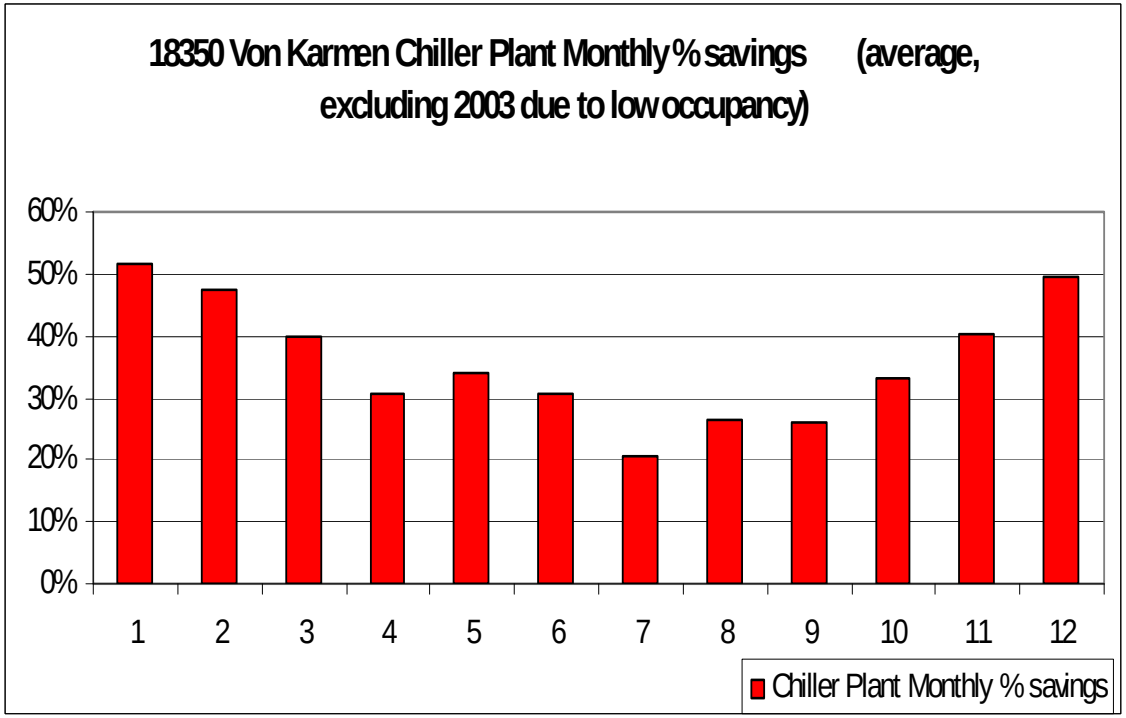
An Overview of Various Cooling Energy Storage Technologies

Net Zero Energy Technologies for Community Systems ERDC-CERL



February 3, 2009
Colorado Springs, CO

Presented by
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Thermal Energy Storage (TES) Overview

- **TES is the process of generating and storing cooling energy in a tank at night when power plants and chiller plants are most efficient, and wind power may be peaking, then shutting down the largest part of the air conditioning systems in the heat of the day and using the cooling energy stored in the TES system to cool the building or process load, reducing site peak electrical demands.**
- **TES has been around for over 100 years.**
- **My Grandmother used TES (lake ice) on the farm in Canada for "Annual TES"**

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TES Overview, continued

- **Properly designed TES systems are the equivalent of a 130% to 150% efficient storage battery for your air conditioning system.**
- When you combine peak day site, power plant and T&D energy efficiency gains, **TES can reduce power plant energy consumption requirements by over 50%** for each kWh shifted to the night, **substantially reducing greenhouse gas emissions.**
- **If wind power is available to power the TES system, the gains can be even greater.**

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Our TES Design Goals

- If TES is cost effective, here are our goals:
  - **Make it operator friendly with proper controls, and a hydraulic design that actually works!**
  - **Ensure occupant comfort and/or process needs are met at all times.**
  - **Shift the maximum cooling load possible.**
  - **Decrease chiller plant energy consumption by at least 5% - shoot for 25% or more.**

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Energy Storage Air Conditioning Definition

- **Many old TES system designs used more energy than their base case comparison systems – the main benefit was shifting the electrical load to the night to:**
  - **Reduce peak electrical demands and kWh consumption**
  - **improve generation system efficiency**
  - **improve T&D system efficiency,**
  - **reduce overall ratepayer costs**.
- **The definition we are proposing for ESAC is that the ESAC system has to be energy neutral or better at the site, when compared to the existing or proposed base case system that it would replace.**

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Energy Storage Air Conditioning

- There is a relatively new subset of Energy Storage, focused on being “energy neutral or better” at the site.
- **We are calling this subset “Energy Storage Air Conditioning” (ESAC) for lack of a better term.**
- Properly designed, installed, controlled and commissioned chilled water based TES systems can exceed a site energy efficiency improvement of 30%.
- **Ice Storage Air Conditioning (ISAC) typically ranges between energy neutral and greater than a 10% site improvement.**
- **Other ice storage systems can be made to qualify for the ESAC designation, but as with chilled water storage, the system design and equipment selections are extremely important.**

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Why is There a Need for ESAC

- Compared to “business as usual”, properly designed and implemented Energy Storage Air Conditioning systems accomplish these tasks:
  - **Reduces pollution and greenhouse gasses**
    - **(up tp a 2:1 day to night ratio)**
  - **Reduces ratepayer costs – shifts lifecycle cost burden onto Energy Storage users, away from ratepayers**
  - Reduces non-renewable energy consumption at the generation source – base load vs. peaker plants
  - Energy neutral or better at the site
  - Reduces T&D congestion & costs
  - Reduces T&D line loss inefficiencies – 3% or more
  - Reduces environmental damage
  - Allows effective use of wind energy resources that peak at night
  - Can augment Demand Response Programs
  - Helps utilities meet AB32, SB1368 and AB2021

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The Effects of Energy Storage on CO2 Production

- California Example, using data for the three major Investor Owned Utilities
- Shifting summertime energy use to the night from the afternoon can reduce CO2 production by nearly 50%.
- **“Energy Neutral Or Better” Energy Storage Air Conditioning systems can improve on this figure.**

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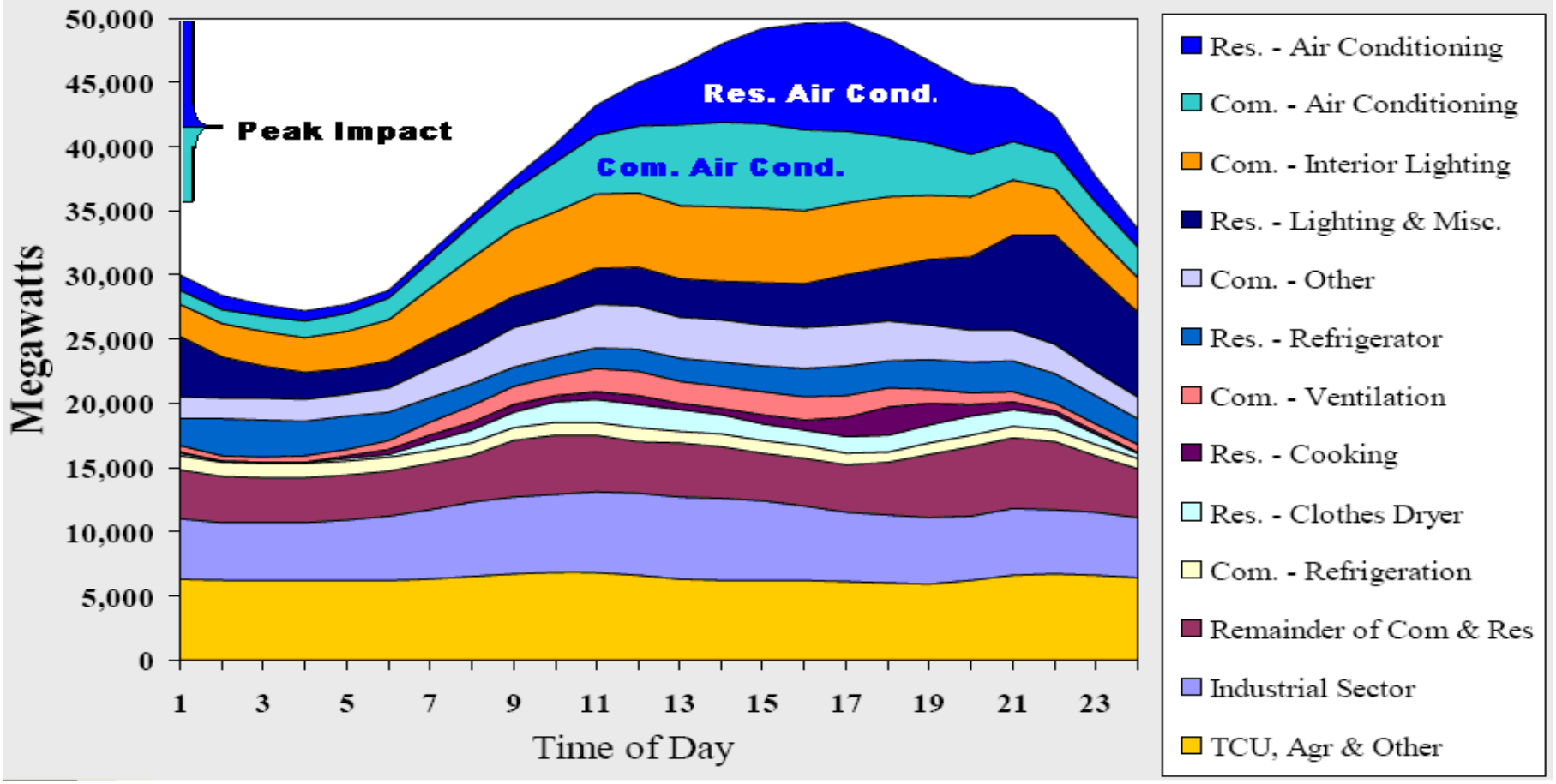
Residential and Commercial HVAC loads are major contributors to the peak demand problem, Properly implemented ESAC systems can reduce this impact.



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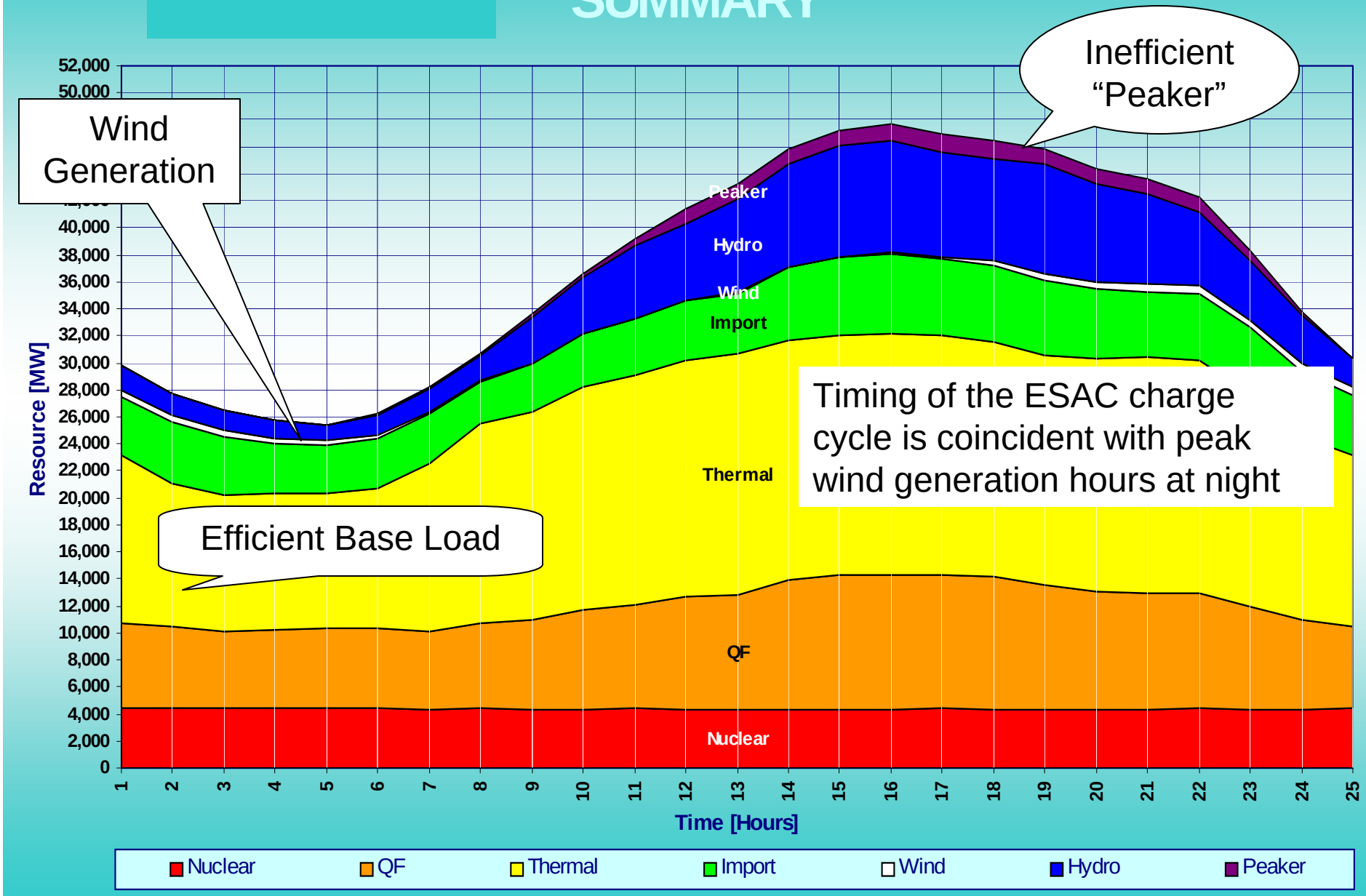
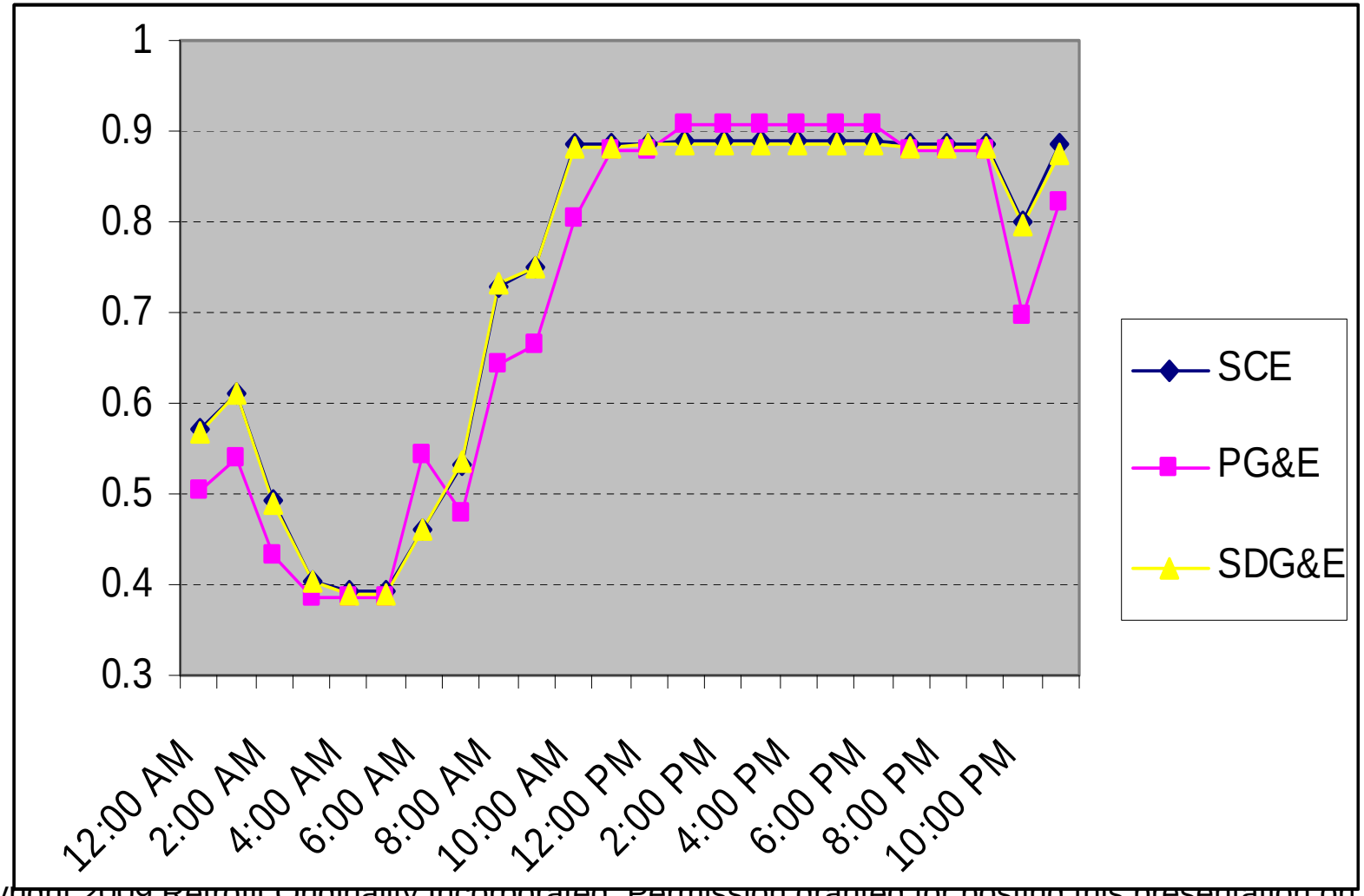
2001 ESTIMATED PEAK DAY RESOURCE SUMMARY



Tons of CO2 per MWH, by hour and utility, summer day



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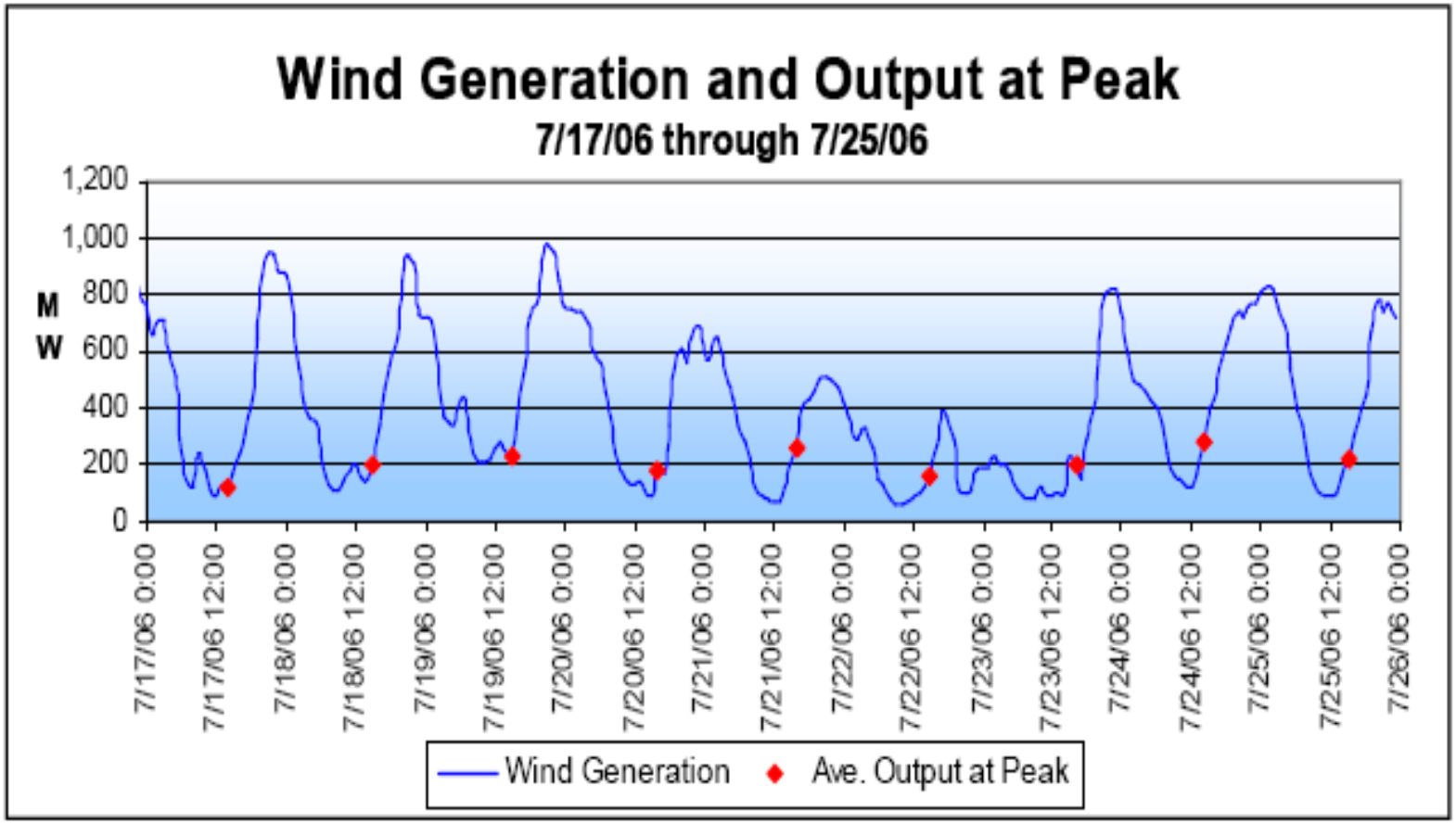

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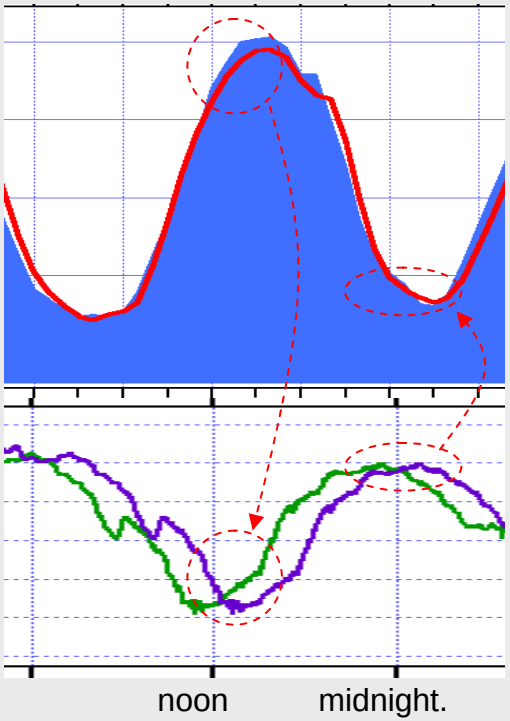
Efficient Storage A/C Delivers on the Promise of Wind

Typical Hot Summer Day

System Demand

Wind Power Generation



- ~ 5% of wind generation available during peak demand period

- Majority of wind generation occurs at periods of lowest demand

Source: 2006 California Energy Commission report

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ESAC Summary

- **Energy Storage Air Conditioning (ESAC) is energy neutral or better at the site.**
- **ESAC promotes source energy savings, reducing non-renewable energy use.**
- **ESAC reduces energy consumption during high pollution production hours and moves it to low pollution production hours.**
- **ESAC can help to reduce daily summer CO2 production by up to 50%.**
- **ESAC reduces ratepayer costs by shifting lifecycle costs to ESAC owners.**

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Thermal Energy Storage System Types

- Air Conditioning Load Related (TES)
  - **Chilled Water Storage**
    - **Normal**
    - **Large Temperature Differential (greater than 25 deg TD)**
    - **Low Temperature Fluid (30°F to 36°F water)**
  - **Ice Storage Air Conditioning**
  - **Ice on Coil**
  - **Ice Harvesters**
  - **Encapsulated Ice**
  - **Energy Storage Air Conditioning**

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Ice Storage

- **Ice storage is typically used on smaller systems, or where space is very constrained, although some of the largest TES systems in the country are based on ice storage.**
- Ice storage requires chillers and other equipment specifically designed for low temperature duty.
- Ice storage relies mainly on the latent heat of ice for the storage capacity.
- **Ice storage systems can be designed to reduce site energy consumption.**

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Ice Storage Air Conditioning ISAC

- **Ice Storage Air Conditioning uses small volumes of liquid refrigerant as the heat transfer fluid**
- Current designs replace the on-peak cooling for 5 to 7.5 ton DX systems
- Energy neutral or better at the site
- In California's Title-24 software

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Chilled Water Storage

- Chilled water storage is typically used on larger systems, or where there is a small footprint available for the TES tank, as the TES tank can be a tall vertical structure.
- **Chilled water storage can use "standard" air conditioning duty chillers.**
- Chilled water storage systems can be designed to substantially reduce site energy consumption.
- Chilled water storage relies on the temperature differential between the chilled water supply and return from the loads for its capacity – the greater the temperature differential, the greater the storage capacity for any given tank size.

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Chilled Water Storage Large Temperature Differential (LTD) Designs

- Typical CHW storage designs use a 15°F temperature differential (TD), but many systems we have designed have been installed using **TD's of 30°F or more**. The higher the TD, the smaller the tank volume can be, so spending extra $$$ on cooling coils can save substantial tank construction costs.
- Large TD's can reduce overall project costs and reduce infrastructure costs.
  - **Your piping will now carry twice as many BTU's as before.**

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Chilled Water Storage Large Temperature Differential (LTD) Designs

- Using a 30°F temperature differential instead of a 15°F TD can help cut pumping and chiller plant energy dramatically – series chillers, 50% less CHW flow…
- Airside pressure drops are the same as, or lower than a typical design strategy.

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Chilled Water - Low Temperature Fluid

- **Freezepoint depressants, such as SoCool™ can be added to chilled water systems to increase their storage capacity.**
- **The typical LTF project will drop the chilled water tank temperature down to 34F, which can increase the BTU storage capability of the TES tank by 30% to 50%, depending upon the original design conditions.**
- The chillers may have to be reselected to be able to drop the CHWS temperature down to 34F, due to the increased lift conditions.

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TES Overview, continued

- Ice based and chilled water based systems can use either buried or above grade storage tanks to contain the ice or chilled water.
- **There are many different ice storage system manufacturers.**
  - Some have had better success than others.
- **There are many different chilled water storage system designers.**
  - Some have had better success than others.

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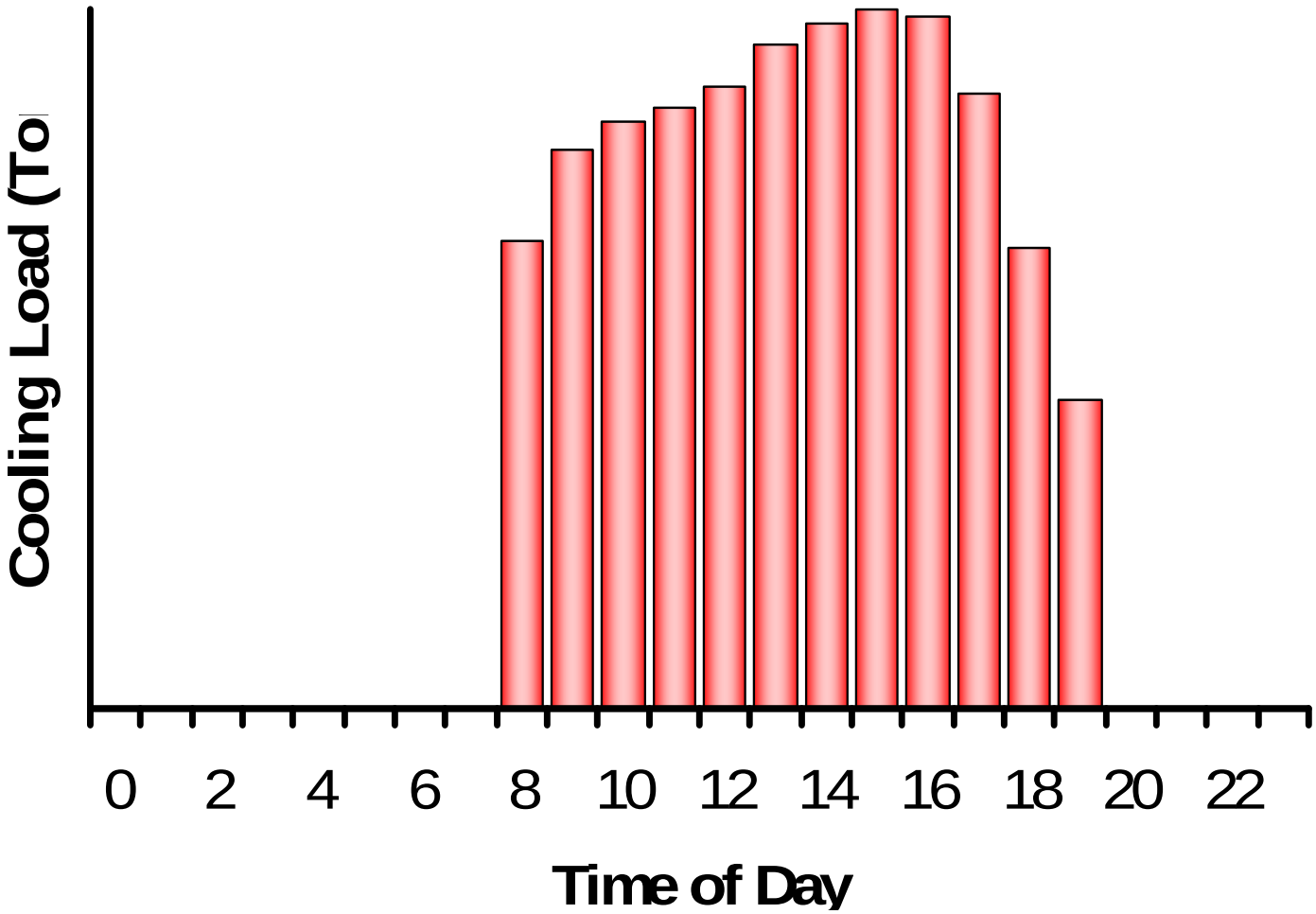
Air Conditioning Load Profile



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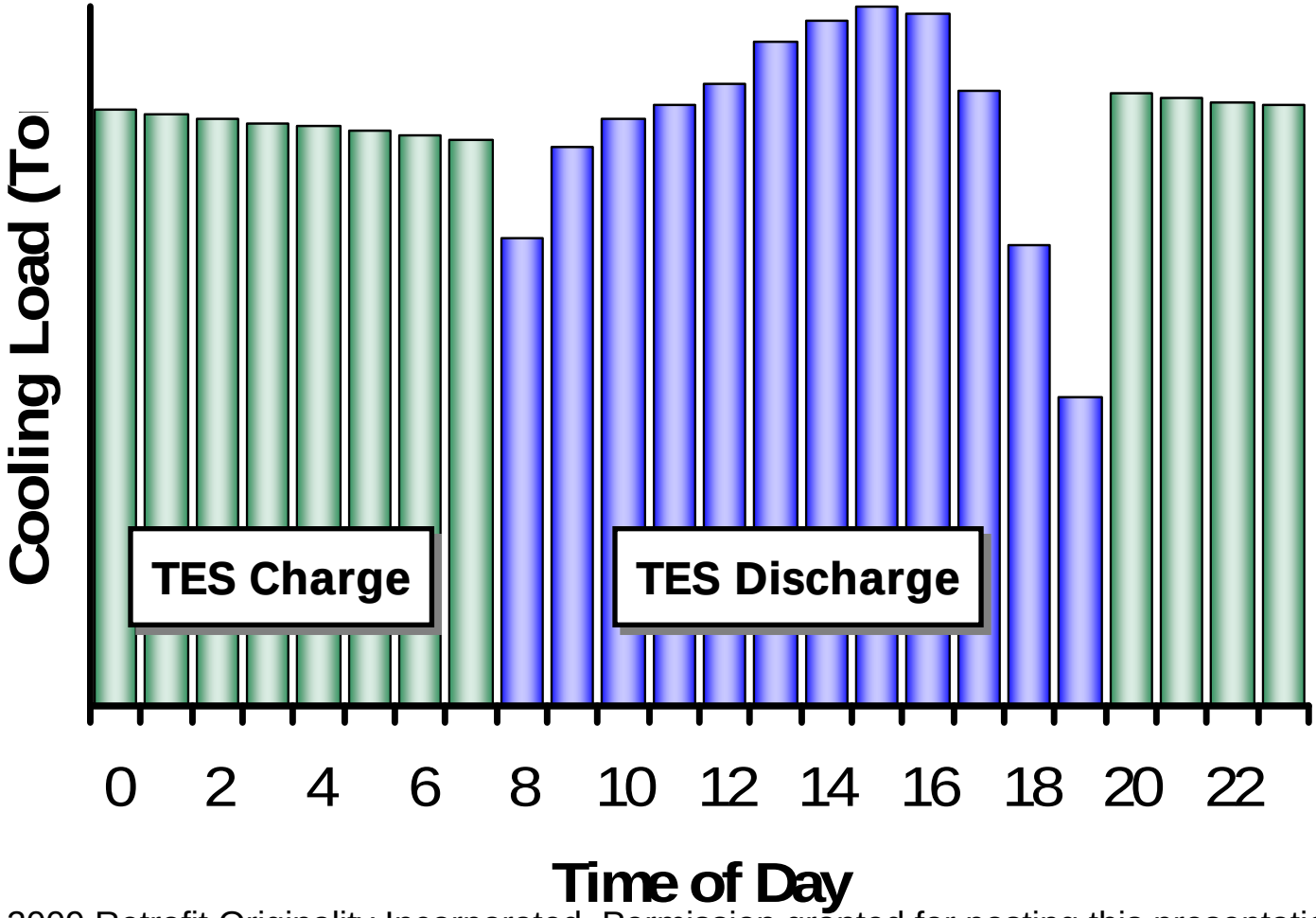
Full Storage
Air Conditioning Application



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Thermal Storage System Design Full Storage

Advantages

- Best suited for short, peak demand periods and/or high, peak loads
- **Shifts largest electrical demand that provides the lowest operating cost**
- Provides system standby capability and operating flexibility

Disadvantages

- Largest storage volume required
- Larger chiller required
- Most expensive thermal storage design

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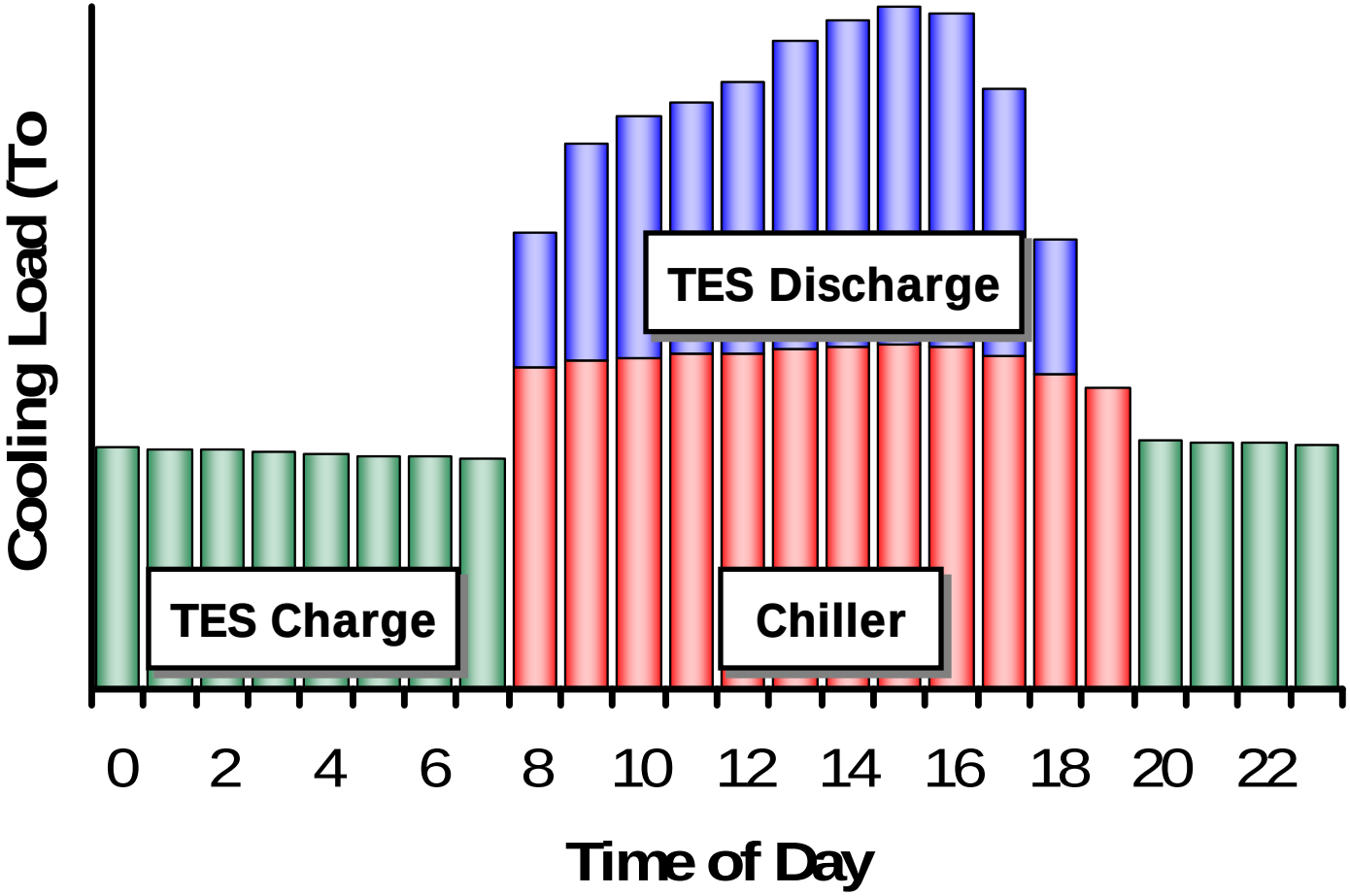
Partial Storage
Air Conditioning Application



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Thermal Storage System Design Partial Storage

Advantages

- Best suited for long cooling periods
- Lower first cost due to reduced storage volume and potentially smaller chiller
- Provides system operating flexibility

Disadvantages

- Less standby capability
- Less electrical demand shifted to off-peak
- Easy to “fudge” on performance

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Bigger is Better!

- I have **never** heard the Owner of a properly designed TES system state:
  - "I wish we had a smaller TES system"
- We are proponents of full storage TES systems, but we realize that there are many instances that partial storage systems can make economic sense.
- There are many ancillary benefits of TES that are tough to quantify financially.

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Acknowledgements

- We have received slides and other information from various manufacturers during the course of assembling this presentation.
- We have not provided acknowledgements on each slide, but we have tried to provide acknowledgements on each slide that depicts a certain type of system.

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Ice Storage Systems

- The next few slides depict several different ice storage system types.

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Ice Thermal Storage
Ice-on-Coil Technology

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Ice Storage Systems
Modular Units or Custom Coils

Photo Courtesy
Baltimore Aircoil
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Thermal Storage Tank

Photo Courtesy CALMAC

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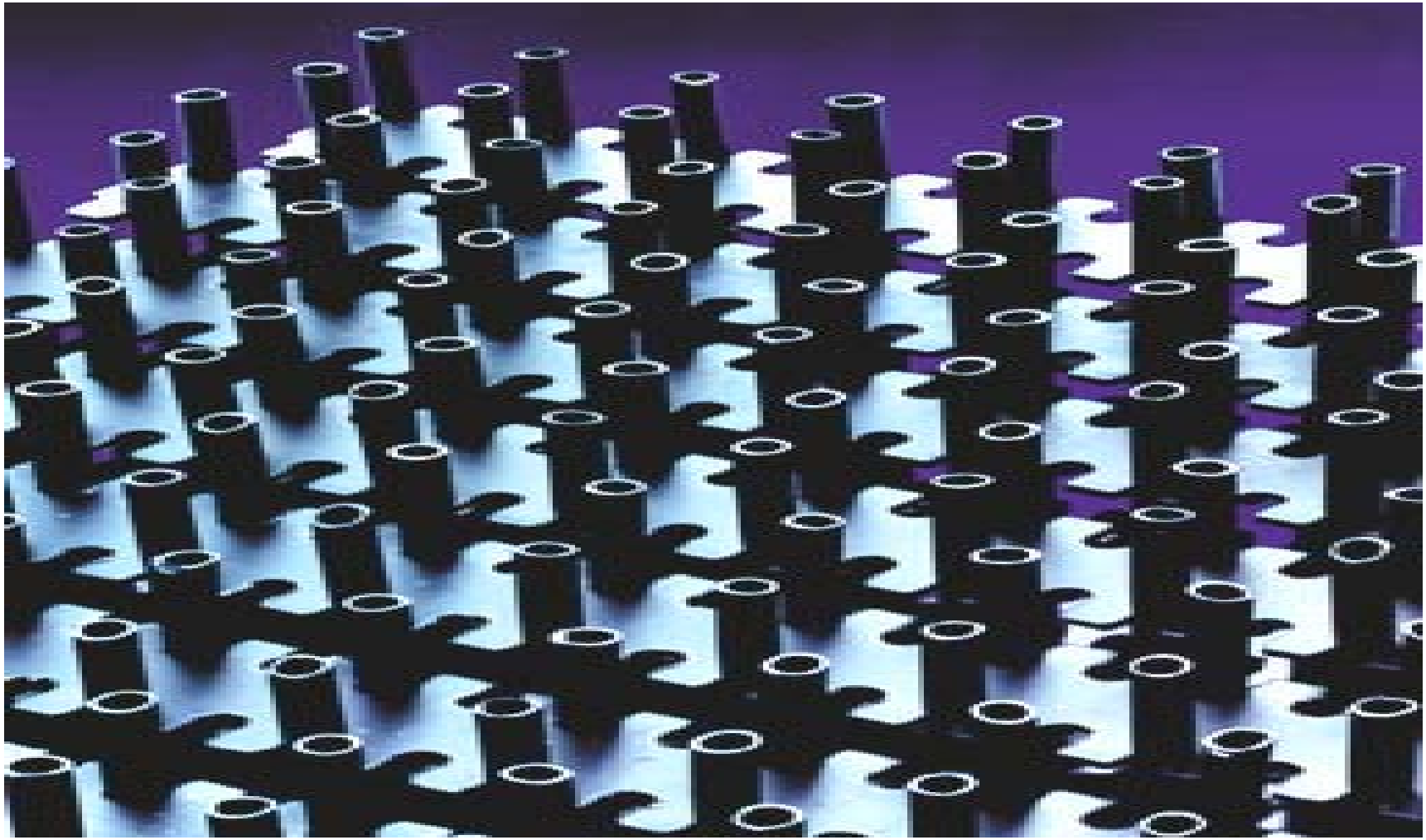

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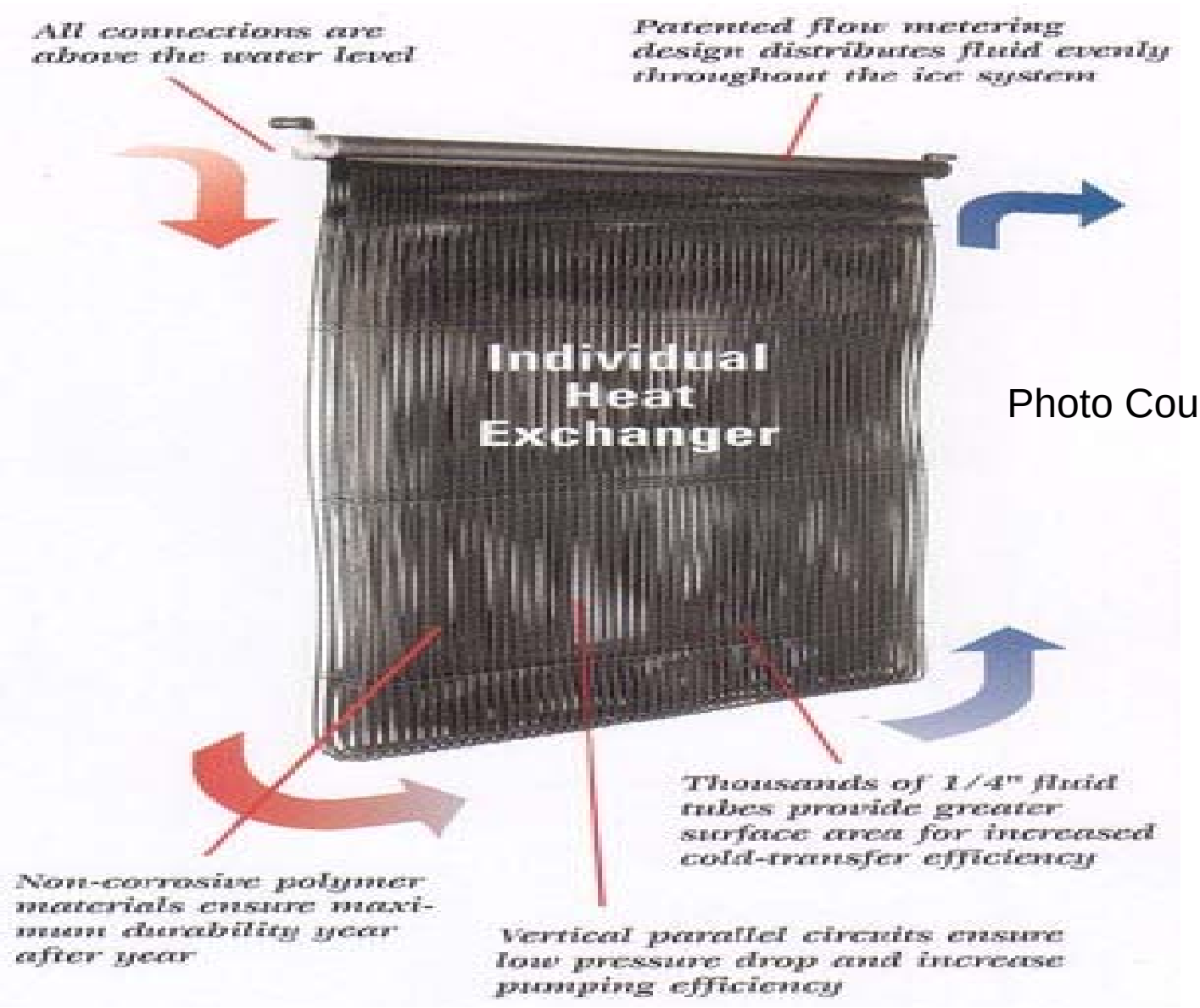

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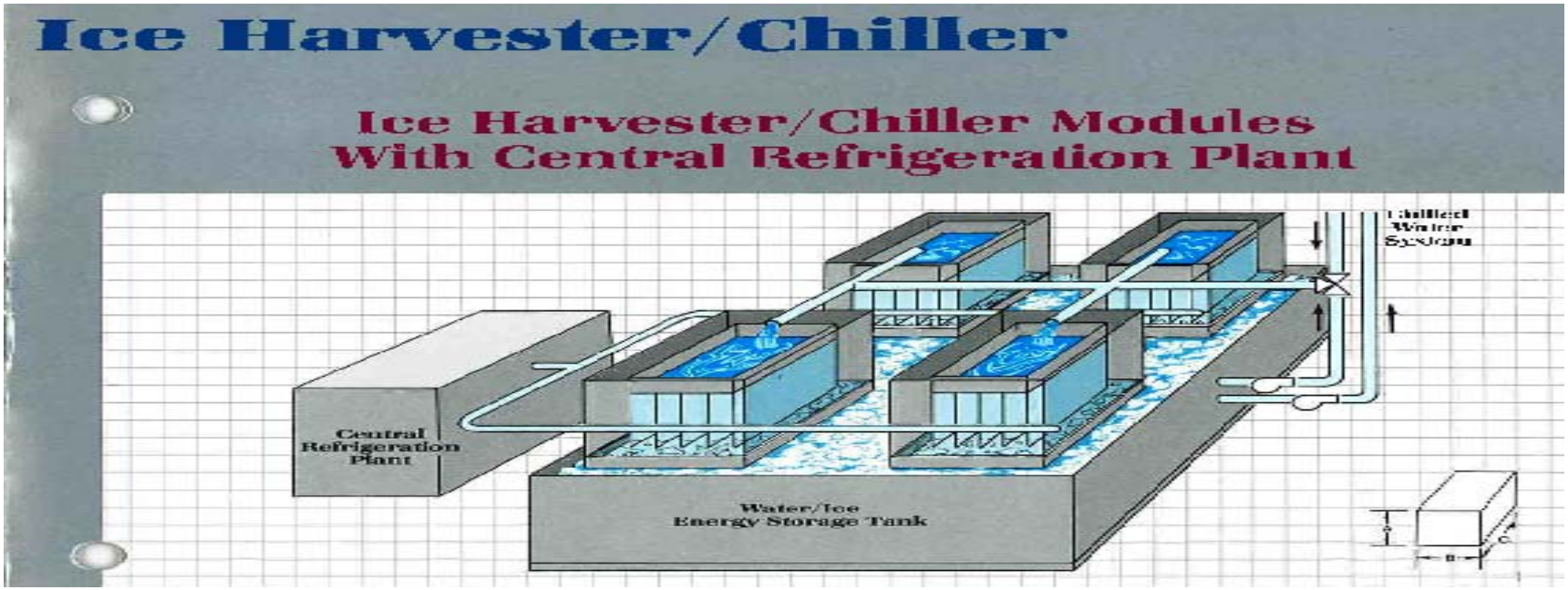

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ISAC Applications

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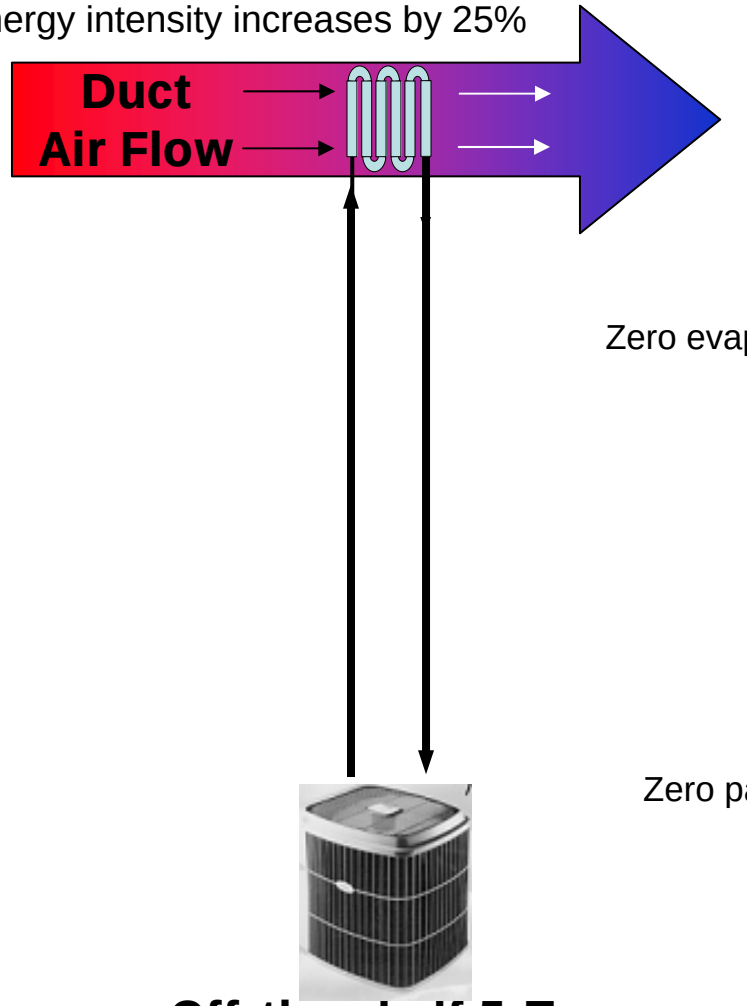
Photos Courtesy Ice Energy

Before….

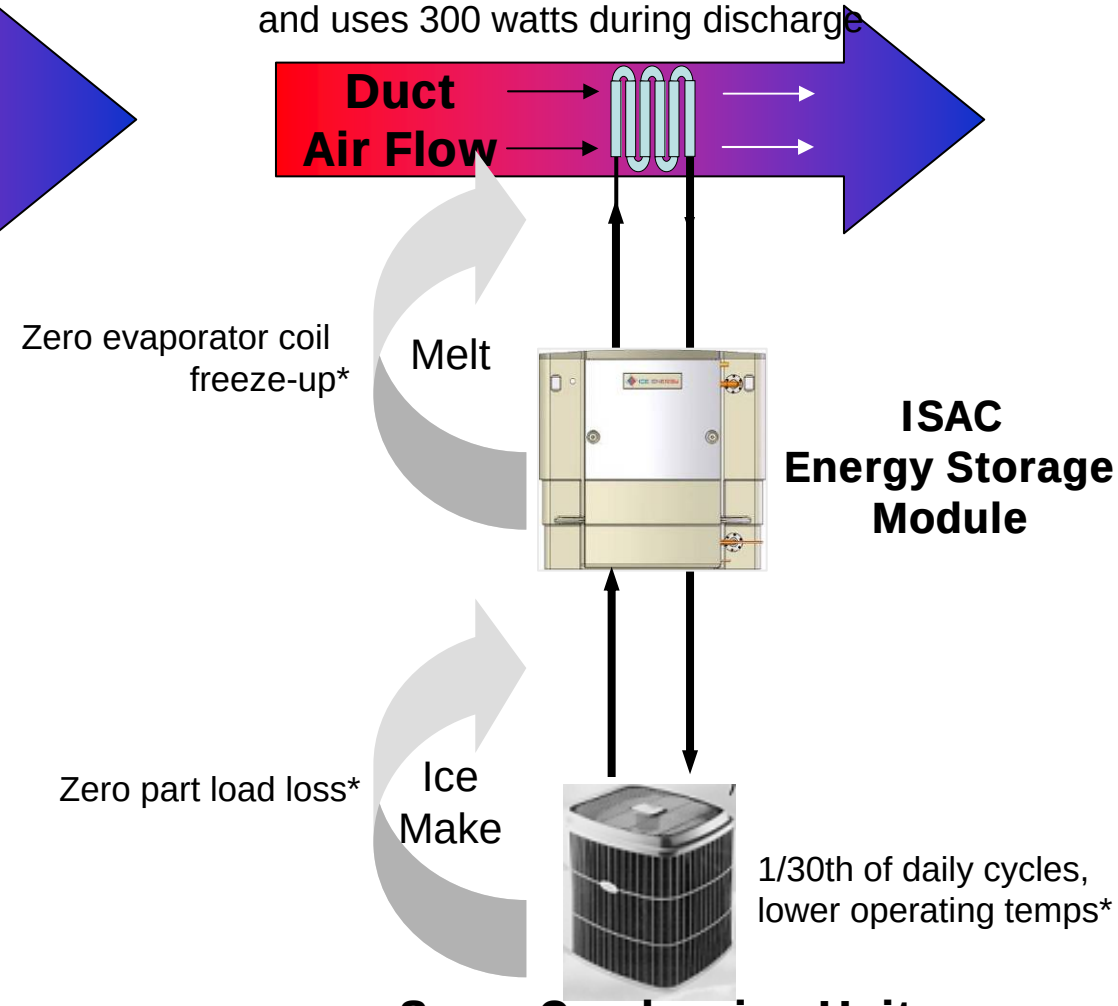
Delivers 4 Tons on hottest days and energy intensity increases by 25%



Off-the-shelf 5-Ton Condensing Unit

After….

Delivers 7.5* Tons on hottest days and uses 300 watts during discharge



Same Condensing Unit (locked-out during peak)

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***Improves Condensing Unit Effectiveness & Reliability**

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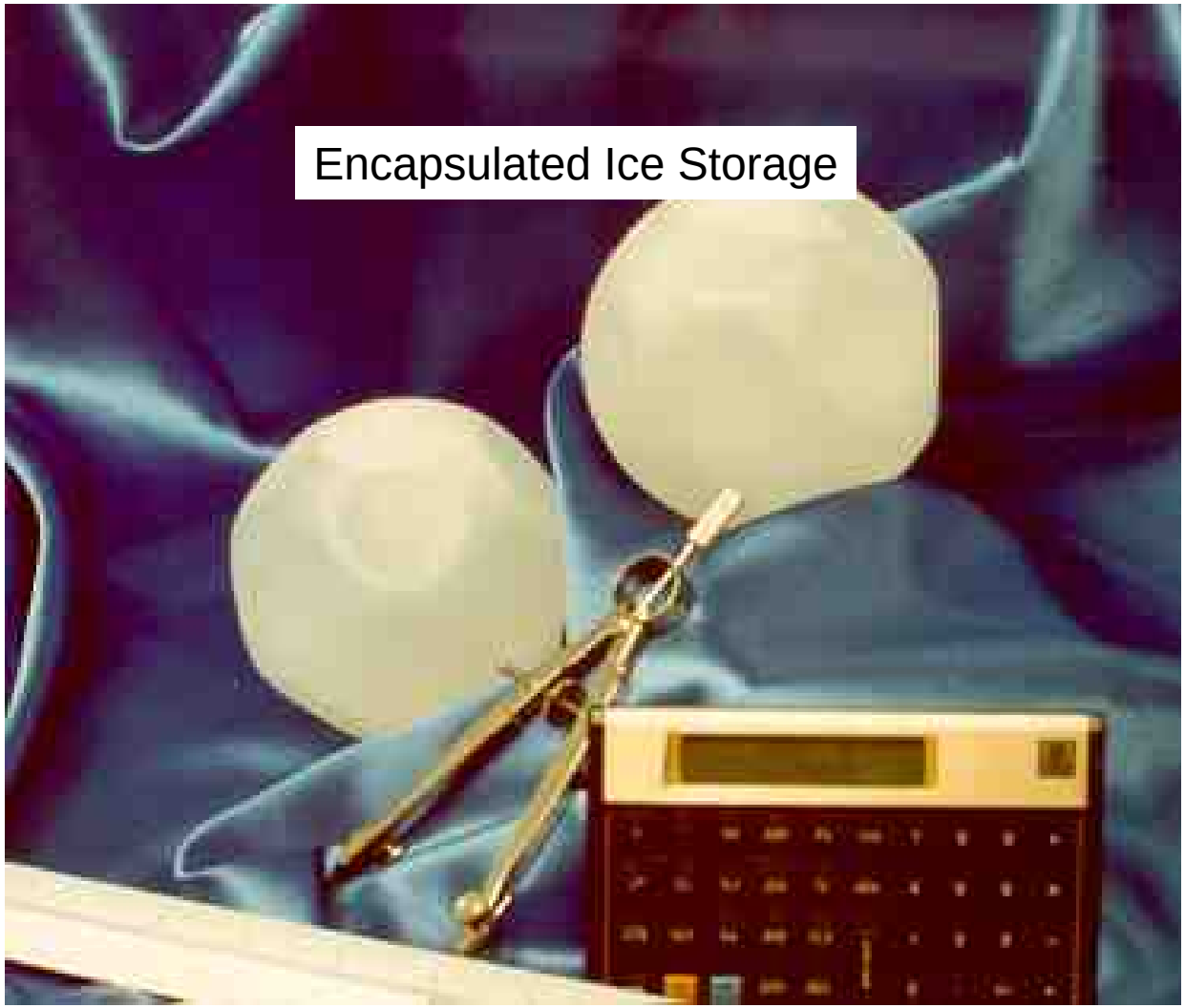

Ice Balls - Four Inch Diameter Plastic Spheres Filled With Water

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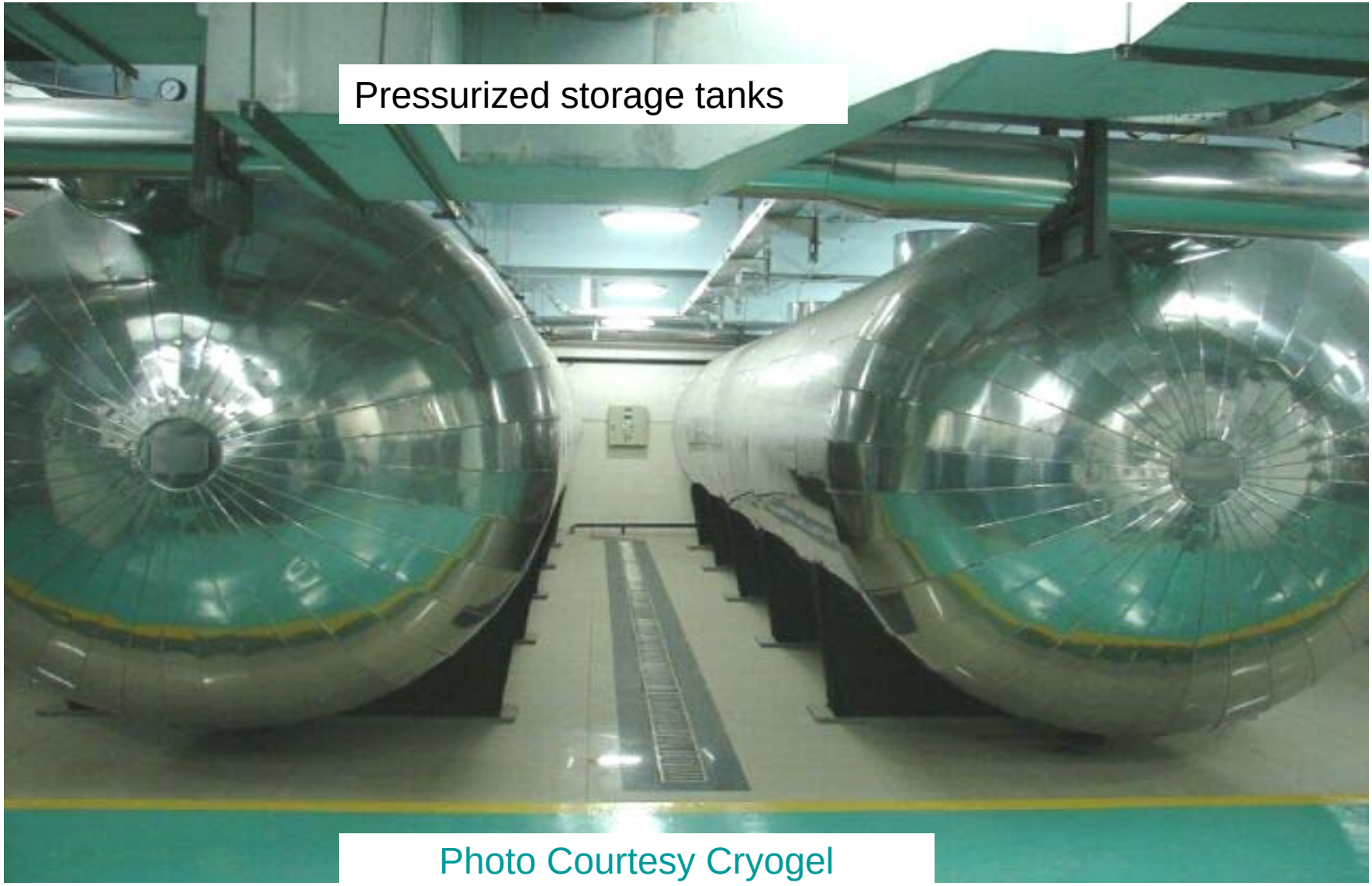

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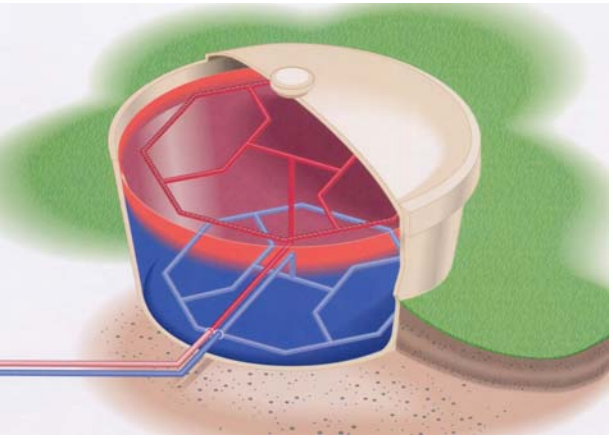
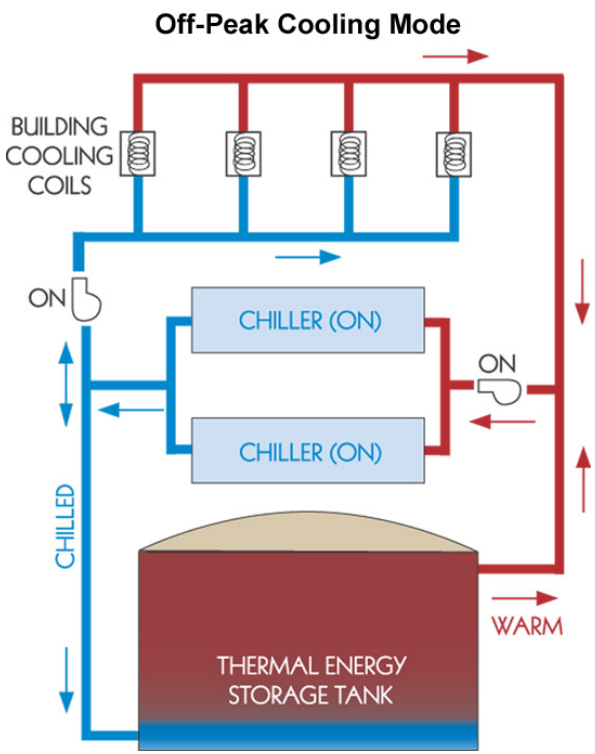
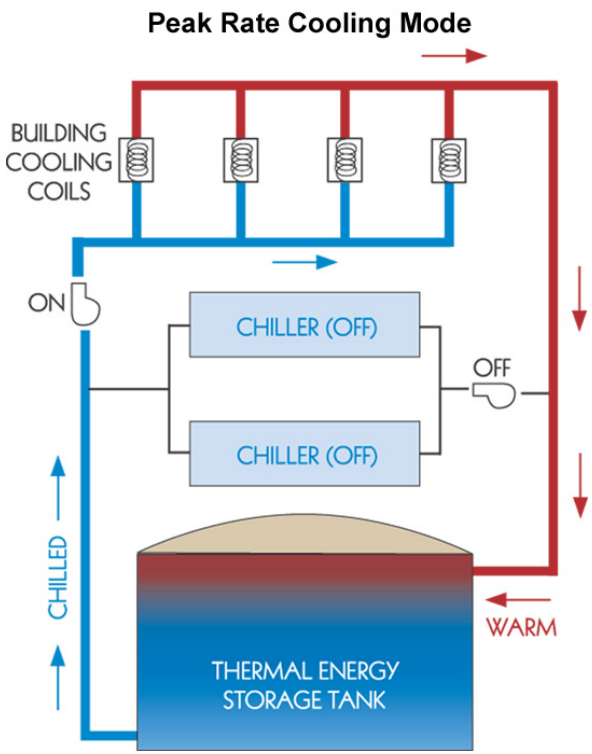
Chilled Water TES Concept







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Concrete Tank Flexibility

Above ground

Fully buried

Fully buried (heavy loading)

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Photos Courtesy Natgun

Numerous Enhancement Alternatives

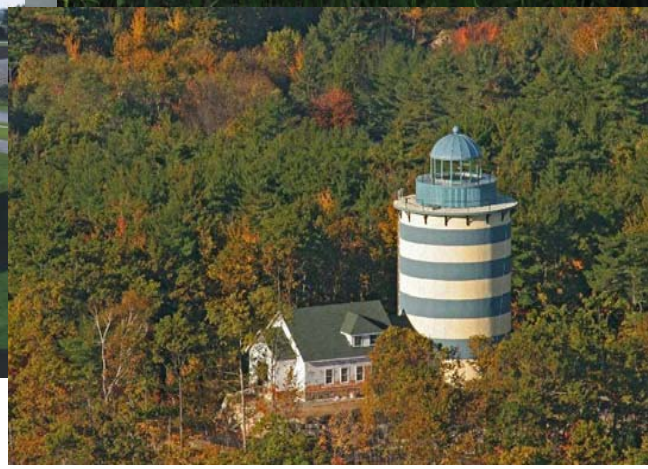

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Photo Courtesy Chicago Bridge and Iron

Above Grade Steel TES tank

Photo Courtesy Chicago Bridge and Iron

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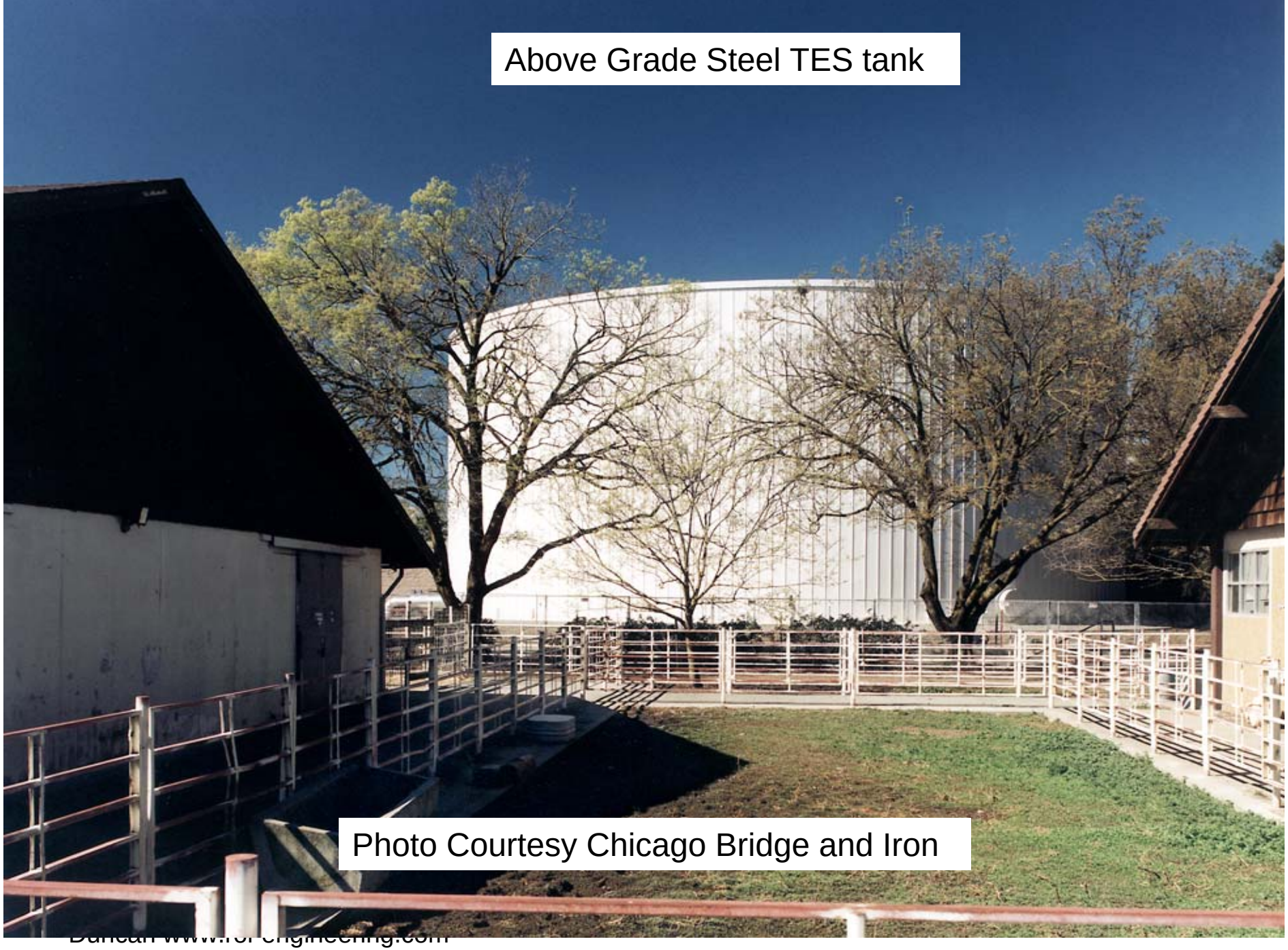
Above Grade Steel TES tank
Photo Courtesy Chicago Bridge and Iron

The Basics Don't Change

- **To save substantial energy with any system, you have to incorporate central plant and airside optimization software.**
- **Central Plant Optimization routines must look at the site loads –** without knowing AHU load information, you cannot maximize energy savings, or respond to load changes properly.
- **AHU Optimization routines are required** to minimize energy use and promote occupant comfort!
- The System Must be Operator Friendly – If the person that designed it cannot explain it in an understandable manner to the people that operate the facility, it is too complex.

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Operating Staff are the Key to Long Term Success

- To ensure long term project success, the operating staff must understand the system, and they must buy into the design philosophy.
- We have all seen facilities with $500,000 DDC system time clocks because the operating staff was not made a part of the process, they did not understand how it was supposed to work, and they could not easily “tweak” the system to suit their tenant needs.
- **Once a system goes to “man-u-matic” operation instead of automatic operation, savings go out the window.**

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Load Based Optimization is the Key to Project Success

- **If the central plant reset routines are not evaluating the loads at the AHU's and responding to those loads, you either waste central plant energy by running the chilled water too cold, or you waste AHU energy by running the chilled water too warm.**

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Load Based Optimization Saves "Invisible" Energy

- **If you run the chilled water too warm to try and make the chiller plant efficiency look good, you won't meet supply air temperature setpoints, causing the fans to speed up to deliver more air, since the VAV boxes will open up.**
- **This wastes a boatload of "invisible" fan energy. It is invisible, since most engineers ignore it, so it must not exist.**

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TES Optimization

- With a chilled water storage system, you don't always need to store 39 degree water.
- On mild days, you can get away with warmer chilled water temperatures in the tank, which will increase chiller energy savings.
- When you don't use a full tank, you can start using it earlier the next day, and start the charge cycle later, when the air temperature is lower and the chiller/CT system will be more efficient.
- These functions should be automated to reduce the need for operator intervention.

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Common Sense Control

- **The chiller plant and TES systems should respond to the AHU loads, like a VAV box responds to the zone loads.**

- **If you controlled a VAV box independently of the load it serves, how many complaints do you think you would get each hour?**

- **How can you control a chiller plant or TES system independently of the loads it serves?**

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Primary/Secondary Plants Are OK!

- If you already have a primary/secondary chiller plant, **you do not have to convert it** to Primary Only Variable Flow (POVF) to get cost effective savings.

- Variable Primary Flow/ Variable Secondary Flow (VPF/VSF) is a far more cost effective alternate.
- Adding VFD's and the appropriate controls to the primary pumps will get you an efficient pumping system at a relatively low cost.

- Variable Primary/Variable Secondary (VPF/VSF) plants are easier to control in a TES environment than POVF plants.

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POVF vs. VPF/VSF Case Study

- We recently evaluated a facility that had been told to convert their plant to POVF from P/S.
- POVF chilled water pumping energy totaled approximately $26,500 per year.
- VPF/VSF chilled water pumping energy totaled approximately $28,000 per year.
- Net savings for POVF over VPF/VSF is approximately $1,500 per year.
- Construction cost bid for conversion: approximately $400,000.

- Incremental costs/savings to convert to POVF resulted in a 200+ year payback period when compared to VPF/VSF.

- VPF/VSF is a much better option for most P/S plants.

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TES can Augment Demand Response Programs

- **If you size a TES system to make it a dispatchable DR system, it can be used to increase the demand reduction over and above the demand reduction from shutting off the chillers, CDWP's and CTF's.**
- **You can use the TES capacity to drop the air handling unit supply air temperature, drop the static pressure and reduce the fan speed, and cut fan energy by more than 50%, while still maintaining comfort.**

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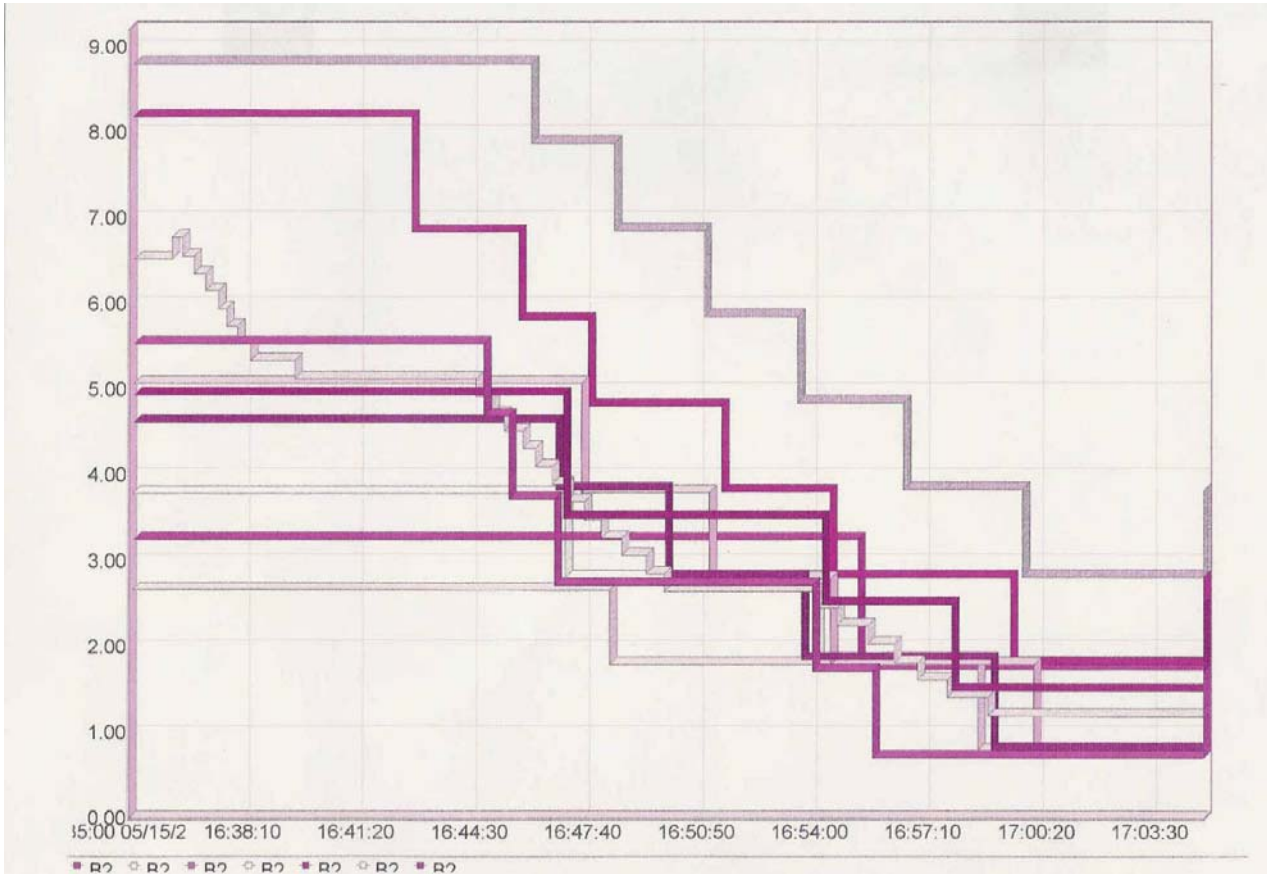
Fan kW Reaction to Demand Response Command



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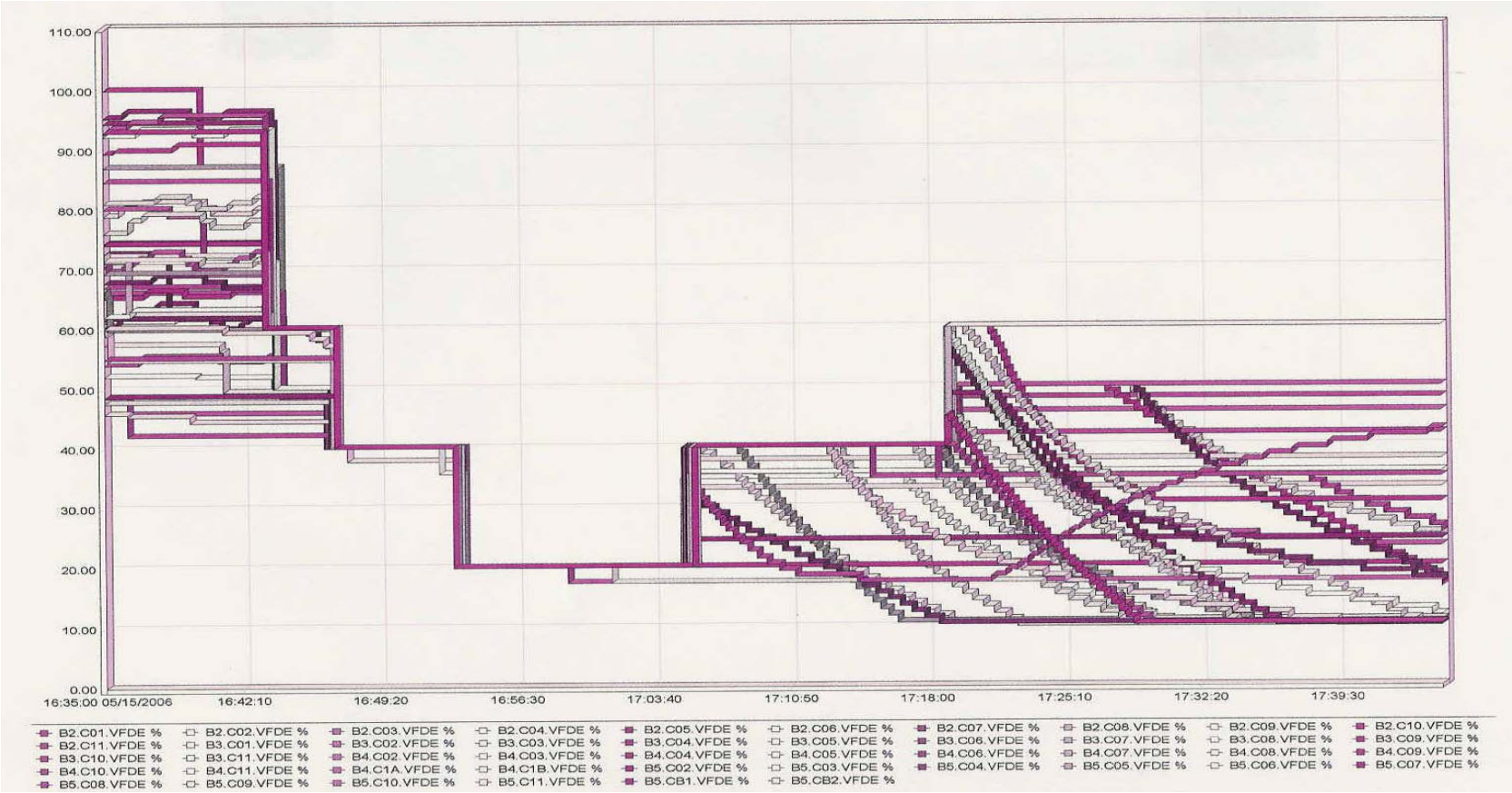
Demand Response and Automatic AHU Coasting Cycle Showing Fan Speed Reduction after 5:00 PM Based on Floor Loading



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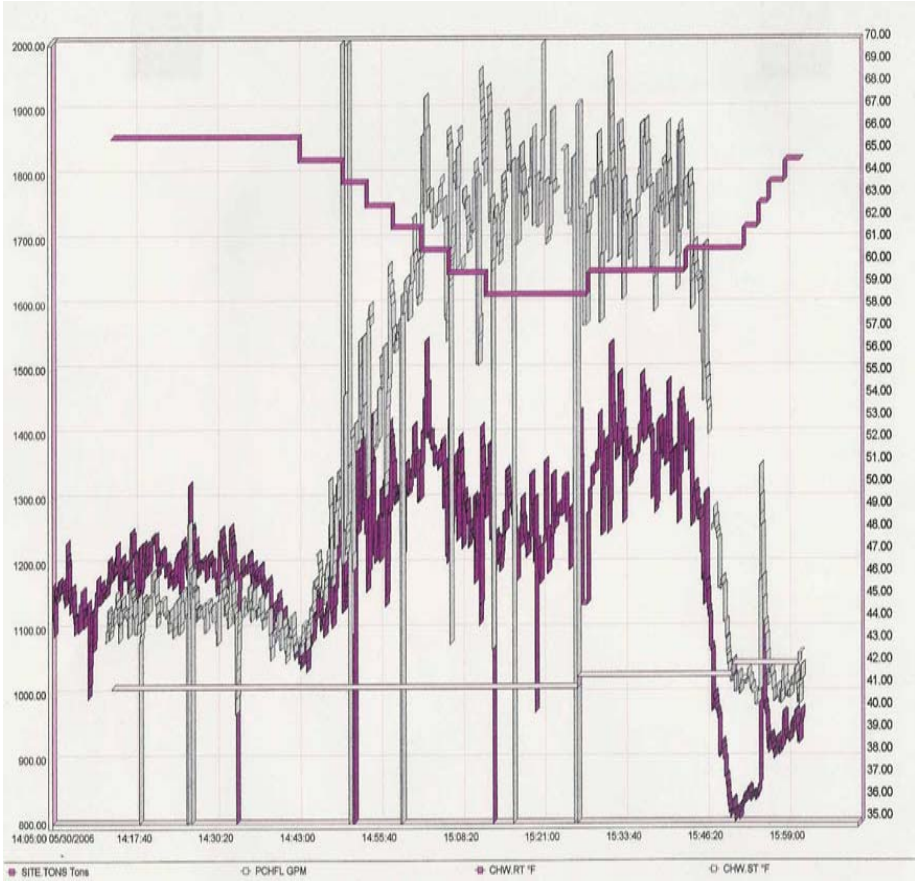
Using TES as Demand Response Tool Maintains Comfort, but Burns Capacity – Size it Right!



- **TES temperature differential drops from 26F down to 18F.**
- TES flow rate goes from 1200 GPM to 1800 GPM
- TES tonnage being discharged goes from 1200 to 1350.

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TES Energy Efficiency Benefits at the Site

- Most of the efficiency items we discuss can be applied to non-TES systems to improve their efficiency as well.
- TES has the added benefit of operating at night when the wet bulb temperature can be 15F to 20F below peak day temperatures, and the dry bulb temperature can be 25F to 40F below peak day temperatures.
- TES systems are typically base loaded when charging, reducing “parasitic” losses from auxiliary equipment.

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TES Energy Efficiency Benefits at the Site

- **Chiller systems running at night when it is cool are more efficient than chiller systems running in the heat of the day.**
- These cooler nighttime temperatures can more than offset the need to run the chillers for chilled water storage systems at 3F to 5F below daytime supply temperatures and ice storage chillers at 20F to 25F below daytime temperatures.
  - This is a system specific savings – some systems will save more than others.

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TES Energy Efficiency Benefits at the Power Plant/T&D System

- **A California Energy Commission Study showed power plant energy savings of between 20% and 43% by shifting energy consumption to the night, using highly efficient base loaded plants when it is cool, when compared to using inefficient peaker plants in the heat of the afternoon to provide power for air conditioning loads.**

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TES Energy Efficiency Benefits at the Power Plant/T&D System

- The **T&D system losses are reduced by 3% or more at night** when T&D equipment is lightly loaded and cool.
- These efficiency gains can result in substantially reduced greenhouse gas emissions, reducing the potential for Global Warming.
- TES can be used to reduce T&D bottlenecks, saving substantial ratepayer costs.

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Draw thru vs. Blow thru Cooling Coils

- **Can cooling coil placement (upstream of fan vs. downstream of fan) affect chilled water system temperature differential?**

- **Can cooling coil placement affect total cooling loads?**

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Importance of Coil Selection and Fan Orientation

- 8 row 12 fin per inch 300 to 350 FPM cooling coils are desired for peak system efficiency.
- Blow through coil design to increase CHW system TD and reduce peak loads due to reduction in over-dehumidification.
- Blow thru coils can increase CHW system TD by 30% to 40%+ and decrease peak day cooling loads by 5% or more.

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Draw Thru Coil Performance

With draw thru coils, the fan heat is sent to spaces instead of being trapped in the coil, this requires over-dehumidification, increases peak loads and hurts the chilled water system temperature differential.

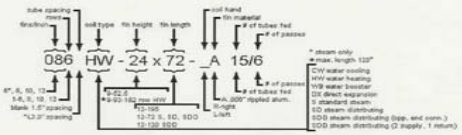
PACE Coil Selection V4.2 DRAW THRU Tue Mar 07, 2006 10:40:13

COIL MODEL #: 128 CW - 24.0 X 120 - _A- 16/8 52.5°F OFF COIL
77° RAT
18° CHW ΔT
26.25 TON LOAD

NAME: Pace
COMPANY: PACE
JOB NAME: No Job Name
DESCRIPTION: Preliminary
JOB #: 97-0000-00

	Specified Capacity	Total Capacity
Standard Air Flow Rate (CFM)	8000	8000
Actual Air Flow Rate (CFM)	8000	8000
Elevation (Ft)	0	0
Density Ratio	1.000	1.000
STANDARD CFM		
Entering Air Dry Bulb Temp. (F)	77.0	77.0
Entering Air Wet Bulb Temp. (F)	64.8	64.8
Leaving Air Dry Bulb Temp. (F)	55.0	52.5
Leaving Air Wet Bulb Temp. (F)	54.0	51.5
Entering Water Temperature (F)	44.0	44.0
Leaving Water Temperature (F)	62.6	62.6
Total BTU/HR for Each Coil	261289	314661
Sensible BTU/HR for Each Coil	191840	213861
Gallons per Minute Each Coil	28.1	33.8
Gallons per Minute per Feed	1.9	2.1
Coil Face Area (Sq. Ft.)	20.00	20.00
Standard Face Velocity (FPM)	400.0	400.0
Actual Face Velocity (FPM)	400.0	400.0
Rows Required for 8 -FPI	7.23	10.14
Rows Required for 10-FPI	6.35	8.88
Rows Required for 12-FPI	5.75	8.00
Air Pressure Drop @12-FPI (In WG)		0.89
Number of Tubes Fed (circuits)		16
Number of Passes		8
Water Pressure Drop (Ft. of H2O)		6.80
Connection Size (In)		1.50

Rated in accordance with ARI Standard 410-91.



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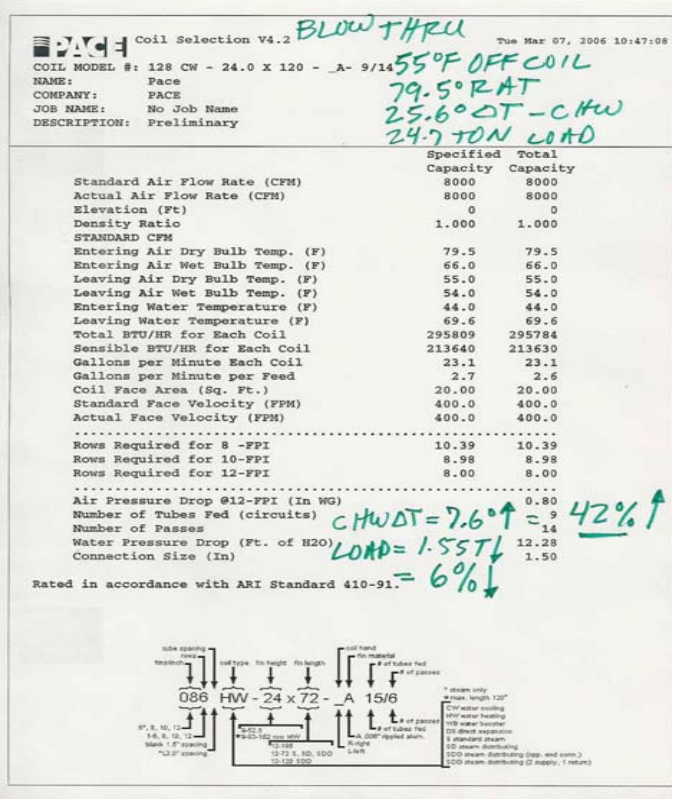
Blow Thru Coil Performance

CHW Temperature Differential increased by 42%, peak design cooling load dropped by 6% when compared to same coil in a draw thru configuration



PACE Coil Selection V4.2 BLOW THRU Tue Mar 07, 2006 10:47:08

COIL MODEL #: 128 CW - 24.0 X 120 - _A- 9/14 55°F OFF COIL

NAME: Pace
COMPANY: PACE
JOB NAME: No Job Name
DESCRIPTION: Preliminary

79.5° RAT
25.6° ΔT - CHW
24.7 TON LOAD

	Specified Capacity	Total Capacity
Standard Air Flow Rate (CFM)	8000	8000
Actual Air Flow Rate (CFM)	8000	8000
Elevation (Ft)	0	0
Density Ratio	1.000	1.000
STANDARD CFM		
Entering Air Dry Bulb Temp. (F)	79.5	79.5
Entering Air Wet Bulb Temp. (F)	66.0	66.0
Leaving Air Dry Bulb Temp. (F)	55.0	55.0
Leaving Air Wet Bulb Temp. (F)	54.0	54.0
Entering Water Temperature (F)	44.0	44.0
Leaving Water Temperature (F)	69.6	69.6
Total BTU/HR for Each Coil	295809	295784
Sensible BTU/HR for Each Coil	213640	213630
Gallons per Minute Each Coil	23.1	23.1
Gallons per Minute per Feed	2.7	2.6
Coil Face Area (Sq. Ft.)	20.00	20.00
Standard Face Velocity (FPM)	400.0	400.0
Actual Face Velocity (FPM)	400.0	400.0
Rows Required for 8 -FPI	10.39	10.39
Rows Required for 10-FPI	8.98	8.98
Rows Required for 12-FPI	8.00	8.00
Air Pressure Drop @12-FPI (In WG)		0.80
Number of Tubes Fed (circuits)		9
Number of Passes		14
Water Pressure Drop (Ft. of H2O)		12.28
Connection Size (In)		1.50

CHW ΔT = 7.6° ↑ = 42% ↑

LOAD = 1.55T ↓ = 6% ↓

Rated in accordance with ARI Standard 410-91.

086 HW - 24 x 72 - _A 15/6

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Cooling Coil Performance Comparisons

- The following pages compare blow thru and draw thru cooling coil selections.
- “Normal” design coils are compared to coils that reduce air and water pressure drop, increase chilled water system temperature differentials and reduce AHU, chiller and pumping system energy consumption.

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Draw Thru 6 R 550 FPM Coil vs. **Blow Thru 8 R 350 FPM Coil**

	Typical Design	**Near-Optimal Design**
	Draw Thru 6 Row 550 FPM	Blow Thru 8 Row 350 FPM
Mixed Air Temperature DB	77°F	77°F
Mixed Air Temperature WB	64.8°F	64.8°F
Coil Entering DB	77°F	79.5°F
Coil Entering WB	64.8°F	66°F
Coil Leaving DB	52.5°F	55°F
Coil Leaving WB	51.5°F	54°F
Air Handling Unit Supply Air Temperature (°F)	55	55
Chilled Water Supply Temperature (°F)	45	45
Chilled Water Return Temperature (°F)	54	69.2
Chilled Water Temperature Differential (°F)	9	24.2
CHW TD % Change		**169%**
Cooling Coil Water DP (ft H2O)	10.1	8
CC DP % Change		**-21%**
Cooling Coil Air DP (InWC)	1.26	0.64
CC DP % Change		**-49%**
Coil GPM Required	70	24.4
Coil GPM % Change		**-65%**
Total BTUH	315000	293000
BTUH % change		**-6%**

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6R 550 FPM Draw thru vs Blow thru and 8R 550 FPM Draw thru vs Blow thru

	Draw Thru 6 Row 550 FPM	**Blow Thru 6 Row 550 FPM**		Draw Thru 8 Row 550 FPM	**Blow Thru 8 Row 550 FPM**
Mixed Air Temperature DB	77°F	77°F		77°F	77°F
Mixed Air Temperature WB	64.8°F	64.8°F		64.8°F	64.8°F
Coil Entering DB	77°F	79.5°F		77°F	79.5°F
Coil Entering WB	64.8°F	66°F		64.8°F	66°F
Coil Leaving DB	52.5°F	55°F		52.5°F	55°F
Coil Leaving WB	51.5°F	54°F		51.5°F	54°F
Air Handling Unit Supply Air Temperature (°F)	55	55		55	55
Chilled Water Supply Temperature (°F)	45	45		45	45
Chilled Water Return Temperature (°F)	54	60.2		58.6	64.7
Chilled Water Temperature Differential (°F)	9	15.2		13.6	19.7
CHW TD % Change		**69%**			**45%**
Cooling Coil Water DP (ft H2O)	10.1	9.9		9.1	8.7
CC DP % Change		**-2%**			**-4%**
Cooling Coil Air DP (inWC)	1.26	1.14		1.58	1.43
CC DP % Change		**-10%**			**-9%**
Coil GPM Required	70	39		46.4	30
Coil GPM % Change		**-44%**			**-35%**
Total Peak Load BTUH	315000	296000		315000	296000
BTUH % change		**-6%**			**-6%**

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6R 350 FPM Draw thru vs Blow thru and 8R 350 FPM Draw thru vs Blow thru

	Draw Thru 6 Row 350 FPM	**Blow Thru 6 Row 350 FPM**		Draw Thru 8 Row 350 FPM	**Blow Thru 8 Row 350 FPM**
Mixed Air Temperature DB	77°F	77°F		77°F	77°F
Mixed Air Temperature WB	64.8°F	64.8°F		64.8°F	64.8°F
Coil Entering DB	77°F	79.5°F		77°F	79.5°F
Coil Entering WB	64.8°F	66°F		64.8°F	66°F
Coil Leaving DB	52.5°F	55°F		52.5°F	55°F
Coil Leaving WB	51.5°F	54°F		51.5°F	54°F
Air Handling Unit Supply Air Temperature (°F)	55	55		55	55
Chilled Water Supply Temperature (°F)	45	45		45	45
Chilled Water Return Temperature (°F)	60	65.5		63.1	69.2
Chilled Water Temperature Differential (°T)	15	20.5		18.1	24.2
CHW TD % Change		**37%**			**34%**
Cooling Coil Water DP (ft H2O)	10.5	8.5		8.2	8
CC DP % Change		**-19%**			**-2%**
Cooling Coil Air DP (InWC)	0.56	0.51		0.71	0.64
CC DP % Change		**-9%**			**-10%**
Coil GPM Required	42	28.9		34.9	24.4
Coil GPM % Change		**-31%**			**-30%**
Total Peak Load BTUH	315000	296000		3'5000	295000
BTUH % change		**-6%**			**-5%**

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Draw Thru 6 R 550 FPM vs. Blow Thru 8 R 350 FPM for 100% OSA system

100% OSA System	45°F CHWS Temp	45°F CHWS Temp		42°F CHWS Temp	42°F CHWS Temp
	Draw Thru 6 Row 550 FPM **(Cannot meet required temps)**	Blow Thru 8 Row 350 FPM		Draw Thru 6 Row 550 FPM	Blow Thru 8 Row 350 FPM
Mixed Air Temperature DB	92°F	92°F		92°F	92°F
Mixed Air Temperature WB	78°F	78°F		78°F	78°F
Coil Entering DB	92°F	94.5°F		92°F	94.5°F
Coil Entering WB	78°F	79.2°F		78°F	79.2°F
Coil Leaving DB	53.1°F	55.8°F		52.5°F	55°F
Coil Leaving WB	52.1°F	54.6°F		51.5°F	54°F
Air Handling Unit Supply Air Temperature (°F)	55.6	55.6		55	55
Chilled Water Supply Temperature (°F)	45	45		42	42
Chilled Water Return Temperature (°F)	49	69		48.5	71
Chilled Water Temperature Differential (°F)	**4**	24		**6.5**	29
CHW TD % Change		**500%**			**346%**
Cooling Coil Water DP (ft H2O)	44	12.6		19.2	13.5
CC DP % Change		**-71%**			**-30%**
Cooling Coil Air DP (InWC)	1.38	0.83		1.38	0.82
CC DP % Change		**-40%**			**-41%**
Coil GPM Required	**362**	59.5		226	50
Coil GPM % Change		**-84%**			**-78%**
Total BTUH	724000	714000		735000	726000
BTUH % change		**-1%**			**-1%**

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TES Optimized Cooling Coil

32 Degree TD – 200% of Typical TES System Designs – Doubles the Capacity of Most Installed TES Systems

	TES Duty 8R 350 FPM
Mixed Air Temperature DB	77°F
Mixed Air Temperature WB	64.8°F
Coil Entering DB	79.5°F
Coil Entering WB	66°F
Coil Leaving DB	55°F
Coil Leaving WB	54°F
Air Handling Unit Supply Air Temperature (°F)	55
Chilled Water Supply Temperature (°F)	40
Chilled Water Return Temperature (°F)	72.1
Chilled Water Temperature Differential (°F)	32.1
CHW TD % Change	
Cooling Coil Water DP (ft H2O)	4.9
CC DP % Change	
Cooling Coil Air DP (InWC)	0.61
CC DP % Change	
Coil GPM Required	18.4
Coil GPM % Change	
Total BTUH	296000
BTUH % change	

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30 Degree + TD's are Possible!

- **We have many systems that operate with 26°F to 32°F chilled water system temperature differentials.**
- It can be done, and it is not that big of a deal to make it happen.
- Big coils allow lots of leeway for energy conservation, and also allow loads to change without having to spend $$$ to upgrade the HVAC system.

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TES Design Strategies

- Some of these strategies will be controversial, and many of them will irritate chiller manufacturers, cooling coil manufacturers and TES component manufacturers. **Oh Well**.
- They are our professional opinions, and are based on many years experience in fixing "broken" TES systems that have been designed by other engineers that had the best of intentions and wanted to do the best they could.
- **We strive for cost effective energy efficiency** on every project, so many strategies will add first cost, but there are corresponding operating cost savings that will accrue to the owners.

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TES Design Strategies, Continued

- Use an Engineer that has a successful track record with TES and energy efficiency.
  - They could be the best TI designer or rooftop packaged unit guys around, but **TES is different**.
  - **There are many, many places to go wrong when designing a TES system**.
- Include Load Based Optimization and TES Optimization control and design strategies to reduce energy consumption and improve annual savings.
- Serving the absolute peak day cooling loads with the highest cost effective energy efficiency and greatest peak demand reduction should be goals of the project.

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TES Design Strategies, Continued

- **Have redundant TES related pumps.** One will eventually die and this can cost you a small fortune if you don't have a backup pump.
- Design the tank structure for several degrees below your expected lowest temperature.

- **Include overnight loads, larger than you expect, in the load profile when you are determining how much charge capacity you have available**. Loads always seem to grow over time, and this can turn a good system into a marginal system.

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TES Design Strategies, Continued

- **Design for a shorter charge period than you think you will have available.**
  - Hot day loads require many buildings to start up at 4:00 AM to 5:00 AM
  - **If you have to serve the building load at 4:00 AM, this capacity is not available to charge the TES system, so your charge cycle can become very short!!**

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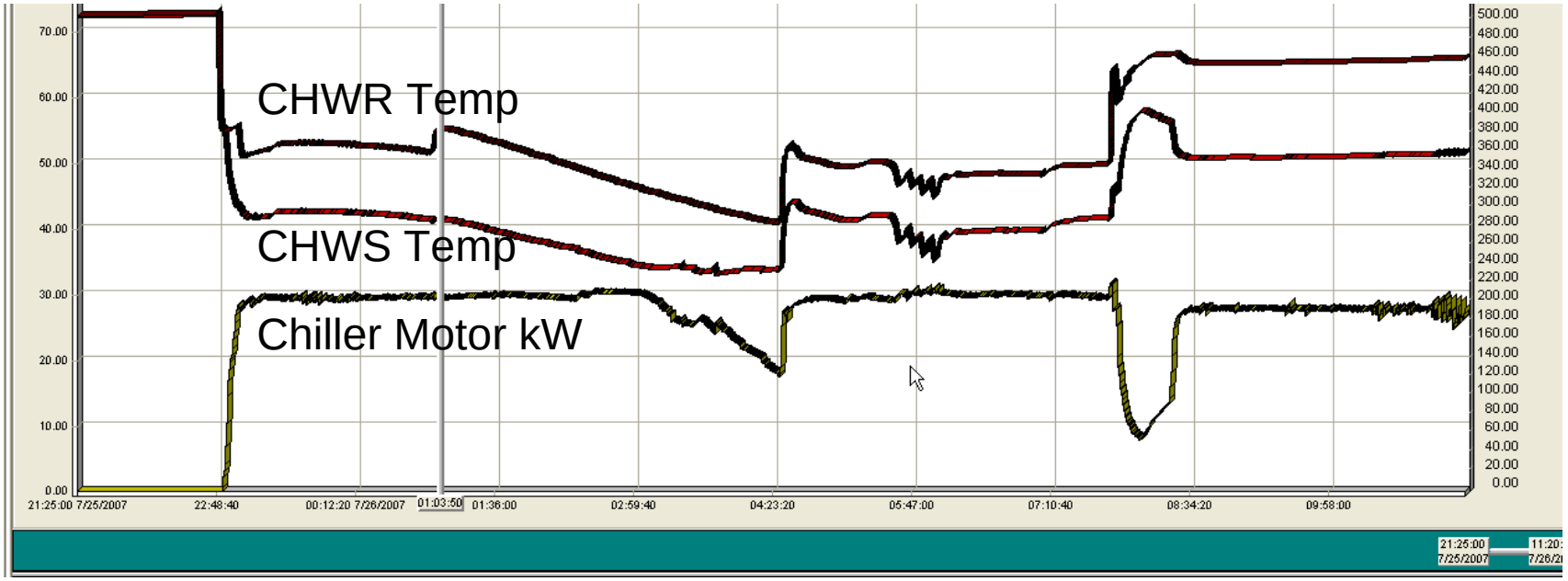
Retrofitted Labyrinth Style TES tank chiller load profile



Charge cycle starts at 11:00 PM, chiller runs flat out until 3:00 AM, CHWS temp drops to 34°F, building load hits the plant at 4:20 AM, chiller runs flat out again, serving TES and building loads until TES charge cycle ends at 7:45 AM, chiller unloads until economizers shut down, chiller runs flat out again (one chiller is down for repairs here)

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TES Design Strategies, Continued

- **Design for absolute peak day loads.**
  - **ASHRAE 0.1% does not cut it!** If you have to start a chiller on peak, you will blow a huge % of your potential savings (rate dependent).
- **Design for a flat, level load profile based on the highest peak cooling load, until 15 minutes past the last minute of the on-peak period.**
- **Size the system to start 30 minutes before on-peak and run for 15 minutes after on-peak**, all with the same chilled water supply temperature and the same discharge rate available as you require for the peak load.

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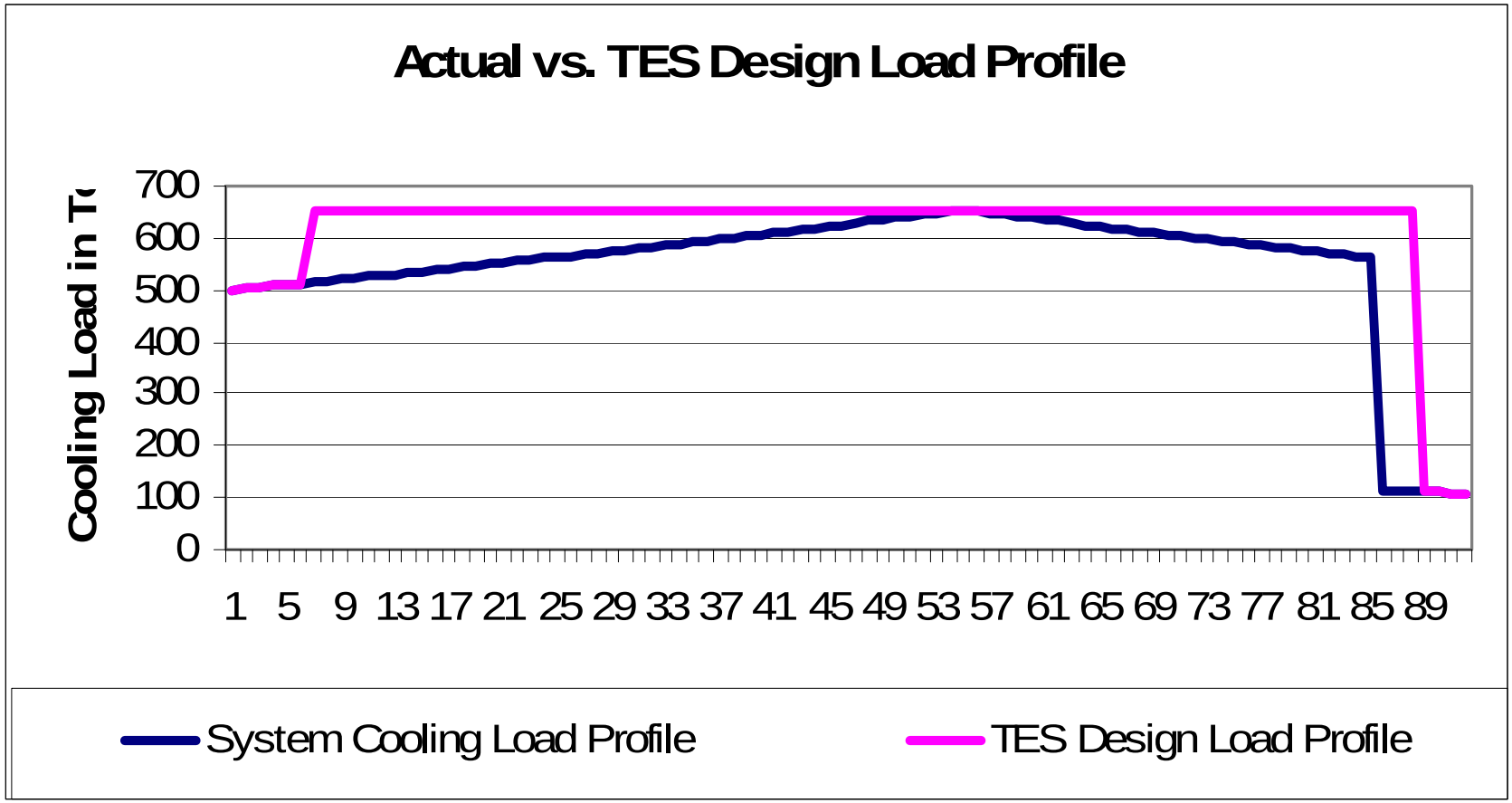
TES Design Day Profile



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TES Design Strategies, Continued

- **Incorporate blow thru fan/coil designs to increase chilled water temperature differential, reduce peak cooling loads and save pumping energy.**
- **Incorporate 8 row 12 fin per inch cooling coils at 300 to 350 feet per minute face velocity to increase water temperature differential, and decrease pumping and fan energy.**
- **Design your cooling coils for 50°F entering CHW and as close as possible to 70°F leaving CHW temps.**
  - **These temperatures will increase chiller efficiency by 25% or more and increase chiller capacity by 20% or more, depending upon motor, shell, compressor and orifice sizing, and the hydraulic design of the system.**

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TES Design Strategies, Continued

- **Design for lower system temperature differentials than you think you will get.**
  - **If you design your cooling coils for 70°F return water, you will typically get around 65°F to 66°F in the summer and 55°F to 60°F in the winter.**
  - **Most times the actual return air conditions do not match the calculated RAT conditions – if the RAT DB is lower than design, or the RAT WB is higher than design, the coil leaving water temperature will be reduced, reducing the TD.**
  - **Cooling coils have ratings tolerances too.**
  - **There are a wide variety of cooling coil selection programs. Many of them are pretty aggressive, so beware.**

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TES Design Strategies, Continued

- If you want to build a bit of “CYA” **(Cover Your Assets)** into your coil selections you can raise the entering wet bulb temperature by 2°F to 4°F to allow for future coil performance degradation.
- **Design for series chiller operation if you have a large enough CHW temperature differential.**
- **Chillers do not stay fully loaded for the entire charge cycle, especially for some ice storage systems – design more charge cycle capacity than you think you will need.**

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Large Temperature Differential Designs

- **We have many chilled water TES systems that operate with 26°F to 32°F temperature differentials during the summer.**
- Chilled water based TES system storage capacity is nearly directly related to the chilled water temperature differential.
- **A 30 degree TD system will have a tank size 50% smaller than a 15 degree TD system – you can spend some money on coils, and save huge $ on tanks.**
- Large Temperature Differential (LTD) system designs can also save significantly in infrastructure costs.
- **LTD cooling coil designs can also improve ice storage system capacity and efficiency and reduce infrastructure costs.**

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Strategy to Increase Chilled Water System Temperature differential

- Many facilities are plagued by low chilled water system temperature differentials. AKA **"Low Delta T Syndrome"**
- **This hurts energy performance, whether or not the system has TES installed.**
- **If the system is equipped with properly selected two-way throttling type cooling coil control valves, and the chilled water supply temperature and differential pressure are properly controlled, the most likely culprit for low system temperature differentials is undersized or "worn-out" cooling coils.**

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Increase CHW System TD's

- **When we started working for USC in the early 1990's, the peak day chilled water system temperature differential in the summer was in the 8°F to 9°F range.**
- Over a 16 year period we have been able to work with the Campus to raise the peak day TD to between **24°F and 26°F.**
- This has been accomplished by replacing cooling coils and control valves, and properly controlling the loads.
- This work cut the required TES tank size from 8,500,000 gallons down to 3,000,000 gallons, **saving approximately $6,000,000 and making TES cost effective.**

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Chilled Water Storage Chiller Selections

- **Each new centrifugal chiller should have a VFD on it.**
- Many existing chillers are VFD capable, some are not.
- **We typically design our chiller systems for 34 to 35 degree chilled water supply temperatures, with 80 to 85 degree condenser water temperatures. This accommodates muggy days, or a cooling tower failure.**

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Chilled Water Storage Chiller Selections

- **The lower (34°F to 35°F) CHWS design temperature allows the chillers to stay fully loaded for most of the charge period.**
- **We design very large heat transfer surface areas to keep the refrigerant temperature above freezing at 34°F CHWS temperatures.**
- If you have a large enough cooling coil system, you can pipe the chillers in series to obtain substantial benefits.
- **Without additives (like SoCool ™, from CoolSolutions), you cannot store water at below about 39 degrees if you are using a stratified TES tank design.**

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Ice Storage Chiller Selections

- **Each new centrifugal chiller should have a VFD on it.**
- Some centrifugal chillers will have difficulty meeting our design lift conditions.
- Many existing centrifugal chillers are VFD capable, some are not.
- **Existing centrifugal chillers can be expensive to modify for ice storage duty.**
- **Existing recip and screw chillers are not as bad, but modifications must typically be made to oil cooling systems and other items like that.**
- Select the HX system for a two degree approach temperature.

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Ice Storage Chiller Selections, continued

- We typically design our ice chiller systems for 20/27 degree glycol supply/return temperatures, with 80 to 85 degree condenser water temperatures. This accommodates muggy days, or a cooling tower failure.
- If possible, the chillers should be capable of providing 16°F glycol.
- The lower design temperature allows the chillers to stay fully loaded for much/most of the charge period.

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Ice Storage Chiller Selections, continued

- We typically use an ice storage chiller capacity ratio of 50% of nominal to account for chiller de-rating during ice making. This number can get smaller, depending upon the system design.
- If you have a large enough cooling coil system, (a large heat transfer surface area) you can pipe the chillers in series to obtain substantial benefits during non-TES operation.
  - You can have series/parallel switchover valves to let the chillers run in parallel during the charge mode.

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Ice Storage Air Conditioning

- **There are guidelines for vertical lift and total refrigerant piping lengths that must be adhered to.**
- The manufacturer provides design engineer training classes, and will review proposed system designs to ensure that they are correctly implemented.
- This is to be commended, as a high success rate will help all involved parties.

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Get What you've Paid For

- **To be sure that you get what you've paid for, the TES system must be thoroughly tested by performing Charge cycle and Discharge cycle tests.**
- Substantial performance monitoring equipment should be included with every system.
- **Detailed performance tests must be included in the specifications, along with performance penalties, should that be an option you can invoke.**
- Firms that understand TES performance tests should be used to perform the tests.

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Summary

- Thermal Energy Storage and Energy Efficiency are not mutually exclusive!
- Properly designed TES systems can **reduce power plant energy consumption and the resultant pollution by 50%** on peak cooling days.

- The TES and HVAC system Load Based Optimization Routines must look at 100% of the HVAC energy consuming devices, and the comfort conditions of the facility, **not just the chiller plant.**
- Load Based Optimization System (LOBOS) controls, combined with TES Optimization has been the key to many successful projects.
- Operating Engineers are the key to long term success. If they don't understand it, or they can't "tweak" it to suit their facility, it won't work for long!

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ESAC Case Study #1

- An 8 story building was equipped with water cooled **DX floor by floor** units that were a maintenance and energy problem.
- A 12 story building next door was equipped with a failed ice storage system that had been “fixed” by another engineering firm by converting it to an empty tank CHW storage system.
- **It had to run both chillers and the TES system on peak and did not meet warm day cooling loads for the 12 story building.**

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ESAC Case Study #1, continued

- Our project removed the DX system from the 8 story building, replaced all of the cooling coils in both buildings with 30 degree TD chilled water coils, re-piped the plant with chillers in series, added headers and valves to convert the TES to a stratified design, and added Load Based Optimization controls for the entire HVAC system.

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ESAC Case Study #1, continued

- The TES system is now a full storage system for both buildings, instead of being partial storage for one building.
- **The combination of the TES project and Loadside Optimization Control strategies reduced total annual site energy by approximately 18%.**

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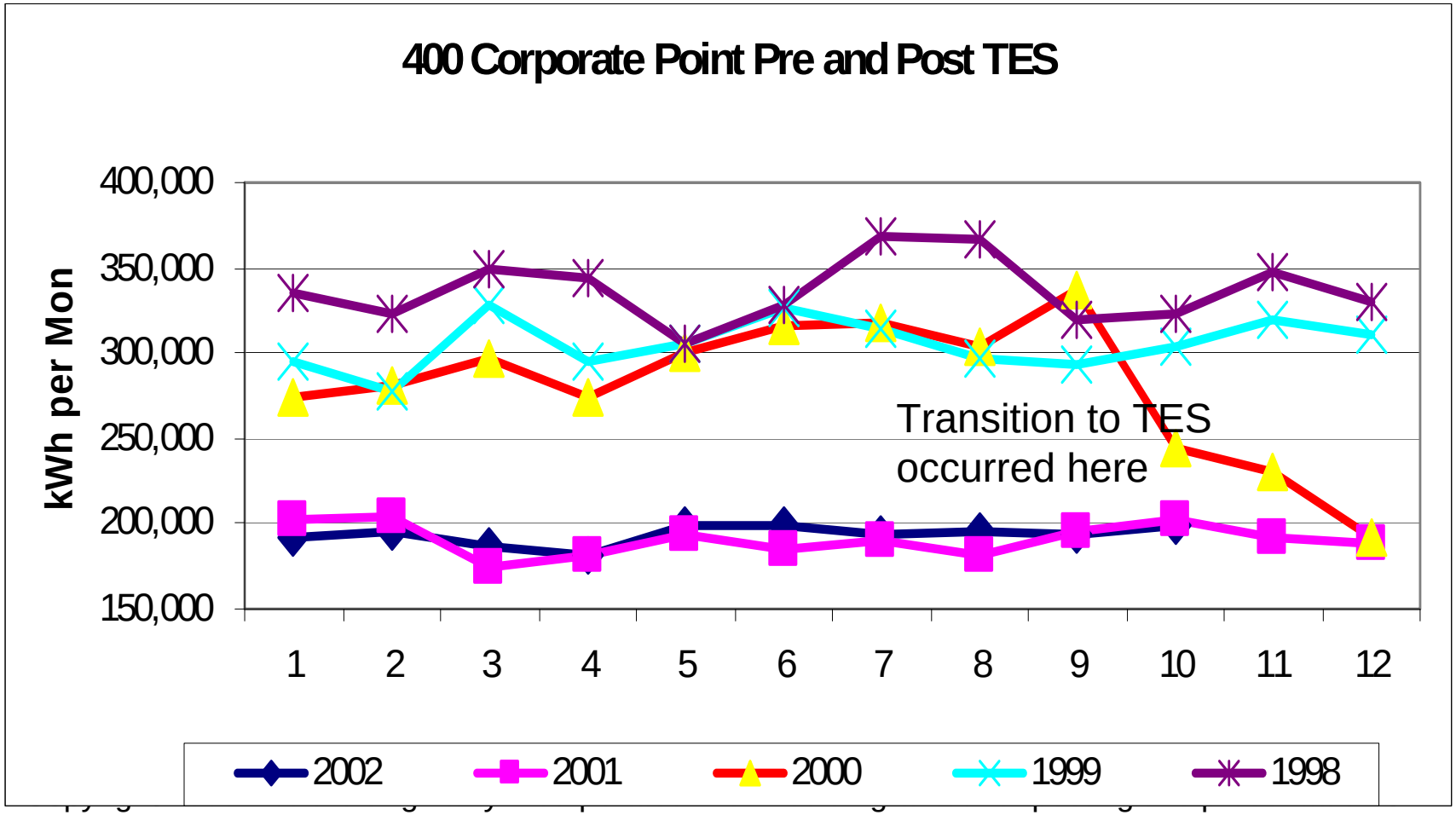
Obvious TES Startup Date in October 2000. Energy Consumption of this building dropped 38% Cooling Load Was Shifted to the 600 Building TES Plant



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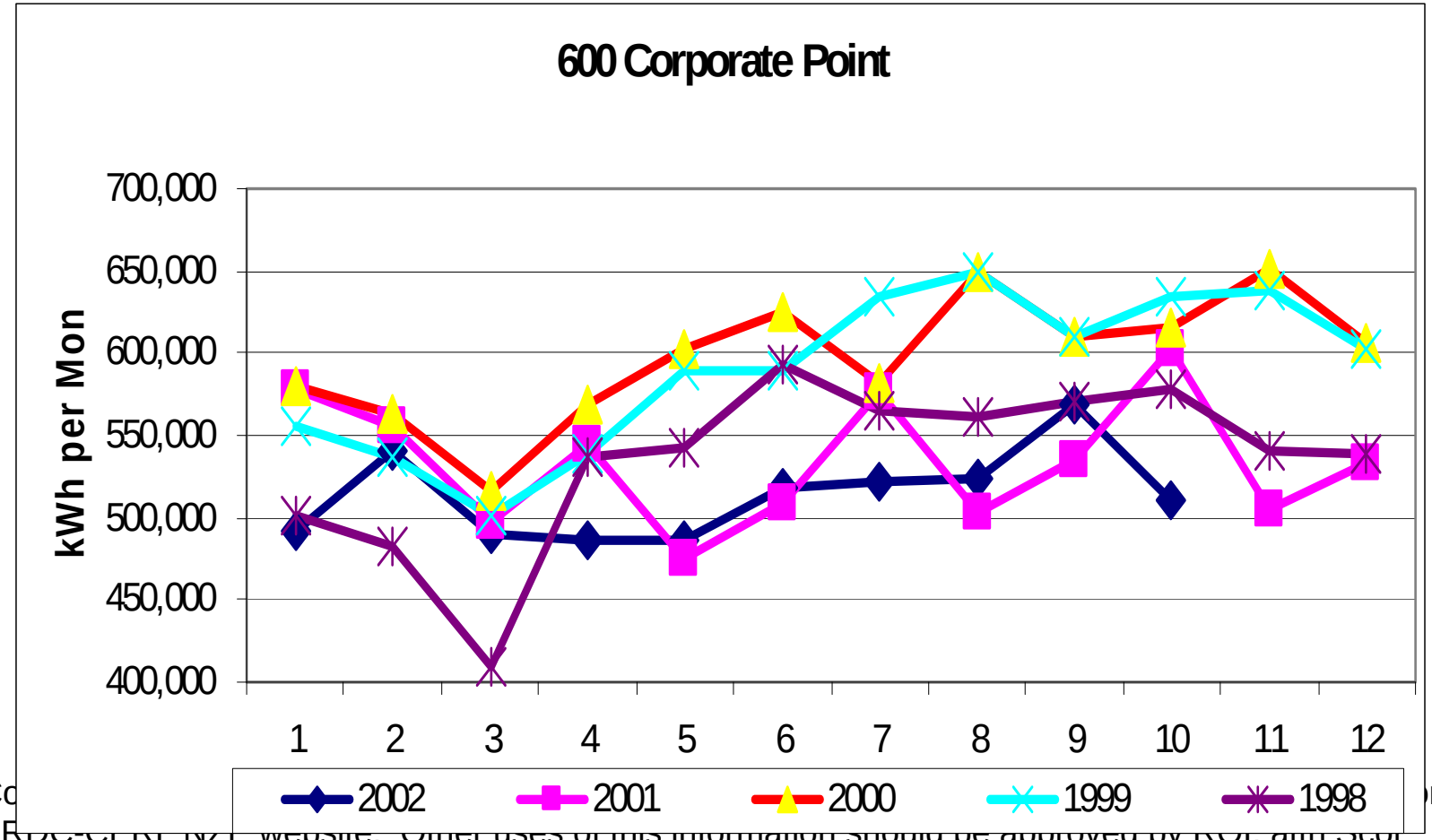
TES was upgraded in this building, and the chiller plant now serves two buildings instead of one, and the energy consumption of the TES chiller plant building still dropped.



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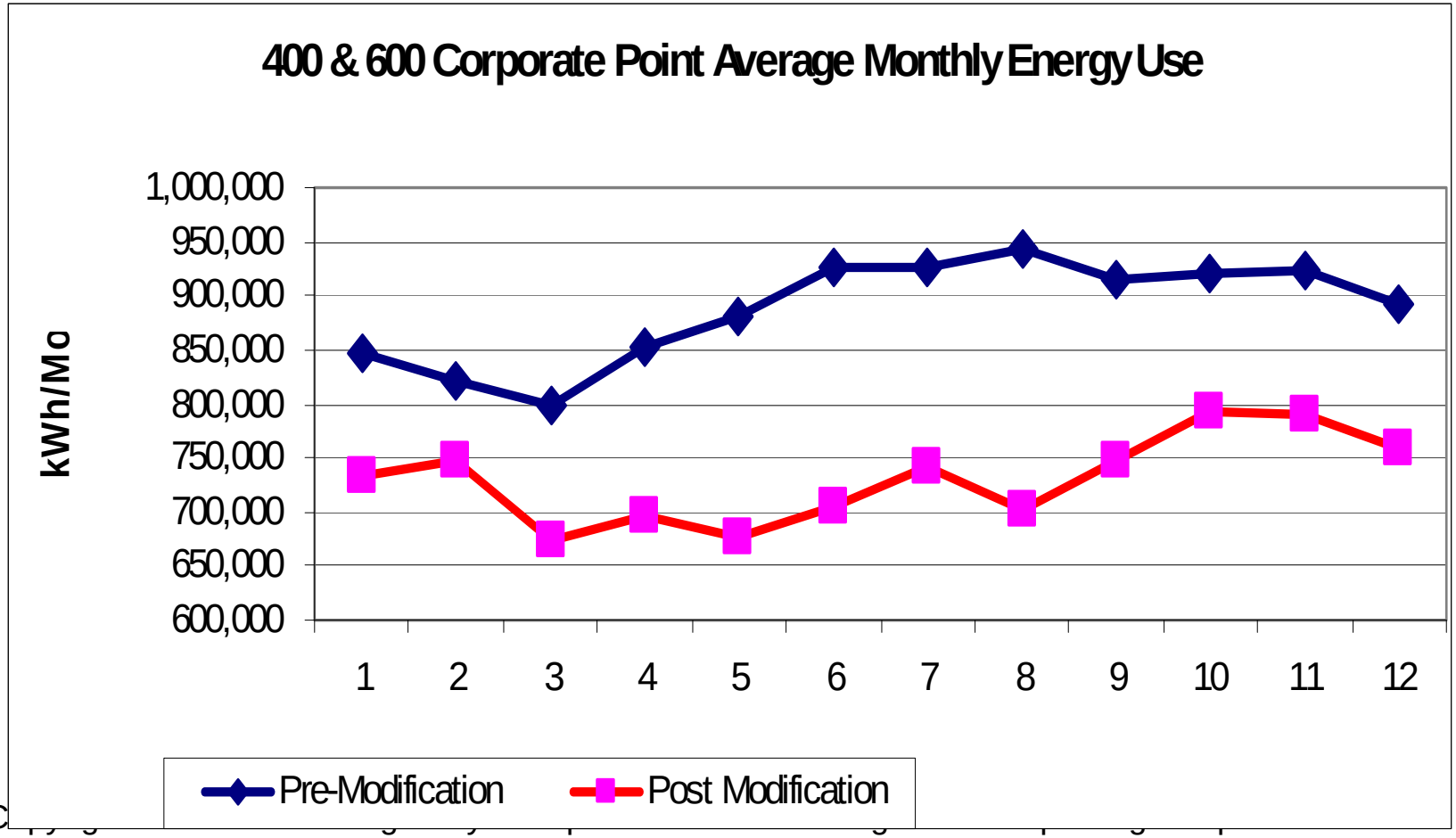
Average Monthly Energy Consumption Combined Facility Pre and Post TES System Upgrade



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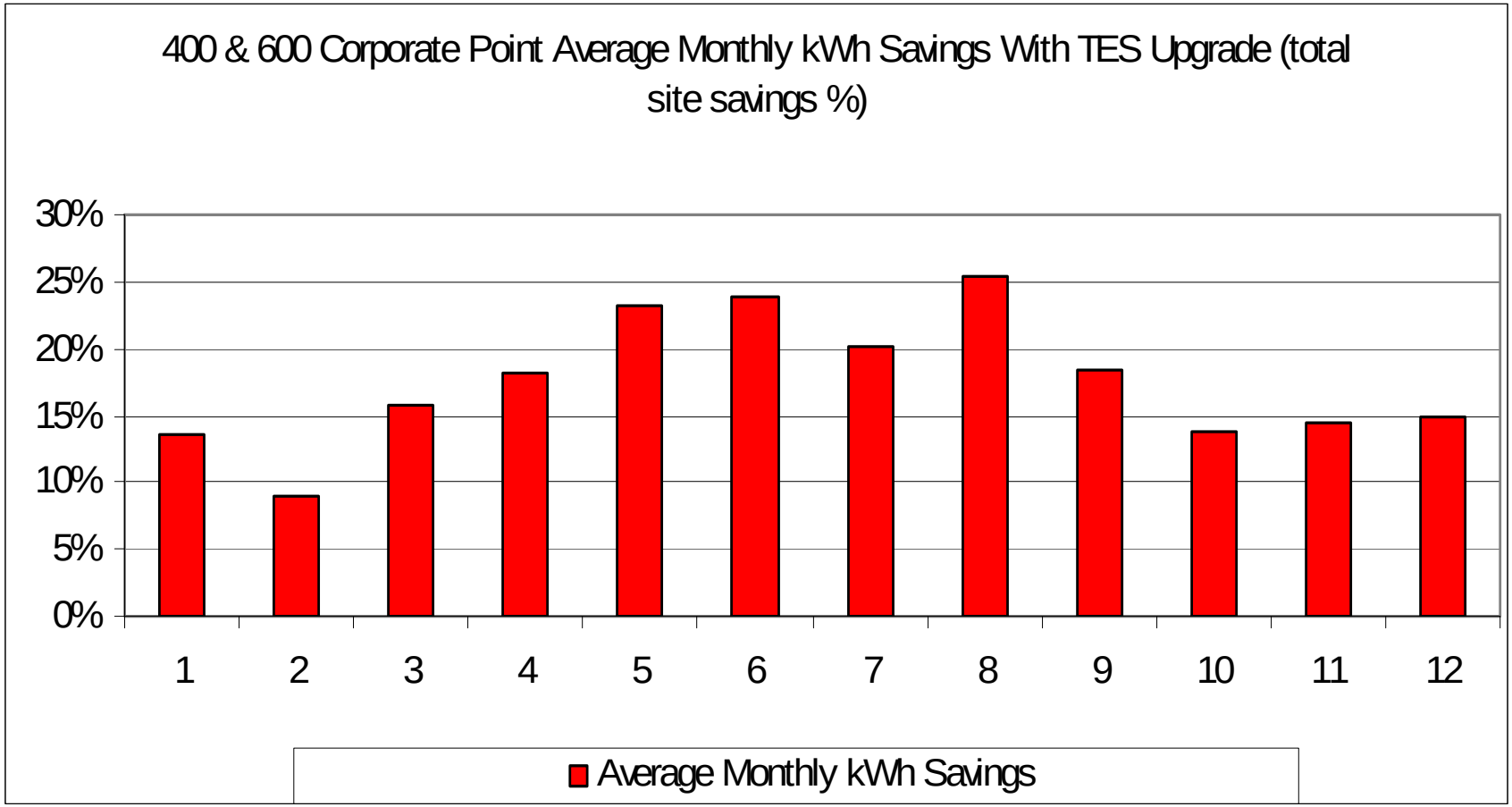
Average Monthly Site Savings Compared to Pre- TES Upgrade Use



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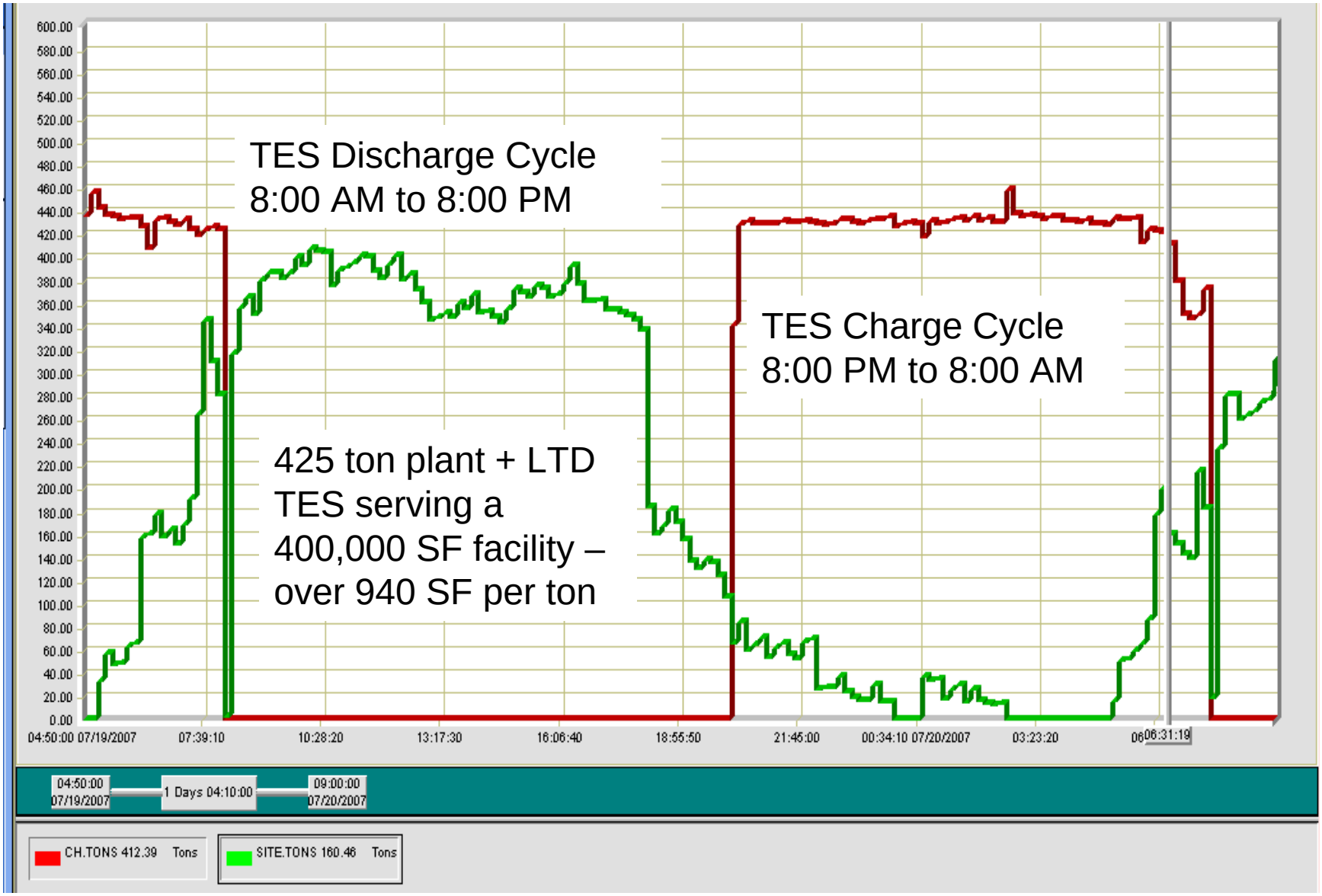

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MAIN TES Tes Adj AH Setpoints AH Select B6 boiler B4 boiler

TES OSA Temp 83.6 °F

Air Handler Status

	Mode	Fan Speed	Duct Static	Sup Temp	Ret Temp	Valve Pos.
FL-12	OCCUPIED	71.9 %	0.46 IWC	57.9 °F	73.6 °F	98.3 %
FL-11	OCCUPIED	75.0 %	0.50 IWC	59.3 °F	74.1 °F	98.4 %
FL-10	OCCUPIED	72.9 %	0.47 IWC	57.8 °F	73.5 °F	28.7 %
FL-09	OCCUPIED	74.4 %	0.53 IWC	60.1 °F	74.7 °F	62.6 %
FL-08	OCCUPIED	78.9 %	0.48 IWC	57.9 °F	74.1 °F	82.8 %
FL-07	OCCUPIED	49.4 %	0.52 IWC	61.3 °F	75.4 °F	66.7 %
FL-06	OCCUPIED	58.7 %	0.42 IWC	58.4 °F	72.8 °F	84.5 %
FL-05	OCCUPIED	62.6 %	0.53 IWC	57.8 °F	75.4 °F	57.9 %
FL-04	OCCUPIED	72.2 %	0.54 IWC	56.2 °F	74.6 °F	98.9 %
FL-03	OCCUPIED	67.4 %	0.49 IWC	59.8 °F	76.2 °F	59.9 %
FL-02	OCCUPIED	57.5 %	0.49 IWC	61.8 °F	74.7 °F	54.9 %
FL-01	OCCUPIED	69.4 %	0.58 IWC	66.8 °F	74.6 °F	53.6 %

Bldg 600

	Mode	Fan Speed	Duct Static	Sup Temp	Ret Temp	Valve
FL-08	OCCUPIED	55.6 %	0.55 IWC 0.59 IWC	56.2 °F	73.9 °F	39.6
FL-07	OCCUPIED	56.3 %	0.49 IWC 0.56 IWC	56.9 °F	74.6 °F	29.2
FL-06	OCCUPIED	53.8 %	0.44 IWC 0.58 IWC	57.9 °F	74.2 °F	27.9
FL-05	OCCUPIED	57.8 %	0.11 IWC 0.53 IWC	63.0 °F	73.2 °F	21.5 %
FL-04	OCCUPIED	57.6 %	0.46 IWC	58.2 °F	73.4 °F	29.5 %
FL-03	OCCUPIED	56.3 %	0.51 IWC	60.0 °F	74.0 °F	22.6 %
FL-02	OCCUPIED	61.6 %	0.57 IWC	62.0 °F	74.3 °F	27.8 %
FL-01	OCCUPIED	60.3 %	0.46 IWC	62.0 °F	73.3 °F	25.2 %

Bldg 400

B600 CHW FLOW 291.6 GPM

MECH CLG READY
NONE
ON

B400 CHW FLOW 185.9 GPM

LOBOS AHU Resets at work.

OSA Temp is 83.6°F at 3:07 PM.

Average AHU Static pressure is around 0.50”.

Average Supply air temperature is 59.6°F.

Old setpoints were 1.5” to 2” and 48°F to 55°F.

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ESAC Case Study #2

- 900,000 SF facility
- Pre-modification on-peak cooling loads – 1,800 tons + partial storage TES system.
- AHU coils replaced with 28 degree TD coils
- 2,400 ton P/S plant upgraded to VFD-everything, and to 2,600 tons, with “Oversized” cooling towers.
- Chillers re-piped in series.
- TES system re-piped with new tank headers and plant piping – 350% more TES capacity.
- **New DDC control system installed with Load Based Optimization System logic.**

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ESAC Case Study #2, continued

- The owner added a 230,000 SF building, and removed an 800 ton constant speed chiller, and installed a new 1,420 ton VFD chiller, **piped in series at 0.37 kW per ton at full load.**
- **The 1,420 ton chiller uses less power than the 800 ton chiller it is replacing, while producing 78% more cooling capacity.**
- All 1,100,000 square feet is still being served by the now full storage TES system.

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ESAC Case Study #2, continued

- The project saved over $1,000,000 by not having to build a new chiller plant to serve the new building.
- The project reduced operating costs by an average of $500,000 per year.

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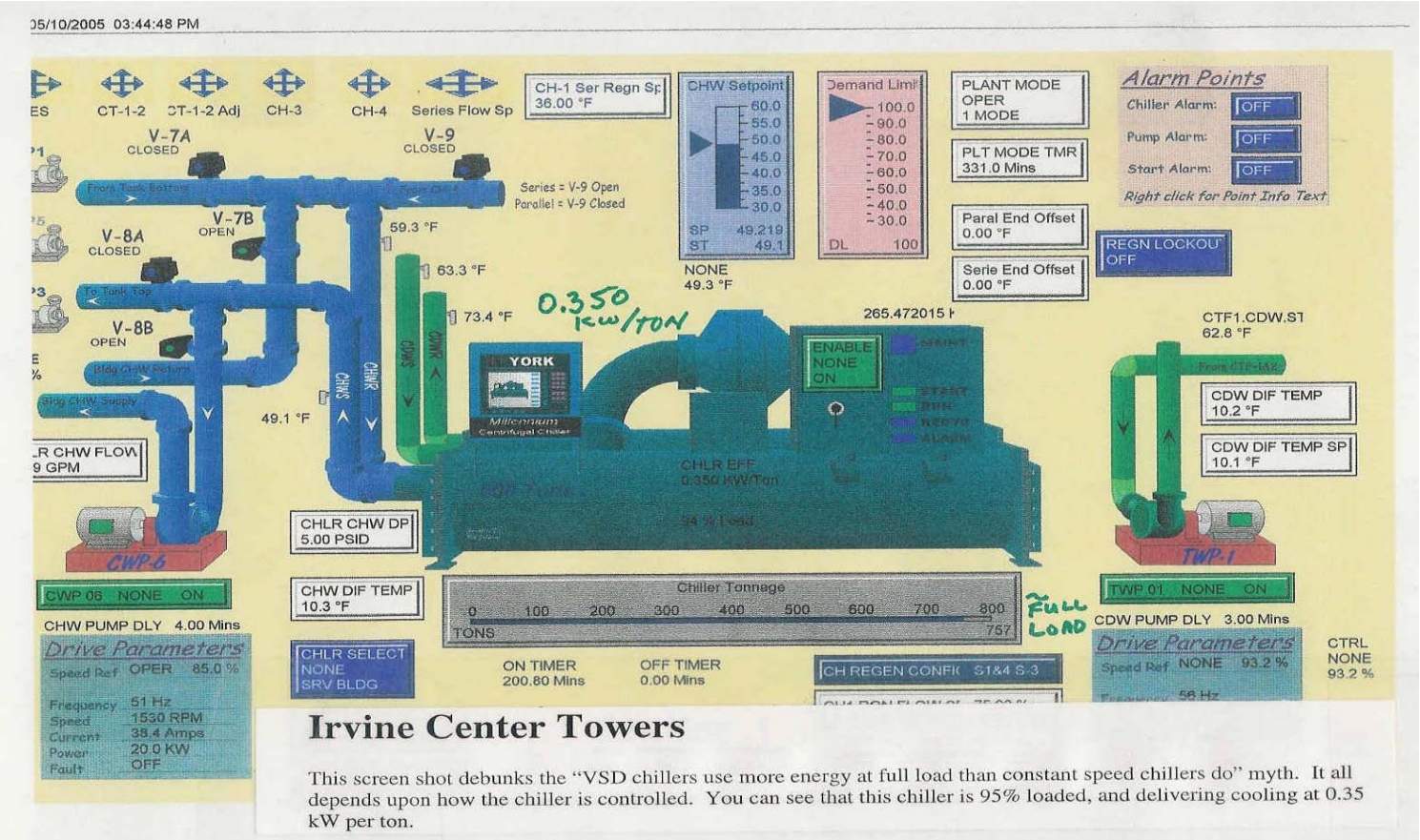
Fully Loaded VFD chiller @ 0.35 kW/Ton – VFD Chiller Efficiency is more related to lift than to load



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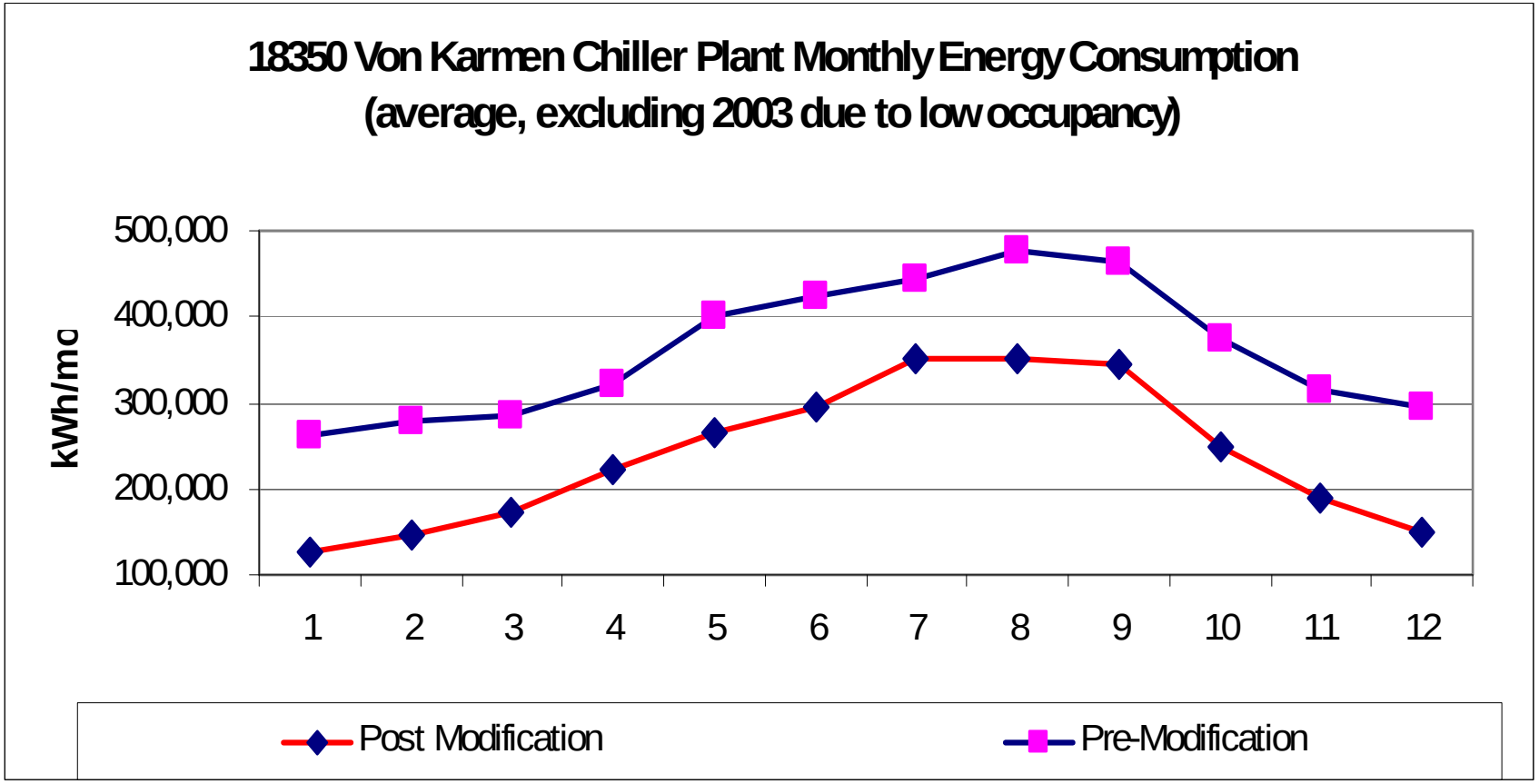
Monthly Average kWh Consumption for Pre and Post TES System Upgrade
35% Annual Savings (From Utility Meter)



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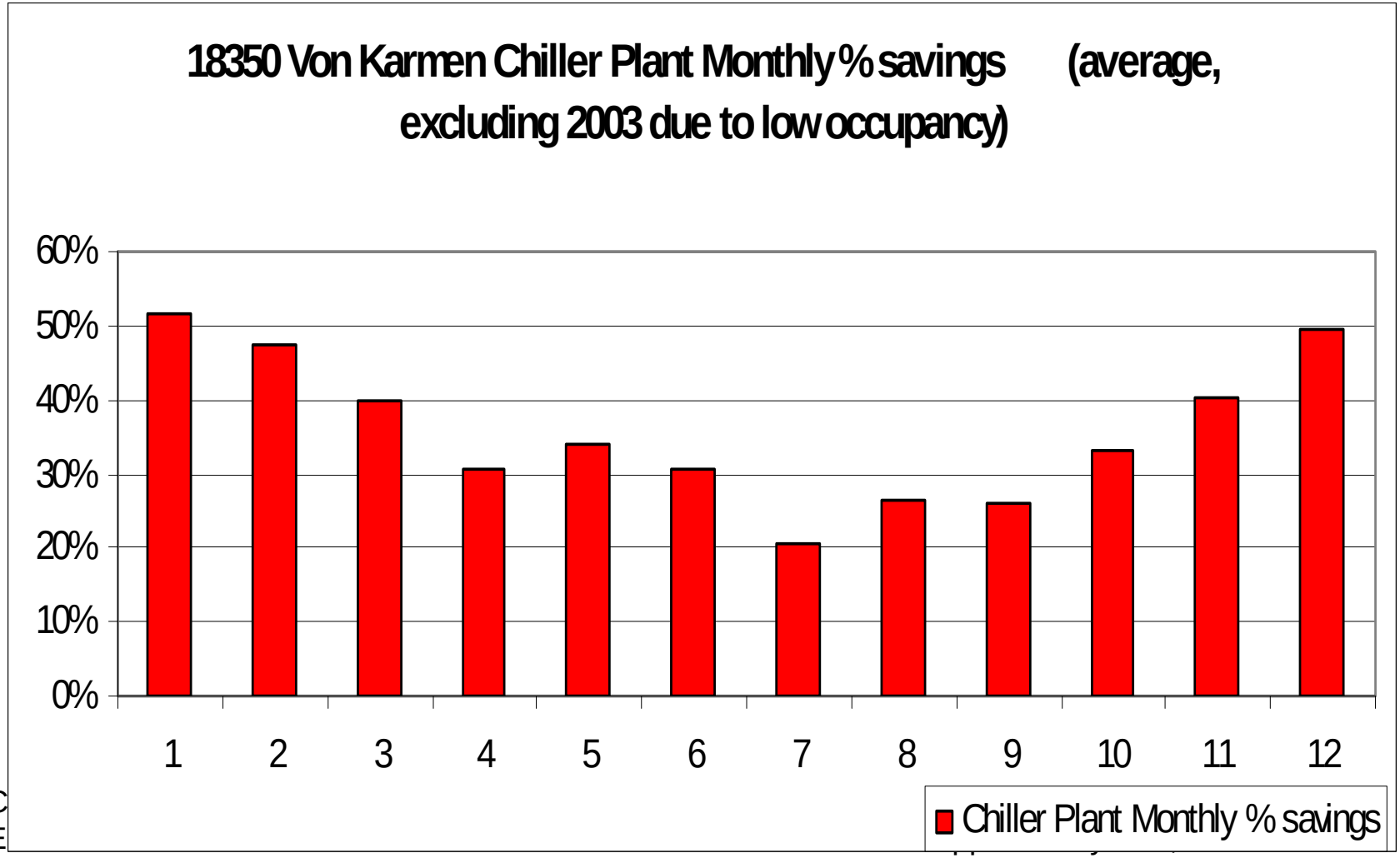
Average Monthly Energy Savings From TES Upgrade Project



Summary

- **Properly designed and applied TES systems can reduce site and source energy consumption and enable wind power to become a viable part of the energy infrastructure.**
- **Energy Efficiency, Demand Response and Demand Reduction Technologies are not mutually exclusive!**
- **You can start with the most efficient design possible, which reduces demand, then add demand response software and Thermal Energy Storage (TES) if/when appropriate.**
- **With TES, airside Demand Response controls can be added to further reduce site demand, over and above the central plant peak demand savings.**

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