigative work for the following:

- HVAC power module
- HVDC distribution system
- Mechanical housing system and its components
- Cooling system and its components
- Software management components for both electrical and cooling

Phase II focused on system integration, testing, design refinement, and validation of these technologies, products, and solutions under simulated IT loads. Our integration testing included, but was not limited to, the mechanical, electrical, power, cooling, and controls and monitoring subsystems. We validated the mechanical enclosure and cooling subsystems with regard to ease of assembly, ease of deployment, operation, and performance.

We demonstrated the power distribution and CMS technologies. Eaton did not ship the UPS from Raleigh, NC to Houston for integration because the unit is too large and heavy for the given lab space. Eaton tested this function, including a simulated solar source, in Raleigh. Eaton also tested the solar array function using an actual array at North Carolina State University.

Background

High-voltage DC power distribution

In a typical data center, AC power from the utility goes through multiple stages to convert it to DC power, which is generally used by IT equipment. Figure 2 shows the total power conversion process from 480 V AC utility voltage to 12 volts DC in a server, depending on the equipment efficiency. In a typical double conversion uninterruptible power supply (UPS) design, incoming AC power is converted to DC to charge the backup batteries and remove anomalies (power spikes, dips, etc.). The UPS then converts DC back to AC for distribution throughout data center infrastructure. These conversions from AC to DC and back to AC result in power loss.

Additional power conversions by the power distribution unit (PDU) and server power supply unit (PSU) result in more power losses. The severity of each power loss depends on the efficiency of

the equipment. In a typical data center, this end-to-end power distribution chain has an overall efficiency of about 64%, if you include the DC-DC converters on the server system board.

Figure 2: Typical efficiencies in a data center power distribution chain

Two methods to increase efficiency are to reduce the number of power conversion stages and to operate each stage near its highest efficiency point, including good efficiency at light loads. Eaton has been working on cycloconverter technology that will enable AC to DC conversion in a single three-phase stage that provides a regulated output voltage with unity input power factor. This technology provides superior efficiency levels for an isolated output topology (Figure 3). The benefit of this DC power distribution scheme is that it eliminates the reconversion from DC to AC by the UPS and it eliminates the conversion from AC to usable DC (+12 V) by the server PSU. Also, eliminating power conversion stages (equipment) can increase available space and reduce capital and operating expenses because there is less equipment to buy and repair.

Figure 3: HVDC data center power distribution chain

There have been a limited number of demonstrations of high-voltage DC power distribution systems in data centers—the most notable was conducted by Lawrence Berkeley National Laboratory (LBNL) at a SUN data center. Previous studies have revealed a number of issues related to the supply chain, implementation, and market acceptance that currently hamper commercialization of these solutions. Another major concern about HVDC power distribution is safety. With AC power, data center personnel can connect or disconnect servers with fewer concerns about safety. However, DC current does not have a zero crossing like AC current, thus disconnection can cause arcing and other safety issues. Therefore, part of this project involved the design of new safety systems.

High-efficiency cooling systems

Raised floor data centers are the most common cooling system configuration found in modern data centers. In a raised floor configuration, computer room air conditioner (CRAC) units draw in warm air from the top of the room, cool the air, and discharge it into a supply plenum beneath the floor. The static pressure in the supply plenum pushes the air up through perforated floor tiles into the cold aisles in front of the racks. Ideally, the warm exhaust air in hot aisles at the rear of the racks rises to the ceiling where it is drawn in by the CRAC units to repeat the cycle. Raised floor configurations have a lot of inefficiency due to airflow direction, layout design, uneven distribution of chilled air, and hot air infiltration into cold areas.

The cooling system for the MDC builds on the technology developed for the HP Modular Cooling System and Performance Optimized Data Center container solutions. The basic concept is to use air in a closed-loop, contained environment and only supply what is needed for the IT equipment, thus preventing any cool air from being distributed where it is not needed and thereby wasted. The MDC uses the hot and cold aisle design but without a raised floor. Also, the controls and monitoring subsystem is designed to match the actual inlet air temperature and pressure of the cooling subsystem with those in the IT equipment specifications. This closely coupled design can react very quickly so that the inlet air temperature of IT equipment can be raised and the system fan power can be reduced to what is required to maintain proper operation.

Results and Discussion

In this section, we describe the R&D efforts for each MDC subsystem, including our technical approach, test procedures, characterization methods, results, and recommendations.

Mechanical Infrastructure

The MDC provides an enclosed environment for standard racks that hold 19” wide IT equipment (mostly servers). The enclosed environment provides a sealed system so that heat escape into the external environment is minimized. The MDC (see Figure 4) provides four to eight IT bays for 42U racks up to 600 mm (23.62”) wide x 1200 mm (48”) deep and it provides space for power distribution. The MDC includes an independent cooling unit (CU) bay in the center that provides the necessary cooling capacity for all enclosed IT racks up to 100 kW. That equates to an average cooling capacity of 25 kW/rack for 4 racks or 12.5 kW/rack for 8 racks.

Figure 4: MDC enclosure IT bays and CU bay

Each bay holds one rack and provides space in front for cold air supply and space in the rear for hot air return to the center CU bay (Figure 5). Easily removable (tool-less) brackets in the front help to seal the front from the rear and provide flexibility for removing the IT racks from each bay. Sliding rails in the bottom of each bay provide the correct fit for the IT racks. Additional basket trays in the top rear provide space for simple cable management within the system. Additional space in the top front is available for components, such as control equipment, that does not require active cooling. The front bay doors have a large viewing area. The doors provide an almost 180° opening for easy access and visibility of the installed components. Insulated rear doors and side walls minimize the amount of heat conducted into the room.

Figure 5: IT bay design

Figure 6 shows the completed enclosure with eight IT bays and the CU bay in the center. The mechanical design is based on Hewlett-Packard Patent 7,688,578: <http://www.patents.com/Modular-high-density-computer-system-7688578.html>.

Figure 6: Photo of MDC

Design requirements

An important requirement of the MDC design was to enable two people to install it within three days with the aid of an electrician and plumber to connect the power and cooling water inputs. We used the following checklist to validate and demonstrate that the CMS mechanical design met this and other design requirements for integrity, installability, serviceability, and safety.

A. Enclosure

- ☒ Quick installation time
- ☒ Installable with common tools
- ☒ Modular
- ☒ Sound mechanical integrity
- ☒ Sealed

B. Cooling Bay

- ☒ Easy hook-up of plumbing

C. Rack roll-in and roll-out capability

- ☑ Quick roll-in, roll-out the front with low degree of difficulty
- ☑ Rack guidance and parking is solid and rack is easy to anchor
- ☑ Easy full front-service and rear service of IT equipment
- ☑ Supports safe service under live operation

Results

We designed the MDC as a highly efficient self-enclosed IT structure that supports high-density and high-performance computer applications. The MDC design allows it to be installed in a conventional data center or in non-conventional environments such as a warehouse, office space, etc.

It can take years to design and build out a conventional data center, including the room, flooring, ceiling, power infrastructure, and cooling system. The MDC provides all of these components pre-configured in a self-contained environment. This reduces the time it takes to plan, design, and build out the room and reduces the effects on the surrounding infrastructure. In addition, the ability to remove and install entire IT racks with the least amount of work allows faster upgrades to the IT system itself.

The MDC provides precise cooling of targeted areas and prevents the over-cooling of other areas in the enclosure. Enclosing the hot and cold air sections saves energy by preventing the cooling of non-air conditioned areas in the surrounding environment and avoiding hot air mixing in cold areas.

The system's energy efficiency is made possible by high-performance cooling based on higher chilled water temperature and the latest and most efficient fan technology, managed by an optimized control system.

Recommendations

The current design includes additional space at the top front portion of each bay for 19-inch rack equipment that doesn't need active cooling or can survive in a hotter environment. If it is determined that this type of equipment will not be deployed, the space could be used as a path to run cables from front to rear. This space can also be used for other purposes, but it was not fully defined.

The IT bays are designed to fit the HP 10000 G2 1200-mm deep rack. The bays can be modified to support racks that are not as deep, but it will require additional brackets that accommodate this depth.

On the rear of the IT racks, cable management and PDU mounting brackets could be included for easier installation.

High-Efficiency Power Distribution System

The power distribution system feeds redundant 480 V, 3-phase power to each rack bay to the Eaton cycloconverter (Figure 7). The cycloconverter converts 480 V AC to 370 V DC for input to the High-Voltage DC Power Distribution Unit (HVDC PDU) for each IT rack. The PDU distributes HVDC power (360-380 V DC) to the IT loads.

This distribution system has an integrated HVDC Battery Backup Unit (BBU). The BBU is a high efficiency, battery charging and discharging solution. The purpose of the BBU is to maintain the HVDC voltage by automatically switching between the input supply (Eaton Rectifier) and the backup batteries in the event of a main power loss. The BBU uses lithium iron phosphate (LiFePO₄) battery technology.

The power distribution system also accepts alternative energy power sources such as solar and with more development, wind. Eaton developed a PV-UPS system based upon their existing transformer-less technology. The output of the PV-UPS is paralleled with protected output derived from the utility.

The distribution system required the design of new conductor/connector/safety systems to ensure efficiency and safety.

Figure 7: Power architecture and division of responsibility

AC Distribution Subsystem

Technical approach

Eaton designed the AC distribution subsystem (Figure 8) to integrate into the multi-bay mechanical enclosure assembly (Figure 9). The subsystem includes branch circuit monitoring, a display, and communication features. The transfer tray located above the Heat Exchanger (HEX)

supplies power to the system fans.

Figure 8: AC distribution subsystem prior to shipping from Eaton (arms in down position)

Figure 9: Illustration of MDC with integrated AC distribution subsystem

Initially, Eaton assembled a prototype of the AC distribution subsystem using existing equipment—a display, controls, monitoring, and switchgear—and shipped it to HP for integration testing in the lab. The unit shipped with pre-wired AC outlets and with the cable arms placed in the down position. At the HP site, installers raised and secured the arms to the multi-bay enclosure assembly. The arms accommodate direct connection of the IT equipment power supplies in the bays (maximum 8 bays, 4 bays on each side of the HEX). Based on the integration testing, the design was modified and Eaton built a second AC distribution subsystem. The AC distribution method within the multi-bay enclosure assembly was improved and the arms were redesigned to allow additional routing space for communication cables in the top of the bays (Figure 10).

Figure 10: AC distribution subsystem installation

This version also included a power transfer switch tray (see Figure 11). The transfer tray provides a single 480 V 3-phase output to power the system fans. It includes a full-featured transfer switch that provides power to the fans from the primary power source. When the primary source fails, the tray transfers the fan load to the secondary or backup power source. When the primary power source becomes available, the tray transfers the fan load back to the primary source. The transfer switch can be configured through a user interface panel. The transfer tray also provides redundant power supplies to support the controls and monitoring system power needs. The tray houses the individual fan circuit protection in accordance with safety requirements.

Figure 11: Transfer tray views from the front and the rear

The distribution subsystem uses an auto-zigzag transformer to supply a limited amount of power at a lower voltage (240 V AC single phase) for convenience needs. The main characteristic of the auto-zigzag transformer configuration is that it does not require a source neutral connection. The configuration provides a single-phase 240 V AC independent of the percent loading or balancing, and it distributes the load on the 3-phase delta source. The auto-zigzag transformer configuration is smaller than a conventional transformer at the same power level, allowing it to fit in the bottom of the AC distribution subsystem. The auto-zigzag transformer configuration was provided for both the primary and secondary AC sources so the outlets for convenience needs can support redundant power configurations.

Eaton Rectifier

The Eaton Rectifier consists of the major electrical blocks shown in Figure 12. The EMI filter complies with FCC regulations on electromagnetic interference. The full bridge cycloconverter is the main innovative component. It switches at a frequency varying from 80 to 250 kHz and is controllable to draw power factor corrected current from the input line while regulating the output DC voltage. The voltage transfer function from the cycloconverter to the output is achieved through the LLC resonant tank and integrated isolation transformer. A low-loss diode bridge rectifies high-frequency sine wave current. A boost holdup circuit maintains the output voltage for a short time when the utility voltage is lost so that other parts of the subsystem have time to respond.

This converter topology avoids the need for multiple active power conversion stages, while integrating galvanic isolation and an output voltage below the peak of the input voltage.

Figure 12: Eaton rectifier and electrical blocks

Two rectifiers (parallel or redundant) are housed in a Power Shelf as shown in Figure 13. The Power Shelf is rack-mountable, allowing the rectifiers to be easily swapped out if necessary. Features of the Power Shelf include:

- Delta or Wye inputs, 380-480VAC
- Inherent phase balancing, <2%
- Parallel-capable A and B feeds, non-sync
- Scalable output power, ~12 kW per rectifier
- Isolated, 380 V DC nominal output
- Ethernet Communication

Figure 13: Illustration of Power Shelf with one Eaton Rectifier partially inserted

Results

The original target for the Eaton Rectifier was 97% efficiency at 50% load for a 20-kW converter. Because of limitations of the state-of-the-art power semiconductors available, the 20-kW rating is not practical. Therefore, the rating was changed to 12 kW. With nominal input voltage, Eaton achieved a peak efficiency of 95.7% (Figure 14) due to extra power losses not accounted for in original projections.

Figure 14: Eaton Rectifier – Instantaneous efficiency over input voltage range

The primary contributions to the extra power losses were:

1. EMI filter devices—not originally anticipated—were required in order to meet FCC specifications.
2. Over-voltage and inrush current protection devices were required to help compensate for best available components that had low safety margins in this application.
3. There was insufficient material and assembly lead time to modify the resonant tank, which was designed for 20 kW, to suit the 12 kW target.

Eaton did not have the opportunity to enable the power factor correction algorithms that can be used with this topology. The main impact is that input current harmonic distortion is too high to commercialize the existing product. However, the measured power factor is high, in the range of 0.92 to 0.94. While original research targets were not met, it is important to note that this rectifier has an approximate power density of 13.55 watts per cubic inch and a peak efficiency of 95.7%.

The results of this research and the expected increases in efficiency are promising. However, the topology chosen is highly complex and difficult to control. Once higher performance components are available, it will likely be simpler and possibly more cost effective to use a standard topology with other enhancements to provide isolation and delta voltage source operation.

Recommendations

One area for improvement of the rectifier is the 900 V rating of the silicon-based, super-junction (SJ) MOSFETs that were used to form the AC switches of the cycloconverter. To have good voltage margin, the SJ MOSFETs should be rated at 1200 V. Low on-resistance 1200V silicon SJ MOSFETs are not available. New silicon carbide-based switching devices that offer low on-resistance at 1200 V (with a very small die size) are just becoming available. One such device called a silicon carbide (SiC) JFET can be used in a cascode arrangement to form the AC switch. Eaton tested an equivalent on-resistance 1200V SiC JFET cascode AC switch in the cycloconverter and successfully produced 12 kW at the same efficiency. A new version of the device with half the on-resistance will soon be available in the same package. This will allow the power level to increase to 20 kW and should increase efficiency by 1%.

High Voltage DC Power Distribution Unit

The HVDC PDU distributes the output of the cycloconverter rectifier (Figure 15). It also provides protection by detecting short circuits, limiting current inrush, as well as detecting series and parallel arc faults and automatically isolating these issues from the rest of the DC bus in positive grounded systems. These protections are provided by electronic interruption means with a DC rated fuse backup in accordance with safety requirements.

Figure 15: HVDC PDU in rear of rack

The PDU has four groups of 20-A loads with eight total active circuits each protecting two HVDC load outlets. This provides 16 load connection points in each HVDC PDU (Figure 16). The PDU:

- Distributes HVDC within the rack
- Allows electronic disconnects, fault isolation, anti-arcing technologies
- Provides distribution and protection for attached loads
- Protects the HVDC bus from notching caused by:
 - Inrush from newly connected load
 - Overcurrent
 - Arc-faults series and parallel types

Figure 16: HVDC PDU

Results/Discussion

Eaton delivered eight final HVDC PDU prototypes. These PDUs protect the downstream load from overcurrent and arc-fault failures as well as protect the HVDC bus from new load connections that might droop the bus. Eaton tested each PDU for functionality and performed the following tests during qualification of the HVDC PDU:

- Arc-Fault detection functionality tested for series arcs (Figure 17)
- Short circuit (Desat) protection tested for parallel arcs and short circuits (Figure 18)
- Short circuit protection tested against load pickup (input capacitor charging)
- Form factor prototype tested for short circuit and arc fault protection
- Hipot testing of circuits to chassis

Figure 17: Arc fault detection scope capture

Figure 18: Desat detection scope capture

Improvements

These protection techniques are not limited to positive grounded systems. The techniques can be extended to negative and ungrounded systems by adding a semiconductor switch. The response time of the microprocessor to a Desat event can be shortened by using a dedicated hardware circuit. This was due to the latency of the microprocessor to read and react to the sampled desaturation input. Late in the program, Eaton became aware of such a hardware circuit that is built into a commercially available MOSFET drive circuit. This circuit, along with the developed algorithms, can reduce the size of the current-limiting inductor in each protection circuit. The net result will be to improve the response time to power off the MOSFET after the desaturation event is detected. This will result in a smaller, lower cost inductor that can be used to limit the rate of current rise.

Direct Current Battery Backup Unit

The BBU is a high efficiency, battery charging and discharging solution that automatically switches between the input supply (rectifier) and the backup batteries. The purpose of the BBU is to maintain the HVDC bus in the event of a main power loss or during the brief transition between the main power source and a backup energy source. The BBU uses lithium iron phosphate (LiFePO_4) battery technology. Conventional UPSs use lead acid batteries, which are cost-effective but increasingly unpopular due to their size, weight, toxic contents, maintenance requirements, and relatively short lifespan. In contrast, LiFePO_4 batteries have a smaller form factor, provide higher power, generate less heat, and an average life cycle of five years.

As shown in Figure 19, each BBU shelf contains four hot swappable battery trays. Each BBU tray contains two LiFeSO_4 battery packs, a battery management unit, and a power board with a bi-directional HVDC-DC converter. Each battery pack consists of several battery cells that are connected to provide the rated power and current. The battery management unit (BMU) monitors the voltage levels of the individual battery cells and the entire battery pack. The bi-directional HVDC-DC converter charges and discharges the batteries. The Advanced Power Manager (APM) module is an existing HP product that performs monitoring, control, and discovery functions.

Figure 19: BBU shelf containing four hot-swappable trays

Under normal conditions, the bi-directional converter charges the batteries at constant current until the voltage level reaches the maximum for each battery pack. Then the converter reduces the current to maintain each battery pack at constant voltage and to minimize heat generation. The converter has a nominal input voltage of 360 V DC and can withstand input voltages up to 420 V.

When the main power source fails or is being switched to a renewable or alternate power source, the converter discharges the batteries to maintain the HVDC bus. When the converters in the trays discharge, they operate in parallel to share the load (Figure 20). The batteries are designed to hold the specified output load versus output voltage characteristics to ensure that each converter produces the specified amount of current. We developed and tested the bi-directional converter as per the requirements and specifications listed in “HVDC-DC Converter Results” below.

Each BBU shelf can provide over 6000 W. Two BBU shelves supply the equivalent power of each rectifier. To provide the rated output power of one Power Shelf (two Eaton Rectifiers), four BBU shelves (sixteen battery trays) are required.

Figure 20: Bi-directional converters operate in parallel to share the load

HVDC-DC Converter Results

HP developed and tested the HVDC-DC converter (HP Server HVDC Input in Figure 7) according to the requirements and specifications. Below is a summary of the results of the R&D effort.

HVDC Input

The module operates with a nominal input voltage of 360 V DC. The primary return is isolated from secondary ground. The module is capable of start-up (power-on) with rated output load, at line voltages as low as 330 V DC measured at the input of the module. The module can withstand input voltages up to 420 V.

Brownout, Under Voltage

The module is protected such that the application of an input voltage below the minimum does not damage the module or cause failure of the input fuse. If the module shuts down due to extended brownout, the module automatically restarts once the DC input is restored to the limit.

Output

The output voltage, when measured at the host side of the module output connector, is +12V for the rated input voltage of +360V as per the specification. The maximum steady state output DC current from the +12V bus does not exceed 100Amps.

Inrush Current

Inrush current limiting is provided by circuits connected external to the module inside the enclosure. Start-up inrush current drawn by the module shall not trip the external input current limiting circuits.

Parallel Operation

The module can operate in parallel with similar modules. All specifications except output load current, power, input current, power, and output load transient pertaining to a single module apply to all modules operating in parallel.

Output load sharing in parallel operation

The modules share the load when operating in parallel. Each module is designed to hold the specified output load vs. output voltage characteristics to ensure the specified amount of current is sourced from each module (Load / no. of parallel modules).

Maximum Output Power

A single unit consisting of two HVDC-DC modules is capable of continuously supplying 1200 W output power from the +12V output under all specified conditions.

Output Voltage Regulation

Output voltage regulation was observed for the rated power at input voltage range of 330 to 420 V DC. The results match the expectations for the specified input voltage range.

Alternative Energy Interface

Technical approach

Eaton led the effort to research and develop an efficient alternative energy interface into the system. The interface uses currently available power conversion stages as much as possible. An Eaton 9395 modular UPS was used as a starting point to develop a PV-UPS integrated system (see Figure 21). Input from the PV installation is fed into one power module and its output is paralleled with protected output derived from the utility. A protection cabinet is installed at the PV input. The cabinet houses ground fault detection equipment as well as safety and disconnect devices to isolate a ground fault.

Figure 21: PV-UPS system overview

Eaton successfully conducted tests using a PV simulator in the lab and using a live PV array (Figure 22). The system operation was tested in both double conversion and battery modes. The UPS extracts as much power as it can from the PV under given solar intensity. The extracted power is limited only by the maximum point of PV power or the load, whichever is smallest. The load is served uninterrupted under all conditions whether the PV power is greater than or less than the load. The system adapted well to changing PV power availability as clouds passed overhead while complying with the Uninterruptible Power Supply Standard UL1778 for back-feed protection. Transfers from double conversion mode to battery mode and back were smooth without interrupting the load.

Eaton implemented a previously designed ground fault protection scheme to disconnect the PV panels from the system in case there is a ground fault anywhere in the PV array. The scheme was tested by applying a ground fault at different locations in a PV array and verifying that it was able to isolate the fault each time. The results support a patent Eaton previously applied for known as “Systems and apparatus for fault detection in DC power sources using AC residual current detection.”

Figure 22: Views of the live solar array used to test the system

Results

The results obtained during testing serve as a proof of concept that a UPS can be interfaced with a fluctuating renewable energy source like solar energy and still serve a varying load. Transfers between double conversion and battery modes were seamless.

There is an ongoing effort to develop firmware that makes the system work in Energy Saver System (ESS) mode to achieve the highest possible efficiency. Some fine-tuning is required to make the system more efficient and robust to deal with large load steps and sudden large PV power changes.

Recommendations

Commercializing this technology would require a full validation and qualification test cycle. The test cycle should include any regulatory testing and the testing of all operating points of the system. The protection cabinet design also needs to be completed and tested for functionality and regulatory compliance.

Cooling System

The cooling system research and development effort involved designing a system that can match the cooling requirements to the IT load. This will allow data centers to increase the inlet supply temperature of the servers to the manufacturers' upper limit. The increased inlet supply temperatures allow for warmer supply water temperature and a significant reduction in external cooling.

This section presents the results of MDC thermal tests performed to characterize the heat exchanger, efficiency metrics, and system stability. After introducing the hardware and instrumentation utilized, we explain the theoretical method used to characterize the heat exchanger. The main portion of this section describes the test plan and execution of thermal tests that produced the thermal characterization curve and conductance curves. In "System stability tests," we include tests running the system with various scenarios to test for steady state stability. The details of the testing procedure, including charts are included in Appendix A.

Hardware and Instrumentation

We used two types of server enclosures to simulate the heat generated by servers: HP ProLiant SL 6500 enclosures and HP BladeSystem C7000 enclosures. We deployed twenty SL 6500 enclosures and eight BladeSystem C7000 enclosures in four racks (Figure 23). We populated the server trays of these enclosures with a series of resistors to simulate the heat load of ProLiant SL and BL servers, respectively, to reach the simulated IT load of 100 kW.

Figure 23: Four-bay configuration with SL and BladeSystem enclosures

We placed the sensors listed in Table 1 on the water side and air side of the system as shown in Figure 24 and Figure 25, respectively. We connected the data acquisition (DAQ) equipment to a PC where data were gathered by the Agilent VEE data acquisition software and saved to an Excel worksheet for analysis.

Table 1: List of sensors deployed in the system

Equipment	Type (Company, Type)	Measurement	Uncertainty
Temperature sensors	Omega - RTD	Inlet and Outlet water temperatures – $T_{w_{in}}$ and $T_{w_{out}}$	$\pm 0.35^{\circ}\text{C}$
Pressure transducers	Honeywell – Strain gauge	Inlet and Outlet water pressure – P_{in} and P_{out}	$\pm 0.5\%$
Flow meter	Flow Technology – Positive displacement flow meter	Water flow rate – V	$\pm 0.05\%$
Heat exchanger motor controlled water valve	Belimo – Characterized control ball valve	Position of the valve	
CU fan inlet rings with differential pressure sensors	Bapi sensors – Silicon capacitive pressure sensor	Air flow rate through CU	$\pm 0.5\%$
Thermocouples	Omega – J type	Inlet and outlet of CU and Rack air temperature	$\pm 1.1^{\circ}\text{C}$
PDU	HP iPDU	IT load	
DAQ	Agilent Technologies	Data acquisition from all sensors	

Figure 24: Schematic of instrumentation utilized in the water loop

We installed a dedicated chiller to support the MDC. Note that incorporating a dedicated air cooled chiller for a single MDC unit is not a typical product configuration.

On the air side of the system, we placed 56 thermocouples (indicated by red dots in Figure 25) in the front and rear of each server enclosure to measure inlet and outlet air temperatures. We placed 12 total thermocouples in front of the CU fans and the heat exchanger. The thermocouple beads were less than 1 inch away from the target locations.

Figure 25: Position of thermocouple beads in front and rear of the system

We installed pressure tubing on high and low pressure locations of inlet rings around the circumference. We measured three differential pressure readings for the three fans using Bapi pressure sensors (Figure 26), which were connected to the pressure tubes. Then we calculated the air flow rate by using the following equation.

—

Where:

= Air flow rate in CFM

= Differential pressure in Inches of water

Figure 26: Bapi pressure sensor

Theoretical considerations

The aim of the thermal experiments is to achieve high thermal effectiveness of the air-to-water heat exchanger for a particular set of conditions. We developed characterization curves for the heat exchanger based on three variables: thermal effectiveness, heat capacity ratio, and number of transfer units (NTU). These variables represent all working possibilities of the heat exchanger.

We considered two approaches to characterize the thermal performance of the heat exchanger: the ϵ -NTU method and the P-NTU_t method.

ϵ -NTU method

Total heat transfer rate in the heat exchanger from hot to cold fluid is calculated by

Where:

ϵ = Heat exchanger effectiveness

= Lowest heat capacity rate among both of the fluids

= Temperature of hot fluid at the inlet of heat exchanger

= Temperature of cold fluid at the inlet of heat exchanger

$$\begin{aligned} & \text{_____} \\ & \text{_____} \\ & \text{_____} \end{aligned}$$

The ϵ -NTU method required us to develop a thermal effectiveness curve with the two variables ϵ (heat exchanger effectiveness) and C_r (heat capacity rate ratio). This method presented a problem with tracking the minimum heat capacity rate ($C_{\min}$) among the two fluids. Therefore, we found that possible errors during transitional periods can be avoided by employing P-NTU_t method.

P-NTU_t method

The P-NTU_t method is typically used for shell and tube heat exchangers and the concept can be utilized in this particular application with shell containing fluid as air and tube containing water.

Heat transfer rate, $q = PC_t (T_1 - t_1)$

Where:

P = Thermal Effectiveness

C_t = Heat capacity rate at tube-side fluid

t_1 = Temperature of the hot fluid at the shell inlet

T_1 = Temperature of the cold fluid at the tube inlet

Thermal Effectiveness (P)

We plotted thermal effectiveness charts $P-C_r$ and $P-NTU_t$ based on the experimental data acquired (see “Test overview” section).

P can also be derived from ϵ -NTU method with the following conversion.

Heat capacity ratio (R)

Number of transfer units (NTU_t)

Unlike ϵ -NTU method, doesn't need to be tracked since the ratio is fixed for all cases despite hot and cold fluid location in shell or tube. Thermal effectiveness charts $P-C_r$ and $P-NTU_t$ are plotted based on the experimental data acquired which is explained in the next section.

Technical approach

We tested two heat exchangers with coil models 14/8x4x915/14-16 and 14/8x4x915/14-17 in succession. For convenience, we refer to these coil models as Proto1 and Pilot1, respectively. To minimize redundancy, we discuss only the results for the Pilot1 heat exchanger in detail.

Proto1 test procedure

The inlet temperatures, outlet temperatures, and flow rates of the two fluids (air and water) are required to plot the P-Cr chart. We adjusted the thermal server load fans to a speed that results in an air temperature difference between inlet and outlet of about 20°C and the speed of the server fans remained fixed for a particular IT load. We created the test plan to estimate the minimum water flow rate required for a given set of inputs – IT load, air flow rate, cold aisle temperature, and inlet water temperature (see Table 2).

Table 2: Variables involved in the test plan

Inputs	Outputs
IT load	Water flow rate
Air flow rate – CU fans	
Cold aisle temperature	Outlet water temperature
Inlet water temperature	

We used a proportional-integral-derivative (PID) controller to regulate water flow rate with a Belimo control water valve to achieve a specific cold aisle temperature for a specific air flow rate and inlet water temperature (Table 3).

Table 3: Test matrix to estimate minimum water flow rate and outlet water temperature for 100 kW IT load

Inlet water temperature		8°C H ₂ O			12°C H ₂ O			16°C H ₂ O		
Cold aisle temperature		20°C	25°C	30°C	20°C	25°C	30°C	20°C	25°C	30°C
Cooling unit fan flow rate	11160 cfm									
	10600 cfm									
	9400 cfm									

Table 4 shows the air flow rates used to cool three IT loads—100 kW, 75 kW and 50 kW—generated by the racks of SL servers and BL blade servers: When the IT load decreased to 75 kW or 50 kW, the thermal load was distributed evenly among the racks. We decreased the server fan flow rates to maintain the same temperature difference of 20°C across the servers.

Table 4: Air flow rates to maintain a 20°C difference across 100-kW, 75-kW, and 50-kW IT loads

IT load	100 kW	75 kW	50 kW
Air flow rate	11160 cfm	11160 cfm	10600 cfm
	10600 cfm	10600 cfm	9400 cfm
	9400 cfm	9400 cfm	7800 cfm

Proto1 test results

We performed a total of 57 test cases. Table 5 provides the estimated minimum water flow rate required to cool 100 kW IT load while achieving the required cold aisle temperature. The remaining data, including IT load of 75 kW and 50 kW, are included in Table 22 and Table 23 in Appendix A. It was not possible to perform test cases for inlet water temperature at 16°C and cold aisle temperature at 20°C because the heat exchanger tubes have a flow rate limitation of up to 200 lpm. Other issues also occurred at very low water flow rates (< 10 gpm), which made it difficult to gather data due to the transitional effects of three way valve with less uncertainty.

Table 5: Minimum water flow rate (GPM) required to cool 100 kW

Inlet water temperature		8°C H ₂ O			12°C H ₂ O			16°C H ₂ O		
Cold aisle temperature		20°C	25°C	30°C	20°C	25°C	30°C	20°C	25°C	30°C
Cooling unit fan flow rate	11160 cfm	20	14	11	35	19	13	--	27	15
	10600 cfm	22	15	12	36	21	13	--	28	17
	9400 cfm	23	15	12	41	21	14	--	35	19

We calculated the values for Thermal effectiveness (P) and Heat capacity rate ratio (R) using the following equations from the P-NTU_t method.

—————
= Outlet water temperature

= Inlet water temperature

= Inlet air temperature (hot aisle temperature)

—————
= Heat capacity rate of water

= Heat capacity rate of air

The curve-fit equation generated in Excel as seen in the Figure 27 with R-squared value of 98.5%. At R=1 and P=0.6, the heat capacity rates of the both fluids are equal and R>1, heat capacity rate of air dominates over water and vice versa.

Eq 1

$R = f(P)$

Figure 27: P-R thermal effectiveness chart

Pilot1 test procedure

It was assumed that the performance of the Pilot1 heat exchanger is identical to Proto1. A thermal control algorithm based on Proto1 thermal effectiveness curve is developed replacing PID controller (used in Proto1 testing) to predict water flow rate needed to a specific test configuration.

= Eq 2

=

Eq 4

—

Substituting Eq 4 in Eq 3

$$\frac{\text{---}}{\text{---}} = \frac{\text{---}}{\text{---}}$$

Replacing in Eq 6

$$\frac{\text{---}}{\text{---}}$$

Expanding R with Eq 1

The variable L_f can be evaluated at any instant during the test at every data acquisition point. Thermal effectiveness, P can be solved from the fourth order polynomial above and eventually Heat capacity rate ratio, R from Eq 1. The minimum water flow rate required to cool the IT load can be easily calculated from R . In Figure 28, the water valve characterization curve is used to attain the required flow rate by regulating the valve position.

Figure 28: Belimo water valve characterization curve

The control scheme is based on the evaluation of the water flow rate equation which is recalculated every iteration (30 seconds). A basic representation of the algorithm is shown in Figure 29. This chart captures the workflow of the algorithm from the start of the data acquisition process to solving for P and R to determining C_w and converting this value to the corresponding water valve voltage.

Figure 29: Flow chart representing the basic concept of the controller used in Pilot1 testing

Ten test cases were designed to check if Pilot1 heat exchanger yields same thermal effectiveness curve as in Figure 30 of Proto1. The test cases were created to gain data that includes the entire range of thermal effectiveness from 0.3 to 0.9.

Pilot1 test results

Table 6 presents the minimum water flow rate required for the specific IT load applied to the Pilot1 heat exchanger. The thermal effectiveness curve is plotted comparing Pilot1 data with Proto1 data in Figure 30. Pilot1 data follows Proto1 data in the water-dominated heat capacity rate ratio region ($R > 1$), but the thermal effectiveness (performance) of Pilot1 is lower in the air-dominated region ($R < 1$).

Table 6: Pilot1 heat exchanger thermal test results

Test case no.	Inputs				Outputs		
	IT load (kW)	Cold aisle set point (°C)	Inlet water temperature (°C)	Air flow rate (cfm)	Water flow rate (gpm)	Thermal Effectiveness, P	Heat capacity rate ratio, R
1	160	25	12	13290	38.7	0.483	1.305
2	150	20	8	12450	36.8	0.471	1.361
3	138	20	8	11126	33.4	0.483	1.323
4	127	25	8	11015	17.4	0.706	0.671
5	122	20	12	9604	36.7	0.432	1.526
6	105	20	16	11072	55.6	0.293	2.575
7	104	20	8	9468	23.3	0.535	1.158
8	103	25	12	9428	24.8	0.524	1.218
9	99	25	8	9348	14.6	0.678	0.749
10	49	30	16	9123	11.1	0.703	0.686

Consider Test case no. 4 from Table 6 as an example to quantify the decreased performance. In Figure 30, the thermal effectiveness of Pilot1 for Test case 4 is 0.706, but the equivalent thermal effectiveness for Proto1 is 0.762. This lower effectiveness of Pilot1 will result in either increased water flow rate or colder inlet water temperature to maintain the specific test conditions. Proto1 can achieve the same conditions of Test case no. 4 with less water flow rate and warmer water as shown in Table 7.

Figure 30: Pilot1 thermal effectiveness curve comparison with Proto1

Table 7: Decreased performance of Pilot1 heat exchanger when compared to Proto1

	Effectiveness	Water flow rate (gpm)	Δ gpm	Water inlet temp (C)	Δ C
Pilot1	0.706	17.44	1.3 (8%)	7.67	-2.93
Proto1	0.762	16.14		10.6	

Thermal conductance curves

The thermal control algorithm used in Pilot1 testing can be applied to conductance curves, which are widely used in the industry for selecting a heat exchanger. It can transform the control system efficiently by streamlining the control process using readily available conductance data from the manufacturer.

Thermal conductance of a heat exchanger can be calculated by dividing the heat load with temperature difference of the two fluids at the inlets. The chart in Figure 31 shows the thermal conductance versus air flow rate at different water flow rates. For a given heat load, the thermal conductance increases with decreased temperature difference of the fluids at inlets. This results in need of higher flow rates for the two fluids in order to maintain the heat load-as observed in Figure 31. A control can be developed to attain lower flow rates possible for a given IT load to

minimize power consumption of the cooling unit fans and chiller. Table 8 shows two test cases with different inlet water temperatures that can cool an IT load of 100 kW. However, the power consumption of the CU fans is much higher for Test case 2 resulting in a higher PUE value.

Table 8: Comparison of two sample test cases PUE can be minimized using thermal conductance curves

Test case	Load (kW)	IWT (°C)	Hot aisle temp (°C)	Conductance (kW/C)	CU fan flow rate (cfm)	CU fan power (kW)	Chiller power (kW)	PUE
1	100	8	35	3.7	8,700	2.8 kW	27.5	1.3
2	100	16	35	5.26	13,400	7.6 kW	25.5	1.33

Figure 31: Pilot1 heat exchanger conduction curves

Power Usage Effectiveness (PUE) data

PUE is a widely used metric to measure the effectiveness of facility infrastructure used to cool IT load in a data center. PUE is defined as the ratio of total facility power to IT equipment power. This study is extended to another metric called partial PUE (pPUE), which excludes the chiller in the system (Figure 32).

Figure 32: Power consumption breakdown in Pilot1 test set-up

The power consumed by the server loads were read directly from the PDU. This reading includes the power consumed by the server fans, power supply, and heat loads. The power consumed by the chiller is a sliding block average of 60 readings over 15 minutes, while the power readings for the cooling fans were acquired individually.

Table 9 shows two test cases we performed to estimate the minimum and maximum PUE of the four-bay Pilot1 test setup with a dedicated chiller.

Table 9: PUE test configuration

Test case	Test type	Load (kW)	Inlet water temperature (°C)	CU fan flow rate (cfm)	Cold aisle set point (°C)
1	Min PUE	100	16	9420	25
2	Max PUE	100	8	11050	20

We performed the Min PUE test on December 12, 2010 for 5 hours beginning at 11:30 AM. The average ambient temperature was 7.53°C. The chiller was unable to maintain the inlet water temperature—it averaged 16.3°C with an uncertainty of 2°C. Due to this anomaly, the thermal control algorithm in Agilent VEE needed to adjust the water flow rate to reach the cold aisle set point temperature of 25°C as seen in Figure 33. Oscillatory data from chiller power (Figure 34) was reflected in the PUE with an uncertainty of 0.043 with an average value of 1.23 (Table 10). Chiller power is the most significant factor affecting PUE, and it reached its minimum due to low ambient temperature and higher inlet water temperature. However, pPUE was unaffected since the chiller power was not considered.

Figure 33: Temperature data acquired during the test

Figure 34: Power and PUE data from the Min PUE test configuration

Table 10: Averages and Uncertainty values of variables in the test

	IT load (kW)	Chiller power (kW)	CU fan power (kW)	Inlet water temperature (°C)	PUE	pPUE
Average	103.5	21	2.8	16.3	1.231	1.027
Uncertainty	0.02	2.18	0.02	2.03	0.04	0

We performed the Max PUE test on December 15, 2010 for 7 hours beginning at 9:30 AM. The average ambient temperature was 22.3°C. Unlike the Min PUE test, the chiller maintained the inlet water temperature (Figure 35), resulting in a uncertainty in PUE as seen in Figure 36 and Table 11. From Figure 35 it is observed that the data acquired is steady state with two sample cold aisle temperatures plotted. The Max PUE is achieved with higher power consumption by the chiller and cooling unit fans. Higher ambient temperature and lower inlet water temperature resulted in the maximum chiller power consumption possible.

Figure 35: Temperature data acquired during the Max PUE test

Figure 36: Power and PUE data from the Max PUE test configuration

Table 11: Average and Uncertainty values of variables in the Max PUE test

	IT load (kW)	Chiller power (kW)	CU fan power (kW)	Inlet water temperature (°C)	PUE	pPUE
Average	101.9	31.3	4.5	8.53	1.352	1.045
Uncertainty	0.03	2.3	0.02	0.23	0.022	0.0002

System stability tests

This section presents the results of three types of tests to ensure that the system is reliable in case of failure of any individual component, unusual placement of racks, or asymmetric placement of IT loads.

Two cooling fans operation tests

The objective of this test is to confirm the stability of the system when one of the cooling unit fans fails to perform and also to compare with normal three-fan operation to find the extent of inefficiency attributed to the failure. Seven test cases are listed in the Table 12, including the baseline case No. 1. All test cases were run at a CU fan flow rate ranging from 9000 to 9400 cfm. When a CU fan failed, the remaining two fans were forced to run at higher speeds to reach an air flow rate of 9400 cfm, which is the maximum air flow rate feasible by two fans.

Table 12: Test cases for Two cooling fans operation tests

No.	Test cases	
1	Normal operation – Three CU fans - ON	IT load 100 kW, Cold aisle set point temperature 20°C, CU fan flow rate 9400cfm and Inlet water temperature 8°C
2	Top fan – OFF	The failed CU fan is in the system and not blanked
3	Middle fan – OFF	
4	Bottom fan – OFF	
5	Top fan – OFF & Blanked	The failed CU fan is blanked with a cardboard to prevent air movement in either direction
6	Middle fan – OFF & Blanked	
7	Bottom fan – OFF & Blanked	

Figure 37 plots the average rack front and rear air temperatures of the above mentioned test cases. It can be concluded from the figure that the temperatures are unchanged when one fan fails, confirming that the system is reliable (in the same state as normal operation). Figure 38 shows that the blanked cases have a little higher flow rate than unblanked cases. As the air follows the path of least resistance, it is sucked back into the failed fan from the cold aisle and exits through the functioning two fans. However, the performance improvement between the blanked and unblanked cases is negligible.

Figure 37: Average rack front and rear air temperatures when two fans are operating

Table 13 shows that there is no significant change in PUE due to inefficiency when comparing the unblanked test cases to the normal operation test case.

Table 13: PUE comparison of unblanked test cases with normal operation

Test case	IT load (kW)	Chiller power (kW)	CU fan power (kW)	PUE
Bottom fan – OFF	102.1	3.9	28.7	1.319
Middle fan – OFF	102	3.8	28	1.312
Top fan – OFF	101.8	3.9	28.6	1.319
Normal operation	102.4	2.9	29.6	1.317

Figure 38: CU fan flow rate and fan power

Rack removed from the system

This test involves completely removing one of the four racks in the system to check system stability in case a rack of equipment is removed for servicing, leaving an empty bay. It is important to make sure that the system does not become unstable when there is a possibility for air to bypass the filled racks through the empty rack space. We ran two tests to demonstrate these scenarios by removing L1 and R2 racks individually from the system.

For the baseline case in Figure 39, the IT load was 100 kW, the cold aisle set point temperature was 20°C, the CU fan flow rate was 11000 cfm, and the inlet water temperature was 8°C. When we removed the L1 rack, the IT load dropped to 75 kW. The air flow rate required to cool 75 kW and maintain the temperature difference across the server loads was approximately 9500 cfm, a decrease of 1500 cfm. The figure shows that the average “L1 out Front” and “L1 out Rear” temperatures are slightly lower than the baseline case. The remaining 1500 cfm is consumed partly in slightly lowering the cold aisle and the rest is bypassed through the open rack space.

Figure 39: Average rack air front and rear temperatures when Rack L1 is removed comparing with the baseline test case

Figure 40 shows that the average front temperatures are similar for the “R2 out” test case and the baseline case. Temperature difference across the R1 rack decreased from 21°C to 15°C. Using an energy balance equation, we calculated that the air flow rate required to obtain this temperature drop is 1200 cfm. Out of 1500 cfm, the remaining 300 cfm appears to bypass R1 to the inlet of heat exchanger.

Figure 40: Average rack air front and rear temperatures when Rack R2 is removed comparing with the baseline test case

The results of these two tests demonstrate that removing a rack from the system will not create instability. Because the IT load dropped to 75 kW, a higher PUE value will be observed. But the PUE can be improved by decreasing the speed of the CU fans to lower the redundant air flow rate.

Asymmetric placement IT loads

We conducted two tests with asymmetric IT loads. For one asymmetric test, a 100 kW load is divided into 30 kW on the SL racks in bays L1 and L2 and 70 kW on the Blade racks in R1 and R2. In the other test, the load distribution is reversed. The RPM of the server fans are increased or decreased accordingly to maintain the temperature difference across the server to be around 20°C as shown in Table 14. For the baseline case, the IT load was 100 kW, the cold aisle set point temperature was 20°C, the CU fan flow rate was 11000 cfm, and the inlet water temperature was 8°C.

Table 14: SL and Blade server fan speed for asymmetric tests

Test case	SL fan speed (PWM)	Blade fan speed (RPM)
100(50, 50) kW	40%	11000
100(30+70) kW	25%	18000
100(70+30) kW	60%	5500

No instability is apparent in Figure 41 when the IT load is distributed unevenly. The server fan speeds were adjusted by trial and error method and, due to human inaccuracy, lower hot aisle temperatures are observed in some cases. It can be concluded that the CU fans adapted to asymmetric heat load effectively by distributing more cold air to low-pressure areas.

Figure 41: Average rack front and rear air temperatures of asymmetric tests comparing with baseline case

Running the system with an unevenly distributed IT load decreases cooling performance. In general, server fans are selected based on the server's most efficient power consumption at a particular range of fan speeds. If the fan exceeds the optimized range, power consumption jumps according to the cube law with respect to fan speed. For a simple study, consider the system is populated with blade servers—only because the power consumption for blade fans is known experimentally but unknown for SL fans. Table 15 shows that the asymmetric test case consumed 7.1 kW more server fan power compared to the symmetric case. Blade fans are highly inefficient at very high speeds (i.e., 18000 rpm), which resulted in power consumption of 11.8 kW. Therefore, we recommend distributing the IT load symmetrically across the racks to achieve the lowest energy consumption.

Table 15: Blade fan power consumption comparison between symmetric and asymmetric IT load distribution

Test case	IT load + Server fan power (kW)	Blade fan speed (RPM)	Blade fan power (kW)	Pure IT load (kW)	Δ
100(50,50) kW	100	11000	5.4	94.6	7.1
100(30, 70) kW	100	7000, 18000	0.7 + 11.8 = 12.5	87.5	

Discussion

The purpose of characterizing the heat exchanger is to quantify the relationship between thermal effectiveness, heat capacity ratio, and number of transfer units. In order to develop a set of explicit equations, it is necessary to determine the amount of water and air flow needed to maintain a constant cold aisle temperature during transient periods where inlet water temperature or system load are changing.

We developed three charts to identify the behavior of the heat exchanger under fixed load conditions. Figure 42 shows the thermal effectiveness to heat capacity ratio based on results obtained from variable load testing of Pilot1. The chart in Figure 69 (see Appendix A) shows a correlation between P and NTU_t . This chart provides a method for determining the overall heat transfer coefficient. Figure 43 shows heat exchanger conductance as a function of water and airflow and can be derived from either the P - R or the P - NTU_t charts.

In Figure 42 below, we plotted the test data labeled “Actual” as solid symbols. The solid line labeled “Predicted” represents a curve fit to test data. This chart provides the bases for understanding the relationship between system load to water and air flow rates. Knowing this correlation, it is possible to determine what combination of water flow and air speed are needed to transfer the required heat load away from the system. This becomes possible by monitoring the changes in air and water temperature due to changes in system load and then correlating these changes to changes in water and air flow rates. This relationship results in an equation that will determine the ratio of air-to-water flow needed to satisfy any possible change in system load.

Figure 42: P-R thermal effectiveness chart for Pilot1

P-R

The relationship between thermal effectiveness and heat capacity ratio for the Pilot1 heat exchanger is represented in Figure 42. A third order polynomial shows a good fit to actual data with a R-squared value of 98%. The polynomial represents P as a function of R or P(R) with a standard error fit (S_{yx}) of 0.025 and a curve fit uncertainty of 0.05 at a 95% confidence. The system showed an upper and lower limit for thermal effectiveness and heat capacity ratio with P ranging from 0.3 to 0.7 and R from 0.7 to 2.60. The equation below is a representation between thermal effectiveness and heat capacity ratio for the Pilot1 heat exchanger.

Eq 7

Thermal Performance Curve

Figure 43 shows a characterization of the performance of the Pilot1 heat exchanger based on thermal conductance as a function of air and water flow rate. Thermal conductance is defined as the heat transferred through the heat exchanger over the maximum temperature difference between the two fluids.

Eq 8

These performance curves show the ability of the heat exchanger to conduct heat away from the system and can be used to predict the future state of the MDC based on current system conditions. Figure 43 displays air flow rate on the x axis and conductance on the y axis with each curve representing a constant water flow rate. System load, return air temperature (RAT) and supply water temperature (SWT) are needed in order to determine heat exchanger conductance. Based on these properties, we can determine the water and air flow rates.

Knowing the capability of the heat exchanger enables the thermal control system to determine the best combination of flow rates that reduces system power consumption and maintains environmental conditions. In Figure 43 for example, at a constant conductance of 3.5 kW/°C there are several combinations of water and air flow rates that would produce the necessary heat removal rate through the heat exchanger. However, only one combination will produce the lowest system and facility power usage. This minimum power value or PUE value is determined by understanding the capability of the facility. For instance, if the facility uses a dedicated chiller in a closed loop system to provide cooling to the MDC unit, the chiller would need to be run continuously. This approach increases the facility power usage compared to a facility capable of utilizing warmer water provided by a river or stream.

Figure 43: Thermal performance chart for Pilot1

A further examination of the dedicated chiller case is used to understand the effects proper thermal control has on reducing system power usage. However, we must first discuss the variables affecting chiller performance in order to determine a method for minimizing chiller power.

The properties affecting the power usage of a dedicated chiller are supply water temperature, system load, and ambient temperature. For a dedicated chiller, we assume water is pumped at a fixed rate that maintains a minimum water flow rate through the evaporator at all times. This is accomplished by placing a three-way valve between the chiller and the destination. The chiller loop maintains minimum flow while allowing the destination loop to vary (see Figure 43). Varying the water flow rate at the MDC unit has little to no effect on pump power. However, maintaining the maximum water flow rate through the heat exchanger allows for reduced air flow rate. Figure 43 shows at a constant conductance the maximum water flow rate curve of 50 gpm requires a lower air flow rate compared to different water flow rates at the same conductance. This reduction in air flow rate leads to a decrease in fan power. In addition to reducing fan power, the maximum water flow rate can support an increase in water temperature while maintaining system load requirements. In the case of a chiller providing external cooling, the use of warmer water will reduce chiller power and improve the overall PUE for the system.

Cooling System Objectives

At the beginning of the project, the research and development effort was aimed at developing a design capable of reducing the energy needs of a data center by reducing the power supplied to supporting hardware. The intent of this design was to match the cooling requirements to the IT load, allowing the inlet supply temperature of the server to rise to the manufacturer's upper limit. The increased inlet supply temperature allows for warmer supply water temperature and a significant reduction in external cooling.

Design Goals

In 2009 when this project began, the average PUE ratio for data centers was 2.0. The design goal of this project was to develop a self-contained modular cooling unit that meets a PUE of 1.25. This research and development effort focused on reducing the energy needed to support the IT load by increasing the efficiency of the power distribution and cooling systems. The next section discusses the current capability of the MDC and a comparison between actual and estimated PUE.

Current System Capabilities

Table 16 shows an energy breakdown for the Pilot1 design. The IT load is considered the total power at the PDU. The IT load is the power consumption of the server, including its power supply and fans. For simplicity we assumed the rack PDU losses were negligible. We excluded upstream power losses from the room PDU and we also excluded lab lighting. This compares more favorably with the HVDC configuration where the rack and room PDUs are not present and the 480 V distribution losses within the MDC are less.

Table 16: Minimum PUE recorded for Pilot1 using a 35T dedicated chiller

IT Load	Chiller Power @ 16C SWT, 8C Ambient	Fans	Infrastructure Load	Total Data Center Load	PUE
103.5 KW	21.3 KW	2.7 KW	24 kW	127.5 kW	1.23
* Supply air temperature was maintained at 25 °C					
* Delta air temperature across the servers was held constant at 19 °C					

We measured the power consumed by the dedicated chiller for the MDC. Incorporating a dedicated air-cooled chiller for a single MDC unit is not a typical product configuration. An alternative external cooling method was explored to reduce chiller power consumption (i.e., free cooling from lakes or streams).

The possible PUE for a warm water case where the chiller was turned off and only pump power was considered is shown in Table 17. The pump power for this test was estimated based on maximum pump power values reported for the lab chiller (see Figure 70 in Appendix A). The major difference when comparing this data to Table 16 is the difference in supply air temperature, water temperature and chiller power. The rise in SAT to 31°C enables the use of

warmer water approach, which eliminated the need for external cooling.

Table 17: Minimum PUE using warm water

IT Load	Pump Power @ 25C SWT	Fans	Infrastructure Load	Total Data Center Load	PUE
101.7 kW	4 kW	2.7 kW	6.7 kW	108.4 kW	1.07
* Supply air temperature was maintained at 31 °C					
* Delta air temperature across the servers was held constant at 18 °C					

Areas of Energy Savings/Contribution

Chiller Power

The variation in chiller power to ambient temperature at a fixed load of 100 kW is represented in Figure 44. Each test point represents steady state values at unique ambient and supply water temperatures. Although the system load was fixed, the supply temperature varied between 8°C and 16°C. Therefore, these points represent a collection of discrete curves each at a different supply water temperature. However, due to complications in monitoring chiller power, only a few points were collected. Therefore, this chart should be viewed as a general interpretation of chiller performance. A comparison between typical chiller trends reported by manufacturers⁴ and the data collected from the lab chiller show similarities in variable dependency, such as chiller power as a function of ambient temperature, system load, and supply water temperature.

⁴ Carrier, AquaForce 30XA080-500 Air-Cooled Liquid Chillers,
<http://www.docs.hvacpartners.com/idc/groups/public/documents/techlit/30xa-8pd.pdf>

Figure 44: Lab chiller power chart to ambient temperature

We found that improving the MDC PUE from 1.31 to a value below 1.25 involves minimizing external cooling by adopting a greener approach to thermal control. For instance, river water can be used through the heat exchanger if the water temperature is below the maximum temperature for thermal conductance. This maximum water temperature can be determined by monitoring heat load and supply air temperature. These key factors determine the allowable water temperature through the MDC.

The ASHRAE provides an allowable range for supply air temperatures of 15 °C to 32°C for an A1 class IT equipment. ASHRAE recommends consulting the manufacturer’s specification sheet to verify compliance with this range.

The system load as a function of supply water temperature (SWT) at two different ASHRAE-allowable supply air temperatures is shown in Figure 45. The blue curve (right) represents the warmest SWT allowed for a range of system loads at a constant SAT of 32°C. Based on this curve, it is safe to assume that a water temperature at or below 25°C will satisfy any load condition at or below 100 kW. Knowing the system’s upper limit of 25°C for SWT allows for alternate water sources, such as lakes or streams, to be used for a portion of the year. This will minimize the need for continuous chiller operation by increasing the use of greener cooling sources that lower energy costs and improve the PUE ratio.

Figure 45: System load vs. inlet/supply water temperature at the ASHRAE allowed supply air temperature limits of 15°C and 32 °C for a constant water flow rate of 50 GPM

System fans

Figure 46 shows performance curves for an EBM-Pabst Inc. (blower or system) fan, model M3G150-FF, at three different RPMs: 2040, 1695, and 1345. The series of system curves over the fan curves show a decrease in system pressure at varying enclosure fan speeds. The purple curve represents a baseline case with only the system fans powered On (server fans powered Off). This case showed the greatest impedance for the system with a decrease in air flow and an increase in static pressure compared to the other curves. There were some anomalies found in the system. The BL14000,SL68% case showed an increase static pressure and drop in airflow compared to the BL15000,SL60% case. This condition appears to be caused by the SL server enclosure increasing impedance corresponding to a drop in server performance at greater server RPMs within the system. However, more work needs to be done in characterizing both the BL and SL enclosure fans to determine any possible degradation in fan performance. Though a drop in airflow was recorded for the BL14000,SL68% case, the overall system impedance improves with increased enclosure fan RPM. This reduction in impedance increases the capability of a single system fan from 4200 CFM to 4700 CFM. This is an increase of 400 CFM per system fan or a total system increase of 1200 CFM.

Figure 46: Systemfan curve for a single fan operating at 1335, 1695, and 2040 RPM

The relationship between system fan power to airflow is shown in Figure 47. The baseline case shows the power of the system fans alone. This case sets a benchmark for the other cases showing the maximum fan power compared to the other test cases. The other curves show a decrease in system fan power at increased server fan RPM. We measured a maximum power of 2.5 kW at 4200 CFM per system fan for the standalone system fan case. While the BL15000,SL60% case produced a system fan power of 2.1 kW at 4200 CFM per system fan. This is a total reduction of 1.2 kW in system fan power.

Figure 47: The relationship between systemfan power and server air flow

Accuracy

Accuracy of Measured Quantities

Two sources of error in the equipment used to monitor thermal properties in the system is shown in Table 18. The first source of error is instrument error, which is reported as a combination of the manufacturer's stated bias and precision error. The second source of error is temporal. This error comes from variations in steady-state values over time.

It became apparent during testing that the internal environment of the MDC is dependent on the ability of the chiller to maintain supply water temperature and water flow rate. Analysis shows that relatively small perturbations in temperature and water flow by the chiller leads to moderate changes to system properties. For example, we performed a test at an IT load of 100 kW with a supply air temperature of 20°C and an airflow of 9,811 CFM. The chiller was set to maintain a supply water temperature of 9°C and a water flow rate of 23 GPM. For this case, the temporal uncertainty (the amount of unsteady behavior in the data) resulted in an IT load uncertainty of 9.9 kW, a supply air temperature uncertainty of 0.43°C, an air speed uncertainty of 52 CFM, a supply water temperature uncertainty of 0.17°C, and a water flow rate uncertainty of 2.19 GPM.

The IT load indicated above is a calculated value dependent on the temperature difference and

flow rate of water. Using Kline-McClintock⁵ second power relation to determine the uncertainty in IT load shows a greater dependency to variations in water flow over any other property. This implies that a 9.5% variation in water flow rate as indicated in the example causes a 9.9% variation in IT load. This is typical in calculated uncertainty values. The variable with the largest uncertainty dominates the resulting value and the calculated uncertainty approaches the largest uncertainty value. A table of uncertainty data for the Pilot1 test matrix shows IT load uncertainty dependent on water flow instability (see Table 21 in Appendix A). The cause of this instability was traced to a cyclic response in temperature and water flow rate from the chiller during fixed lab conditions (see Figure 33). The cyclic response by the chiller was treated as a known source of error and steps were taken to eliminate test cases with IT load uncertainty values above 10%.

All measured variables were assumed to have a Gaussian distribution. An infinite statistical approach⁶ was used with the total population for each measured variable being 100 instances after steady state is reached. To determine the uncertainty of a measured variable twice the standard deviation was used for a probability of 95%. Measured uncertainty is represented by the following equation where Z_1 is equal to 2.

Eq 9

Table 18: The average uncertainty for various measured quantities

Equipment	Type (Company, Type)	Measurement	Error Sources	
			Instrument	Temporal
Temperature sensors	Omega - RTD	Inlet and Outlet water temperatures – Twin & Twout	± 0.35°C	± 0.49°C
Pressure transducers	Honey well – Strain gauge	Inlet and Outlet water pressure – Pin and Pout	± 0.5%	-
Flow meter	Flow technology – Positive displacement flow meter	Water flow rate – V	± 0.05%	± 2.21 GPM
Heat exchanger motor controlled water valve	Belimo – Characterized control ball valve	Position of the valve		-
CU fan inlet rings with differential pressure sensors	Bapi sensors – Silicon capacitive pressure sensor	Air flow rate through CU	± 0.5%	± 77 CFM
Thermocouples	Omega – J type	Inlet and outlet of CU and Rack air temperature	± 1.1°C	± 0.42°C
PDU	HP iPDU	IT load	± 1.0%	± 9.69 kW
DAQ	Agilent technologies	Data acquisition from all sensors	± 1.0°C	-

⁵ Richard S. Figliola, Donald E. Beasley, “Theory and Design for Mechanical Measurements,” Third Edition, Ch. 5.6 pg 161.

⁶ Richard S. Figliola, Donald E. Beasley, “Theory and Design for Mechanical Measurements,” Third Edition, Ch. 4.3 pg 116.

Accuracy of Calculated Quantities

Similar to the measured quantities, we determined a temporal uncertainty for the instabilities recorded in the calculated quantities. Table 19 shows a list of key variables and their corresponding uncertainty values. The second Error Source column in the table indicates the *standard error of the fit* (S_{xy}) to the various curve fits produced relating Thermal Effectiveness (P) to Number of Transfer Units (NTU_t) and Thermal effectiveness to heat capacity ratio (R).

Table 19: Uncertainty for calculated quantities

Calculated	Error Source	
Variable Name	Temporal	Curve fit
Effectiveness (P)	± 0.05	± 0.05
Log Mean Temperature Difference (LMTD)	± 1.35	-
Ovall Heat transfer Coeffecient (UA)	$\pm 1.01 \text{ kW/}^\circ\text{C}$	-
Number of Transfer Units (NTU_t)	± 0.21	± 0.08
Heat Capacity Ratio	± 0.1	± 0.1

We estimated all calculated uncertainties using Kline-McClintock second power relation⁷. The relation can be derived from the linearized approximation of the Taylor series expansion of a multivariable function. A sensitivity index, θ_i , results from the Taylor series expansion and is shown in the following equation.

$$\theta_i = \frac{\partial R}{\partial x_i} \frac{x_i}{R} \quad \text{Eq 10}$$

The sensitivity index relates how changes in each x_i affect R. The propagation of uncertainty in the variables to the result will yield an uncertainty estimate by using the equation below.

$$\Delta R = \sqrt{\left(\frac{\partial R}{\partial x_1} \Delta x_1\right)^2 + \left(\frac{\partial R}{\partial x_2} \Delta x_2\right)^2 + \dots + \left(\frac{\partial R}{\partial x_n} \Delta x_n\right)^2} \quad \text{Eq 11}$$

⁷ Richard S. Figliola, Donald E. Beasley, "Theory and Design for Mechanical Measurements," Third Edition, Ch. 5.6 pg 161.

Conclusion

In summation, we conducted tests to characterize the performance of the cooling unit and determine methods to reduce overall system load. After extensive testing, we generated a series of charts to determine the thermal performance of heat exchanger designs used in the development of the MDC. The analysis shows that system power can be reduced by increasing the supply air temperature to the ASHRAE allowed limit of 32°C and maintaining a supply water temperature below 25°C for an IT load of 100 kW. By running a warmer system, the dependence on external cooling decreases and the PUE ratio improves.

We tested the system fans, which provide bulk airflow to the servers, to determine maximum flow through the system as well as system impedance. An analysis of the system fans' performance shows that the maximum airflow through the system will increase with increasing server fan RPM. The assistance of the server fans lowers system impedance and reduces system fan power. However, without direct power measurements from the server fans, it is not clear how much power they consume and the overall contribution they have on infrastructure load.

Estimating the IT load at the PDU shows that for a Pilot1 heat exchanger operating a dedicated chiller, the system PUE produced was 1.31. A warm water approach showed an improved PUE of 1.14 at an ASHRAE-allowed supply air temperature of 32°C and a supply water temperature below 25°C.

In conclusion, the current MDC design has surpassed the proposed PUE objective of 1.25 by efficiently utilizing the cooling hardware to reduce system power, while maintaining environmental conditions.

Controls and monitoring subsystem

The controls and monitoring subsystem (CMS) manages the delivery of cooling to meet the demand of the IT equipment and improves the overall cooling efficiency. The chilled water cooling unit housed within the enclosure ensures that the IT equipment receives adequate cooling (Figure 48). The cooling unit consists of a heat exchanger (HEX) and three centrifugal system fans. The system fans provide air circulation from the cold aisle to the hot aisle and back through the HEX. The HEX extracts heat from the hot exhaust air that passes through it. The Dedicated Controller actuates a chilled water valve to control the amount of chilled water flowing through the HEX. See Appendix B for a functional description of the Dedicated Controller.

Figure 48: Models for IT, cooling, and facility loads

We developed a hierarchical system for the MDC with rich sensing mechanisms and multiple actuators for cooling provisioning and optimization. As shown in Figure 49, a cell aggregator is installed in each bay for data acquisition. The cell aggregator collects data from the air temperature sensors and the local actuators, such as door openers. The heat exchanger is equipped with temperature and pressure sensors. The Dedicated Controller actively controls the air flow rate and the water flow rate by tuning the system fan speed and controlling the chilled water valve, respectively. The Primary Controller optimizes overall cooling efficiency and provides functions such as alarms.

The Controls and Monitoring System is implemented using Beckhoff™ industrial PC (IPC), Programmable Logic Controller (PLC) and their related TwinCAT software. The Human Machine Interface (HMI) system provides a graphical user interface for the operation status of the MDC. The HMI also provides interfaces for end users to configure the control systems and access data locally or remotely.

Figure 49: Controls and Monitoring based on Beckhoff IPC and PLC

The specifications for the Primary Controller and the Dedicated Controller are shown in Table 20 and Figure 50.

Table 20: Primary Controller and Dedicated Controller specifications

	Primary Controller and Dedicated Controller
Model No.	Beckhoff™ CX5020
Processor	Intel® Atom™ 1.1 GHz
Memory	1 GB DDR2 RAM
Network connectors	2 x 10/100/1000BASE-T Ethernet connectors
Hard Disk Drive	32 GB Compact Flash
Ports	4 USB 2.0 ports
Operating System	Microsoft Windows XP Embedded

Figure 50: Dedicated and Primary Controllers

Implementation and Evaluation

Feedback control

The chilled water valve and system fans (Figure 51) are controlled through two PID controllers, both implemented in the Dedicated Controller. The PID controller for the chilled water valve is driven by the set point of the supply air temperature (SAT) at the outlet of the heat exchanger. It maintains the SAT at or below the threshold based upon changes in the IT load, air flow rate, supply water temperature, etc. The set point of the SAT may be further tuned by the inlet air temperatures of the IT racks, or by the optimizer in the Primary Controller.

Figure 51: Chilled water valve and system fans

The second PID controller automatically tunes the speed of the three system fans uniformly. The controller can be driven using either of the two metrics: the pressure differential between the

cold aisle and the hot aisle or the temperature differential between the RAT at the inlet of the heat exchanger and the average SAT. In the first case, the differential pressure is maintained close to zero but always at a positive level as shown in Figure 52. This PID controller meets the air volume demand of the IT equipment and it responds to the fan activities of the IT equipment in the aggregate level by tuning the system fan speed. In the second case shown in Figure 53, the RAT is maintained below a threshold that is allowed by the IT equipment.

Figure 52: Fan control based on differential pressure

Figure 53: Fan control based on differential temperature

The system fan control loop and the valve control loop work together to meet both the air volume demand and thermal requirement of the IT equipment. They interact with each other based on conditions inside the IT equipment and heat exchanger. For instance, when the IT load increases the IT fans run faster, thus increasing the demand for the air volume and raising the system fan speed. This increases the valve opening to maintain the supplied air temperature. We applied different time scales to the two loops to ensure stability.

We tuned the two feedback controllers based on experience, automatic tuning approaches, and a modeled tuning approach. Experimental validation shows that the controllers can converge to the set points upon disturbances from inside or outside, e.g., set point changes, heat load variance, or SWT variance.

Our design can also tolerate the failure of components such as a fan. For example, if one or two fans fail, the differential pressure will change. Consequently, the PI loop will speed up the remaining fan(s) to attempt to regain differential pressure set point.

Figure 54 shows a scenario where two simultaneous step changes on supply air temperature set point (from 20°C to 25°C) and differential pressure set point (0.25 inH₂O to 0.3 inH₂O) were made (as disturbances) to the control system. The pressure and temperature control loops converged to their new set points within 10 seconds and 50 seconds, respectively. This result implies that both control loops can operate independently and sufficiently adapt to new disturbances.

Figure 54: Two PID loops tracking cold aisle static pressure and SAT output to set points

We introduced the metric Coefficient of Performance (COP_G)⁸ in place of PUE for the efficiency of the MDC cooling system.

The main difference between COP_G and PUE is that COP_G considers the server fan power as part of the cooling overhead. The COP_G metric is a function of multiple factors such as heat load, ambient air conditions, and the operation configuration of MDC, such as the SAT and SWT set points. The Primary Controller collects all the sensor data along with current environmental conditions (ambient temperature), and determines the optimal configuration for MDC operation. In our implementation, we used the IT models, the heat exchanger model, the blower model, and the chiller model described previously. IT models utilized in the work were developed in the series of work by HP Labs⁹. We developed optimization algorithms to find the optimal SAT set point value for the valve controller and/or the SWT set point for the chiller.

Our approach is illustrated using Figure 55, which shows the variation of COP_G and PUE with SAT and SWT. The IT load is close to 100 kW and outside air is assumed to be 22°C. The achievable set points are limited by the heat exchanger at low SAT range. Higher SAT is not achievable because of the airflow capacity of server fans and system fans or the maximum load that the chiller can handle. It is seen that the maximum COP_G of 3.45 is obtained for SAT=19.5°C and SWT=13 °C for the chosen conditions. The PUE for the corresponding set points is 1.23.

Figure 55: Lowest supply air temperature based on highest water temperature yields the best COP_G and PUE

Conclusions

The implementation of the CMS serves as a proof of concept of a consolidated thermal control

⁸ Amip Shah, Thomas Breen, Edmond Walsh, Jeff Punch, Cullen Bash, Niru Kumari, Tahir Cader, “On the Need for Energy Efficiency Metrics that Span Integrated IT-Facility Infrastructures.” ASME Pacific Rim Technical Conference and Exhibition on Packaging and Integration of Electronic and Photonic Systems, MEMS, and NEMS (IPACK2011-52140), Portland, OR, July 6-8, 2011.

⁹ Zhikui Wang, Cullen Bash, Niraj Tolia, Manish Marwah, Xiaoyun Zhu, Parthasarathy Ranganathan, Optimal fan speed control for thermal management of servers, in Proceedings of the ASME/Pacific Rim Technical Conference and Exhibition (InterPACK’09), San Francisco, CA, July 2009.

and optimization framework developed by HP Labs^{10,11,12}. This framework integrates controls and monitoring components, adaptive control, and optimization techniques that are scalable and adaptable to existing and future product lines. The outcome obtained during testing shows promising results, indicating that the system can adapt to environmental changes while sustaining the most efficient operating parameters.

Recommendations

We developed cooling optimization using two approaches:

1. Model-based approach – We used realistic models of IT equipment as well as an actual heat exchanger and chiller. We ran a software program to show consistent optimization conditions from which to derive an optimal set point. We applied operational constraints to the model, such as chilled water temperature range and supply air temperature range, to yield more realistic results.
2. Blackbox hybrid approach – Based on some known models, we ran tests to collect additional data to correlate changes affecting different power buckets to characterize system energy consumption. The results of this method were inconclusive due to multiple factors:
 - i. The IT loads used for testing were not real servers so the power consumption was insensitive to changes in air temperature and airflow.
 - ii. Disturbances such as chiller power fluctuation and water temperature fluctuation prevented more accurate characterization of the system.

We recommend further work to study trends in power consumption of various components over a longer period (allowing the system to settle to a steady state) and using real equipment for the IT load.

¹⁰Thomas Breen, Edmond Walsh, Jeff Punch, Amip Shah, Cullen Bash, Niru Kumari, Brandon Rubenstein, Scot Heath “From Chip to Cooling Tower Data Centre Modeling: Influence of Air-Stream Containment on Operating Efficiency.” 8th ASME/JSME Thermal Engineering Joint Conference (AJTEC), Honolulu, HI, March 13-17, 2011.

¹¹ Thomas Breen, Edmond Walsh, Jeff Punch, Amip Shah, Cullen Bash, Niru Kumari, Tahir Cader, “Experimental Validation of Multi-Scale Data Center Energy Efficiency Models.” ASME Pacific Rim Technical Conference and Exhibition on Packaging and Integration of Electronic and Photonic Systems, MEMS, and NEMS (IPACK2011-52030), Portland, OR, July 6-8, 2011.

¹² Thomas Breen, Edmond Walsh, Jeff Punch, Amip Shah, Cullen Bash, Niru Kumari, Brandon Rubenstein, Scot Heath, “From Chip to Cooling Tower Data Centre Modelling: Chip Leakage Power and its Impact on Cooling System Efficiency.” ASME Pacific Rim Technical Conference and Exhibition on Packaging and Integration of Electronic and Photonic Systems, MEMS, and NEMS (IPACK2011-52141), Portland, OR, July 6-8, 2011.

Human Machine Interface

The HMI is a built-in, industrial quality Panel PC located on the front door of the Cooling Unit bay (Figure 56). It has a 15-inch touch screen TFT display with a resolution of 1024 x 768 (see other specifications below).

Figure 56: HMI on front door of Cooling Unit bay

HMI Specifications

Housing	Aluminum front with steel sheet rear cover Drives easily accessible All connectors at the bottom of the rear side 1 slot for one 2½-inch hard disk or SSD and 1 slot for one Compact Flash card accessible from the rear side Lithium battery of the system clock accessible from the rear side Operating temperature 0 to 55 °C
Features	Processor: Intel® Atom™ 1.1 GHz (TC3: 40) 3½-inch motherboard for Intel® Atom™ 1 Mini PCI slot free for cards installed ex factory 1 GB DDR2RAM, expandable to 2 GB On-board graphic adapter, Intel® GMA 500, DVI-D connector On-board dual Ethernet adapter with 2 x 10/100/1000BASE-T connector 64 MB Compact Flash card, extended temperature range 4 USB 2.0 ports 24 V DC power supply Operating system Microsoft Windows XP Embedded, English

The HMI is implemented with a graphical user interface (GUI) using the InduSoft Web Studio software (Figure 57). The HMI has a local database (DB) for data collection, data archiving and data trending. Optionally, the HMI could have local authentication or domain authentication using Microsoft Active Directory (AD) LDAP. The HMI could also have remote data archiving (a dedicated database server). The HMI can forward/integrate the data collected with HP Insight Control software or a Building Management System (BMS).

Figure 57: Human Machine Interface

HMI Pages

The Active Alarms page displays alarms associated with sensors installed in the system. For example, Figure 58 shows alarms that were triggered when we opened several bay doors immediately after one of the IT load tests.

Figure 58: HMI – Active Alarms

The screenshot shows the 'Enterprise Web Application' interface with the 'Alarms' tab selected. The 'Active Alarms' sub-tab is active, displaying a table of alarms. The table has columns for Message, Event Time, Norm Time, Value, and Station. The alarms listed are related to bay doors being open or offline. At the bottom of the page, there are two buttons: 'Acknowledge An Alarm' and 'Acknowledge All Alarms'.

Message	Event Time	Norm Time	Value	Station
Bottom fan is offline	03/22/2012 14:51:11		0	Garage_HMI
R3 bay front door is open	03/22/2012 14:50:53		1	Garage_HMI
R4 bay front door is open	03/22/2012 14:50:52		1	Garage_HMI
Middle fan is offline	03/22/2012 14:50:49		0	Garage_HMI
R2 bay front door is open	03/22/2012 14:50:45		1	Garage_HMI
R1 bay front door is open	03/22/2012 14:50:43		1	Garage_HMI
L1 bay front door is open	03/22/2012 14:50:28		1	Garage_HMI
Top fan is offline	03/22/2012 14:50:26		0	Garage_HMI
L2 bay front door is open	03/22/2012 14:50:07		1	Garage_HMI
L3 bay front door is open	03/22/2012 14:50:04		1	Garage_HMI
L4 bay front door is open	03/22/2012 14:50:03		1	Garage_HMI
L4 bay rear left door is open	03/22/2012 14:41:50		1	Garage_HMI
L4 bay rear right door is open	03/22/2012 14:41:50		1	Garage_HMI
L3 bay rear right door is open	03/22/2012 14:41:48		1	Garage_HMI
L3 bay rear left door is open	03/22/2012 14:41:47		1	Garage_HMI
L2 bay rear right door is open	03/22/2012 14:41:45		1	Garage_HMI
L2 bay rear left door is open	03/22/2012 14:41:43		1	Garage_HMI
L1 bay rear right door is open	03/22/2012 14:41:40		1	Garage_HMI
L1 bay rear left door is open	03/22/2012 14:41:39		1	Garage_HMI
R1 bay rear left door is open	03/22/2012 14:41:27		1	Garage_HMI
R1 bay rear right door is open	03/22/2012 14:41:27		1	Garage_HMI
R2 bay rear right door is open	03/22/2012 14:41:25		1	Garage_HMI
R2 bay rear left door is open	03/22/2012 14:41:25		1	Garage_HMI
R3 bay rear right door is open	03/22/2012 14:41:24		1	Garage_HMI
R3 bay rear left door is open	03/22/2012 14:41:23		1	Garage_HMI
R4 bay rear left door is open	03/22/2012 14:41:22		1	Garage_HMI
R4 bay rear right door is open	03/22/2012 14:41:22		1	Garage_HMI

The screen in Figure 59 allows the user to choose between Optimized, Manual, and Test modes. In Manual and Test modes, the user can input the set points for Supply Air Temperature, fan RPM, and the Chilled Water Valve position (in percent open). The user can also input the PID (proportional-integral-derivative) values for CWV and Fan Control using the Settings button.

Figure 59: HMI – Operating Mode

The screenshot shows the 'Enterprise Web Application' interface with a navigation bar (Home, Alarms, Logs, Control, Maintenance, Settings) and a 'Logout' button. The 'Operating Mode' section is currently set to 'Optimized Mode'.

Mode	Parameter	Actual	Setpoint / Recommended
Optimized Mode	Supply Air Temperature	21.85	20.00
	Supply Water Temperature	24.73	-
Manual Mode	°C SAT	21.85	28.00
	RPM	0.00	1100
Test Mode	% Open	13.33	50.0

The AHU Data View page (Figure 60) displays various data points for the Air Handler Unit. These include:

- SWT
- RWT
- CW flow rate
- CWV position
- relative humidity (H_{rel}) behind the heat exchanger
- average RAT
- average SAT
- system fan RPM set points
- actual RPM

Figure 60: HMI - AHU Details View

Figure 61 shows the Front View (main) page of the HMI. This page displays various data collected from the Primary Controller and the Dedicated Controller. The data includes the following:

- IT loads for each bay (including the fans)
- Air Handler Unit data (chilled water valve actual and set point position, supply and return water temperature, and water flow rate)
- fan data (actual and set point RPM, total air flow, average cold and hot aisle temperature)
- environmental data (internal and external humidity)
- operating mode (optimized, manual, or test)
- PUE
- total IT power
- total facility power
- total number of alarms
- the status of the Primary and Dedicated Controllers
- cold aisle and hot aisle temperatures for each bay

Figure 61: Main HMI screen

Figure 62 shows the Top View page of the HMI. It displays the following information:

- status of the front and rear bay doors (open or closed)
- presence of an equipment rack in each bay
- IT load of each bay
- cold aisle and hot aisle temperatures for each bay
- internal humidity
- external humidity
- external temperature of the enclosure

Figure 62: HMI – Top View

The DC Air Trend page (Figure 63) displays values from different data sources in graphical format. The top graph displays data points from the Air Handler Unit, such as delta T actual, average Supply Air Temperature, average Return Air Temperature, and delta T set point. The bottom graph displays data points from the fans, such as fan RPM.

Figure 63: HMI – Dedicated Controller Air Trend

The DC Water Trend page (Figure 64) displays values from different data sources in graphical format. The top graph displays data points from the Water/CWV, such as the average SAT, SAT set point, SWT, and RWT. The bottom graph displays data points from the water flow rate (GPM) and the CWV position (percent open).

Figure 64: HMI – Dedicated Controller Water Trend

The Fan Information page (Figure 65) displays various static data for the three system fans in the enclosure.

Figure 65: MHI – Fan Information

Fan 1 (top)		Fan 2 (Middle)		Fan 3 (Bottom)	
Fan Info		Fan Info		Fan Info	
Modbus Slave ID:	66	Modbus Slave ID:	67	Modbus Slave ID:	68
Serial Number:	BF0B	Serial Number:	BB0B	Serial Number:	7R07
S/W Bus Controller Name:	45062.000000	S/W Bus Controller Name:	45062.000000	S/W Bus Controller Name:	45062.000000
S/W Bus Controller Version:	33.000000	S/W Bus Controller Version:	33.000000	S/W Bus Controller Version:	33.000000
S/W Comm. Controller Names:	45062.000000	S/W Comm. Controller Names:	45062.000000	S/W Comm. Controller Names:	45062.000000
S/W Comm. Controller Version:	7.000000	S/W Comm. Controller Version:	7.000000	S/W Comm. Controller Version:	7.000000
Setpoint Source:	1	Setpoint Source:	1	Setpoint Source:	1
Internal Parameter:	Parameter Set 1	Internal Parameter:	Parameter Set 1	Internal Parameter:	Parameter Set 1
Operating Mode:	Closed Loop Speed Control	Operating Mode:	Closed Loop Speed Control	Operating Mode:	Closed Loop Speed Control
PID Parameters: K1 6.250000 Kp 99.609375		PID Parameters: K1 6.250000 Kp 99.609375		PID Parameters: K1 6.250000 Kp 50.000000	
Status:	Enable On	Status:	Enable On	Status:	Enable On
Position:	TOP	Position:	MIDDLE	Position:	BOTTOM
Fan Data		Fan Data		Fan Data	
Max Speed:	2040.000000	Max Speed:	2040.000000	Max Speed:	2040.000000
RPM:	0.000000	RPM:	0.000000	RPM:	0.000000
Set Point:	0.000000	Set Point:	0.000000	Set Point:	0.000000
Motor Status:	0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0	Motor Status:	0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0	Motor Status:	0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0
DC Link Voltage Reference:	900.000000	DC Link Voltage Reference:	900.000000	DC Link Voltage Reference:	900.000000
DC Link Amperage Reference:	0.000000	DC Link Amperage Reference:	0.000000	DC Link Amperage Reference:	0.000000
DC Link Voltage:	0.000000	DC Link Voltage:	0.000000	DC Link Voltage:	0.000000
DC Link Amp:	0.000000	DC Link Amp:	0.000000	DC Link Amp:	0.000000
Module Temp:	0.000000	Module Temp:	0.000000	Module Temp:	0.000000
Electronic Temp:	0.000000	Electronic Temp:	0.000000	Electronic Temp:	0.000000
Power:	0.000000	Power:	0.000000	Power:	0.000000
Fan Direction:	R	Fan Direction:	R	Fan Direction:	R

The Control Configuration Settings page (Figure 66) allows the user to input the PID values for the CWV and Fan Control.

Figure 66: HMI – Control Configuration Settings

hp Enterprise Web Application Logout

Home Alarms Logs Control Maintenance Settings

Control Configuration Settings Fan Configuration Settings Control Parameter Limits

Chilled Water Valve Control

PID Mode

P 0.24
I 0.3000
D 0.01

JG Mode

Use Manual or Optimized SAT setpoint

Advanced Mode

(to be implemented in future)

Fan Control

Delta T Mode

Setpoint °C
19.00

P 10.00
I 0.3000
D 0.00

JG Mode

Setpoint °C
19.00

Delta P Mode

Setpoint
1.01

P 50.00
I 0.1000
D 0.00

Back To Control Page

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The Fan Configuration Settings page (Figure 67) allows the user to re-initialize all the fans, configure a fan, or reboot the Dedicated Controller. The user can press the Re-Initialize All Fans button if all the fans in the enclosure lose power or when the Dedicated Controller loses communication to the fans. The user can press the Configure a Fan button when replacing an old fan with a new fan. The user can press the REBOOT the Dedicated Controller button to reboot the Dedicated Controller if it hangs or to reboot the Dedicated Controller remotely.

Figure 67: HMI – Fan Configuration Settings

The screenshot shows the HP Enterprise Web Application interface. At the top, there is a navigation bar with the HP logo, the text "Enterprise Web Application", and a "Logout" link. Below this is a menu bar with tabs: "Home", "Alarms", "Logs", "Control", "Maintenance", and "Settings". The "Settings" tab is selected, and within it, the "Fan Configuration Settings" sub-tab is active. The main content area is titled "Fan Configuration" and features a blue fan icon. It contains three sections: 1. "Click to Re-Initialize All Fans" with a button labeled "Re-Initialize All Fans". 2. "Configure A Fan" with the instruction "Select a fan to configure then click OK", a dropdown menu showing "Configure Fan 1", and an "OK" button. 3. "Click to REBOOT the Dedicated Controller" with a button labeled "REBOOT the Dedicated Controller (DC)".

The Control Parameter Limits page (Figure 68) allows the user to input the lower and upper limits for the SAT, Delta T, and Delta P that will be used in the Control page.

Figure 68: HMI – Control Parameter Limits

HP Enterprise Web Application

Home Alarms Logs Control Maintenance Settings Logout

Control Configuration Settings Fan Configuration Settings Control Parameter Limits

Control Parameter Limits

Parameter	Lower Limit	Upper Limit
Supply Air Temperature	10.	30.
Delta T	10.	30.
Delta P	0.8	2.0

Benefits Assessment

Because the MDC achieved a PUE better than the target of 1.25, it is possible to exceed the estimated electrical energy savings of approximately 1.70 billion kWh by year 2015 as determined by the ICT Benefits Spreadsheet (Appendix C). This is a reasonable assumption based on the commercialization of the technology in 2012. In addition to the energy savings, the technology has the potential to significantly reduce the creation of greenhouse gases. By 2015, the technology could displace more than the original estimate of 284 metric tons of CO₂.

Commercialization

The increasing emphasis on energy efficient power and cooling solutions is demonstrated by customers trying to improve the PUE and TCO of their data centers. We believe that closed-loop IT systems have good market potential because solutions such as the MDC can be utilized inside existing data centers, and solutions such as the HP POD can be used externally for larger deployments. HP and Eaton will continue to explore and develop these power and cooling markets and will bring product to market given sufficient demand. Our policy however is not to discuss details on unreleased products.

At the time of this writing, no decision has been made whether to productize the complete solution as a general offering. Near term we have plans to move the current DOE/HP MDC prototype into the new HP Discovery Lab (opening in 2013) to power and cool the next generation of HP IT architectures as part of our announced Moon Shot program. This working

lab will be used for testing and evaluating the latest HP technologies where both customers and HP development partners will have an active role in the evaluations. It will also serve as a showcase for HP technology during regular customer tours. Customer interest and feedback in this MDC solution will play strongly in the final go-to-market decision.

It's important to note that we are leveraging our findings from this development effort into enhancing existing and future products, specifically on the following products, which form a large part of our go-to-market business:

HP Modular Cooling System (rack level) – cooling and cooling management

HP POD (multi-rack level) – cooling, cooling management, and power distribution

Rack level solutions – power conversion and distribution

Also, for customers that want to connect a solar array, Eaton will provide a UPS with the integrated UPS-PV feature. Initially, this product will be by request only. However, if sufficient demand occurs, Eaton will make this a standard offering. We are also working with HP Labs on their Net-Zero initiative—announced in May of 2012—which has an alternative energy component associated with its architecture. This initiative aligns well with what we have researched as part of the DOE grant program.

Although the cycloconverter topology is compelling, the semiconductor devices to make this a robust commercial offering are just being introduced. We will assess the semiconductor device maturity in the coming years and will re-evaluate commercialization of the cycloconverter topology. During the development of the controls for the cycloconverter, we have discovered new control techniques that may find their way into future products to make them more robust.

DC power in the datacenter is not yet generally accepted and the efficiency improvement over the most modern AC systems is small. However, we believe there are other benefits to DC power distribution, especially at the rack and rack row level. These benefits are related to more efficient implementations and freeing up space in servers due to the smaller size of a DC supply as well as providing more efficient back-up solutions. We will continue the development of HVDC distribution and backup techniques. We will introduce such products if and when the market makes this viable.

Significant work remains to productize these elements, but many of the high-risk factors have been addressed and mitigated during development under this grant. This should speed up the time-to-market considerably when the time comes.

Accomplishments

Summaries of the accomplishments for each subsystem are as follows:

- **Mechanical Enclosure/Thermal Subsystems** – We developed a modular enclosure, scalable to 4, 6, or 8 bays of roll-in, roll-out IT racks. The enclosure is rugged and semi-sealed to minimize heat loss, and it is easily assembled by two people within three days. The system includes a Cooling Unit with an efficient water-to-air heat exchanger and a

set of three highly efficient system fans, as well as a moisture eliminator. A plumber is required to make the chilled water connections.

- **AC Power Distribution Subsystem** – We developed an AC power distribution system that is mechanically integrated to the system. The system distributes redundant 480 V AC to each IT bay as well as a limited amount of redundant 240 V AC to each bay. The unit has branch circuit monitoring and Ethernet communications. An electrician is required to hook up the service AC mains.
- **Alternative Energy Unit/UPS Subsystems** – We developed an alternative energy interface to the UPS. We demonstrated operation with actual solar arrays and were able to reduce mains power by supplementing with solar power.
- **Power Supply and HVDC Power Distribution Subsystems** – We developed a high-efficiency, single-stage power supply, with a 480 V AC input and 360 V DC output. Two of these units fit inside a Power Shelf with redundant AC feeds. We also developed a HVDC power distribution bus (mountable in the rack) to distribute the 360 V DC from the power supply to the servers.
- **HVDC-DC Converter and Battery Backup Unit Subsystems** – We developed a high-efficiency, high-voltage DC to DC converter in a common slot form factor to fit inside HP servers. We also developed a smaller, lighter, more efficient Battery Backup Unit that operates from the 360 V DC bus and fits in the rack to maintain server operation in the event of mains failure.
- **Controls and monitoring subsystem** – We developed a controls and monitoring subsystem that optimizes the overall cooling efficiency. Users can set the system set points and control modes and the system will achieve the most energy efficient operation.

Conclusions

A close-coupled, fully contained cooling system provides the opportunity to save significant energy over a traditional raised-floor data center. It can improve a typical PUE of 2.0 down to 1.25 or lower.

Additional savings can be found by improving the method of power distribution. Using a UPS in eco-mode, for example, offers much better efficiency (>99%) than a traditional online model at 93%. Including solar or wind power as a supplement further increases savings.

Reducing the number of power conversions also has a significant impact on energy efficiency. Distributing 480 V AC versus the traditional 208 V AC distribution saves power and copper. It eliminates the need for a 98% isolation transformer, which saves not only the power consumed by the transformer but also the space and the cost of the unit. Distributing 480V AC reduces the size and number of power feeds for a given load, saving panel space for breakers and saving copper wiring. Higher efficiencies are also achieved using a pooled power setup with rectifiers with 480 V AC inputs and 360 V DC outputs. HVDC requires less copper and can be distributed to the servers where a DC-DC converter drops the voltage down to the traditional 12 V DC.

In addition to the technical knowledge gained from working on the MDC project, there were

several unexpected benefits to the organization and to the larger product offerings of HP.

- In parallel with the MDC research, HP continued to develop other power and cooling solutions as part of the normal business/development cycle. Our product development teams were able to apply lessons learned from the MDC program and, in some cases, directly implement technologies based on MDC research in shipping products to make them more energy efficient.
- We learned and developed new techniques and processes to measure power and cooling efficiency for all elements of the design from component selection to the geometry of the enclosure. We will carry these techniques and processes forward on future product and solution development work.
- Valuable trial-and-error lessons allowed newly hired employees and contractors to become proficient in skills in an environment that was very conducive to learning and that allowed them to apply the lessons learned in a productive and structured manner. Due to the sometimes aggressive pace of product development at HP, we do not always have the time to perform product development in this manner.
- IT and facilities integration is still a new area of development for the industry and HP. So analyzing all aspects of energy efficiency in product development in parallel with more traditional IT requirements provided good insight and knowledge that will be carried forward by multiple HP development teams.

Many of the skills, techniques, and values gained will influence future generations of products. While there has been a trend in making products more energy efficient for some time, there still needs to be a cultural shift in organizational thinking about how products are developed. The deep immersion in the MDC project has been a good launching pad in driving this change.

Recommendations

Any recommendations for follow up research and development of the individual subsystems are included in their respective section.

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Appendix A: Cooling System Test Procedure Overview

Table 21: The variation in IT load uncertainty as a function of water flow rate uncertainty

Test Case	Water flow GPM	IT Load KW	Water flow Uncertainty GPM	IT Load Uncertainty KW	Water flow Uncertainty %	IT Load Uncertainty %
100kw_8Ch20_20csp_6.8V_VC_Pilot1_111711_Analysis.xlsm	14.63	99.36	0.36	2.65	2.46	2.67
100kw_8Ch20_26csp_6.8V_VC_Pilot1_111711_Analysis.xlsm	23.30	104.30	0.57	2.64	2.44	2.53
100kw_16Ch20_22csp_8V_VC_Pilot1_111711_analysis.xlsm	55.62	104.58	1.12	2.24	2.01	2.14
120kw_8Ch20_27csp_8V_VC_Pilot1_111811_Analysis.xlsm	17.44	127.08	1.39	11.04	7.98	8.69
120kw_12Ch20_22csp_6V_VC_Pilot1_heat_flux_111811_analysis.xlsm	36.71	121.90	0.89	3.03	2.43	2.49
130KW_20C CSP_8C H2O_8V_Pilot1_112211_Steady state_Analysis.xls	33.37	138.02	0.73	3.07	2.20	2.23
160KW_25C CSP_12C H2O_10V_Pilot1_112311_Steady state_Analysis.xlsm	38.68	168.64	0.91	4.06	2.35	2.41
100KW_20C CSP_8C H2O_6.8V_Pilot1_120211_Analysis.xlsm	21.35	103.89	0.78	4.01	3.65	3.86
100KW_20C CSP_8C H2O_8V_Pilot1_122111_Pilot1eqn_SS_A1.xlsm	23.69	94.17	1.85	7.79	7.79	8.27
100KW_20C CSP_8C H2O_9V_Pilot1_Mid_fan_Off_010412_A1.xlsm	22.13	89.02	1.02	4.48	4.60	5.04

Figure 69: Pilot1 Thermal Effectiveness vs. NTUt

Figure 70: Summary Performance Report for the 34-ton lab chiller

Summary Performance Report For CH-1													
Project: HP Mechanical Lab Prepared By:	07/15/2010 04:15PM												
<div style="display: flex; justify-content: space-between; align-items: center;"> <div style="text-align: center;"> </div> <div style="text-align: center;"> AquaSnap™ Air-Cooled Scroll Chiller </div> <div style="text-align: center;"> </div> </div> <div style="text-align: center; margin: 20px 0;"> </div>													
Unit Information Tag Name: CH-1 Model Number: 30RAP035 Quantity: 1 Manufacturing Source: Charlotte, NC USA Refrigerant: R410A Independent Refrigerant Circuits: 2 Shipping Weight: 3055 lb Operating Weight: 2850 lb Unit Length: 89 in Unit Width: 93 in Unit Height: 67 in	Performance Information Cooling Capacity: 34.1 Tons Total Compressor Power: 36.3 kW Total Fan Motor Power: 3.85 kW Pump Power: 4.84 kW Total Unit Power (without pump): 40.2 kW Total Unit Power (with pump): 45.0 kW Efficiency (without pump): 10.18 EER IPLV: 15.20 EER A-Weighted Sound Power Level: 90 dbA												
Evaporator Information Fluid Type: Fresh Water Fouling Factor: 0.00010 (hr-sqft-F)/BTU Leaving Temperature: 44.0 °F Entering Temperature: 51.5 °F Fluid Flow: 109.1 gpm Pressure Drop: 38.9 ft	Accessories and Installed Options Cooler Heater Non-Fused Disconnect Micro Channel, E-Coat Low Sound Dual Pump, 7.5 HP Digital Compressor Low Ambient Head Pressure Control Wind Baffle Chilled Water Storage Tank BACnet Translator Control												
Condenser Information Altitude: 0 ft Number of Fans: 3 Total Condenser Fan Air Flow: 29646 CFM Entering Air Temperature: 95.0 °F	Electrical Information Unit Voltage: 460-3-60 V _{Ph} -Hz Connection Type: Single Point												
Integrated Pump Information Dynamic Head At Pump: 145.0 ft Internal Chiller Head Loss: 38.9 ft Balancing Valve Head Loss: 6.1 ft Dynamic Head External To Chiller: 100.0 ft	<table border="1" style="width: 100%; border-collapse: collapse; text-align: center;"> <thead> <tr> <th style="width: 30%;">Amps</th> <th style="width: 35%;">Electrical Circuit 1</th> <th style="width: 35%;">Electrical Circuit 2</th> </tr> </thead> <tbody> <tr> <td>MCA</td> <td>91.7</td> <td>---</td> </tr> <tr> <td>MOCP</td> <td>100.0</td> <td>---</td> </tr> <tr> <td>ICF</td> <td>185.7</td> <td>---</td> </tr> </tbody> </table>	Amps	Electrical Circuit 1	Electrical Circuit 2	MCA	91.7	---	MOCP	100.0	---	ICF	185.7	---
Amps	Electrical Circuit 1	Electrical Circuit 2											
MCA	91.7	---											
MOCP	100.0	---											
ICF	185.7	---											

All performance efficiency data are without pump.
 Note: Weights and electrical values do not include separately shipped accessories.
 Certified in accordance with the ARI Water-Chilling Packages using the Vapor Compression Cycle Certification Program, which is based on ARI Standard 550/590-2003.
 Sound power measured in accordance with ARI 370-2001.

Figure 71: Method used to determine water flow rate though the heat exchanger

Mathcad - [NTU method_12-17-11.xmcd]

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Normal Arial 10 B I U

Inputs:

$q := 158$ IT Load

$Thout := 20$ Cold Aisle set point

$Tcin := 9$ Inlet water temperature

$DT := 23.16$ Air Delta Temperature across HEX

Solution:

$HAT := Thout + DT$ $HAT = 43.16$

$Cair := \left(\frac{q}{DT} \right)$ $Cair = 6.822$ Thermal Capacity of Air

$flow_{cfm} := Cair \cdot \frac{2118.88}{(1.185 - 1.005)}$ $flow_{cfm} = 1.214 \times 10^4$ Air CFM

$Cratio(\epsilon) := -16.083 \cdot \epsilon^3 + 33.776 \cdot \epsilon^2 - 25.459 \cdot \epsilon + 7.5746$ Heat Capacity Ratio

$CrP(\epsilon) := -\epsilon \cdot \sin\left(7 \cdot \frac{3.14}{180}\right) + Cratio(\epsilon) \cdot \cos\left(7 \cdot \frac{3.14}{180}\right) + 0.25 - 0.08256$

$\epsilon_{min} := \frac{q}{Cair \cdot (Thout - Tcin)}$ $Lf := \frac{1}{1 + \frac{1}{\epsilon_{min}}}$ $Lf = 0.678$

$\epsilon_{min} = 2.105$

Note : The HEX effectiveness has a range between 0.25 to 0.90.
This range is based on test data.

$\epsilon_{ww} := Cratio(\epsilon) \cdot \epsilon = Lf$ $\left| \begin{array}{l} \text{assume, } \epsilon > .25 \\ \text{solve, } \epsilon \end{array} \right. \rightarrow 0.4404972958791510129$ Effectiveness

$Cratio(\epsilon) = 1.539$ $CrP(\epsilon) = 1.641$

$Cw := CrP(\epsilon) \cdot Cair$ Thermal Capacity of Water

$Cw = 11.198$ $mdotwater := \frac{Cw}{4.18}$ $mdotwater = 2.679$ kgps

$mdotwater \cdot 60 = 160.74$ LPM

$\frac{mdotwater \cdot 60}{3.785} = 42.468$ GPM

Table 22: Minimum water flow rate (GPM) required to cool 75 kW

Inlet water temperature		8°C H ₂ O			12°C H ₂ O			16°C H ₂ O		
Cold aisle temperature		20°C	25°C	30°C	20°C	25°C	30°C	20°C	25°C	30°C
Cooling unit fan flow rate	10600 cfm	18	12	--	28	15	11	54	18	14
	9400 cfm	16	11	--	23	14	10	54	18	13
	7800 cfm	14	11	--	26	12	--	54	17	12

Table 23: Minimum water flow rate (GPM) required to cool 50 kW

Inlet water temperature		8°C H ₂ O			12°C H ₂ O			16°C H ₂ O		
Cold aisle temperature		20°C	25°C	30°C	20°C	25°C	30°C	20°C	25°C	30°C
Cooling unit fan flow rate	9400 cfm	12	8	--	18	11	--	23	12	9
	7800 cfm	11	8	--	15	11	--	32	12	9
	6300 cfm	9	8	--	14	10	--	20	12	--

Appendix B: Dedicated Controller functional description

The Dedicated Controller (DC) is a PLC-based controller from Beckhoff (Model CX5020). Its main function is to control the chilled water valve of the heat exchanger based on a desired SAT set point and a SAT thermocouple sensor reading. The DC also controls the three system fans based on a user-defined average delta T between inlet and outlet of the heat exchanger, or a user-defined average pressure between all pressure sensors throughout the enclosure.

The functionality of the dedicated controller is divided into the following blocks:

- **Chilled water valve control** – Operates the chilled water valve control algorithm. The control algorithm will open or close the valve based on the difference between the current SAT reading and the desired SAT set point. The user has the choice between PID control and a mathematical model estimated control. The feedback control algorithm ensures that the valve operation will come to a steady state over time.
- **Fan control** – Operates the fan control algorithm. The control algorithm will increase or decrease the three EC fans RPM depending on how far the calculated average of delta temperatures are from the desired delta temperature set point, or how far the calculated average of air pressure is from the desired internal pressure of the enclosure. The delta temperature is the difference between the heat exchanger's front temperature (supply air) and rear temperature (return air). The choice of either control method is up to the user. The control algorithm utilizes PID or mathematical model estimated control for delta temperature, and PID only for pressure control. The control algorithm ensures that the fan operation will quickly come to a steady state.
- **Get fan configuration** – Modbus acquisition of fan static data (e.g. fan serial number, operating mode, etc.) that occurs once on PLC startup or when requested by the HMI.
- **Get fan real-time data** – Modbus acquisition of fan dynamic data that occurs at set intervals (e.g., speed set point, actual speed, link current, etc.)
- **Fan initialization** – Ensures the fans are configured to respond to Modbus commands upon PLC startup or when requested by the HMI.
- **Add new fan** – Configures a fan preset to factory defaults to the correct mode of operation. Occurs when requested by the HMI.
- **Real time acquisition** – Acquires supply air temperature, return air temperature, average air pressure, water inlet temperature, water outlet temperature, leak detection, humidity, water valve position, air flow rate, and water flow rate. All of these sensors are hard-wired into the Dedicated Controller except for air pressure, which is acquired from the Primary Controller.
- **Heartbeat detection** – Activates a fail-safe mode when the connection to the Primary Controller is lost.

Modes of Operation

The Dedicated Controller operates in one of the following modes:

- **Manual Mode** – The DC receives a SAT set point from the HMI. The DC will validate this set point against a valid SAT scope before applying it to the control logic. In this mode, DC will ignore all heartbeats.
- **Test Mode** – The DC receives test set points (CWV position (%) and Fan speed (rpm)) from the HMI and drives the actuators to these values. This is also called Bypass mode since the DC will simply use the set points to drive the actuators and not care about system feedback. The DC validates these set points against a max/min range prior to any action. In this mode, DC will ignore all heartbeats.
- **Optimized Mode** (default) – The DC receives a SAT set point from the PC given that there are live PC heartbeats. The DC will validate this set point against a valid SAT scope prior to applying the set point to the control logic. In this mode, DC will continuously check for PC heartbeats prior to getting an optimized SAT set point.
- **Fail-Safe Mode** – The DC goes into this mode only when it no longer detects the PC heartbeats changing every second while running in Optimized mode. Fail-Safe set point is a pre-configured FS set point. Default value for this set point is 65°F (18.3°C).

Upon energizing the DC module, the controller will be initialized to its Optimized (default) state. If the Primary Controller optimizer is not running, the DC will shortly move to Fail-Safe state. As soon as PC optimization code is running and updating its heartbeats, it will move back to the Optimized state. This cycle can be repeated as often as needed.

Appendix C: ICT Benefits Spreadsheet

Figure 72 and Figure 73 are from the DOE ICT Benefits Spreadsheet, which is used for energy savings calculations. (IT Load = 100kW. Hours/Year = 8760. New Technology PUE = 1.25. Current Technology PUE = 2.04)

Figure 72: ICT Benefits Spreadsheet: Unit Inputs Worksheet

4	Per Unit Impacts per year			
5		New	Current	Net
6		Technology	Technology	Impact
7				
8	Energy Use			
9	Electricity (billion kWh)	2.19E-04	9.11E-04	6.92E-04
10	Natural Gas (trillion Btu)			0.00E+00
11	Petroleum - Residual Fuel (million barrels)			0.00E+00
12	Petroleum - Distillate Fuel (million barrels)			0.00E+00
13	Petroleum - Liquefied Petroleum Gas (million barrels)			0.00E+00
14	Coal (million short tons)			0.00E+00
15	Feedstock (trillion Btu, please specify)			0.00E+00
16	Biomass (trillion Btu, please specify)			0.00E+00
17	Waste (trillion Btu, please specify)			0.00E+00
18	Other (please specify, trillion Btu)			0.00E+00
19				

Figure 73: ICT Benefits Spreadsheet: Results Worksheet

1	Impacts: New Technology vs Current Technology							
2	0							
3	Impact By Year	2010	2015	2020	2025	2030	2035	2040
4	Cumulative Units Installed	0	2,463	19,275	60,959	88,266	106,558	125,708
5	Annual New Units Installed	0	902	5,850	8,105	4,160	3,591	4,040
6	Annual Installations Total	0	902	5,850	8,105	4,160	3,591	4,040
7	Market penetration	0%	4%	29%	79%	97%	100%	100%
8								
9								
10	ANNUAL SAVINGS							
11	Energy Metrics							
12	Total primary energy displaced (trillion Btu)	0.00	14.09	105.25	332.86	481.96	581.85	686.41
13	Direct electricity displaced (billion kWh)	0.00	1.70	13.34	42.18	61.08	73.74	86.99
14	Direct natural gas displaced (bcf)	0.00	0.00	0.00	0.00	0.00	0.00	0.00
15	Direct petroleum displaced (million barrels)	0.00	0.00	0.00	0.00	0.00	0.00	0.00
16	Direct coal displaced (million short tons)	0.00	0.00	0.00	0.00	0.00	0.00	0.00
17	Feedstock energy displaced (trillion BTU)	0.00	0.00	0.00	0.00	0.00	0.00	0.00
18	Biomass energy displaced (trillion BTU)	0.00	0.00	0.00	0.00	0.00	0.00	0.00
19	Waste energy displaced (trillion BTU)	0.00	0.00	0.00	0.00	0.00	0.00	0.00
20	Other energy displaced (trillion Btu)	0.00	0.00	0.00	0.00	0.00	0.00	0.00
21								
22	Financial Metrics							
23	Energy-cost savings (\$MM/yr)	0.00	101.75	787.25	2547.93	3718.14	4488.68	5295.34
24								
25	Environmental Metrics							
26	CO Displaced (Metric tonnes)	0.00	284.41	2192.29	6933.51	10039.37	12119.91	14297.96
27	Carbon Dioxide emissions displaced (MM TCE)	0.00	0.28	2.05	6.47	9.37	11.31	13.35
28	Other greenhouse emissions displaced (MM TCE)	0.00	0.00	0.00	0.00	0.00	0.00	0.00
29	SO ₂ displaced (Metric tonnes)	0.00	3991.81	27605.94	87308.55	126418.33	152617.11	180043.63
30	NO _x displaced (Metric tonnes)	0.00	2512.97	18125.04	57323.56	83001.60	100202.74	118209.98
31	Particulates displaced (Metric tonnes)	0.00	85.33	587.65	1858.53	2691.06	3248.75	3832.57
32	VOCs displaced (Metric tonnes)	0.00	30.55	236.12	746.78	1081.29	1305.38	1539.97
33	Hydrocarbons displaced (Metric tonnes)	0.00	0.00	0.00	0.00	0.00	0.00	0.00
34	Solid Waste (Metric tonnes)	0.00	0.00	0.00	0.00	0.00	0.00	0.00
35	Other environmental benefits (Metric tonnes)	0.00	0.00	0.00	0.00	0.00	0.00	0.00
36								