# Energy Efficiency Baselines for

# DATA CENTERS

## PG&E's Non-Residential New Construction and Retrofit Incentive Programs

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# Contents

## Incentive Programs

PG&E's incentive programs are designed to help PG&E customers save energy by implementing energy efficiency measures. Many market sectors, such as residential and commercial, are served by well-established calculation methods. The industrial sector – in particular, high tech industrial facilities such as laboratories, cleanrooms and datacenters – are large consumers of energy, yet are poorly targeted by standard incentive calculations.

PG&E's incentive programs for high tech customers are designed to help the customer go beyond selection of incrementally more efficient components, and push designers and owners to consider new design strategies not normally offered in lowest-first-cost situations. Historically, datacenters have not received the same level of attention as commercial projects. This leaves ample opportunity to significantly reduce the energy budget for datacenter facilities by incorporating non-standard but well proven design strategies.

PG&E takes a customized design-assistance approach. Customers are invited to sign up for the Non-Residential New Construction (NRNC) or Non-Residential Retrofit (NRR) incentive program at the beginning of their high-tech project. PG&E hires a consultant with expertise in the customer's type of facility, who then meets with the customer's design team. Potential energy efficiency measures are explored. Any measure that saves energy is open for consideration. The consultant then analyzes the measures the customer wishes to pursue, estimating the annual energy savings, the implementation cost, and the PG&E incentive the customer is eligible for.

## High Tech Industrial Facilities

Datacenters, laboratories, and cleanrooms are referred to as "high tech industrial" spaces. These spaces are largely exempt from Title 24 compliance. For more definition of the term "datacenter", refer to Datacenter Definition on page 6.

The typical high tech industrial facility (i.e., the industrial space and the HVAC systems serving it) operates continuously – 24 hours/day, 7 days/week, 365 days/year – even if the space is not occupied by staff continuously.

Offices, conference rooms, auditoriums, cafeterias, restrooms and so forth are referred to as "commercial" space, and are governed by Title 24.

Buildings that contain industrial space often contain commercial space as well. Such buildings are referred to as "hybrid".

## Modeling Approach

Energy calculations for commercial space are typically performed using EnergyPro or an equivalent Title 24 performance compliance program. These programs take into account the construction of the building shell (roof, walls, glazing), solar orientation, and shading to determine the extent of HVAC loads due to outside conditions. These "shell loads" are often comparable to or even exceed the internal loads (people, lights, and office equipment).

The internal and process loads in high tech industrial spaces are typically far greater than any HVAC loads that migrate through the building shell. Additionally, the HVAC systems serving high tech industrial spaces are often not easily modeled in software programs designed for commercial spaces. As a result, the energy calculations for high tech industrial facilities typically ignore shell loads, and are typically performed with custom software packages.

Performing an energy analysis for a hybrid building usually requires a two-pronged modeling approach (EnergyPro or similar for the commercial space, and custom software for the industrial space.) Care must be taken to integrate the two analyses properly to avoid double-counting.

## Baselines

To determine the energy savings due to a particular energy efficiency measure, the proposed situation is compared to a baseline situation.

For commercial spaces in the PG&E service territory, the baseline is defined by California's Title 24 Non-Residential Building Energy Efficiency Standard.

This Title 24 standard does not apply to HVAC systems that serve "process loads."

From the 2005 Standard, Section 141 (c) 3. A. (page 79):

> 3. **Energy excluded**. The following energy shall be excluded:
> A. Process loads; and
> B. Loads of redundant or backup equipment, if the plans submitted under Section 10-103 of Title 24, Part 1, show controls that will allow the redundant or backup equipment to operate only when the primary equipment is not operating, and if such controls are installed; and
> C. Recovered energy other than from space conditioning equipment; and
> D. Additional energy use caused solely by outside air filtration and treatment for the reduction and treatment of unusual outdoor contaminants with final pressure drops more than one-inch water column. Only the energy accounted for by the amount of the pressure drop that is over one inch may be excluded.

There is some debate as to what constitutes a process load, and as to how far "downstream" in the HVAC system the exemption from the Title 24 standard applies. High tech industrial spaces are currently interpreted to be those spaces and associated infrastructure where the dominant design criteria are to satisfy the needs of processes rather than human comfort.

Some portions of the HVAC systems that serve industrial spaces, and are therefore exempt from Title 24, must still adhere to the American Society of Heating, Refrigeration, and Air Conditioning Engineer's (ASHRAE) Standard 90.1. For example, a chilled water plant delivering chilled water at a standard temperature must meet minimum performance criteria, regardless of the activity in the space being cooled.

This document does not intend to establish new baselines in any case where Title 24 and/or ASHRAE 90.1 have already defined them. Although Title 24 and ASHRAE 90.1 do not cover process load HVAC systems as a whole, both standards may have baselines that are appropriate to any individual component, such as equipment selection. In that case, designers are expected to abide by the relevant Title 24 or ASHRAE 90.1 baseline. This document only aims to provide baselines where neither Title 24 nor ASHRAE 90.1 gives guidance.

The starting point for defining high tech industrial baselines is to determine what current standard design and operating practice is for these types of facilities in the PG&E service territory. These practices are a moving target, particularly in the high tech industry. Periodic research is needed to update the descriptions of standard practice.

Rumsey Engineers has conducted much of this research. In some instances we have performed direct benchmarking measurements in the field. We indicate where this is the case. In other instances we have conducted literature searches or phone surveys. Finally, we rely on our extensive experience as mechanical designers in the high tech industrial field to assess current standard practice.

The second step in defining industrial baselines is to set the bar slightly higher than current typical practice (in the same way that the Title 24 standard pushes the envelope of typical commercial construction.) This is intended to encourage adoption of new technologies and operating strategies in addition to selecting more energy efficient system components. In general, the incentive programs are designed to reflect trends in design practice and recognize those designs that are above the norm in terms of energy performance.

Energy savings for a given measure are determined by comparing the estimated energy use of the proposed new system to the estimated energy use of a baseline system that serves the same load.

HVAC equipment operating efficiencies depend not only on technology of the devices used in their construction but also the fluid temperatures and flow rates involved in the heat transfer process. For instance a water-cooled chiller, regardless of compressor technology, will transfer heat more efficiently as the chilled water temperatures approach the condenser water temperatures. ARI Standards prescribe test conditions under which

HVAC equipment efficiencies are determined. The standard test conditions allow for equipment to be rated uniformly, enabling fair comparisons.

California Title 24 and ASHRAE 90.1 energy efficiency standards both mandate minimum non-residential air conditioner and electric water chiller efficiencies with respect to ARI Standards 340/360 and 550/590, respectively. The test conditions under these ARI standards, however, do not necessarily coincide with typical operating conditions for these HVAC equipment in PG&E territory for high-tech and bio-tech facilities. When there is a difference between ARI standard test conditions and typical operating conditions for air conditioners or electric chillers, the minimum equipment efficiency mandated by California Title 24 is used to establish one point on an equipment efficiency curve but this curve is then modified, via DOE 2 default efficiency formulas, to construct the baseline efficiency curve at typical operating conditions in PG&E territory.

## *Metrics*

We define baselines not just for equipment efficiency, but overall system efficiency. Incentives are based on the degree to which the entire proposed system (the air delivery system, the cooling system, humidity control system, etc.) out-performs its baseline counterpart. The federal EPA is currently working to develop metrics for data center energy usage. Meanwhile, this baseline document serves to encourage data center energy efficiency improvement on a project-by-project basis.

## *Title 24 Equipment*

Unless otherwise stated, we follow Title 24 standards for the efficiency requirements of individual pieces of equipment (DX cooling units, chillers, boilers, etc.) High tech industrial spaces generally have stringent requirements for safety, redundancy, control of space temperature & humidity, and so forth, but these requirements can usually be met with off-the-shelf HVAC equipment.

## *Retrofit Program*

In the baseline case for retrofit projects, all equipment should be assigned a baseline efficiency as defined by this document, with the exception of equipment that meets Early Retirement criteria as defined by the NRR Procedures Manual. The Early Retirement Program feature is intended to accelerate the retirement of less efficient equipment by offering increased incentives (see NRR Procedures Manual, p1-9).

For equipment that meets Early Retirement criteria:

> The baseline case can either incorporate the equipment's existing efficiency or baseline efficiency.

If the existing equipment efficiency is worse than baseline efficiency (which is the assumption of the NRR Procedures manual) modeling the existing efficiency in the baseline case will increase the calculated savings and incentive.

If the existing equipment efficiency is better than baseline efficiency, modeling the existing efficiency in the baseline case will decrease the calculated savings and incentive. This case is not accounted for in the NRR Procedures Manual. In this case PG&E recommends assigning a baseline efficiency in the baseline case, as would be done if the equipment was not eligible for Early Retirement.

In the baseline case for retrofit projects, all equipment should be modeled with the existing operating setpoints (temperatures, flow rates, etc.) and configurations (quantity and capacity of equipment).

## Incentives

The incentives offered by PG&E under their NRNC and NRR programs are directly proportional to the estimated electric energy (kWh) and gas energy (therm) savings in the first five years of operation. The rates vary from year to year. In 2009, the NRNC rates are:

| | Electric | Gas |
|---|---|---|
| | $/kWh | $/therm |
| Daylighting Systems | 0.04 | |
| Lighting Systems (Interior and Outdoor) | 0.05 | |
| HVAC Systems | 0.15 | 1.00 |
| Service Hot Water | | 1.00 |
| Process Systems | 0.09 | 1.00 |

In 2009, the NRR rates are:

| | Electric | Gas |
|---|---|---|
| | $/kWh | $/therm |
| Lighting Systems (including daylighting) | 0.05 | |
| Air Conditioning and Refrigeration I, (Includes Datacenter Economizers) | 0.15 | |
| Air Conditioning and Refrigeration II, (Includes non-Datacenter Economizers) | 0.09 | |
| Motors and Other Equipment, including Process Systems | 0.09 | |
| Natural Gas, including Service Hot Water | | 1.00 |

Additionally:

- There is an incentive of $100 per kW of peak electric power reduction.
- There is no minimum required threshold of energy savings.
- The total incentive for any one project may not exceed $500,000 in the case of New Construction, and may not exceed $3,600,000 in the case of Retrofit.
- In the case of New Construction, the total incentive for any one project may not exceed 50% of the *incremental* implementation cost of all the energy efficiency measures adopted for the project. Labor costs can be included in the implementation cost.
- In the case of Retrofit, the total incentive for any one project may not exceed 50% of the *total* implementation cost of all the energy efficiency measures adopted for the project. Labor costs can be included in the implementation cost.
- There is no incentive provided for fuel switching (e.g., installing an absorption chiller in place of an electric chiller).
- There is no incentive provided for cogeneration systems.
- Adjustments to the incentive may be applied if a measure increases energy use of one type while decreasing energy use of another type. (Example: A heat recovery system that saves gas heating energy but increases electric fan energy.)

## Format of this Document

This is a living document, to be updated as more data is gathered on actual field operations, as technologies evolve, and as new energy efficiency measures are considered. Suggestions for possible enhancements of selected baseline values appear in several places in the document.

### *Categories*

The baselines presented in this document are arranged in the following categories:

- Loads
- Redundancy
- Space Design Conditions
- Air Delivery Systems
- Hydronic Systems
- Cooling Systems
- Heating Systems
- Humidity Control Systems
- Process Systems

### *Subcategories*

Each category is further divided as follows:

**System Configuration**
Description of the baseline system, what components it contains, how it is generally operated.

**System Efficiency Metric**
What is the baseline operating efficiency of the entire system?

**Economizing & Heat Recovery**
Are there any economizing or heat recovery schemes that are considered baseline?

**Pressure Drop**
Baseline pressure drops for air delivery and hydronic systems.

**Component Efficiency**
Baseline efficiencies for individual system components.

**Thermal Storage**
Are there any thermal storage components that are considered baseline?

**Control Sequences**
Baseline methods of system control.

# Datacenter Definition

Datacenters are spaces specifically designed to accommodate dense arrangements of computer equipment. This currently includes telephone company "central offices" or "telcos", and computer labs. Any space where dedicated HVAC is installed to handle computing equipment load is likely to be considered a datacenter.

## *Size Categories for Datacenters*

**Small Datacenters**

These are facilities provided with up to and including 360 tons total installed capacity, not including any cooling system redundancy.

**Large Datacenters**

These are facilities provided with more than 360 tons total installed capacity, not including any cooling system redundancy.

# Loads

The baseline cooling load in a datacenter is set equal to the proposed cooling load. If the proposed load is expected to start at less than the full capacity load and grow over time, the estimated average load during the first 5 years of operation is used.

# Redundancy

- We use the Uptime Institute's definitions of redundancy Tiers. See the white paper "Tier Classifications Define Site Infrastructure Performance", by Turner, Seader, & Brill, 2006, available from www.upsite.com.
- The baseline datacenter is modeled at the same redundancy tier as the proposed datacenter is designed to.
- If the redundancy level of the proposed datacenter is not stated by the customer, it is assumed to be Tier II.
- For N+1 and N+2 redundancy, we apply the requirement to every 10 CRAC units. For example, if an N+0 requirement resulted in 50 units, then N+1 would result in 55 units and N+2 would result in 60 units.
- For information on control of redundant fans and pumps, see Air Delivery Systems\Control Sequences and Hydronic Systems\Control Sequences, respectively.

## Space Design Conditions

There are three baseline air management schemes for datacenters, depending on the actual load density of the IT equipment.

### Baseline Air Management Schemes

| ID | Name | Supportable Load Density | | Return Air Drybulb Temp. Setpoint | Supply Air Temp. | Airside Delta-T | Location of RH Sensor | Humidity Control Range | | | Drybulb Temp. at RH Sensor | Average Temp. of Air Entering IT Equip. | Temp. Rise of Supply Air Before Entering IT Equip. | Dew-point Temp. at RH Sensor | Total Static Pressure |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Min | Max | | | | | Min DP | Max RH | Assumed RH Setpoint and Tolerance | | | | | |
| | | W/sf | W/sf | F | F | F | | | | | F | F | F | F | in. w.g. |
| I | Hot Aisle/Cold Aisle, Open | 0 | 100 | 70 | 62 | 8 | Return Air Opening of CRAC | 41.9 | 60% | 50% +/- 10% | 70 | 65 | 3 | 51 | 1.6 |
| II | Hot Aisle/Cold Aisle, Ducted Return | 101 | 220 | 74 | 64 | 10 | Cold Aisle | 41.9 | 60% | 50% +/- 10% | 66 | 66 | 2 | 47 | 1.9 |
| III | Hot Aisle/Cold Aisle, Fully Enclosed | 221 | 400 | 82 | 67 | 15 | Cold Aisle | 41.9 | 60% | 50% +/- 10% | 68 | 68 | 1 | 49 | 2.4 |

Notes

Air Management Scheme III: A fully enclosed cold aisle scheme is assumed to be identical to a fully enclosed hot aisle scheme, from the standpoint of temperatures, humidity, and total static pressure drop.

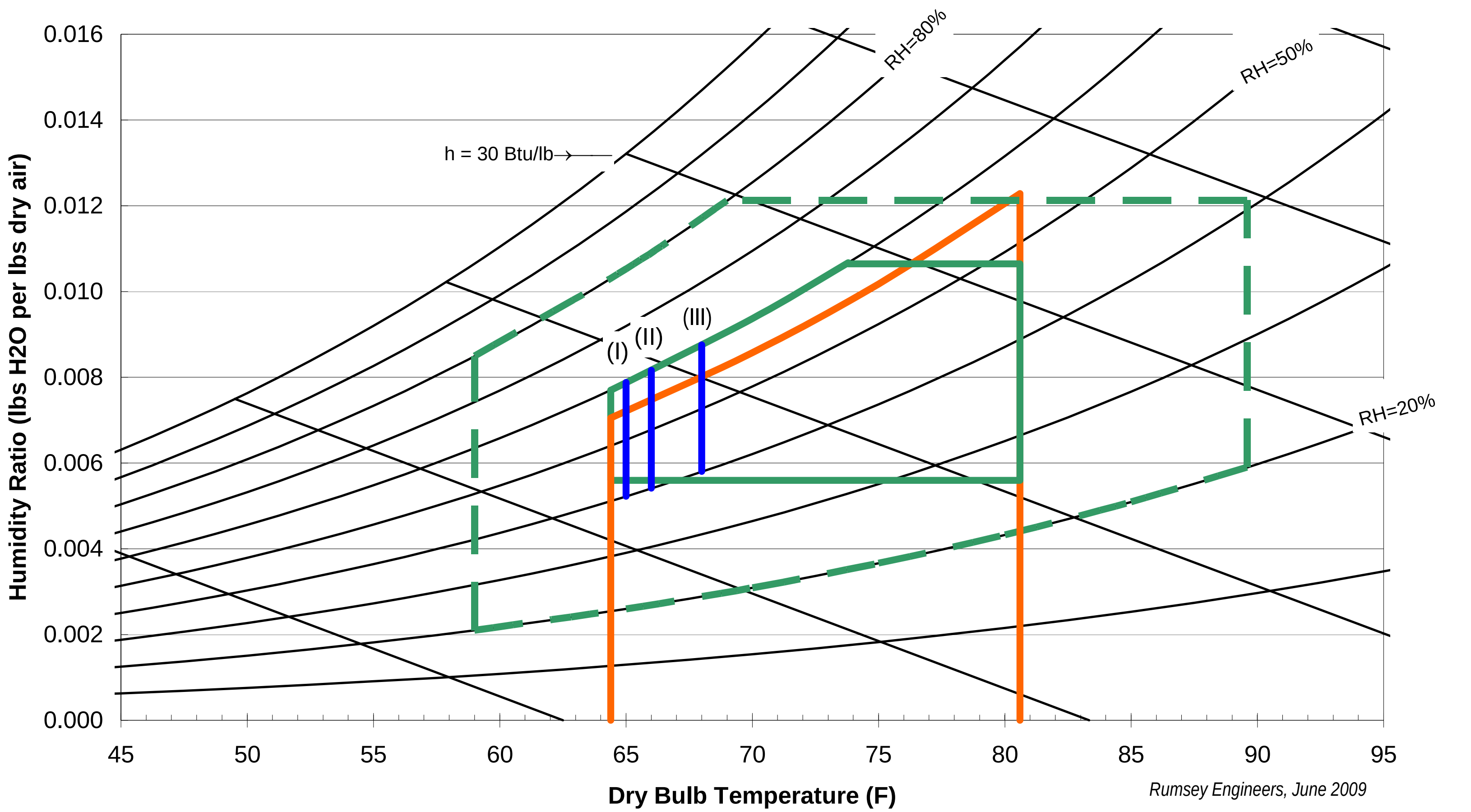
Load Density is actual density, not design density, and is based on total data center floor area (not rack footprint).

Load densities above 400 W/sf (actual) are assumed to not be successfully addressed by any of these air management schemes.

Location of RH Sensor: For CRAC units with humidity control capability, the RH sensor is typically located in the return air opening of the CRAC. In Scheme I, we assume the RH sensor is left in this location. In Schemes II and III we assume the RH sensor is relocated to the cold aisle.

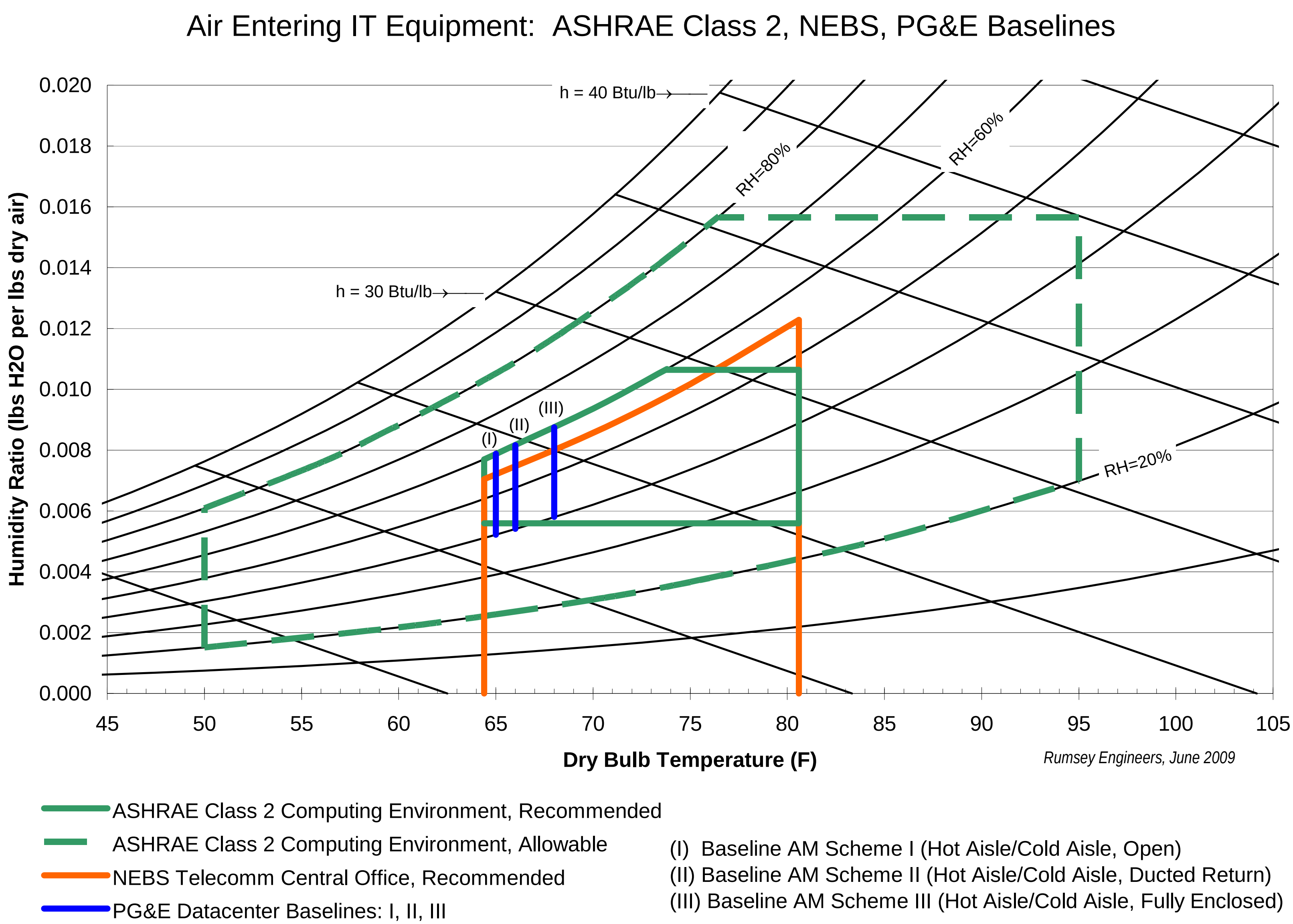
Humidity Control Range: "Thermal Guidelines for Data Processing Environments, Second Edition", ASHRAE, 2009. The minimum dewpoint temperature and maximum relative humidity shown in this table is the "Recommended" range for Class 1 and 2 facilities. The values apply to the air entering the computer equipment. We assume that baseline facilities employ RH sensors, not dewpoint sensors. We assume the relative humidity setpoint and tolerance are set as shown in the table. The humidity control range in this table does not apply to telecomm facilities. See the following psychrometric charts.

## Air Entering IT Equipment: ASHRAE Class I, NEBS, PG&E Baselines




ASHRAE Class 1 Computing Environment, Recommended
ASHRAE Class 1 Computing Environment, Allowable
NEBS Telecomm Central Office, Recommended
PG&E Datacenter Baselines: I, II, III

(I) Air Management Scheme I (Hot Aisle/Cold Aisle, Open)
(II) Air Management Scheme II (Hot Aisle/Cold Aisle, Ducted Return)
(III) Air Management Scheme III (Hot Aisle/Cold Aisle, Fully Enclosed)


Air Entering IT Equipment: ASHRAE Class 2, NEBS, PG&E Baselines
Humidity Ratio (lbs H2O per lbs dry air)
0.020
0.018
0.016
0.014
0.012
0.010
0.008
0.006
0.004
0.002
0.000
45
50
55
60
65
70
75
80
85
90
95
100
105
Dry Bulb Temperature (F)
h = 40 Btu/lb
h = 30 Btu/lb
RH=80%
RH=60%
RH=20%
(I)
(II)
(III)
Rumsey Engineers, June 2009
ASHRAE Class 2 Computing Environment, Recommended
ASHRAE Class 2 Computing Environment, Allowable
NEBS Telecomm Central Office, Recommended
PG&E Datacenter Baselines: I, II, III
(I) Baseline AM Scheme I (Hot Aisle/Cold Aisle, Open)
(II) Baseline AM Scheme II (Hot Aisle/Cold Aisle, Ducted Return)
(III) Baseline AM Scheme III (Hot Aisle/Cold Aisle, Fully Enclosed)

**Ventilation**
In the CA Title 24 Non-Residential Alternative Calculation Method (ACM) Manual for 2008, Table N2-5 (page 2-34) lists minimum outside air ventilation rates for a variety of occupancy types but does not include datacenters. We consider datacenters as part of the "All Others" occupancy type, for which CA Title 24 requires 0.15 cfm/sf, and we use it as the baseline.

**Exhaust**
Exhaust volume is assumed to be same as ventilation volume (no in/exfiltration in space).

**Baseline Occupancy**
Human occupancy is assumed to add a negligible load to the HVAC system that serves the datacenter. The IT equipment load is assumed to be present 24/7/365.

# Air Delivery Systems

## *System Configuration*

### Recirculation

**Small Datacenters**
Recirculation is provided by air-cooled DX CRAC units equipped with constant-speed fans. Manufacturer's specifications for the Liebert Deluxe System 3 series of DX CRAC units are used as the baseline. The nominal capacity of the unit is selected based on IT equipment load density as shown in the table on the following page.

*Exception 1*: For an existing or new small datacenter in an existing building served by a chilled water plant, and the plant has sufficient cooling and pumping capacity, and running new CHW pipes to the datacenter is feasible, then the baseline consists of tapping in to the existing CHW system to run CHW CRACs with constant-speed fans.

*Exception 2*: For telecomm facilities, chilled water air handlers may be considered baseline practice where feasible. The baseline TSP (at peak design airflow) is ___.

**Large Datacenters**
Recirculation is provided by chilled water CRAC units equipped with constant-speed fans. Manufacturer's specifications for the Liebert Deluxe System 3 series of chilled water CRAC units are used as the baseline. The nominal capacity of the unit is selected based on IT equipment load density as shown in the table on the following page.

*Exception*: For telecomm facilities, chilled water air handlers may be considered baseline practice where feasible. The baseline TSP (at peak design airflow) is ___.

## Baseline CRAC Units

| 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Technology Type | Load Density | Liebert CRAC Unit | Sensible Cooling Capacity (tons) at Various Return Air Drybulb/Dewpoint Temps (deg F) | | | Nominal Air Flow | Nominal External Static Pressure | Nominal Total Static Pressure | Supply Fan Motor Nominal Size | Supply Fan Motor Efficiency | Supply Fan Drive Efficiency | Supply Fan Efficiency |
| | W/sf | | 72/52 (AM I) | 80/47 (AM II) | 90/49 (AM III) | cfm | In. w.g. | in. w.g. | hp | | | |
| Air-Cooled DX | <= 40 | DH/VH125A | 9.1 | | | 5,650 | .3 | 0.610 | 2 | 87.5% | 95% | 41% |
| Air-Cooled DX | > 40 | DH/VH380A | 24.6 | 29.5 | 31.3 | 15,200 | .3 | 1.660 | 10 | 91.7% | 95% | 57% |
| CHW | <= 100 | FH/UH600C | 28.1 | | | 17,100 | .3 | 1.550 | 10 | 91.7% | 95% | 60% |
| CHW | > 100 | FH/UH740C | | 47.2 | 58.8 | 16,500 | .3 | 1.600 | 10 | 91.7% | 95% | 60% |

The data in columns 3, 4, 7, 8, and 10 was drawn from Liebert specifications for their Deluxe System 3 series. (Standard fan in each case.)
We use Liebert units to represent baseline as they appear to have the largest share of the data center "precision cooling" market.
Columns 4, 5, and 6 apply to air management schemes **I**, **II**, and **III**, respectively. The gray cells indicate combinations of CRAC model and air management scheme that are intended not to occur in a baseline case.
Column 1: The current data center baseline assumes air-cooled DX CRAC units for small data centers, and CHW CRACs for large datacenters, so we include only those two types of CRAC unit in this table.
Column 2: We assume that the nominal cooling capacity of the CRAC units selected for a given data center is based on the full build-out load density.
Column 3: Standard L:iebert models.
Columns 4, 5, 6: CHW units: The CHW supply temperature is assumed to be 45 F in all cases. The cooling capacities are assumed to be net, after fan motor heat has been accounted for.
Columns 5 and 6: These are extrapolated from the Liebert specs.
Column 7: From the Liebert specs.
Column 8: From the Liebert specs.
Column 9: This is based on the information in columns 7 and 8, and estimates of fan system efficiency (see columns 11, 12, 13).
Column 10: All units have only one supply fan motor.
Column 11: Estimated.
Column 12: Liebert specs indicate non-cogged V-belt drives on all models. This model assumes constant 95% efficiency for such drives.
Column 13: Estimated.

## Ventilation

In hybrid facilities, ventilation for the datacenter is often provided by the “house air” system; ie, the system that serves the commercial space. However, to simplify energy calculations we define the baseline ventilation air system as a dedicated MUAH equipped with a constant-speed fan. Excess air is assumed to be removed from the space via a constant-speed exhaust fan.

The MUAH is assumed to deliver 0.15 cfm/sf at a drybulb temperature of 55F. Energy is required to temper the outside air, and this air provides a cooling effect that supplements the main cooling system. This tempering energy and the cooling effect are addressed in the Cooling Systems section.

## Exhaust

The exhaust system is not equipped with any heat recovery devices.

### *System Efficiency Metric*

**The operative metric for air delivery systems is the volumetric flow rate of air being delivered divided by the fan motor power, in units of cubic feet per minute per kilowatt, or cfm/kW.**

For a given volumetric air flow rate, the efficiency of the air delivery system is dictated both by the total static pressure drop (TSP) of the system, and by the efficiency of the fan system (the fan/drive/motor combination).

Baseline values for TSP and for the fan system are defined below, but exceeding baseline practice for one of these two aspects does not necessarily provide better-than-baseline overall system efficiency. It is the combination of the two aspects that determines the resulting value of cfm/kW.

The baseline cfm/kW value for the ventilation, recirculation, and exhaust systems are currently being studied further and will be included in future releases of this baseline document.

### *Pressure Drop*

The total static pressure drop of the air delivery system is the sum of the pressure drops of the components that make up the system while under peak design airflow conditions – the filters, coils, fans, duct system, silencers, dampers, grills, and any other devices the air flows through.

Baseline pressure drops are defined below for many of these components, but it is the total static pressure drop that influences the efficiency of the air delivery system. In other words, reducing the pressure drop below the baseline value for just one or two components of the system does not necessarily provide a better-than-baseline TSP. Note that the following values apply only at the peak design airflow.

## Ventilation

**MUAH Face Velocity**

The baseline MUAH coil face velocity is 500 fpm. This is a long-standing design rule of thumb. Reducing the face velocity decreases the fan energy required to deliver a given air volume.

**MUAH Internal Static Pressure Drop**

The internal pressure drop of a baseline MUAH (total of all components) is 2.0”.

**Ventilation Duct Static Pressure Drop**

The baseline duct static pressure drop is 0.10” w.g. per 100 ft. This is also a long-standing design rule of thumb.

## Recirculation

The baseline total static pressure drop for the three baseline air management schemes involving DX CRACs and chilled water CRACs are shown in the rightmost column of the Baseline Air Management Schemes table on Page 9.

## Exhaust

The baseline exhaust path pressure drop is 1.0” w.g. for the exhaust duct, up to and including a vertical run of 3 floors. We add 0.5” for every additional floor beyond a vertical run of 3 floors.

## *Component Efficiency*

### Fans

| Nominal Fan Motor Horsepower | Baseline Efficiency |
|---|---|
| 1 | 0.145 |
| 1.5 | 0.270 |
| 2 | 0.345 |
| 3 | 0.440 |
| 5 | 0.500 |
| 7.5 | 0.533 |
| 10 | 0.556 |
| 15 | 0.587 |
| 20 | 0.608 |
| 25 | 0.624 |
| 30 | 0.638 |
| 40 | 0.658 |
| 50 | 0.675 |
| 60 | 0.686 |
| 75 | 0.698 |
| 100 | 0.715 |
| 125 | 0.727 |
| 150 | 0.736 |
| 200 | 0.750 |

These fan efficiency values apply to all the fans -- MUAH, CRAC and exhaust.

### Fan Drives

Baseline: V-shaped belt drive, non-cogged. 95% average belt lifetime efficiency.

| *Drive Type* | *Average Lifetime Efficiency* |
|---|---|
| V-shaped belt drive, non-cogged | 95% [1] |
| V-shaped belt drive, cogged | 98% [1] |
| Direct drive | 100% |

[1] US Department of Energy Industrial Technologies Program (2008, September). *Motor Systems Tip Sheet #5.*

### *Control Sequences*

#### Recirculation

Constant speed fans, balanced at startup, run 24/7.

#### Ventilation

Baseline is a constant ventilation rate (no demand-controlled ventilation).

#### Exhaust

Constant exhaust rate.

#### Redundant Fans

For air delivery systems with redundant fans, the baseline model assumes the redundant fans are never needed and never run. The energy savings calculations do not model failure events of fans.

## Hydronic Systems (Chilled Water, Condenser Water)

### *System Configuration*

The baseline chilled water pump configuration is a constant flow primary loop with constant-speed primary pump motors, and a variable flow secondary loop. ASHRAE 90.1, section G3.1.3.10 states that a VFD on the secondary pump is considered baseline only for facilities of more than 120,000 sf. In this case, the pump speed is controlled to maintain a constant pressure delta at a far point in the secondary loop. For facilities of 120,000 sf or less, the secondary pump is assumed to be constant speed and to “ride its curve” as the CHW valves open & close. We adopt ASHRAE’s definition for use as the baseline.

Triple duty valves are considered baseline. Removal of TDVs is incentable, if the removal results in higher-than-baseline system efficiency.

The baseline condenser water pump configuration is one constant speed condenser water pump dedicated to each chiller.

### *System Efficiency Metric*

**The baseline total chilled water pumping energy at peak design load is 0.044 kW/ton.**

## *Pressure Drop*

Water distribution system total head pressure: This is currently being studied further and will be included in future releases of this baseline document.

## *Component Efficiency*

### Pumps

| | | Head (ft) | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | 20 | | 40 | | 60 | | 80 | | 100 (%) | |
| | | hp | eff (%) | hp | eff (%) | hp | eff (%) | hp | eff (%) | hp | eff (%) |
| GPM | 100 | 1 | 58 | 2 | 69 | 5 | 59 | 5 | 51 | 7.5 | 56 |
| | 500 | 5 | 72 | 7.5 | 84 | 15 | 79 | 20 | 71 | 20 | 75 |
| | 1000 | 7.5 | 79 | 15 | 79 | 25 | 80 | 30 | 84 | 40 | 86 |
| | 1500 | 15 | 73 | 25 | 77 | 40 | 80 | 50 | 80 | 60 | 85 |
| | 2000 | 15 | 76 | 40 | 62 | 50 | 79 | 60 | 82 | 75 | 81 |
| | 2500 | | | 40 | 77 | 75 | 63 | 100 | 65 | 100 | 72 |
| | 3000 | | | 50 | 73 | 75 | 75 | 100 | 82 | 100 | 86 |
| | 3500 | | | 60 | 70 | 100 | 71 | 100 | 81 | 125 | 84 |
| | 4000 | | | 60 | 78 | 100 | 74 | 125 | 78 | 150 | 81 |
| | 4500 | | | 75 | 73 | 125 | 72 | 125 | 84 | 150 | 86 |
| | 5000 | | | 100 | 75 | 125 | 72 | 150 | 77 | 200 | 81 |

The above table is considered baseline practice for pump efficiency based on system pressure drop and flow rate. The table was derived by selecting the least expensive option for a given condition from the Bell & Gossett product selection software. All selections in the table are sized not to exceed 90% of the rated power at the given condition.

For baseline facilities, pump selections are tailored to the project via impeller trimming.

### Pump Motors

Baseline motor efficiencies are tabulated in the Electrical section on page 27.
Baseline pump motors are not equipped with VFDs, except the secondary CHW pump motors as described elsewhere in this document.

## *Control Sequences*

If the secondary pumps are equipped with VFDs, it is baseline practice to control the pump speed to maintain a constant differential pressure setpoint.

For hydronic systems with redundant pumps, the baseline model assumes the redundant pumps are never needed and never run.  The energy savings calculations do not model failure events of pumps.

It is baseline practice to stage the condenser water pumps on and off with the chiller they serve.  Baseline condenser water pumps do not run when their associated chiller is off.

For datacenters without air-side or water-side economizing, with a continuous non-zero cooling load, served by CHW, the baseline is that the CHW pumps run continuously.

For datacenters served by a CHW plant that is expected to experience periods of zero load due to air-side economizing, the baseline is that all CHW pumps turn off when the outside air drybulb temperature is 5 deg F or more below the required supply air drybulb temperature, and turn on when the outside air drybulb temperature is higher than this.

For datacenters served by a CHW plant that is expected to experience periods of zero load due to water-side economizing, the baseline will vary depending on the particular water-side economizer implementation.  At least one CHW pump and one CW pump will need to keep running.  But the CHW pumps and CW pumps that serve the chillers are assumed to turn off when the cold condenser water temperature is 5 deg F or more lower than is needed to serve 100% of the cooling load, and turn back on when the cold condenser water temperature is higher than this.

## Cooling Systems

**Baseline cooling system efficiency varies by system type (DX, air-cooled chiller, water-cooled chiller), and by system capacity. In all cases the efficiency is expressed in units of annual average kW/ton – the annual combined electric energy use (kWh/yr) of all fans, compressors, and other system components that help provide cooling, divided by the annual ton-hours of cooling delivered. Unless otherwise stated, the incentive for a cooling system energy efficiency measure (or suite of measures) is based on the degree to which the measure(s) improve upon the baseline.**

Substituting one or more heat-driven chillers (e.g., absorption chillers) in an effort to reduce the kW/ton efficiency of the proposed system compared to baseline does not qualify for an incentive.

There are many different aspects of cooling systems that can be addressed to improve efficiency. Baseline practice for many of these aspects are defined below, but exceeding baseline practice for one or more of these aspects does not necessarily provide better-than-baseline overall system efficiency. It is the resulting overall kW/ton efficiency of the cooling system that determines the resulting savings and incentive.

### *System Configuration*

**Small Datacenters**

The baseline cooling system for datacenters up to and including 360 tons total installed capacity (not including any cooling system redundancy) is uniformly-sized air-cooled DX, constant-speed fan CRAC units.

Exception: For an existing or new small datacenter in an existing building served by a chilled water plant, and the plant has sufficient cooling and pumping capacity, and running new CHW pipes to the datacenter is feasible, then the baseline consists of tapping in to the existing CHW system to run CHW CRACs (or CHW air handler units for telecomm facilities) with constant-speed fans.

**Large Datacenters**

The baseline cooling system for datacenters above 360 tons total installed capacity (not including any cooling system redundancy) is a water-cooled chilled water plant serving uniformly-sized CHW CRAC units (or CHW air handler units for telecomm facilities) equipped with constant-speed fans.

## Air-Cooled DX CRAC Units

The baseline air-cooled DX CRAC configuration has:

- Redundancy = N+1, if not otherwise specified in the proposed design.
- Safety factor on capacity = 98% design condition * 1.20.
- All units are equally sized.
- Condensers are air-cooled. Add-on evaporative cooling devices for condenser coils are not baseline.

## Water-Cooled Chilled Water Plant

The baseline chilled water plant has:

- No CHW storage.
- No water-side economizing (aka "free cooling")
- Redundancy = N+1 on chillers, cooling towers, and pumps, if not otherwise specified in the proposed design.
- Safety factor on capacity = 98% design condition * 1.20.
- All chillers are identical.
- Idle chillers are staged on after operating chillers exceed 80% load factor
- The cooling load is assumed to be shared equally among all active chillers.
- Baseline chillers are electric (not absorption or adsorption).
- Baseline chiller performance is displayed in the table: *Chilled Water Plant Performance (kW/ton) vs Load Factor,* which is presented later in this document.

Electric chiller technology type (screw, scroll, centrifugal; constant-speed vs variable speed; etc.) tends to vary with capacity, but PG&E's incentive program does not dictate technology type. If a chiller of any technology type can be shown to produce annual energy savings over the defined baseline chiller in its capacity class – and using the same fuel -- then it is eligible for an incentive.

## All CRACs

The baseline number of CRAC units is determined by finding the maximum of:

- the number of CRACs needed to meet the airflow requirement
- the number of CRACs needed to meet the cooling requirement with a safety factor
- the number of CRACs needed to meet the cooling requirement with redundancy

(CRAC redundancy is not addressed in the Uptime Institute's definition of Tier II. The CRAC redundancy level discussed here is to address general mechanical reliability.) The baseline quantity and size of CRACs are calculated for the expected build-out load. The table on page 13 shows baseline CRAC units.

**Number of CRACs Needed to Meet Airflow Requirement**

This is determined by first establishing the baseline air management scheme based on the proposed build-out rack load density. This in turn determines the nominal baseline supply and return air temperatures. These temperatures and the proposed total load are used to calculate the needed cooling airflow. The nominal baseline CRAC size is selected based on the proposed load density. The air that the CRAC unit can deliver is dependent on the expected total static pressure drop it will see with the given air management scheme. Once the airflow per CRAC is determined, the number of CRACs needed to meet the airflow requirement is calculated by taking the total cooling air flow requirement divided by the airflow per CRAC and rounding up to the nearest integer.

**Number of CRACs Needed to Meet Cooling Requirement**

This is determined first establishing the baseline air management scheme based on the proposed rack load density. This in turn determines the nominal baseline return air temperature. The nominal baseline CRAC size is selected based on the proposed load density. The CRAC cooling capacity is dependent on the return air temperature. (The cooling capacity increases with the return air temperature, all other factors held equal.) Once the cooling capacity per CRAC is determined, two values are calculated:

1. The number of CRACs needed to meet the cooling requirement with a safety factor. This is calculated by dividing the expected maximum cooling load times a safety factor by the cooling capacity per CRAC and rounding up the result to the nearest integer.

2. The number of CRACs needed to meet the cooling requirement with redundancy. This is calculated by taking the number of CRACs needed to meet the cooling requirement *without* a safety factor, then applying the redundancy requirement to every ten CRAC units.

## *System Efficiency Metric*

The operative metric for baseline datacenter cooling systems is annual average kW/ton, as defined at the beginning of this section. If the ventilation system provides cooling that supplements the main cooling system, it is included in the kW/ton calculation.

## Air-Cooled DX CRAC Units

The cooling efficiency of baseline air-cooled DX CRAC units is described below under Component Efficiency.

## Water-Cooled Chilled Water Plant

If the entire CHW plant (chillers, cooling towers, CW pumps, CHW pumps) is being considered in the analysis, then the proposed plant efficiency is compared against the corresponding baseline curve in the following table.

**Chilled Water Plant Performance (kW/ton) vs Load Factor**

| *Load Factor* | *<150 tons* | *>=150 tons and <300 tons* | *>=300 tons* |
|---|---|---|---|
| 0.10 | 1.096 | 0.827 | 1.297 |
| 0.20 | 0.810 | 0.698 | 0.855 |
| 0.30 | 0.741 | 0.670 | 0.717 |
| 0.40 | 0.729 | 0.668 | 0.655 |
| 0.50 | 0.738 | 0.677 | 0.625 |
| 0.60 | 0.760 | 0.693 | 0.611 |
| 0.70 | 0.790 | 0.712 | 0.607 |
| 0.80 | 0.876 | 0.785 | 0.658 |
| 0.90 | 0.916 | 0.812 | 0.667 |
| 1.00 | 0.959 | 0.840 | 0.679 |

These curves are for plants consisting of one chiller. For plants with multiple chillers, the chillers are assumed to stage on in series. The resulting performance curves will have better efficiency.

**Redundancy**

The baseline large datacenter (served by water-cooled chillers), is assumed to have N+1 redundancy in the number of chillers, if not otherwise specified in the proposed design.

# *Economizing & Heat Recovery*

## Air Side Economizing

Air-side economizing is not baseline practice for datacenters. Because the baseline CRAC units typically have low static pressure drop, air delivery systems with economizing should be designed with as low pressure drop as possible in recirculation mode to avoid a summer peak demand spike with respect to the baseline.

*Exception*: Air-side economizers for telecomm facilities are considered standard practice.

## *Component Efficiency*

### DX CRACs

We use Liebert's specifications for their Deluxe System 3 series of DX CRAC units as our baseline, but the specifications do not address the efficiency of the on-board DX cooling system. To model the cooling system performance, we use the calculation method specified by the CA Title 24 2008 Alternative Calculation Method (ACM) Manual. This method relies on the DOE 2.1 modeling engine. The calculated efficiency depends on the effective air temperature at the condenser coil, the air condition (temperature and humidity) at the cooling coil, and the system's rated efficiency at the ARI standard condition. For the latter parameter, we assume the CRAC cooling efficiency is equivalent to a CA Title 24 minimally-compliant DX package unit of the same nominal capacity (see table on page 18.)

### CHW CRACs

For the cooling efficiency of chilled water CRACs, refer to the water-cooled chilled water plant system efficiency metric, above.

### Air-Cooled DX Package Units

We borrow the baseline efficiencies for these units from the 2008 California Non-Residential Title 24 Standards.

**Table 112A – Electrically Operated Unitary Air Conditioners and Condensing Units – Minimum Efficiency Requirements**

| *kBTU/hr* | *Tons* | *EER* | *kW/ton* |
|---|---|---|---|
| >=65 and <135 | >=5.4 and <11.25 | 11.2 | 1.07 |
| >=135 and <240 | >=11.25and <20 | 11.0 | 1.09 |
| >=240 and <760 | >=20 and <63.3 | 10.0 | 1.20 |
| >=760 | >=63.3 | 9.7 | 1.24 |

The values in the above table for air-cooled air conditioners with electric resistance heating or no heating.

Unlike chillers, part-load efficiency curves are typically not available from package unit manufacturers. Therefore we use the calculation method specified by the CA Title 24 2008 Alternative Calculation Method (ACM) Manual as described in the previous section on DX CRACs. The efficiency values shown here include the supply fan energy as well as the condenser fan and compressor energy.

## Chillers

**Chiller Efficiency in General**

As described in the Baselines section at the beginning of this document, the efficiency of a baseline chiller that is providing chilled water in a temperature range that is typical for space cooling needs (42 to 50 deg F) is expected to meet the CA Title 24 minimum efficiency standard. The efficiency of baseline chillers that provide chilled water temperatures lower than this (for example, making ice or maintaining low humidity levels) or higher than this (for example, serving water-cooled industrial tools), are currently not addressed by this document.

Chiller manufacturers typically describe the efficiency of their products with a single number (EER, COP, or kW/ton) that corresponds to full load operation at specific conditions. Some may offer a single efficiency number that is an average over a well-defined, limited number of operating conditions (SEER or IPLV). Title 24 follows suit, by assigning minimum allowable efficiencies to chillers that are grouped by their nominal, full-load capacity (and technology type).

However, the operating efficiency of virtually all chillers varies significantly with the load imposed on them, ambient air conditions, the chilled water supply temperature setpoint, and if water-cooled, the condenser water temperature setpoint. The efficiency typically decreases as the load decreases, as the chilled water supply temperature decreases, and as the ambient air temperature and/or the condenser water temperature increases. The shape of this efficiency-vs-load, (performance) curve also usually differs by chiller technology type.

Furthermore, chilled water systems for high tech facilities are typically and deliberately oversized by designers, to provide redundancy and increased safety factors.

Therefore, chilled water systems for high tech facilities typically operate most of the time at something less than 100% capacity. To accurately estimate the energy use of a given chiller, we first obtain part-load efficiency data from the manufacturer. We then combine these curves with the estimated load imposed by the facility, a typical meteorological year of hourly weather data appropriate for the project site, and chilled water and condenser water temperature setpoints, to determine the chiller's annual energy use.

Because Title 24 does not address part-load chiller performance, or performance at other than standard ambient conditions, we have created several baseline chiller performance curves that match Title 24 at full load and have shapes that are characteristic of the given technology type. We run these curves through the same type of analysis as described in the previous paragraph, to estimate the annual energy use of a baseline chiller appropriate to the project at hand.

Baseline chiller curves are modeled on single-compressor chillers.

**Baseline Quantity and Size of Chillers**
All chillers in a baseline chilled water plant are identical and rotated equally. Baseline chillers have one compressor. Cooling load is shared equally among all active chillers. The baseline chiller redundancy requirement is N+1 for data centers, if not otherwise specified in the proposed design. The maximum nominal cooling capacity of a single baseline chiller is assumed to be 2,000 tons. Chillers larger than this are not commonly commercially available and are, therefore, not considered typical design.

Baseline chillers are selected first based on the peak expected cooling load. The first rule when selecting the number and size of non-redundant chillers is to take the greater of:

- The minimum number of chillers necessary to meet the peak design cooling load, including a baseline safety factor (1.2). In many cases only one chiller is needed to meet this requirement, but if the peak design cooling load is greater than 1,670 tons, more than one chiller will be needed.
- Two chillers.

The second rule to determine the baseline quantity and size of chillers is to ensure that the active chiller load factor at the minimum expected cooling load is at least 50%. If after applying Rule 1 the active chiller load factor at the minimum expected cooling load is less than 50%, the number of chillers should be increased and their capacity decreased, ensuring Rule 1 is not violated.

**Water-Cooled Chillers**

CHWST Setpoint
The baseline chilled water supply temperature setpoint is 44ºF, constant.

CHW Loop Delta-T
The baseline chilled water loop ΔT is 10ºF.

CCWT Setpoint
The ARI test standard is a cold condenser water temperature of 85ºF, but in PG&E territory 80ºF is a more common specification point. A CW temperature reset is required by CA Title 24, but is typically not implemented in critical facilities. The baseline cold condenser water temperature setpoint is 80F, constant.

CHW Flow Rate
The baseline chilled water flow rate is the ARI test standard flow rate of 2.4 gpm/ton.

CW Flow Rate
The baseline condenser water flow rate is the ARI test standard flow rate of 3.0 gpm/ton.

Minimum Chiller Load Factor
The baseline minimum continuously-operating chiller load factor is 20%.

Water-Cooled Chiller Efficiency

**Chiller Performance (kW/ton) vs Load Factor**

| *Load Factor* | *<150 tons (screw compressor)* | *>=150 tons and <300 tons (screw compressor)* | *>=300 tons (centrifugal compressor)* |
|---|---|---|---|
| 0.20 | 1.422 | 1.291 | 0.797 |
| 0.30 | 1.060 | 0.962 | 0.657 |
| 0.40 | 0.895 | 0.813 | 0.593 |
| 0.50 | 0.809 | 0.735 | 0.560 |
| 0.60 | 0.763 | 0.693 | 0.543 |
| 0.70 | 0.740 | 0.672 | 0.533 |
| 0.80 | 0.731 | 0.664 | 0.530 |
| 0.90 | 0.731 | 0.664 | 0.530 |
| 1.00 | 0.738 | 0.670 | 0.532 |

The above chiller efficiency curves were obtained by using default chiller curve formulas defined in the CA Title 24 2008 Alternative Calculation Method (ACM) Manual. The minimum 2008 Title 24 chiller efficiency for each chiller type is applied as the ARI standard condition to these formulas. The efficiency curve outputs adjusted for CCWT = 80 °F generate the values in the above table.

The above efficiencies implicitly assume the ARI standard condition CHW and CW flow rates of 2.4 gpm/ton and 3 gpm/ton of chiller capacity, respectively.

## Cooling Towers

### Efficiency

The efficiency of a cooling tower plant (measured in kW of cooling tower fan energy divided by tons of cooling provided by the entire chilled water plant) varies with ambient conditions and the cooling tower specifications. Baseline cooling tower specifications are described below. We model each cooling tower based on the selected cooling tower manufacturer's reported performance data.

### Capacity

The baseline cooling tower plant is sized to handle the total chiller capacity at the 98% design wetbulb condition, with no safety factor. (The safety factor is applied to the chiller capacity.)

### Cold Condenser Water Temperature

The baseline CCWT is 80F.

### Approach Temperature

The baseline cooling tower approach temperature at the "nominal condition" is 10ºF.

By “nominal condition”, we mean:

- A hot condenser water temperature of 95F.
- An ambient wetbulb temperature of 78F.
- A condenser water flow rate equal to the maximum design flow for the given tower. (This is not necessarily the same as the *chiller's* design CW flow rate.)
- Fan speed = 100%.

The actual approach temperature of a given tower at any given moment will vary depending on incoming condenser water temperature and flow rate, tower fan speed, and ambient wetbulb temperature.

**Fan Speed**

- For cooling towers with a fan motor less than 7.5 hp, the baseline tower has a constant speed, single speed fan motor.
- Title 24 2005 requires ‘speed control’ on all cooling towers greater than 7.5 hp. For cooling towers with a fan motor greater than 7.5 hp, the baseline tower has a 2-speed motor – full speed and half speed.

If an existing cooling tower plant is being expanded, and the existing towers are equipped with VFDs, the baseline for the new tower is still as stated above.

**Staging**

Baseline cooling towers are staged sequentially, not in parallel.

**Minimum Condenser Water Flow Rate**

If not otherwise called out in the tower specifications, we assume the minimum allowed condenser water flow in a baseline cooling tower is 50% of the tower's maximum condenser water flow rate.

## *Thermal Energy Storage (TES) Systems*

TES systems are not baseline. They can be configured and applied in different ways. If a TES system will be used to occasionally shed electric demand upon request from PG&E, the customer should inquire with PG&E's Demand Response Program regarding incentives.

If a TES system will be used regularly as part of the cooling system, the difference in annual energy use between the proposed TES system and the baseline (no TES) must be examined. Ice storage systems typically use more total annual energy, even though they can save annual cost by avoiding high-rate peak demand periods. Such a system would not be eligible for an incentive under the Non-Residential New Construction or Retrofit programs. Chilled water storage systems can be designed to save both annual energy and cost, and can therefore earn an incentive.

### *Control Sequences*

Refer to the Baseline Air Management Schemes table on page 7.

- The baseline return air drybulb temperature is assumed to be successfully controlled to the setpoint shown in the table.
- The baseline air flow rate is constant for each CRAC.
- The baseline supply air drybulb temperature is therefore a resultant (not controlled to a setpoint) and is assumed to float with the cooling load.

## Heating Systems

Datacenters are cooling dominated. The only need for heat is in the MUA system for preheating cold outside air, or in the CRAC units for humidifying.

### *System Configuration*

The following configuration applies to small and large datacenters alike.

**Preheat**

Preheat is provided by a natural gas-fired air furnace in a Title 24-compliant air-cooled DX MUAH.

**Humidification**

See next section on Humidity Control Systems.

### *System Efficiency Metric*

**Air Furnace**

Baseline efficiency: [not yet defined]

### *Economizing & Heat Recovery*

The baseline datacenter in a hybrid building (a building containing both datacenter and office space) does not recover heat from the datacenter to heat office space.

### *Component Efficiency*

**Air Furnace**

Same as System Efficiency.

# Humidity Control Systems

We take ASHRAE's "Thermal Guidelines for Data Processing Environments" as our starting point. This document describes the "Recommended" relative humidity (RH) range for Class 1 and Class 2 computing environments as 40% to 55%, and the "Allowable" range for these same environments as 20% to 80%. These RH values apply to the air entering the computer equipment.

For baseline RH values, refer to the Baseline Air Management Schemes table on page 7.

CRAC units with on-board humidity control systems typically have the temperature and humidity sensors factory-mounted in the return air opening of the CRAC. For Scheme I (open aisles), we assume the sensors are left in this position, and that the humidity of the return air is controlled to the ASHRAE Recommended range of 40% to 55%.

For Schemes II and III, we assume the baseline practice is to relocate the CRAC temperature and humidity sensors to a cold aisle, or to install additional temperature and humidity sensors in the cold aisles and disable the original CRAC sensors. The aisle-mounted sensors are assumed to control the CRAC humidity system to provide the ASHRAE Recommended range of 40% to 55% in the cold aisles.

*Exception*: For telecomm facilities, no active humidity control is considered typical practice.

## *System Configuration*

### Dehumidification

In baseline datacenters, the MUAH is assumed to not provide active humidity control. Only default latent cooling occurs, due to the temperature of the cooling coil in the MUAH. Humidity control is provided by the CRAC units.

Dehumidification is accomplished by cooling the return air stream, not by desiccant systems.

**Small Datacenters**
Dehumidification is accomplished by condensation on the CRAC DX coil.

**Large Datacenters**
Dehumidification is accomplished by condensation on the CRAC CHW coil.

A single CHW plant provides the CHWST necessary to accomplish dehumidification requirements; this CHWST is served to all cooling coils in the facility, even if they are not called upon to perform dehumidification.

The CHWST setpoint is constant, set at a value to ensure the relative humidity of the return air never exceeds upper limit.

### Reheat

The baseline datacenter does not employ reheat.  The baseline RH upper limit is high enough that the temperature of the supply air after dehumidifying is close to the nominal supply air temperature when not dehumidifying.

### Humidification

A humidifier is considered baseline equipment in both small and large datacenters.  It uses electric resistance heat to boil water.  This includes infrared lamps.

Adiabatic humidifiers (evaporative, ultrasonic, etc) are not considered baseline.

*Exception*: Typical practice for telecomm facilities is to not have a humidifier.

## *System Efficiency Metrics*

### Dehumidification

The efficiency of this process is determined by the efficiency of the cooling system.

### Reheat

No reheat – not applicable.

### Humidification

The baseline humidifier uses 0.33 kWh to produce 1 pound of steam.  This is the amount of energy required to isobarically bring one pound of 60 ºF liquid water at atmospheric pressure to 212 ºF saturated water vapor.  This energy becomes a load on the cooling system.

## *Economizing & Heat Recovery*

Hot gas bypass is not used as a heat source for reheat or humidification in baseline DX CRAC units.

## *Component Efficiency*

See System Efficiency Metrics, above.

## *Control Sequences*

The humidity is measured at the return air inlet of the CRAC units.  This is where the RH sensor is installed by CRAC manufacturers, and it is baseline practice not to move it.  Note that this implies that the RH range of the air entering the computer equipment will be somewhat higher.  This is taken to be baseline practice.

Each CRAC unit controls RH independently. (In practice, there may be only a subset of CRAC units programmed for active humidity control.) Independent RH control often leads to an energy-wasting struggle among the CRAC units, but this is assumed not to occur in the analysis model of the baseline datacenter.

# Electrical

## *Electric Motors for Fans and Pumps*

Refer to the table below for motor baseline efficiencies, given as a percentage. Premium efficiency motors are not considered baseline. If not otherwise specified, we assume the baseline motor is Open Drip-Proof (ODP), 1200 rpm.

**Baseline Electric Motor Efficiencies in Percent (NEMA EPACT Efficiencies)**

Source: "Induction Motor Efficiency Standards" by John Douglas, PE, 2005, Washington State University Extension Energy Program

| Open Drip-Proof (ODP) | | | | |
|---|---|---|---|---|
| 460 V | | | | |
| Nominal Motor hp | 3600 rpm | 1800 rpm | 1200 rpm | 900 rpm |
| 1 | | 82.5 | 80.0 | 74.0 |
| 1.5 | 82.5 | 84.0 | 84.0 | 75.5 |
| 2 | 84.0 | 84.0 | 85.5 | 85.5 |
| 3 | 84.0 | 86.5 | 86.5 | 86.5 |
| 5 | 85.5 | 87.5 | 87.5 | 87.5 |
| 7.5 | 87.5 | 88.5 | 88.5 | 88.5 |
| 10 | 88.5 | 89.5 | 90.2 | 89.5 |
| 15 | 89.5 | 91.0 | 90.2 | 89.5 |
| 20 | 90.2 | 91.0 | 91.0 | 90.2 |
| 25 | 91.0 | 91.7 | 91.7 | 90.2 |
| 30 | 91.0 | 92.4 | 92.4 | 91.0 |
| 40 | 91.7 | 93.0 | 93.0 | 91.0 |
| 50 | 92.4 | 93.0 | 93.0 | 91.7 |
| 60 | 93.0 | 93.6 | 93.6 | 92.4 |
| 75 | 93.0 | 94.1 | 93.6 | 93.6 |
| 100 | 93.0 | 94.1 | 94.1 | 93.6 |
| 125 | 93.6 | 94.5 | 94.1 | 93.6 |
| 150 | 93.6 | 95.0 | 94.5 | 93.6 |
| 200 | 94.5 | 95.0 | 94.5 | 93.6 |
| 250 | 94.5 | 95.4 | 95.4 | 94.5 |
| 300 | 95.0 | 95.4 | 95.4 | |
| 350 | 95.0 | 95.4 | 95.4 | |
| 400 | 95.4 | 95.4 | | |
| 450 | 95.8 | 95.8 | | |
| 500 | 95.8 | 95.8 | | |

| Totally Enclosed Fan-Cooled (TEFC) | | | | |
|---|---|---|---|---|
| 460V | | | | |
| Nominal Motor hp | 3600 rpm | 1800 rpm | 1200 rpm | 900 rpm |
| 1 | 75.5 | 82.5 | 80.0 | 74.0 |
| 1.5 | 82.5 | 84.0 | 85.5 | 77.0 |
| 2 | 84.0 | 84.0 | 86.5 | 82.5 |
| 3 | 85.5 | 87.5 | 87.5 | 84.0 |
| 5 | 87.5 | 87.5 | 87.5 | 85.5 |
| 7.5 | 88.5 | 89.5 | 89.5 | 85.5 |
| 10 | 89.5 | 89.5 | 89.5 | 88.5 |
| 15 | 90.2 | 91.0 | 90.2 | 88.5 |
| 20 | 90.2 | 91.0 | 90.2 | 89.5 |
| 25 | 91.0 | 92.4 | 91.7 | 89.5 |
| 30 | 91.0 | 92.4 | 91.7 | 91.0 |
| 40 | 91.7 | 93.0 | 93.0 | 91.0 |
| 50 | 92.4 | 93.0 | 93.0 | 91.7 |
| 60 | 93.0 | 93.6 | 93.6 | 91.7 |
| 75 | 93.0 | 94.1 | 93.6 | 93.0 |
| 100 | 93.6 | 94.5 | 94.1 | 93.0 |
| 125 | 94.5 | 94.5 | 94.1 | 93.6 |
| 150 | 94.5 | 95.0 | 95.0 | 93.6 |
| 200 | 95.0 | 95.0 | 95.0 | 94.1 |
| 250 | 95.4 | | 95.0 | 94.5 |
| 300 | 95.4 | | 95.0 | |
| 350 | 95.4 | | 95.0 | |
| 400 | 95.4 | | | |
| 450 | 95.4 | | | |
| 500 | 95.4 | | | |

## *VFDs*

### Efficiency

*2008 ASHRAE Handbook-HVAC Systems and Equipment* reports that VFD efficiency is not constant over the operable speed range of the controlled fan or pump. The table below is an adaptation of the chart published on page 43.13 in the 2008 ASHRAE Handbook and is considered baseline practice.

**VFD Efficiency vs Design Speed**

| Design Speed, % | Efficiency, % | |
|---|---|---|
| | Newer Model | Older Model |
| 0 | 88 | N/A |
| 20 | 91 | 72 |
| 40 | 94 | 83 |
| 60 | 97 | 89 |
| 80 | 98 | 93 |
| 100 | 98 | 95 |

### Turndown Limit

For new VFD compatible motors with new VFDs, there is no practical turndown limit. For systems with older motors or older VFDs, the baseline VFD turndown limit is 12 Hz (20%).

## *Uninterruptible Power Supply (UPS)*

The baseline UPS system is battery-based, double-conversion, non-switching (all power to the devices supported by the UPS flows through the UPS battery system.) Baseline loading depends on the redundancy requirement.

The baseline efficiency curve depends on the size of the UPS. The baseline efficiency criteria are as follows:

| UPS Size | % Load | | | |
|---|---|---|---|---|
| | 25% | 50% | 75% | 100% |
| kVA < 20 | 86.3% | 89.1% | 89.6% | 89.6% |
| 20 <= kVA <= 100 | 88.5% | 90.5% | 91.1% | 91.1% |
| kVA > 100 | 89.4% | 92.2% | 93.1% | 93.3% |

These efficiency values are averages of published UPS efficiency data compiled from several prominent UPS manufacturers. The data set includes several UPS models in each of the listed UPS size ranges.

Different datacenters have different requirements for UPS run-time in the event of a power outage. The baseline run-time requirement is set to match the requirement of the proposed datacenter. In the baseline case, it is assumed that run-time requirement is met by adjusting the number of storage batteries, not by changing the baseline power output capacity of the UPS inverter.

### Baseline Quantity and Size of UPS Modules

For UPS systems with capacities greater than or equal to 100kVA, the baseline UPS system is determined as follows:

1. Determine the fewest number of UPS modules that meet the design load.
2. Applying the baseline safety factor (1.20) and the redundancy required by the proposed facility.

The largest baseline UPS capacity is 750kVA, based on commonly available UPS sizes.

## Calculation Assumptions

The following items are ***not*** baseline targets. Some of them are average values reflecting typical practice, that could possibly be developed into baselines with further research. For now, they are used merely to help round out our calculation models. Other items describe our standard calculation procedures.

## *Datacenter Load*

A rule of thumb is the initial actual energy use of a new datacenter is 50% of the design capacity of the cooling system serving the datacenter. If a customer presents credible evidence for a higher density in their facility, we use it. This is not uncommon, as datacenter load densities are rising as computer technology advances. Also, new datacenters being built to replace existing datacenter space are more likely to operate near the full occupancy. If the load is expected to increase over time we take this in to consideration in our calculations, projecting out to a maximum of five years.

In all cases, the actual loading is verified after construction is complete and the energy savings and incentive is adjusted up or down accordingly.

## *Commercial Space Load*

To accurately model the performance of a chilled water plant that serves a hybrid building (or multiple buildings), it is necessary to account for the load imposed on the plant by the commercial space. If the commercial space is not being analyzed separately with EnergyPro or similar software, we use the following:

| | |
|---|---|
| Typical occupant density (floor area per person) | 150 sf |
| Typical office occupancy | 8 am to 6 pm, M-F |

| | Loads | | | Rates | | Load Densities | |
|---|---|---|---|---|---|---|---|
| | Per Person | | Density | | | Watts/sf | |
| | BTU | Watts | Watts/sf | Occupied | Unoccupied | Occupied | Unoccupied |
| People | 250 | 73 | 0.5 | 100% | 0% | 0.5 | 0.00 |
| Office Equipment | | | 0.5 | 100% | 20% | 0.5 | 0.10 |
| Lights (incl. task lights) | | | 1.2 | 100% | 20% | 1.2 | 0.24 |
| Total | | | | | | 2.2 | 0.34 |

## *Cooling System Performance*

### Chiller Plant Average Efficiency

Average cooling plant efficiency = 0.98 kW/ton. This is an ***average*** for the chilled water ***plant***; i.e., chiller, pumps, cooling tower, not just the chiller). This number is derived from actual performance measurements of 12 chilled water plants serving cleanroom facilities, by LBNL, in 2000 and 2001. This sampling includes old and new plants, water- and air-cooled, running at various load factors. See our May 2003 paper, "Cleanroom Energy Baseline Study", written for PG&E. This number is sometimes used in analyses of individual HVAC components that use chilled water (such as air handlers). It allows approximating the component's annual energy use without having to model the entire cooling plant – assuming the component is served by a typical high tech industrial cooling plant.

### Chiller Performance as a function of CHWST, CWST

The efficiency of water-cooled chillers increases by 1.5% for every 1ºF drop in CW temperature, and increases by 1.0% for every 1ºF increase in CHW supply temperature. We use this rule of thumb to modify chiller performance curves to match proposed operating conditions, in cases where the chiller manufacturer is unable to provide us with chiller performance data at those conditions.

### Chiller Capacity as a function of CHWST, CWST

The *capacity* of water-cooled chillers also increases, about 1.5% for every 1ºF decrease in chiller lift, as compared to the chiller's capacity & lift at nominal conditions.
This can be seen in chiller selection software. There is of course a practical limit to this rule of thumb, as every chiller has a minimum allowed operating lift.

## IT Equipment Measures

The following measures reduce IT equipment power draw. PG&E offers incentives for each. A savings and incentive calculator for each is available directly from PG&E.

### High Efficiency Servers

Some server manufacturers are now offering products that provide significantly more computational power per Watt of input energy. PG&E offers an incentive for replacing existing servers with these high efficiency servers.

### Thin Clients

This computing model replaces desktop computers with smaller, low-power devices that serve primarily as an interface to central servers. Most of the software applications that formerly ran on the desktop computers are moved to the servers, where they can operate more efficiently from a Watts-per-computation perspective.

### Virtualization

Baseline datacenter operation assigns separate computing tasks to specific servers. As a result, some servers are under-utilized. Virtualization software can be installed to intercept all computing tasks and distribute them among all available servers to maximize their utility. It is normally possible to greatly reduce the number of servers needed to handle a given workload.

### MAID

MAID stands for Massive Array of Idle Discs. In some data storage applications, the data is accessed relatively infrequently. The storage array of discs can be designed to power down idle discs when they are not needed, saving energy.

## Abbreviations

| | |
|---|---|
| ACH | Air changes per hour. |
| AHU | Air handling unit. |
| ASHRAE | American Society of Heating, Refrigeration, and Air-conditioning Engineers. |
| BTU | British Thermal Unit. |
| CCW | Cold condenser water. |
| CCWT | Cold condenser water temperature. |
| CFM | Cubic feet per minute. |
| CHW | Chilled water. |
| CHWR | Chilled water return. |
| CHWRT | Chilled water return temperature. |
| CHWS | Chilled water supply. |
| CHWST | Chilled water supply temperature. |
| CRAC | Computer room air conditioner. |
| CW | Condenser water. |
| dP | Delta-P (pressure difference). |
| dT | Delta-T (temperature difference). |
| DB | Drybulb. |
| DP | Dewpoint. |
| DX | Direct expansion. |
| EER | Energy efficiency ratio. |
| FFU | Fan filter unit. |
| HCW | Hot condenser water. |
| HCWT | Hot condenser water temperature. |
| HEPA filter | High efficiency particulate air filter. |
| HHW | Heating hot water. |
| HVAC | Heating, ventilation, and air conditioning. |
| HW | Hot water. |
| HX | Heat exchanger. |
| IPLV | Integrated part load value. |
| MUA | Makeup air. |
| MUAH | Makeup air handler. |
| NRNC | Non-Residential New Construction. |
| NRR | Non-Residential Retrofit. |
| OA | Outside air. |
| OAT | Outside air temperature. |
| ODP | Open drip-proof. |
| RAHU | Recirculation air handling unit. |
| RAT | Return air temperature. |
| RH | Relative humidity. |
| SAT | Supply air temperature. |
| UPS | Uninterruptible power supply. |
| VAV | Variable air volume. |
| VFD | Variable frequency drive. |
| WB | Wetbulb. |
| in. w.g. | Inches of water gauge. |