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FINAL PROJECT REPORT

Demonstration of Alternative Cooling for Rack-Mounted Computer Equipment

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PREFACE

The California Energy Commission Public Interest Energy Research (PIER) Program supports public interest energy research and development that will help improve the quality of life in California by bringing environmentally safe, affordable, and reliable energy services and products to the marketplace.

The PIER Program conducts public interest research, development, and demonstration (RD&D) projects to benefit California.

The PIER Program strives to conduct the most promising public interest energy research by partnering with RD&D entities, including individuals, businesses, utilities, and public or private research institutions.

PIER funding efforts are focused on the following RD&D program areas:

- Buildings End-Use Energy Efficiency
- Energy Innovations Small Grants
- Energy-Related Environmental Research
- Energy Systems Integration
- Environmentally Preferred Advanced Generation
- Industrial/Agricultural/Water End-Use Energy Efficiency
- Renewable Energy Technologies
- Transportation

Demonstration of Alternative Cooling is the final report for the task within the project entitled Data Center Energy Efficiency and Demonstration Projects (contract number 500-09-002, conducted by Lawrence Berkeley National Laboratory. The information from this project contributes to PIER's Industrial/Agricultural/Water End-Use Energy Efficiency Program.

For more information about the PIER Program, please visit the Energy Commission's website at www.energy.ca.gov/research/ or contact the Energy Commission at 916-654-4878.

ABSTRACT

A prototype design for rack-mounted computer equipment cooling using refrigerant at the server/rack level, and conduction heat transfer at the board level (chip and other components) was tested to measure energy efficiency compared to other solutions. The prototype was previously demonstrated through the PIER Energy Innovations Small Grants program. For this project, the technology was evaluated at a larger scale by comparing rack-level performance against a rack of traditional servers. Energy performance was compared with a commercially available refrigerant-based rack-level cooling solution, a standard sever/rack, and an overall cooling performance estimate for a common facility-level cooling design.

A number of energy-efficiency metrics were introduced and used for comparison. The prototype design allows server fans to be removed, saving approximately 13 percent of the power required at higher server air inlet temperatures. For chilled water temperatures of 45°F (7.2°C) to 60°F (15.5°C), the prototype design had 14 to 16 percent, respectively, better energy efficiency than the other refrigerant device tested, when evaluated on a basis of total energy used per computing amount delivered. The prototype was compared to a typical computer room air handler cooling design at 45°F (7.2°C) chilled water temperature and 72°F (22.2°C) server air inlet temperature; the results showed a 13 percent energy-efficiency improvement. In addition to the energy savings found with chilled water temperatures ranging from 45°F (7.2°C) to 60°F (15.5°C), the prototype has the ability to cool effectively with much higher 78°F (25.5°C) chilled water temperatures, potentially providing even more energy efficiency while still providing chip and other component temperatures that meet long-term durability goals. The prototype design can provide significant energy savings in many climates, by using cooling towers, other evaporative cooling techniques, or dry coolers to reduce or eliminate chiller energy use.

Keywords: Server rack cooling, server cooling, direct chip cooling, datacenter cooling, refrigerant rack cooling

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EXECUTIVE SUMMARY

Introduction

A number of new designs to cool rack-mounted computer equipment were introduced in the last several years. In July 2009 a series of energy-efficiency evaluations (project name Chill-Off 2), was initiated in Santa Clara, California, to evaluate a number of these new designs.

The California Energy Commission funded a demonstration study through the Public Interest Energy Research program with the objective to compare cooling energy use of an innovative rack-mounted computer cooling design to other computer-rack cooling designs. This report presents the testing methods and results of the innovative cooling design from Clustered Systems, Inc. Previously, a PIER Energy Innovation Small Grant project supported a proof-of-concept demonstration for this technology. Based on that project's success, a larger-scale demonstration was designed to be included in the Chill-Off 2 series of demonstrations hosted by Sun Microsystems.

The Clustered Systems cooling design uses a low-pressure refrigerant to cool a "cold plate" mounted as the top cover of a server. Heat is transferred from server components by conduction to solid metal blocks or bands connected to the cold plate cover. The heat is then removed from the cold plate cover by a refrigerant to a heat exchanger.

By using the Clustered Systems cooling design, server fans can be eliminated, thus saving the electrical energy to run the fans. This innovative design can achieve further energy savings by reducing, or eliminating central server-room air conditioning.

Purpose

The purpose of this study was to evaluate a new rack-mounted computer cooling method as an alternative to conventional cooling methods, to compare their energy use.

Objective

The Clustered Systems design was compared to traditional servers and representative cooling systems. Seven different combinations of chilled water supply and server air inlet temperatures were developed to investigate how each device tested would perform for different conditions. In addition, results from two chilled water plant models were used to investigate potential energy savings as a function of chilled water temperature. One chilled water plant design was typical, and the other was equipped with a water side economizer.

Six energy-evaluation metrics were developed and used as a basis for comparing this new device; three use coefficient of performance and net cooling concepts that follow the ASHRAE 127-2007 Standard, and three others were developed to more clearly show the significant energy savings compared to the other devices. The second three were used to evaluate a power usage effectiveness-type comparison and the total energy used per server for a constant computing power delivered.

Conclusions and Recommendations

The results showed that the prototype design significantly improves energy efficiency compared to the conventional room-level cooling approaches and the other refrigerant-based design tested.

The prototype system has four key energy efficiency advantages: (1) energy is saved because server chassis-level fans can be removed, (2) fans to move the hot server exhaust air through heat exchangers can either be eliminated or reduced, (3) cooling fluid pumping energy at the rack is minimized because the liquid refrigerant has a very low flow rate compared to water, and (4) warmer chilled water temperatures can be used, allowing significant energy savings.

The researchers found that the prototype design delivers a 14 to 16 percent energy savings using 45°F (7.2°C) chilled water and 72°F (22.2°C) server air inlet and 60°F (15.5°C) chilled water and 80°F (26.6°C) server air inlet temperatures respectively, compared to the other refrigerant design tested and a simulation of a computer room air handler. These findings are based on using a metric that considers the total power needed per compute delivered, including the power for the chilled water plant.

Therefore, this new design consisting of the Liebert XDP (refrigerant-to-water heat exchanger), Liebert XDS (rack components), and Clustered Systems, Inc., design (server chassis modifications) shows significant energy savings compared to conventional server-cooling practices and compared to an in-row, refrigerant-based cooling design. The maturity of the mechanical design was not evaluated as part of this energy use evaluation, but the design may benefit from minor revisions that improve the heat transfer consistency from some components to the cold plate cover. Additional testing of an updated mechanical design evaluating the overall suitability for the industrial market is recommended.

CHAPTER 1: Introduction

In recent years, a number of new approaches to cooling IT equipment in data centers have been introduced. Many of these new designs promise improved energy efficiency compared to conventional methods such as below-floor plenum cold air delivery combined with computer room air handlers. This project's goals were to test an alternative IT equipment cooling system and compare its energy efficiency to common solutions in a side-by-side test. The alternate design by Clustered Systems, Inc., uses a low-pressure refrigerant cycle, chilled water, and conduction heat transfer to transfer the heat from the temperature-critical components, inside the server, to a refrigerant fluid and ultimately dissipated in the building-supplied chilled water system. This project developed energy-efficiency evaluation metrics and used these metrics to measure the new system's energy efficiency against (1) a common cooling method using computer room air handlers (CRAHs) cooled with chilled water, and (2) another low-pressure refrigerant cycle computer rack cooling device. A preliminary test plan included the installation and use of a CRAH; however, this was not implemented. In place of testing a CRAH, energy performance of a representative cooling system was simulated and an evaluation using a ceiling-mounted cooling unit was performed. The evaluation metrics include the incremental energy needed to produce the supplied chilled water.

A thermal schematic of this unique design is shown in Figure 1-1. The vast majority (in some cases close to 100 percent) of the required cooling is provided by a cold plate heat exchanger using conduction heat transfer to cool the temperature-critical components inside the server. The conduction heat transfer from the components to the cold plate uses solid metal blocks or strips made from aluminum or copper and other low heat conduction resistance material.

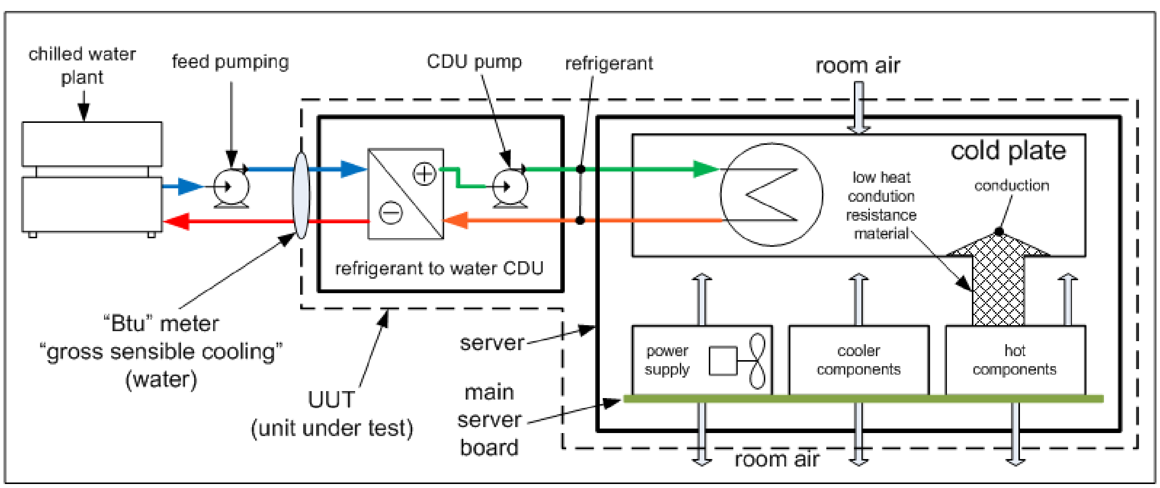

Figure 1-1: Thermal Schematic for the Clustered Systems Design

Source: Author

The remaining heat from lower-power components and the power supply are absorbed by the exposed surfaces of the cold plate or by nearby refrigerant plumbing via natural or forced convection or expelled to the room. This design requires the use of a Liebert XDP refrigerant-to-water cooling distribution unit (CDU). The left side of Figure 1-1 shows how this new system functions, making use of a CDU and chilled water supply. For energy analysis purposes, since the CDU is required, it is included as part of the unit under test (UUT). A custom rack, Liebert XDS, (Figure 1-2) is required with this design to provide the necessary refrigerant plumbing and unique mechanical systems.

Figure 1-2: Custom Rack for the Clustered Systems Design

Source: Author

The Clustered Systems design eliminates the need for the fans usually found inside the IT equipment. The energy saved by removing the fans and other attributes provided significant energy-efficiency gains.

CHAPTER 2: Project Methods

Layout and Test Plan

This project's goal was to compare the energy efficiency of the Liebert XDS rack to other data center cooling designs. A series of tests using seven standard target combinations of chilled water temperature and server air inlet temperature was planned, as shown in Table 2-1. Some difficulty reaching and controlling conditions for Test 1 and Test 7 were encountered due to control limitations with the refrigerant-to-water heat exchange unit supplying refrigerant to the prototype unit and another device being compared. The other two designs used for comparison did not have the piping installed to allow water temperature control; therefore, they were tested using the supplied 45°F (22.2°C) chilled water and a server air inlet temperature of 72°F (22.2°C). Because of the two previously mentioned reasons and a desire to show significant performance differences, only two of the seven tests—Test 2 and Test 6—were chosen to compare the different cooling device designs.

Table 2-1: Target Chilled Water and Server Air Inlet Temperature Combinations

Test ID	UUT Chilled Water Supply Target Temperature (°F)	UUT Server Air Inlet Target Temperature (°F)
1	45	60
2	45	72
3	50	72
4	55	72
5	60	72
6	60	80
7	60	90

Source: Author

The project researchers compared the performance of the Clustered Systems design to two other designs (Other Refrigerant and a Base Case) and to estimated data from a computer room air handler (CRAH). Descriptions of each test setup are detailed below.

- **Clustered Systems design with a Liebert XDS Rack**: This test was performed using one Liebert XDS custom rack containing 36 modified servers (with server chassis-level fans removed and an added chip-level custom heat transfer system). The heat transfer system moves heat from selected components directly to the bottom surface of the top cover using conduction. The base server used was a SUN x4100 configured with two dual-core AMD 285 processors, a 73 gigabyte (GB) disk drive and 16 GB of main memory. The Clustered Systems design allows the server fans to be removed from the SUN x4100 servers. The one rack was populated with 36 of the modified servers, not 40 as in a normal rack, because the Clustered Systems design tested requires more vertical rack space (2.0 inches vertical compared to the standard 1.75 inches) per server than that used for the standard SUN x4100 server in a standard rack.

- **Other Refrigerant Design:** The Other Refrigerant design consists of a Liebert XDP (refrigerant-to-water CDU) connected to a number of Liebert XDH/XDV refrigerant cooling units. The same Liebert XDP CDU was used to test the Clustered System design and the Liebert XDH/XDV-configured test. The IT equipment for this test consisted of eight racks populated with 40 SUN x4100 servers per rack and a number of Liebert XDH/XDV units between the racks and overhead providing the necessary cooling and air flow. The cooling equipment was configured with cold aisle containment. Note that the use of cold aisle containment with servers of this type (low heat rejection compared to required air flow rate) may not be recommended; consult the cooling system manufacturer.

- **Base Case:** This test used a fan cooling unit, sometimes referred to as a *fan coil unit, auxiliary cooling unit,* or *fan unit,* mounted in the ceiling area of the test room. Other than the fan coil unit, there was no other cooling provided in the room during the base case test. The base case is meant to see how the Clustered Systems design performs compared to a conventional data center cooling approach, where the cooling is provided by a device meant to cool a large part of a computer room using a water-cooled heat exchanger and fans moving the room air through the heat exchanger, without any containment other than that provided by the room. This test used one rack filled with a mixture of SUNx4100 and IBM servers running special software simulating a typical maximum compute load. The fan coil unit was a Williams AH-4000.

- **CRAH Estimate:** A common cooling approach in data centers places CRAH units in the same room as the racks of computers requiring cooling. A CRAH manufacturer provided thermal simulation results for the constraints equivalent to Test 2. The CRAH unit selected was a Liebert model CW114D. This model best matched the target IT load of 100 kilowatts (kW). This estimate is used to check the results obtained in the Base Case test as part of establishing the energy used for a common data center cooling design. See Appendix A for more details.

Test 2 conditions were used to compare all four devices. Test 6 conditions were used to compare the Clustered Systems rack to the Liebert XDH/XDV.

The tests were done in a specially prepared, semi-sealed room. Figure 2-1 shows the room layout and component locations for the Clustered Systems test.

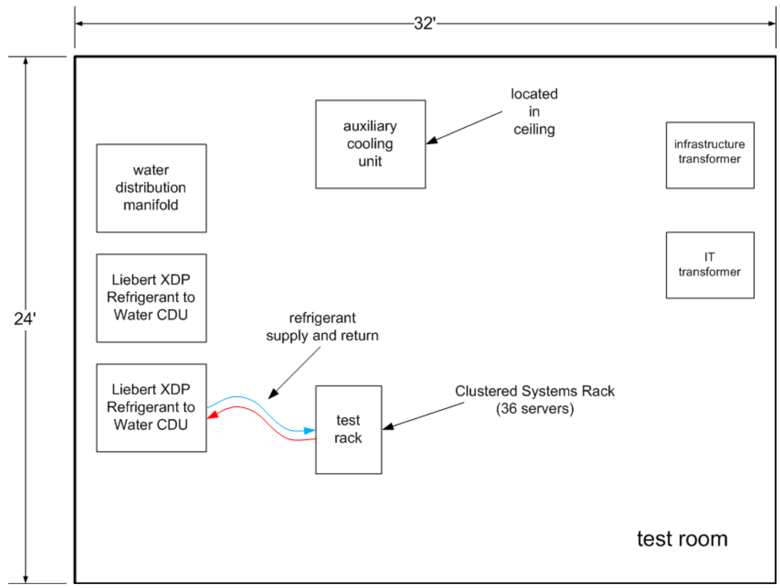

Figure 2-1: Room Layout for Cluster Systems Test

Source: Author

Figure 2-2 shows the room layout, component locations, and approximate dimensions for the Other Refrigerant design testing. During the tests of the Other Refrigerant design the number of active cooling units was adjusted to keep the air flow balanced with the server needs. Only Test 2 and Test 6 were used for comparison against the Clustered Systems rack, as indicated in Table 2-1.

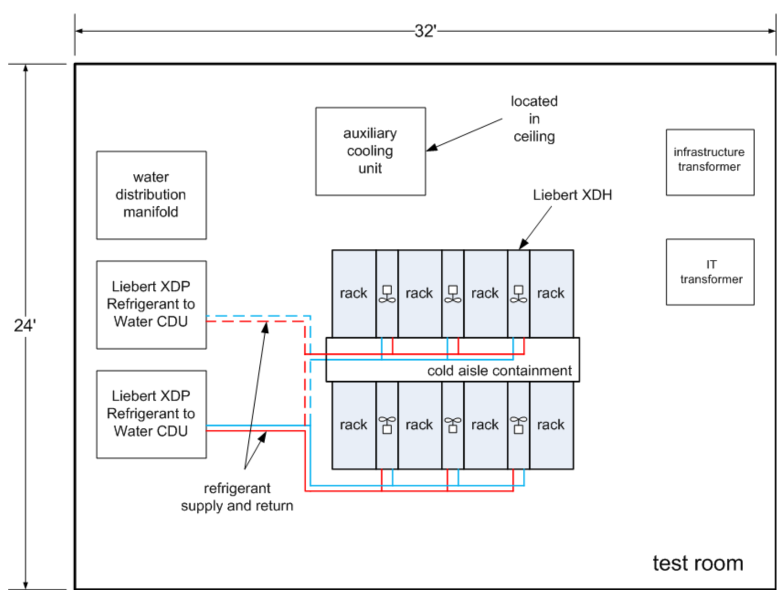

Figure 2-2: Room Layout for the Other Refrigerant Design Test

Source: Author

Data was recorded continuously (approximately every 30 seconds) during a four-hour test and the collected data was averaged, using one-minute intervals to obtain a final per test value for each monitored data point.

Assumptions and Definitions

The following three diagrams show the basic components and items considered part of the unit under test (UUT) for the designs tested or estimated for comparison.

Liebert XDS

Figure 2-3 shows the component definitions and UUT thermal analysis boundary for the Liebert XDS rack. Note that the water-to-refrigerant CDU is considered part of the Clustered Systems solution, and the server does not contain the standard server chassis-level fans.

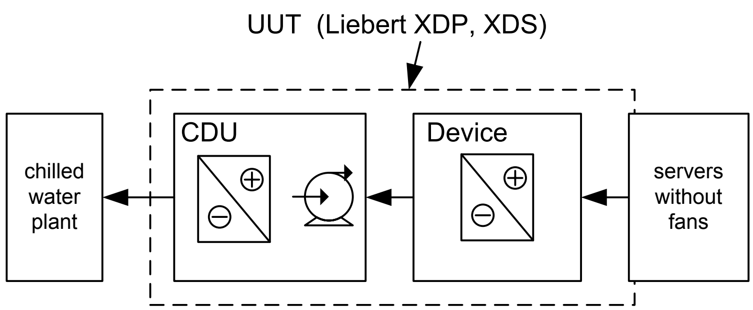

Figure 2-3: UUT Definition Diagram for the Liebert XDP, XDS

Source: Author

Liebert XDH/XDV

Figure 2-4 shows the basic component definitions and UUT thermal analysis boundary for the Other Refrigerant design used for comparison to the Clustered Systems design. Note that the device for this design has fans requiring power, and the servers are unmodified, containing the standard chassis-level fans found in the SUN x4100.

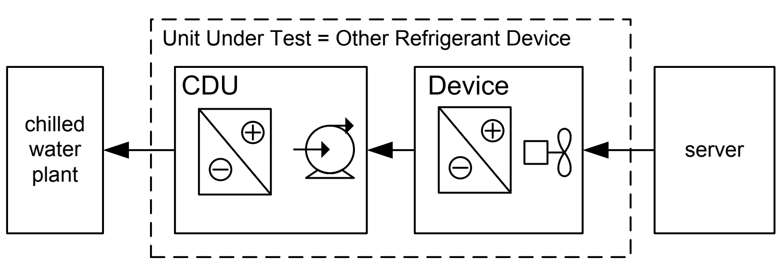

Figure 2-4: UUT Definition Diagram for the Liebert XDH/XDV

Source: Author

Base Case and CRAH Estimate

Figure 2-5 shows the basic component definitions and thermal analysis boundary for the two designs representing two commonly found data center cooling methods. These are called the *Base Case* and *CRAH estimate.* The components considered as part of these UUTs are simplified. A water-to-water CDU is not included as part of the UUT definition because these devices use chilled water supplied directly from the chilled water plant. Note that both the Base Case and CRAH estimate UUT types have power-consuming components (air circulation fans).

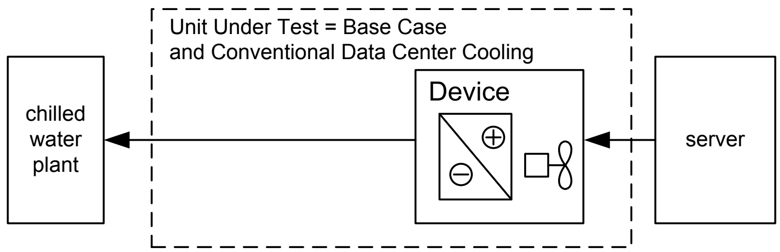

Figure 2-5: UUT Definition Diagram for the Base Case and CRAH Designs

Source: Author

Assumptions for the Energy Use Comparison Calculations

The following assumptions were made for this project:

- **Feed pump power is absorbed in the UUT and water flow.** The feed pumping power, including hydraulic, motor and pump inefficiencies, is released in the UUT and/or is absorbed in the water flow. This assumption accounts for the total electrical power needed for chilled water pumping power and reduces the net cooling available relative to the thermal power measured using the water flow rate and water delta temperature. This power is referred to as *Feed, Feed Power, or Feed Pumping Power*.

- **The refrigerant-to-water CDU prorating factor is the server power divided by 160 kW.** The water-to-refrigerant CDU used in the testing has a 160 kW cooling capacity—well above what was needed for some tests. The refrigerant pump power was constant (approximately 821 watts) and could not be reduced in speed or power to match the low flow desired for best energy efficiency for some tests. Therefore the assumption made was that the CDU power would be prorated using the multiplier of server heat divided by 160 kW, as if an actual installation fully utilized the CDU's maximum performance. The effects of this assumption were not fully characterized and could be an area for further research. The chilled water flow rate for the refrigerant-to-water CDU used the same prorating factor with a flow rate of 120 gallons per minute, equal to 100 percent utilization.

- **The prorated CDU pump power is absorbed in the CDU and refrigerant flow.** The prorated electrical power for the CDU is fully absorbed in the refrigerant flow. During the testing the refrigerant-to-water CDU pump power was a constant. A simplifying assumption is that the difference between the actual test CDU power and prorated power is lost to the room and not part of the calculation of net cooling. This pumping power value is referred to as *CDU power* or sometimes as *CDU.*

- **The device power is part of calculating the net cooling.** If the device has fans or other electrical power-consuming contents, this power is referred to as the *device power* and is subtracted from the Btu meter value as part of calculating the net cooling provided. The assumption for this additional subtraction is that 100 percent of the device power will be absorbed in the water or refrigerant flows, thereby increasing the water or refrigerant delta temperature, thereby reducing the calculation of cooling provided.

Test Quality Indicators

Two calculations were used to check for large amounts of unaccounted-for heat energy that could indicate measurement errors or outside influences that might significantly affect the final results: (1) calculating the test room net power balance, and (2) comparing the net cooling provided to the server power.

1. Test Room Net Power Balance

The room power balance, also called *heat balance*, was calculated by subtracting the thermal power removed by the two water cooling methods in the room, the UUT (when present), and the ceiling-mounted fan cooling unit from the sum of the server electrical power and infrastructure electrical power measured at the room transformer inputs. The electrical power for the server and infrastructure was metered and recorded, and it accounted for the major power inputs to the test room. When the test room thermal power balance was +/-10 percent or better, or the device being tested was close-coupled with the heat from the server rack(s), it was assumed that data collected was valid unless the recorded key test parameters listed in Table 2-1 were not close to the target values. Appendix D lists this calculation, percent room heat lost, for each test.

A few observations were noted:

- There was a noticeable air flow coming under the test room entry door from the adjacent employee office areas. There was no attempt to directly measure this heat loss or gain, or to otherwise characterize the heat flow relative to the other heat power measurements during a test, other than calculating the net room heat loss or gain. The room energy balance calculation should be affected more for one-rack tests than for eight-rack tests.

- During the tests it was noted that the ceiling-mounted fan unit return water temperature was lower than the supply temperature in many cases when the ceiling fan unit was being controlled to supply no cooling. The fan unit cooling was controlled by a three-way valve, therefore the water temperature should rise very slightly or be equal (supply relative to the return temperature) when the fan unit control calls for no cooling. Given that there were no other objects or an environment nearby colder than the building chilled water supply temperature (normally 44°F [6.6°C]–46°F

[7.7°C]), an investigation was conducted to find the root cause of this reading anomaly. After the testing was done, inspection of a contact-type water pipe temperature sensor was found to be not well connected. Additional review of past data indicated that the reading error was approximately 0.3°F (0.17°C) in many cases, resulting in a power accounting error of approximately 550 watts. Because the cooling provided by the ceiling fan unit is not directly part of the net cooling calculation, this error did not affect the metrics calculation, but it did affect the room energy balance.

- The infrastructure transformer supply power to the devices, CDU, and room recirculation pump had power monitoring on both the 480 volt (input) and 208 volt (output) sides. It was observed that in some cases the output power recorded was higher than the input power (efficiency gain), and in many cases there was a much larger than expected efficiency loss recorded. An investigation was undertaken to determine the cause of these anomalies. It was found that the current transformers installed were not sized per manufacturer recommendations. In an attempt to verify the power reading on the 208 volt side (output) of the transformer, different current transformers were obtained and used with the input power meter installed on the 208 volt side (output), to investigate the power reading quality on the output side. The results show that the 208 volt (output) side readings were accurate. This finding indicates that the anomaly is most likely associated with the meter readings on the 480 volt (input) side of the infrastructure transformer. Since the 480 volt side transformer readings are part of the room energy balance calculation, there could be some false results shown for room energy balance values.

2. Compare Net Cooling Provided to the Server Power

For each test, the server power was recorded, and the net cooling provided (UUT bulk cooling minus cooling device power) was calculated. If the net cooling divided by the server power is above 100 percent, this indicates that more cooling than necessary is being provided and is not considered a concern. If less than 90 percent, the cooling device may not be capturing the amount of heat desired, or instrumentation errors are present. Resources were not available to find the source of anomalies and retest. Three different valuations for net cooling divided by server power are listed in the last three columns listed in Appendix D for each test. In all cases the comparison metrics use the net cooling provided as the basis for the calculations.

The recorded data and calculated results are presented for all tests that were successfully completed. In addition to the above two quality indicators the reader should review the tested server air inlet temperature and chilled water temperature when making a comparison of performance differences between designs.

Data Recording

The following data points were continuously recorded for each test. The numbers correspond to the data points in Figure 2-9.

1. power going to the servers (e.g., 36 servers in the Clustered Systems rack test) (kW)
2. UUT chilled water supply temperature (°F)
3. UUT chilled water return temperature (°F)
4. UUT chilled water flow (gallons per minute, gpm)
5. auxiliary unit chilled water supply temperature (°F)
6. auxiliary unit chilled water return temperature (°F)
7. auxiliary unit chilled water flow (gpm)
8. power to auxiliary cooling unit (kW)
9. power to infrastructure (kW) (includes room recirculation pump [4] and UUT power)
10. power to infrastructure transformer (kW)
11. power to IT transformer (kW)
12. cpu0 and cpu1 temperatures – (°C) (both for each server)
13. power to secondary loop (room) circulation pump (kW)
14. server inlet air temperatures (SIAT) for each rack (°F) (18 points per rack)]
15. server leaving air temperatures (SLAT) for each rack (°F) (18 points per rack)
16. room area temperatures (°F) (5 zones, 3 per zone)

Figure 2-9 shows the schematic locations for the above-listed data points. It also shows the layout for the Other Refrigerant cooling design. The Clustered Systems design has the spot cooling function inside the rack and does not have fans.

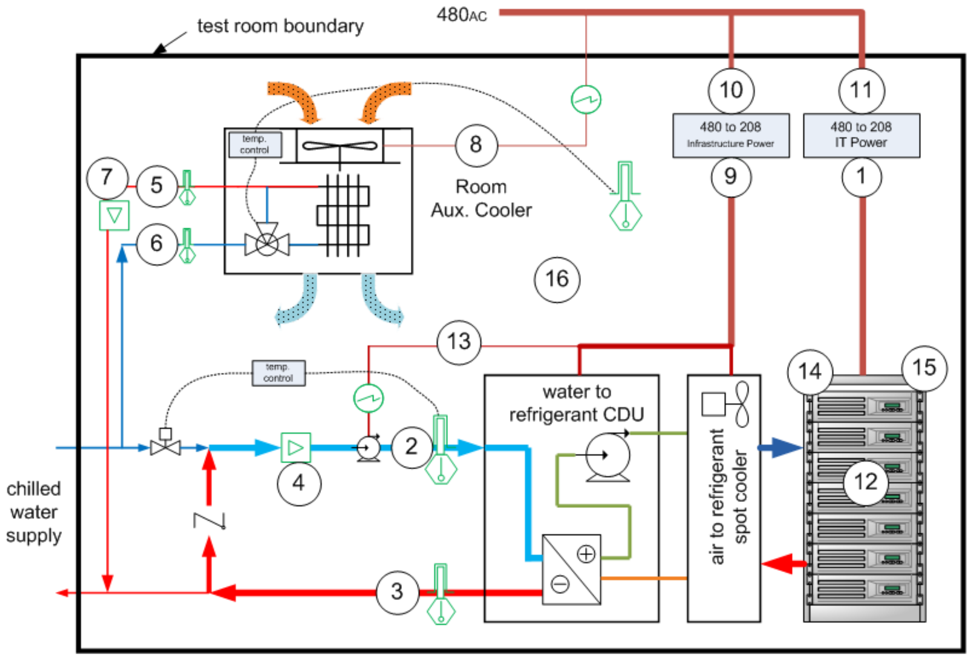

Figure 2-6: Schematic Locations of the Data Points

Source: Author

Not all of the recorded data were directly used in the final comparison calculations. For example, the temperatures in the room or at the server inlets where checked to ensure that they corresponded to the test plan, but those data were not used to make thermal power calculations.

Description of Terms and Metrics Used to Compare Cooling Designs

Six energy-efficiency metrics were developed to evaluate and compare the Clustered Systems design to two other cooling system designs. The following paragraphs first define the terms used in the metrics and then present the metrics themselves.

Chilled Water Plant Power ("CWP")

Chilled Water Plant Power is the electrical power necessary (expressed in kW electrical) to process the chilled water. Two chilled water plant models were used. One was a based on an American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) code minimum design, and one had a water-side economizer feature to evaluate advantages of higher chilled water supply temperatures. The two models did not include the pumping power to distribute the chilled water; only the power needed to cool the chilled water. Both models used a plant size of 600 tons of cooling. The individual pumping power needed for each test was accounted for using the pump calculation methods described below. See Appendix B for more chilled water plant model details.

A chart containing data plots is shown in Figure 2-7.

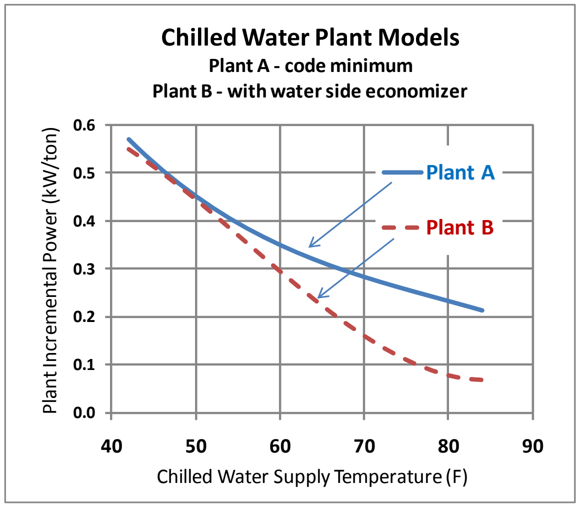

Figure 2-7: Chilled Water Plant Models (A and B) Electrical Power – Not Including Chilled Water Distribution Pumping Power

Source: Taylor Engineering

Chilled Water Distribution Pumping Power ("Feed or ASHRAE Feed")

Feed pump power (expressed in kW electrical) is calculated using two methods:

1. ASHRAE 90.1 Defined Feed Delta Pressure (ΔP) (ASHRAE Feed)

The ASHRAE 90.1 guidelines for chilled water plant design contain a defined pressure differential for chilled water distribution. This value is 75 feet of head, which corresponds to 32.4 pounds per square inch differential, (psid). The feed pump power was calculated using this defined delta pressure and the actual or estimated water flow rate. In the case that a CDU is part of the UUT, the pumping power is calculated using the primary side flow rate for maximum CDU-supported server heat load divided by the actual server heat. This calculation provides a prorated feed or pumping power. In the case where the UUT does not contain a CDU, the feed power is calculated using the defined Δ**P** and the water flow rate measured during the test.

Equation 2-1 defines the ASHRAE Feed Pumping hydraulic power. The required motor electrical power is found by dividing the hydraulic power by 0.65 (the total pump efficiency ratio) to account for the combined pump and motor losses. The 0.65 value for total pump efficiency is obtained from the ASHRAE 90.1 Standard. Equation 2-2 is used to calculate the electrical power stream needed for the chilled water distribution pumps.

ASHRAE Feed Pumping Power (kW hydraulic) = (UUT water flow rate (gpm) x ASHRAE Δ**P** (32.4 psid) x 0.000435) (Eq. 2-1)

ASHRAE Feed Pumping Power (kW pump electrical) = ASHRAE Feed Pumping Power (kW hydraulic) / 0.65 (Eq. 2-2)

2. As-Tested Feed Pump Power

The test setup contained a pump, often referred to as the *room pump* or *red pump,* used to recirculate the water, enabling temperature control of the secondary loop and at the same time providing pressure to feed the UUT. The pump was a variable frequency drive (VFD) type and was speed-controlled as a function of the Δ**P** supplied to the main distribution supply and return pipes. The pressure setting for the pump was changed, sometimes without good record keeping therefore the ASHRAE based Feed pumping power was used for analysis. The room pump power is not presented, but is listed in Appendix C under Test Room Pump Power.

CDU Power ("CDU" or "CDU Prorated")

CDU power is expressed in kW electrical. The test room was equipped with two refrigerant-to-water type CDUs, each containing a refrigerant pump, available to provide cooling fluid to the Clustered Systems rack and the Other Refrigerant design used for comparison testing. The pump control system prohibited the reduction of pump power to match the required heat load; therefore, the measured CDU power was scaled or prorated. (See Eq. 2-3.) The CDU pump power was measured as a constant 821 watts.

CDU Prorated (kW) = 0.821 kW x tested server power (kW) / 160 kW) (Eq. 2-3)

Cooling Device Power ("Device")

The Clustered Systems UUT is considered to include the water-to-refrigerant CDU and the actual cooling device as shown in Figure 2-4. Power use for the device is expressed in kW electrical. The typical device may contain one or more fans or other electrical power-consuming components. In the case of the Clustered Systems design there are no fans or power consuming components assigned to the cooling device, therefore the results in the following sections have zero, or a very small value, listed as the device power for the Clustered Systems design. For the other devices used for comparison there was device power recorded during the tests because these devices contained fans. In those cases the power assigned to the cooling device is the power measured using the test room power meters.

Unit Under Test ("UUT")

The UUT power (expressed as kW electrical) is the sum of the electrical consuming components of all equipment found inside the data center room necessary to support the cooling device coil. For example, in the case of the Other Refrigerant design, electrical power is needed for the refrigerant-to-water CDU and for the fans contained in each of the cooling modules located near the server racks. Both the CDU power and the fan power are combined to obtain the UUT power.

UUT power (kW) = CDU Prorated power + Device power (Eq. 2-4)

Gross Sensible Capacity ("GSC")

The thermal energy (expressed in kW thermal) removed from the test room using water was measured separately and can be assigned to either the UUT or the ceiling-mounted fan unit. The thermal energy was measured using two temperature probes and a flow meter for each. See Eq. 2-5 for determining the value of GSC. The name for this power flow is "gross sensible capacity (GSC)" according to ASHRAE 127-2007.

GSC(kW) = water flow (gpm) x water delta temperature (F) x 0.1464 (Eq. 2-5)

Net Cooling Provided ("NC")

In the case of a device that uses a refrigerant-to-water heat exchanger, the best way to determine the cooling provided by the UUT is to find a method to directly measure the refrigerant fluid enthalpy difference of the flow to and from the cooling device. A direct method was not available, therefore an alternative method using collected data and calculation was used and follows the ASHRAE 127-2007 Standard for calculating net cooling. This method is also followed for device cooled with water only.

The definition of *net cooling* provided for the comparisons is the power determined through use of the GSC thermal power minus the sum of the cooling device power, CDU power, and feed pump power. The actual net cooling provided may contain heat lost or gained from the room environment. For example, some tests showed that the UUT provided more or less net cooling than the server kW, this indicates that the UUT is providing more or less cooling respectively than necessary. The energy efficiency comparison metrics defined below use the net cooling provided.

Figures 2-8, 2-9, and 2-10 provide details for each device type tested and show the calculation of net cooling provided (NC_b type – Eq. 2-7), when the UUT and feed pump power are considered, in a graphical format. Note that the pressure data locations, indicated by circles with an X, were not test measurements; the delta pressure is defined from ASHRAE 90.1.

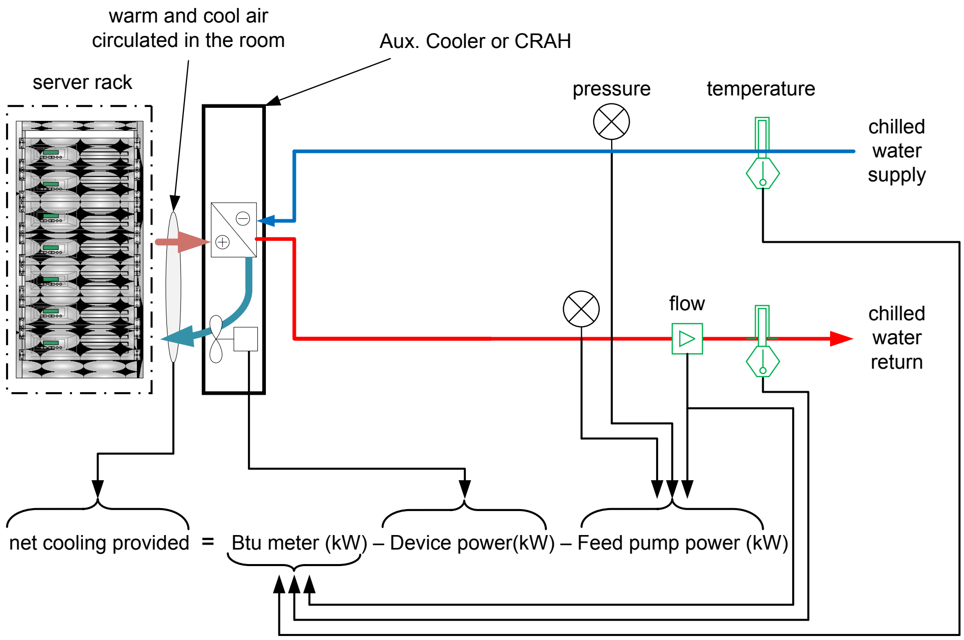

Figure 2-8: Net Cooling Provided (NC_b) – Water Only Cooling Example Shown – Auxiliary Cooler or CRAH

Source: Author

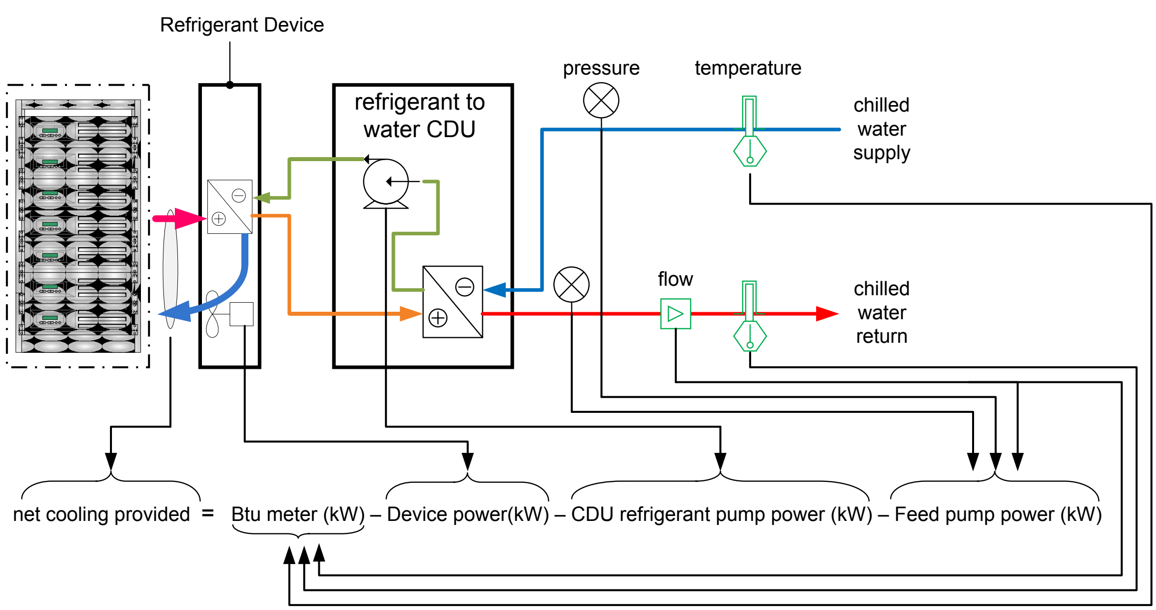

Figure 2-9: Net Cooling Provided (NC_b) – Refrigerant-to-Water Cooling Example Shown – In-the-Row Type Refrigerant Based Device

Source: Author

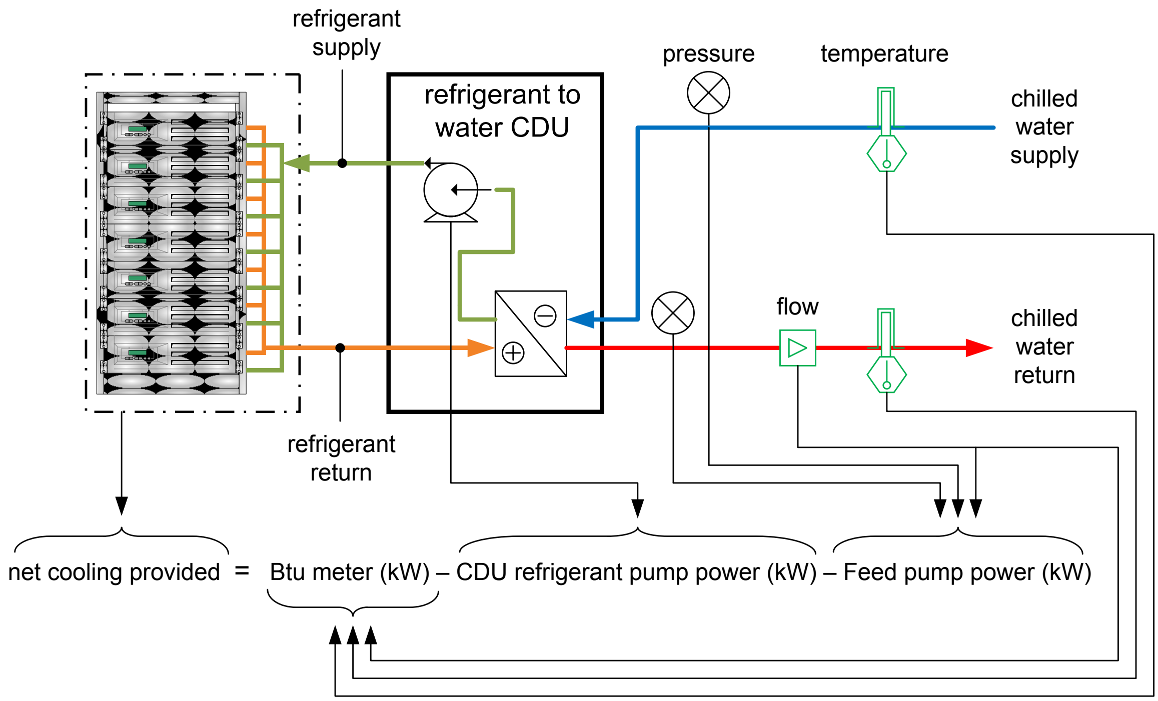

Figure 2-10: Net Cooling Provided(NC_b) – Direct Touch Cooling

Source: Author

Not all of the energy (expressed in units in kW thermal) recorded as GSC for the UUT is available to provide cooling for the IT heat being cooled. The net cooling provided is the cooling power remaining after the local power consumption and flow friction is subtracted. Two different calculations for net cooling are provided. See the discussion and definition of net cooling provided above. Equations 2-6 and 2-7 define NC_a and NC_b, respectively that are used in the following energy efficiency performance comparison metric calculations.

$$NC_a = GSC_{UUT} - \text{UUT power} \qquad \text{(Eq. 2-6)}$$

$$NC_b = GSC_{UUT} - \text{UUT power} - \text{ASHRAE Feed} \qquad \text{(Eq. 2-7)}$$

Server Power – ("IT")

Fortunately, the test room was equipped with a transformer powering all servers used as part of a test with no other loads. The transformer had power meters on the input (480 volt ac) side and the output (208 volt ac) side supplying power to the servers, providing good data on the combined true power used by the servers or "IT." This power is expressed as kW electrical. The power lost from cables going from the transformer output to the power distribution units (PDUs) on the server racks and from the actual power cords was assumed to be negligible.

Figure 2-11 illustrates the makeup of each electrical and thermal power stream defined above.

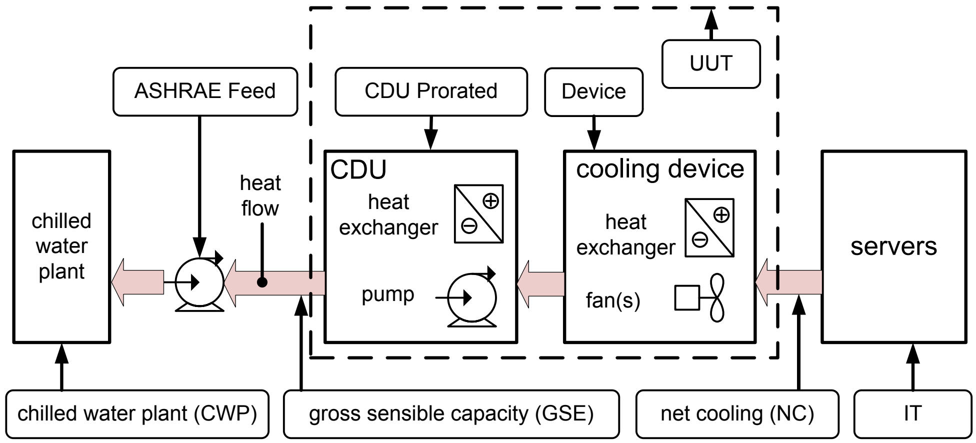

Figure 2-11: Electrical and Thermal Power Streams

Source: Author

Comparison Metrics

The following six metrics were used to compare cooling devices. The definitions of sensible coefficient of performance (SCOP)-type metrics for this report are guided by the ASHRAE 127-2007 Standard. All SCOP metrics defined below have units of kW thermal divided by kW electrical. The air-conditioners covered by the ASHRAE 127 standard include equipment that use chilled water to cool warm air generated by computer equipment. For example, a computer room air handler (CRAH) can be evaluated using the ASHRAE 127 Standard. Although some of the devices described in this report use a different layout and configuration compared to a typical computer room air handler, since all use chilled water for cooling inside the data center, the analysis follows the ASHRAE 127 evaluation approach.

Metric 1: $SCOP_a$

$SCOP_a$ is the electrical power needed for the device and nearby required supporting equipment. Units for SCOP metrics are kW thermal / kW electrical power.

$$SCOP_a = NC_a / \text{UUT power} \qquad \text{(Eq. 2-8)}$$

Metric 2: $SCOP_b$

$SCOP_b$ adds the feed pumping power to $SCOP_a$.

$$SCOP_b = NC_b / (\text{UUT power} + \text{Feed power}) \qquad \text{(Eq. 2-9)}$$

Metric 3: $SCOP_C$

SCOPc is the same as SCOPb but adds the power needed to produce an incremental amount of chilled water. NCb is used because the incremental chilled water plant power is assumed to be dissipated outside the data center room and doesn't directly affect the net cooling provided.

$$SCOP_c = NC_b / (\text{UUT power} + \text{Feed power} + \text{CWP}) \qquad \text{(Eq. 2-10)}$$

Metric 4: Chill-Off 2 Energy Efficiency ("COEE")

The COEE metric is provided for readers who may want to see a metric very similar in concept to the widely accepted power usage effectiveness (PUE) metric. The definition for COEE is total energy used, including chilled water plant power and IT power, divided by IT power (Eq. 2-11). This metric does not include power components commonly found included in PUE evaluations, such as lighting, UPS systems, or distribution losses from the utility connection to the data center computer room. This result can be used to estimate cooling related energy needs as a function of server power. For example: if COEE = 1.5 and the server power is 400 watts, then approximately 600 watts will be needed to operate and cool the server. Lower values are more energy efficient. For processes commonly considered, the lower limit value for this metric is unity. The value of COEE is unitless or can be thought of as kW electrical / kW electrical because all power streams defined in the equation are electrical.

$$COEE = (\text{CWP} + \text{UUT power} + \text{Feed power} + \text{IT power}) / \text{IT power} \qquad \text{(Eq. 2-11)}$$

The Clustered Systems design has the fans removed and therefore provides the same compute effort using less power. Since the COEE metric contains the server power and the Clustered Systems design provides more compute per an given amount of server power an adjusted COEE is need for the Clustered Systems (COEEcs) results to provide a fair comparison.

The adjusted Clustered Systems COEE (COEEcs) calculation method is shown using Eq. 2-11a and Eq. 2-11b.

First the normalized server power is calculated using Eq. 2-11a:

Normalized Server Power = tested Clustered Systems server power / (1- percent server power saved using the Clustered Systems designed server) (Eq. 2-11a)

Second, the adjusted COEEcs is calculated using Eq. 2-11b:

COEEcs = (tested Clustered Systems Server power * tested Clustered Systems COEE) / Normalized Server Power (Eq. 2-11c)

COEEcs values may be below unity when compared to other devices that have low COEE values just above unity.

Note: COEE is directly related to SCOPc by Eq. 2-11d when the net cooling is equal to the IT power.

COEE = (1 / SCOPc) + 1 (Eq. 2-11d)

Metric 5: Power per Server ("PPS")

The PPS metric is useful when servers of a different design are tested as part of an energy efficiency comparison study and the amount of computer-calculated results per unit of time delivered is equal. The Clustered Systems modified server uses less electrical power for a given amount of computing compared to the unmodified design. The reduced power is listed in the comparison results under PPS. The PPS values use units of kW per server.

PPS (kW/server) = total IT power tested / number of servers tested (Eq. 2-12)

Metric 6: Total Power Used per Server ("TPUS")

The TPUS metric calculates the total power used per server, including the power needed to supply cooling. TPUS contains similar information compared to the COEE metric but lists the energy comparison on a per-server basis. This metric fairly accounts for the power saved by removing the server fans. This metric also contains the reduced energy required for the reduced cooling needed because the server heat load is reduced. The TPUS units are kilowatts/server. Care should be used when testing and making comparisons using this metric; for example, in this case the basic server design and compute calculations per unit of time was controlled to be equal.

TPUS = (CWP + Feed power + UUT power + IT power) / number of servers tested (Eq. 2-13)

CHAPTER 3:
Project Results

The Liebert XDS design and two other devices—another refrigerant device Liebert XDH/XDV and a base case—were tested and results calculated as described above to obtain comparisons of energy use efficiency. Because of test equipment limitations, Test 2 (45°F chilled water temperature, 72°F server air inlet temperature) was used to compare the three devices. A comparison between the Clustered Systems design and the Other Refrigerant design was made using the Test 6 (60°F chilled water temperature, 80°F server air inlet temperature) condition to see if there were energy savings at higher chilled water temperatures.

The estimated performance of a CRAH-type device was also evaluated to compare to the base case test at test condition 2. In addition to the planned tests there was an unexpected event resulting in very high temperature water being supplied to the test setup. Fortunately the data collection systems recorded the Clustered Systems design performance during this event. The analysis of this data uncovered unique energy-saving performance advantages of the Clustered Systems design.

Standalone Metric Performance

Five of the seven standard tests (2 through 6) listed in Table 2-1 in Chapter 2 were completed using the Liebert XDS rack. During the course of testing there was an unexpected event causing the chilled water temperature supplied to the UUT to rise. Data from this event is listed as Test 8*. Table 3-1 lists key conditions and the energy consuming contributions for the standard tests along with the data from the unexpected event. Refer to definitions for metric contributions in Chapter 2 to understand the origins of listings in Table 3-1.

Table 3-1: Energy Efficiency Performance Tests

Test ID#	UUT ID	Actual Server Inlet (F)	Actual Chilled Water (F)	Plant A Elect. Power (kW)	Plant B Elect. Power (kW)	ASHRAE Feed Pump Power (kW)	Prorated CDU Power (kW)	Device Power (kW)	Total Server Power (kW)	Server Qty.	$SCOP_a$	$SCOP_b$	$SCOP_c$ Plant A	$SCOP_c$ Plant B	COEE Plant A	COEE Plant B	TPUS Plant A	TPUS Plant B	PPS
2	CS	72.4	45.0	1.59	1.56	0.168	0.053	0.001	10.3	36	196.7	47.2	5.8	5.9	1.04	1.04	0.337	0.336	0.286
3	CS	73.0	50.9	1.50	1.46	0.169	0.053	0.001	10.4	36	217.8	52.4	6.8	6.9	1.03	1.03	0.337	0.336	0.289
4	CS	72.9	54.6	1.19	1.11	0.170	0.054	0.001	10.5	36	189.3	45.3	7.2	7.6	1.00	1.00	0.331	0.328	0.291
5	CS	73.2	59.5	1.06	0.90	0.171	0.054	0.001	10.5	36	188.0	45.2	7.9	9.1	0.99	0.98	0.327	0.323	0.292
6	CS	78.6	60.4	1.08	0.89	0.171	0.054	0.001	10.5	36	196.5	47.1	8.1	9.5	0.98	0.96	0.328	0.323	0.292
8*	CS	69.6	78.4	0.62	0.21	0.170	0.054	0.001	10.5	36	158.4	38.0	10.2	19.5	0.96	0.92	0.314	0.303	0.290
2	Ref. #1	72.1	43.5	16.77	16.32	1.681	0.530	2.896	103.3	320	30.5	20.1	4.7	4.8	1.21	1.21	0.391	0.390	0.323
6	Ref. #1	78.5	59.5	10.21	8.62	1.745	0.551	3.673	107.3	320	22.8	15.8	5.8	6.5	1.15	1.14	0.386	0.381	0.335
2	Base	79.5	44.0	2.63	2.57	0.323	NA	3.680	12.5	NA	3.7	3.3	2.0	2.0	1.53	1.52	NA	NA	NA
2	CRAH Est.	72.0	45.0	7.37	7.23	2.168	NA	10.330	100.0	NA	12.0	9.8	6.2	6.2	1.20	1.20	NA	NA	NA

Source: Author

The data for Test 8* collected during the event show that the chilled water supply failure resulted in 78°F (25.5°C) chilled water to be supplied to the UUT. This event revealed a positive attribute (it was able to provide adequate cooling using very high chilled water temperatures) of the Clustered Systems design and at the same time provided a means to gain some insight regarding energy savings that are potentially achievable, making use of the reduced energy needed to produce the high chilled water temperatures. There was an additional electrical interruption during the chilled water temperature event, which affected all servers being tested and caused the power to drop.

A one-hour period, where the power supplied to the servers was consistent and the water temperature was at or near 78°F, was selected for analysis. This one hour of data was averaged and is shown in Table 3-1 under Test 8*. Test 8* did not appear to have good energy balance results, therefore it is not included with the results reporting. Comments regarding the data from Test 8* are discussed later in the Observations Using Very High Chilled Water Temperature Supply section.

The energy metrics used for comparison have different values, depending on the selection of the options that fit an interest or a particular facility design. Note that the CDU power is listed as prorated. The refrigerant-to-water CDU control system kept the pump power constant. The assumption made was that the number of CDUs deployed in a data center would be such as to operate and make full use of the CDU refrigerant pump power, therefore the maximum performance of the CDU was used calculate the prorated power.

Comparisons to Other Cooling Methods

Data from two other cooling designs that were tested, and estimated data from another design, were compared against those from the Clustered Systems design:

Other Refrigerant Design

The Other Refrigerant design was tested. A description is provided at the beginning of Chapter 2.

The Other Refrigerant design used for comparison was set up to cool eight racks filled with the same base server model, totaling 320 servers. The servers used in the eight racks contained the standard chassis level fans. This other device setup also used one or two water-to-refrigerant CDUs, depending on the load and test temperature targets. The refrigerant-to-water CDUs (model Liebert XDP) used were the same for both the Clustered Systems tests and the Other Refrigerant design type.

Base Case

One test performed was called the Base Case. This test attempted to simulate the case of only a CRAH device cooling the computer rack heat. A CRAH unit was not available for the test so a four-ton fan unit (Williams AH-4000) located in the ceiling area was used as the substitute for the simulation. This unit was supplied with water directly from the building chilled water supply, and therefore adjusting the water supply temperature was not possible (see Figure 2-9). This fan unit had a constant speed fan continuously consuming approximately 3.6 kW. This test setup used a single standard rack filled with a mixture of SUN and IBM servers placed in the test room without any spot cooling type device nearby or connected. The ceiling-mounted fan unit supplied the cooling by circulating the room air. No attempt was made to separate the hot or cold air. Because the Williams unit fan power was constant at approximately 3.6 kW, and the server power in the rack tested was 10.3 kW, the energy use cooling efficiency was low compared to the CRAH simulation. Therefore the CRAH simulation results are primarily used to represent typical data center cooling performance.

CRAH Cooling Design

Data from this design was estimated. See the description at the beginning of Chapter 2 and information listed in Appendix A.

Metric Comparison Calculation Information

The bullets below summarize important calculations assumptions.

- The recorded room recirculation feed pump power was not used in the comparison metrics. During the tests the delta pressure setting for the pump was changed and not always recorded. In some cases the actual room pump power fell close to the prediction using the actual flow rate and ASHRAE fixed delta pressure design guidelines. Therefore the presented results use Feed pump power calculated using the ASHRAE guidelines and tested water flow rate.

- The actual CDU power measured during the tests is not used in the comparison metrics. As mentioned in Chapter 2 the performance metrics are calculated assuming the CDU equipment manufacturer listed maximum performance is fully utilized. The CDU power is prorated using the value of server power divided by the maximum manufacturer listed cooling performance.

- Some small values for device power are listed for the Clustered Systems tests. These values arise from a combination of electrical power measurement methods and subtraction to calculate the cooling device power. If all the measurements were 100 percent correct, the device power for the Clustered Systems tests would be zero.

- Two tests—2 and 6—are used for comparison.
    - Test 2 – Chilled Water Temperature 45°F, Server Air Inlet Temperature 72°F

      Four designs were compared:
        - Clustered Systems Design (Liebert XDP, XDS and Clustered Systems design)
        - Other Refrigerant (Liebert XDP with XDH/XDV)
        - Base Case
        - CRAH Estimate
    - Test 6 – Chilled Water Temperature 60°F, Server Air Inlet Temperature 80°F

      Two designs were compared:
        - Clustered Systems Design (Liebert XDP, XDS and Clustered Systems design)
        - Other Refrigerant (Liebert XDP with XDH/XDV)

- The Base Case was not tested using the conditions for Test 6 because the chilled water temperature could not be adjusted. The CRAH case was not estimated for Test 6 because the performance information using 60°F chilled water was not readily available.

$SCOP_a$ Results

Chart 3-1 shows the results for $SCOP_a$ for the Clustered Systems rack tests 2 through 6. The $SCOP_a$ performance using the prorated feed power and device power is very high because the device power is zero and the prorated CDU power is low. Metric SCOPa does not contain the power needed to make the incremental amount of chilled water or the feed pumping power.

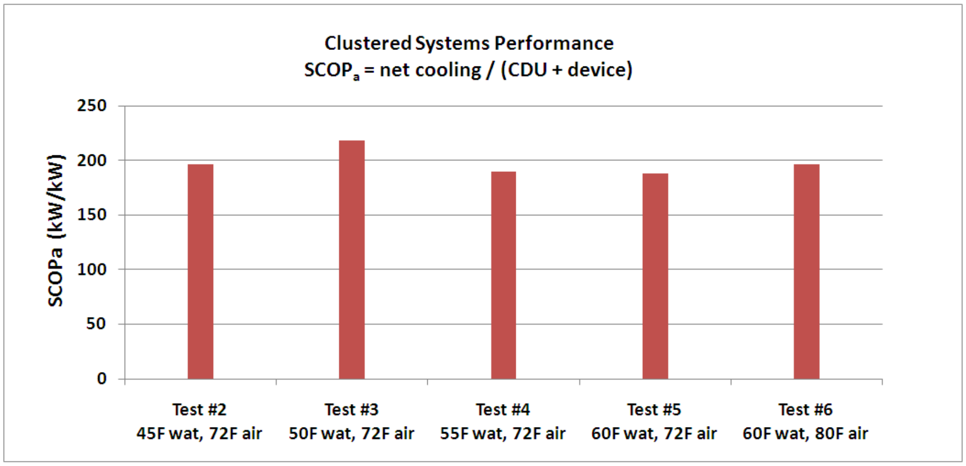

Figure 3-1: Clustered Systems $SCOP_a$ Performance

Source: Author

Figure 3.2 shows the performance compared to the Other Refrigerant design, Base Case, and CRAH estimate. The Clustered Systems rack again has good performance because the prorated CDU power is low (54 watts), and the device power is zero. The other devices have fans and or refrigerant-to-water CDU power.

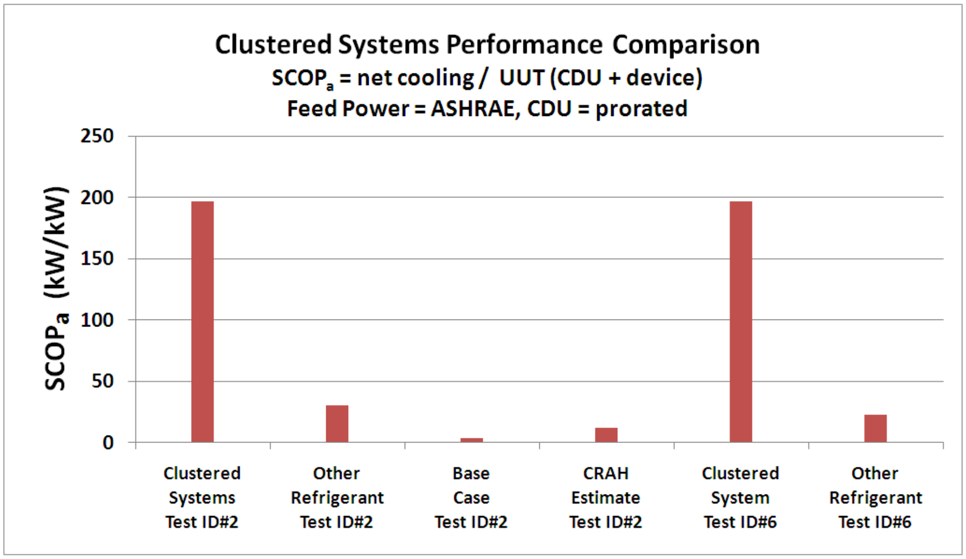

Figure 3-2: SCOP$_a$ Comparison Test 2 and 6

Source: Author

SCOP$_b$ Results

Figure 3-3 shows the results for metric SCOPb for Clustered Systems rack tests 2 and 6. Metric SCOPb is identical to SCOPa but with feed pumping power included. The results are similar to SCOPa but indicate lower performance as expected because an additional amount of power (feed pumping) is being accounted for. Note: the power needed to make the chilled water is not included.

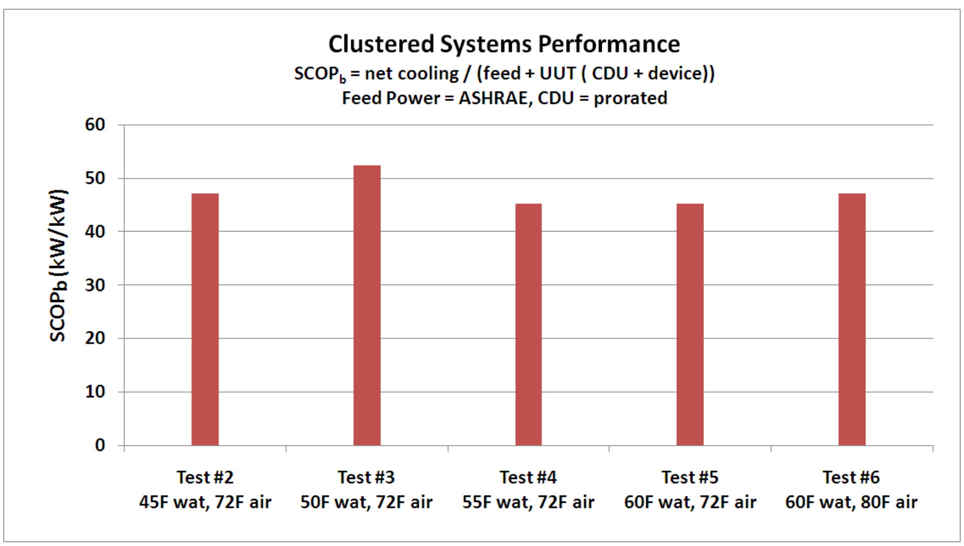

Figure 3-3: Clustered Systems SCOP$_b$ Performance

Source: Author

Figure 3.4 shows the performance compared to the Other Refrigerant design, Base Case, and CRAH estimate. The Clustered Systems rack again has good performance because the prorated CDU power is low (54 watts), and the device power is zero. The other designs have fans.

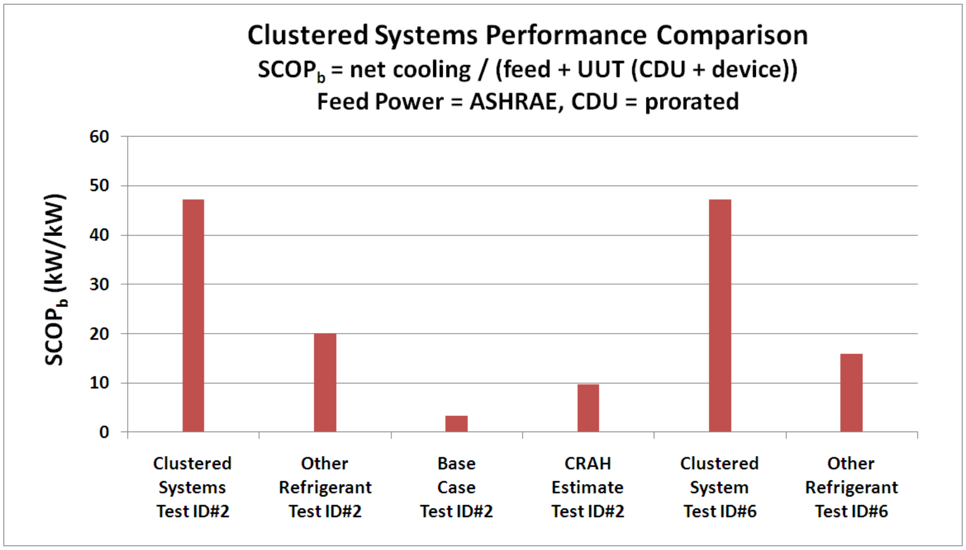

Figure 3-4: SCOPb Comparison Tests 2 and 6

Source: Author

SCOPc Results

Figure 3-5 shows the results for metric SCOPc for Clustered Systems rack tests 2 though 6. Metric SCOPc is identical to SCOPb but with the power added for the chilled water plant. There is a trend toward improved energy efficiency as the chilled water temperature increases. Again the data show the effect of the two different chilled water plant models, with more pronounced improvement as chilled water temperatures rise.

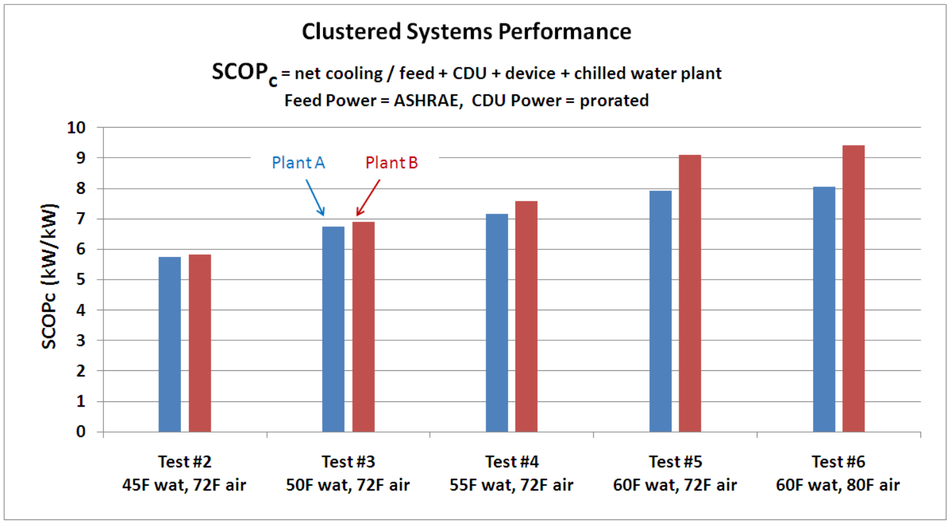

Figure 3-5: Clustered Systems SCOPc Performance

Source: Author

Figure 3-6 shows the results for metric SCOPc for Clustered Systems rack compared to other devices. The CRAH estimate and Clustered Systems performance is similar. The Clustered Systems rack has better efficiency than the Other Refrigerant design and the Base Case.

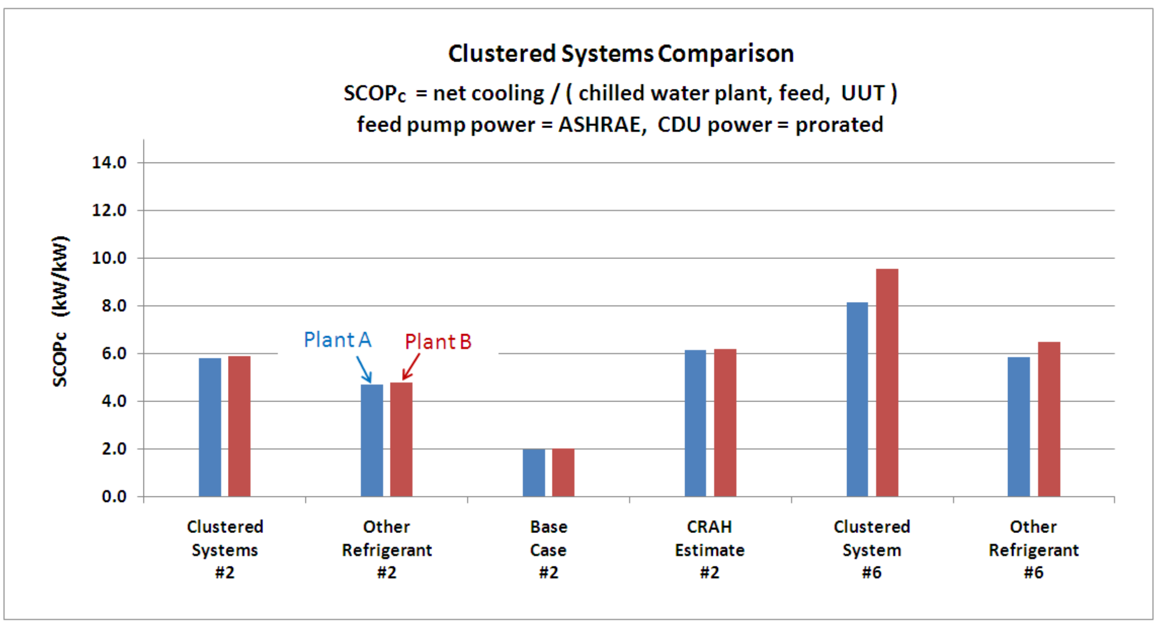

Figure 3-6: SCOPc Comparison Tests 2 and 6

Source: Author

COEE (Chill-Off 2 Energy Efficiency) Results

Figure 3-7 shows the results for metric COEEcs for tests 2 through 6. Metric COEE includes the incremental power needed for the chilled water plant and the power needed to run the IT equipment. As described in Chapter 2, metric COEE is similar in concept to the well-known industry PUE metric. Lower values are more energy efficient. The results show that the major advantage comes from using higher chilled water temperatures. Both plant types provide similar energy savings in the range from 45°F to 60°F; however, as the chilled water temperature rises, we see that plant type B has an increasing advantage as expected. The as-tested results have reduced energy efficiency due to a constant power pump in the CDU and use of the test room recirculation pump. Note: The Clustered System COEE performance is adjusted to COEEcs (Eq. 2-11a, 2-11b) for the purpose of fair comparison to tests using servers containing fans.

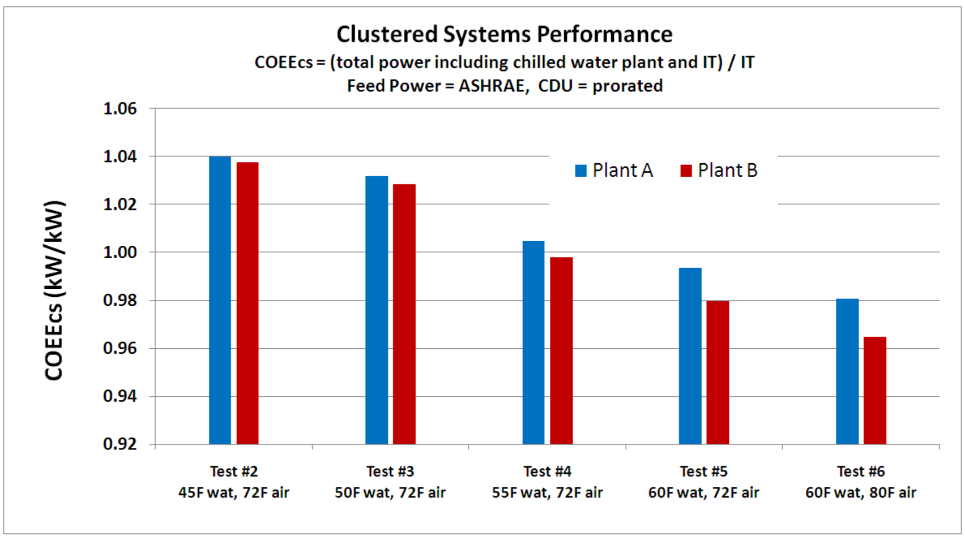

Figure 3-7: Clustered Systems COEE Performance

Source: Author

Figure 3-8 shows the results for metric COEEcs for Clustered Systems rack compared to other devices. Using the COEE metric, combined with the power reduction provided by removing the server fans, and comparing final total power using the measured COEE values, the Clustered Systems design is 13 percent more energy efficient compared to the CRAH and 14.5 percent more efficient compared to the Other Refrigerant design for Test 2 conditions. The total power used comparison for Test 6 showed a 16 percent improvement for the Clustered Systems design compared to the Other Refrigerant design. The energy savings comparing the chilled water plant models was less than 1 percent for a given test number and comparison.

The constant-speed, ceiling-mounted fan unit used 3.6 kW while cooling 10.5 kW of server heat, contributing to the low performance of the base case using the COEE metric. Therefore the Base Case results for COEE are not considered, and the results using the CRAH simulation are considered to be representative of a typical data center.

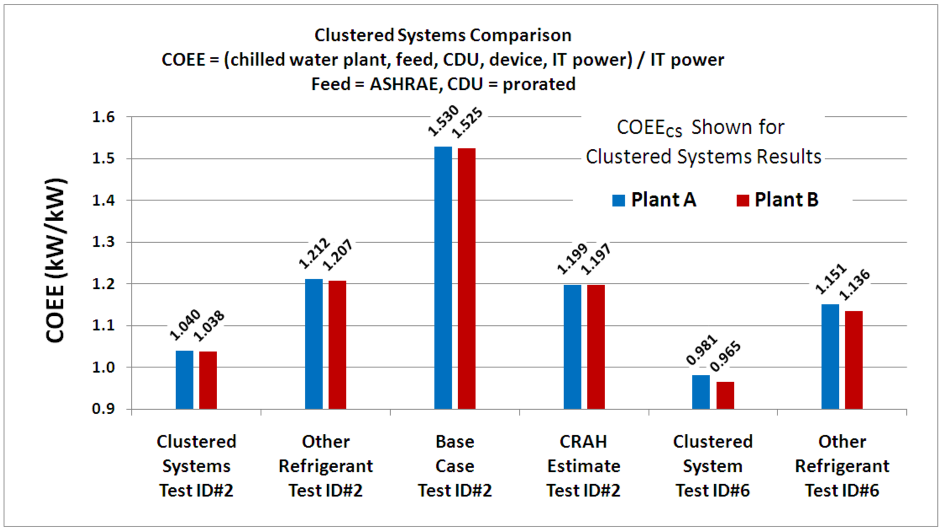

Figure 3-8: COEE Metric Comparison

Source: Author

Power per Server Results

Figure 3-9 shows the results for metric PPS for tests 2 through 6. Metric PPS shows only the power for each server. Server designs typically use more power at higher server inlet temperatures as the fans increase in speed to maintain server component temperatures. The Clustered Systems design shows a small increase in required server power, on the order of 2 percent. The cause of this increase was not determined.

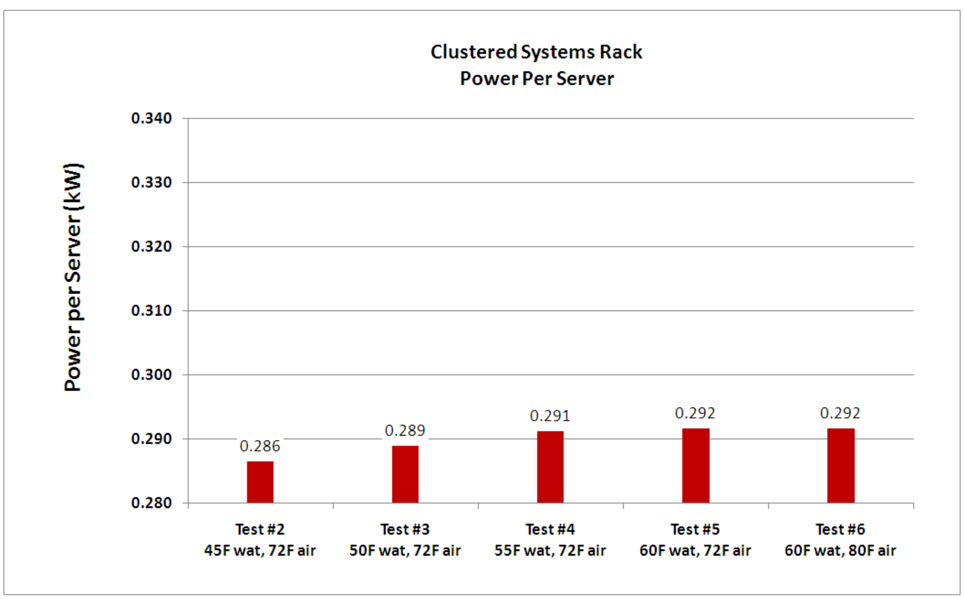

Figure 3-9: Clustered Systems Power per Server

Source: Author

Figure 3-10 shows the results for metric PPS, comparing to the Other Refrigerant design (Liebert XDH/XDV) .

The chart shows two significant advantages of the Clustered Systems design:

- **Lower Server Power -** The Clustered Systems server power is 11.5 percent lower for Test 2 and 12.8 percent lower for Test 6, compared to the Other Refrigerant.
- **Required Server Power is Steady with Increasing Server Air Inlet Temperatures -** The Clustered Systems server power increased 2.1 percent, going from Test 2 to Test 6, while in the Other Refrigerant test the increase was 3.7 percent.

Note: There was an unexpected small rise in power consumed for the Clustered Systems rack as the server air inlet temperature changed from Test 2 to Test 6. The cause of this increase was not determined. The data shows a very small increase (< 0.5 percent) from Test 4 to Test 6; therefore, the cause of the 2.1 percent rise may not be due to a temperature increase in the server air inlet.

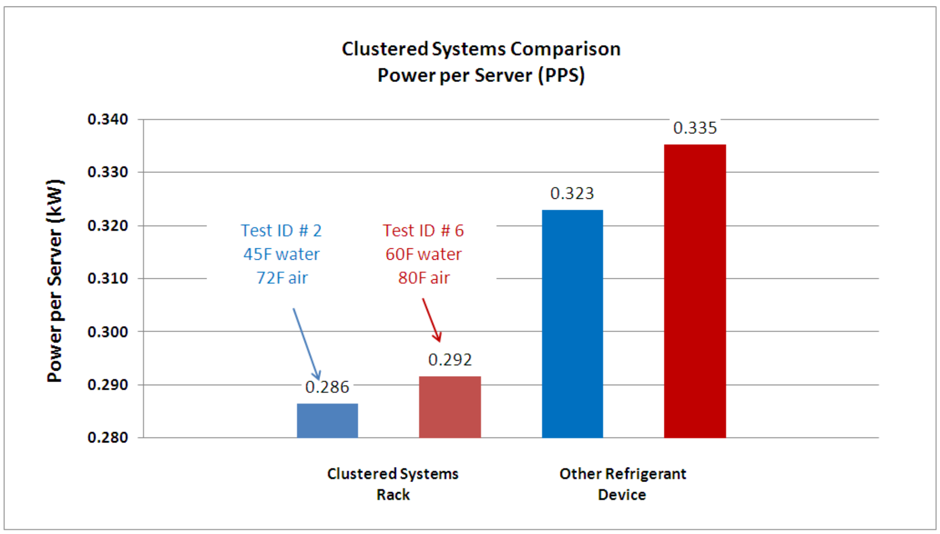

Figure 3-10: Clustered Systems Server Power Comparison

Source: Author

Total Power Used per Server

Figure 3-11 compares metric TPUS (total energy use per server) for tests 2 through 6 for Plant A and Plant B. Lower values are more energy efficient. The energy components used in the sum are: chilled water plant power (CWP), feed pump power, CDU power, cooling UUT device power, and IT power.

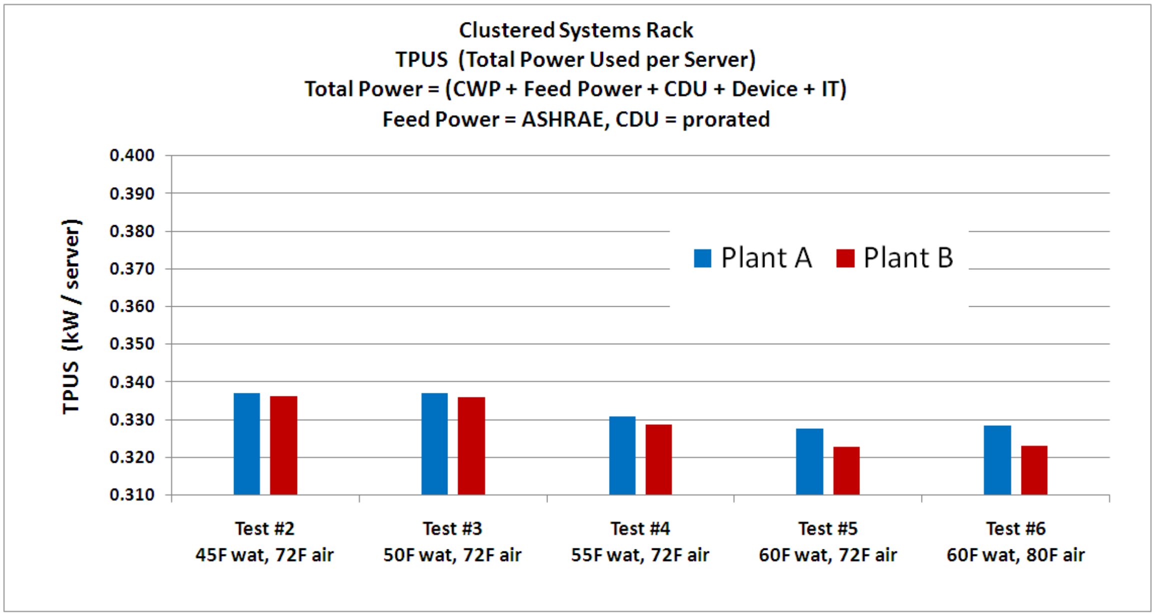

Figure 3-11: Clustered Systems TPUS Performance using Two Chilled Water Plant Models

Source: Author

Figure 3-12 compares the Clustered Systems Rack to the Other Refrigerant design tested using the TPUS metric. For each test condition the compute delivered per server was equal. The data show the significant energy savings provided by the Clustered Systems design compared to the other refrigerant design tested.

The energy saving compared to the Other Refrigerant design is approximately 14 percent for Test 2 and 16 percent for Test 6. The same improvement exists using both plant models. Results will be different if the server power as a function of compute delivered is not held constant.

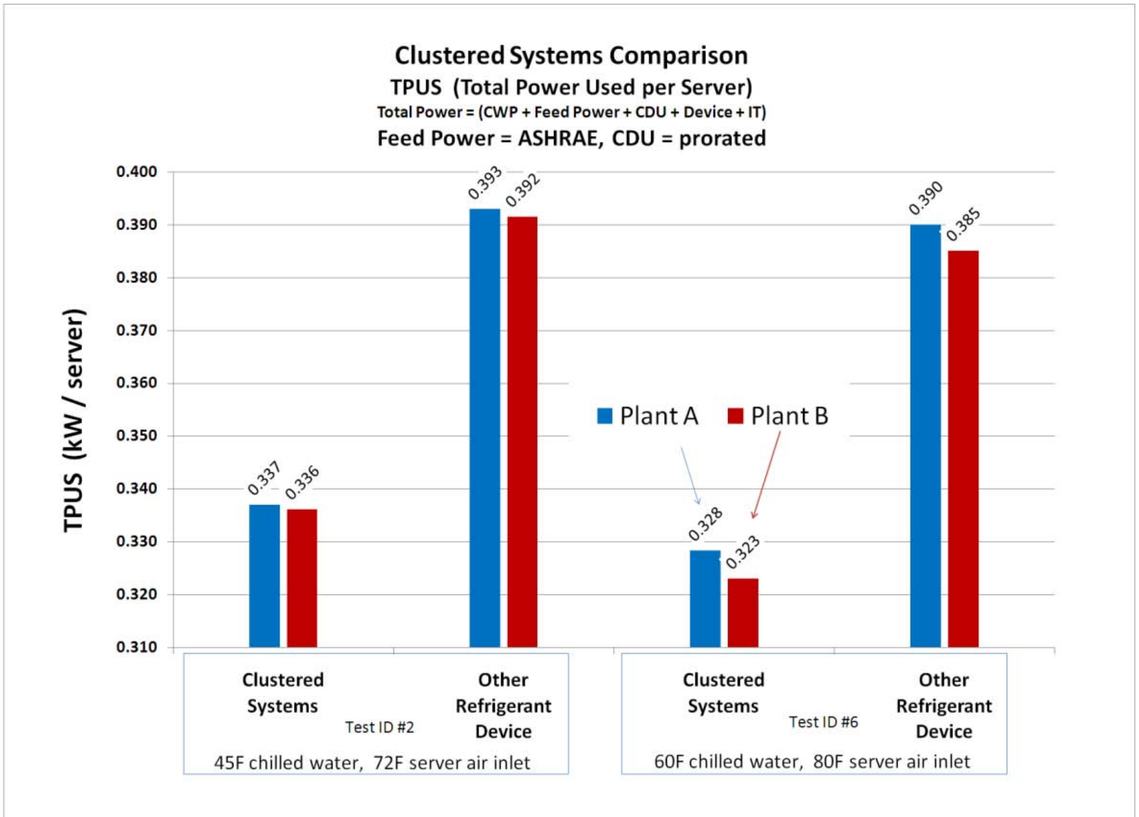

Figure 3-12: Clustered Systems Compared to Other Refrigerant (Liebert XDH/XDV) for Test 2 and Test 6, for Two Chilled Water Plant Models

Source: Author

Observations Using Very High Chilled Water Supply Temperature

The unexpected event that resulted in the test room being supplied with very high temperature chilled water provided some interesting observations (Figure 3-13). The chilled water temperature went from 44°F (6.6°C) to 78°F (25.5°C) during a period of 46 minutes. The server CPU temperatures averaged 99°F (37°C) for CPU 0 and 106°F (41°C) for CPU 1 when the chilled water supply was 44°F. When the chilled water temperature was stable at 78°F, the CPU temperatures were 129°F (54°C) and 136°F (58°C).

During the event there were electrical disruptions of unknown origin that reduced the server power. The first disruption shown in Figure 3-13 starts at minute 50 and the servers automatically recover by minute 90. The second disruption happened at minute 175 and the data analysis was concluded. A key observation is that the CPUs operated for a period of 85 minutes (minute 90 to 175), in an acceptable range (below 158°F [70°C]), while 78°F (25.5°C) chilled water temperature was supplied to the refrigerant-to-water CDU. The power balance during the high temperature period indicated that 25 percent of the server heat was not being removed by the refrigerant system. This result is somewhat expected, considering that the refrigerant temperature was much higher than the room temperature, likely causing a significant amount of heat lost to the room from the refrigerant distribution system. Additional testing to verify the CPU temperatures and find the best room temperature conditions using high chilled water temperatures is recommended.

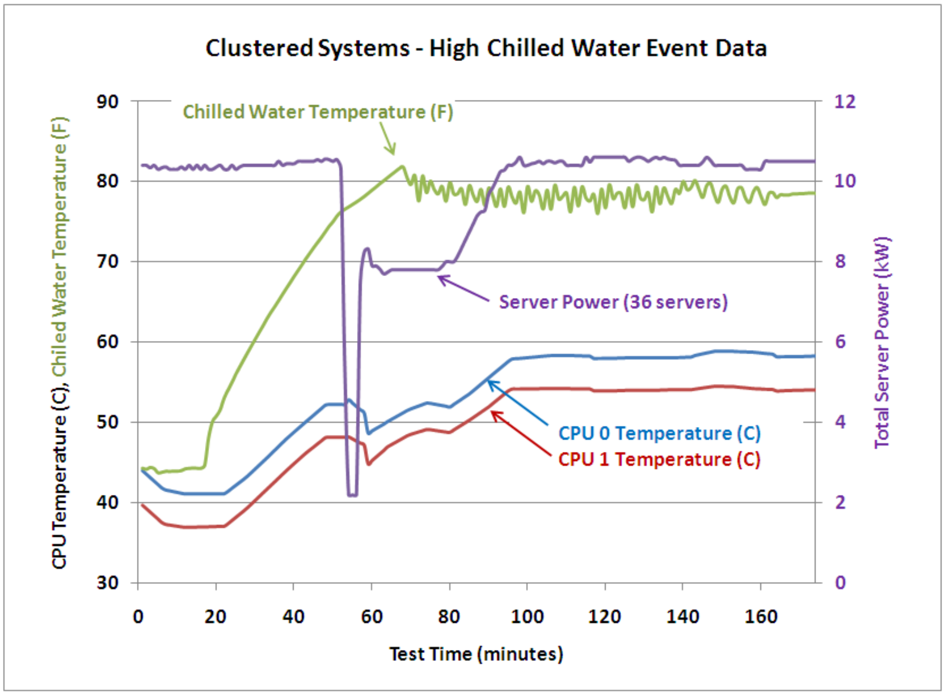

Figure 3-13: Clustered System Performance During the Facility Event, Resulting in Very High Chilled Water Being Supplied

Source: Author

CHAPTER 4: Conclusions and Recommendations

The energy efficiency of the Clustered Systems design was tested and compared to a common data center cooling design and to a currently available refrigerant-based cooling design.

Using the COEE energy efficiency metric, calculating the total power needed and allowing for the server power saved with this new design, the Clustered Systems design is 14 percent more energy efficient than the Liebert XDH/XDV test and CRAH simulation when evaluated at the Test 2 conditions (72°F [22.2°C] server air inlet, and 45°F [7.2°C] chilled water supply temperature). The energy efficiency comparison of the Clustered Systems design improved slightly to 16 percent when evaluated at the Test 6 conditions (80°F [26.6°C] server air inlet, and 60°F [15.5°C] chilled water supply temperature). The overall energy efficiency did not improve significantly, less than 1 percent, comparing the results of the A and B chilled water plant models for either Test 2 or Test 6 conditions.

The results using the TPUS metric showed the same results because when the COEE metric is used to calculate the total power needed, and the server power saved by the Clustered Systems design is accounted for, the results should be and were shown to be equal.

The observations during the use of 78°F (25.5°C) chilled water temperature indicate that the Clustered Systems design potentially can be operated with very low-cost cooling water, providing additional energy savings compared to the compared test results.

Measurements of CPU temperatures indicate that a small difference in heat transfer resistance exists between CPU 0 and CPU 1. The Clustered Systems design team is aware of this issue and is working on an improved design.

This new rack-level cooling concept will provide California with significant energy savings if the supporting mechanical design is suitable for the commercial market. Additional testing of an updated mechanical design evaluating the overall suitability for the industrial market is recommended as part of a plan to interest a major electronic equipment manufacturer to incorporate this innovative design into a standard product.

To bring this design to market, server manufacturers will need to adopt the technology and begin manufacturing servers that incorporate the Clustered Systems-type heat removal system.

CHAPTER 5: References

ASHRAE. 2007. ASHRAE 90.1-2007. ASHRAE Standard for Buildings Except Low-Rise Residential Buildings.

ASHRAE. 2007. ASHRAE 127-2007. Method of Testing for Rating Computer and Data Processing Room Unitary Air Conditioners (ANSI Approved).

CHAPTER 6: Glossary

ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers
Btu	British thermal unit
CDU	cooling distribution unit
CDU power	electrical power consumed by the CDU
chilled water plant power	the amount of additional electrical power needed to produce the chilled water
COEE	Chill-Off 2 Energy Efficiency
COEEcs	COEE for Clustered Systems
cooling device power	the UUT power minus the CDU power, if there is no CDU as part of the UUT, the device power equals the UUT power
CRAH	computer room air handler
CWP	Chilled Water Plant Power
device power	See cooling device power.
Feed or Feed Power	electrical power consumed by pump(s) for the chilled water distribution
GB	gigabyte
GSC	Gross Sensible Capacity
hydraulic power	Hydraulic power is the mechanical energy loss (kW) calculated using pressure and flow rate across heat exchanger The equation used is flow rate (gallons per minute) multiplied by the pressure difference (pounds per square inch differential) and a constant 0.000435 to obtain kilowatts.
IT	information technology
kW	kilowatt
NC	net cooling
PDU	power distribution units

PPS	power per server
PUE	power usage effectiveness
SCOP	sensible coefficient of performance
SIAT	server inlet air temperatures
SLAT	server leaving air temperatures
TPUS	Total Power Used per Server
UPS	uninterruptible power system
UUT	Unit Under Test. This refers to all the equipment being tested, for energy efficiency purposes, that is considered one device. For example the CDU is usually considered as part of the equipment tested.
VFD	variable frequency drive

APPENDIX A:
CRAH Performance Information

Computer Room Air Handler (CRAH) Estimate Information

Information Obtained from Liebert North America – 4/28/2010

Model Number:	CW Model CW114D; Chilled Water
Unit Power Supply:	460/3.60
Internal Filter Class:	Merv 7 Std. -4
Unit Airflow:	16500 cfm
ESP:	0.2 "WG

Cooling Fans

Quantity of Fans:	3
Quantity of Motors:	1
Type:	Centrifugal -FC
Full Load Amps:	21
Locked Rotor Amps:	116
Total Motor HP:	15

Performance – Mechanical Cooling

Enter Dry Bulb °F:	75
Enter Wet Bulb °F:	61.1
Enter Rel Humid %:	45
Air Vol ACFM:	16500
Face Vel FPM:	455
Enter Water Temp °F:	45
Enter Fluid Rise °F:	12
Fluid Flow GPM:	76.7
Total Cool Cap kW:	125.1
Sens Cool Cap kW:	111.9
Total Unit PD ft H_2O:	23.8
Leave Dry Bulb °F:	53.3
Leave Wet Bulb °F:	51.6
Motor kW:	10.33
Motor BHP:	11.77
Motor HP:	15.00

Capacity shown has been reduced by fan motor heat (net)
Test method as defined by ASHRAE 127-2007
Capacity Tolerance is 5%

APPENDIX B:
Chilled Water Plants Descriptions

Chilled Water Plant Model Information
3/17/2010 – Taylor Engineering, Alameda CA 94501

To the extent possible ASHRAE 90.1 Chapter 11 (ECB) Rules are followed.

Note: Chilled water plant models that included chilled water distribution pumping were made but used in this report.

Chillers

Type:	water cooled chillers meeting 90.1-2007 Addendum M Path B (chillers with VSD) minimum efficiencies: COP of 0.6 and IPLV of 0.4 at ARI 550/590 rating conditions
Quantity:	three chillers in the plant, n+1 design (one redundant) each sized for 330 tons to serve an actual 600 ton load (660 ton design load, 600 ton actual load 10% over sizing)
Evaporator Flow:	790 gpm/chiller evaporator
Chiller Condenser Flow:	920 gpm/chiller condenser
Performance:	0.58 kW per ton at design conditions of 44°F chws and 80°F cws.

Cooling Tower

Efficiency:	38.2 gpm/hp at 95°F CWR,85°F CWS, 75°F Twb (90.1 minimum efficiency)
Quantity:	3 cells one redundant each sized for 330 tons of chiller (10% over sizing) Selected for design flow of 925 gpm per cell (10°F DT)

Pumps

Rules:	from 90.1-2008 Chapter 11 rules
Chilled water pump power:	0.019 kW/gpm
Condenser water pump power:	0.022 kW/gpm

Water-Side Economizer

Rules:	criteria in 90.1-2007 Addendum BU
Size:	100% of the design load (660 tons) at 35°F Twb.
CHW Flow:	1580 gpm
CHWR:	62°F
CHWS:	52°F
CW Flow:	1580 gpm
CWS:	48°F
CWR:	58°F
WSE Heat Exchanger:	4°F approach on the water side economizer heat exchanger

Climate
San Jose Airport TMY 3 File

APPENDIX C:
Additional Test Data for Table 3-1

Test ID#	UUT ID	Target Server Inlet (F)	Target Chilled Water (F)	Aux. Cooler Electrical In (kW)	Infrastructure Test Room Electrical In Power (kW)	Test Room Pump Power (kW)	[part of UUT] Refrigerant to Water CDU Power (kW)	[part of UUT] Cooling Device Power (kW)	Computer (IT) Power (kW)	UUT Bulk Cooling (kW)	Aux. Cooler Bulk Cooling (kW)
2	CS	72	45	3.6	1.64	0.822	0.821	0.001	10.3	10.67	1.565
3	CS	72	50	3.6	1.64	0.821	0.821	0.001	10.4	11.94	0.749
4	CS	72	55	3.6	1.64	0.819	0.821	0.001	10.5	10.43	2.534
5	CS	72	60	3.6	1.64	0.818	0.821	0.001	10.5	10.45	3.075
6	CS	80	60	3.5	1.64	0.821	0.821	0.001	10.5	10.87	-0.958
8*	CS	NA	NA	3.6	1.80	0.973	0.821	0.001	10.5	8.77	5.356
2	Ref. #1	72	45	0.0	3.81	0.090	0.821	2.896	103.3	107.85	0.512
6	Ref. #1	80	60	3.3	6.04	0.727	1.642	3.673	107.3	100.45	-0.680
2	Base	72	45	0.0	0.00	0.000	0.000	3.680	12.5	17.18	0.035
2	CRAH Est.	72	45	0.0	0.00	0.000	0.000	10.330	100.0	134.74	0.000

APPENDIX D:
Test Quality Data for Table 3-1

Test ID#	UUT ID	Target Server Inlet (F)	Actual Server Inlet (F)	Target Chilled Water (F)	Actual Chilled Water (F)	Test Room Power Loss (kW)	Percent of Power Lost In Test Room (%)	Aux. Cooler Net Cooling (kW)	(Bulk UUT Cooling - UUT power) / Server Power
2	CS	72	72.4	45	45.0	3.3	26.7%	-2.03	96%
3	CS	72	73.0	50	50.9	2.9	24.6%	-2.81	107%
4	CS	72	72.9	55	54.6	2.7	23.1%	-1.05	92%
5	CS	72	73.2	60	59.5	2.2	20.0%	-0.51	92%
6	CS	80	78.6	60	60.4	5.7	41.2%	-4.47	96%
8*	CS	NA	69.6	NA	78.4	1.8	16.9%	1.72	76%
2	Ref. #1	72	72.1	45	43.5	-1.2	-0.2%	0.51	101%
6	Ref. #1	80	78.5	60	59.5	16.8	17.5%	-3.97	89%
2	Base	72	79.5	45	44.0	-4.7	3.1%	NA	108%
2	CRAH Est.	72	72.0	45	45.0	NA	NA	NA	NA

APPENDIX E:
Raw Test Data for Table 3-1

UUT ID	Test ID#	IT 480 Power (kW)	IT 208 Power (kW)	Infra 480 Power (kW)	Infra 208 Power (kW)	Red Pump Power (w)	Aux. Cooler Power (kW)	Main Water Return (F)	Main Water Supply (F)	Main Water Flow (gpm)	Aux Water Return (kW)	Aux Water Supply (kW)	Aux Water Flow (kW)	Air Temp. Check (F)
CS	2	10.85	10.31	2.267	1.644	782	3.593	50.5	45.0	13.3	44.9	44.1	13.9	72.4
CS	3	10.99	10.40	2.271	1.643	785	3.555	56.1	50.9	15.6	44.3	43.9	14.1	73.0
CS	4	11.00	10.48	2.269	1.641	784	3.581	59.6	54.6	14.2	45.1	43.9	14.2	72.9
CS	5	11.03	10.50	2.293	1.640	788	3.582	64.3	59.5	15.0	45.5	44.1	14.4	73.2
CS	6	11.08	10.50	2.280	1.643	781	3.507	65.6	60.4	14.4	43.8	44.3	14.1	78.6
CS	8*	10.97	10.45	2.403	1.795	789	3.635	81.7	78.4	17.7	46.3	43.8	14.7	69.6
Ref. #1	2	104.50	103.34	3.690	3.807	90	Off	60.8	43.5	42.6	48.3	47.8	7.4	72.1
Ref. #1	6	108.56	107.29	5.439	6.042	727	3.291	71.3	59.5	58.0	45.8	46.2	12.4	78.5
Base	2	13.10	12.53	0.981	0.000	0	3.680	63.9	60.2	0.1	51.9	44.0	14.9	79.5
CRAH	2	NA	100.00	NA	0.000	0	10.330	NA	NA	NA	57.0	45.0	76.7	72.0

APPENDIX F:
Constants/Equations

Calculation of Hydraulic Power

Hydraulic Power(kW) = flow rate (gpm) * delta pressure (psid) * 0.000435

ASHRAE Combined Pump Efficiency

65%

ASHRAE Defined Chilled Water Supply Delta Pressure

75 feet of water (32.4 psid)

Calculation of Pump Motor Electrical Power given Hydraulic Power

Pump Motor Electrical Power = Hydraulic Power / Combined Pump Efficiency

Liters per Gallon

3.7854

Calculation of Water Heating or Cooling Thermal Power Given Flow Rate and Delta Temperature

Thermal Power (kW) = flow rate (gpm) * delta temperature (°F) * 0.1464

Thermal Power (kW) per Ton of Cooling/Heating of Water

3.516

Chilled Water Plant Model A Coefficients

Third Order Equation: kW/ton = f(chilled water supply temperature °F)

constant for X^3 term	-0.00000307
constant for X^2 term	0.0007278
constant for X^1 term	-0.06216206
X^0 term	2.12613527

Chilled Water Plant Model B Coefficients

Third Order Equation: kW/ton = f(chilled water supply temperature °F)

constant for X^3 term	0.00000628
constant for X^2 term	-0.00105494
constant for X^1 term	0.04389858
X^0 term	0.10162531

Refrigerant-to-water CDU (Liebert XDP) Information

Maximum Supported Server Heat	160 kW
Water Flow at Maximum Server Heat	120 gpm
Refrigerant Pump Power	0.821 kW

Water-to-water CDU (Liebert XDP-W) Information

Maximum Supported Server Heat	100 kW
Water Flow at Maximum Server Heat	75 gpm
Water Pump Power	1.25 kW