Energy Efficiency Baselines for

CLEANROOMS

Non-Residential New Construction
and Retrofit Incentive Programs

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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, laboratories have not received the same level of attention as commercial projects. This leaves ample opportunity to significantly reduce the energy budget for lab 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 "cleanroom", refer to Cleanroom 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. In any event, high tech industrial spaces are interpreted to be 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.

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.

Cleanroom Definition

A cleanroom is a manufacturing environment that has a low level of environmental pollutants such as dust, airborne microbes, aerosol particles and chemical vapors. More accurately, a cleanroom has a controlled level of contamination that is specified by the number of particles per unit of air volume, and by maximum particle size.

Cleanrooms are cooling dominated; i.e., little or no comfort heating is required. Reheat of supply air is needed after dehumidification.

Classification Categories for Cleanrooms

In the United States, cleanrooms are classified according to cleanliness level using one of two standards, ‘cleanroom class’ and/or ‘ISO standard.’ For example, a ‘Class 1’ cleanroom (ISO 3) will only contain one particle larger than 0.5 microns per cubic foot of air. A ‘Class 10’ cleanroom (ISO 4) will only contain 10 particles larger than 0.5 microns per cubic foot of air, and so forth. The following table correlates these two most frequently-used cleanroom classifications:

Class 1	ISO 3
Class 10	ISO 4
Class 100	ISO 5
Class 1,000	ISO 6
Class 10,000	ISO 7
Class 100,000	ISO 8

Size Categories for Cleanrooms

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

Large Cleanrooms
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 cleanroom 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

- The baseline cleanroom systems are modeled at the same redundancy level that the corresponding proposed cleanroom systems are designed to.
- If the redundancy level of a proposed cleanroom system is not stated by the customer, the baseline follows current typical engineering practice for the given type of system in a cleanroom context. This is usually N+1.
- For information on control of redundant fans and pumps, see Air Delivery Systems\Control Sequences and Hydronic Systems\Control Sequences, respectively.

Space Design Conditions

Baseline Space Design Conditions

Parameter	Value
Space Drybulb Temperature Setpoint	72F
Space Relative Humidity Range	The baseline range is set equal to the proposed range. If the proposed range is not given, we assume 45% to 55% RH.
Recirculation Rate (assumed to be continuous, 24/7/365)	Depends on cleanliness rating. See Air Delivery Systems, below.
Ventilation Rate	The baseline ventilation rate is assumed to be equivalent to the Title 24 minimum rate for office space of 0.15 cfm/sf, plus any additional ventilation required for fume hoods or similar devices. The ventilation air is assumed to be provided by dedicated makeup air handler (MUAH).
Exhaust Rate	For modeling purposes the exhaust volume is assumed to be same as ventilation volume (no in/exfiltration in space). In reality, the ventilation volume exceeds the exhaust volume in order to maintain positive pressurization of the clean space. This is not yet addressed by this baselines document.

Baseline Occupancy
Human occupancy is assumed to add a negligible load to the HVAC system that serves the cleanroom.

Air Delivery Systems

System Configuration

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 cleanrooms. We consider cleanrooms 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. This is an appropriate minimum to ensure human comfort and dilution of non-process air-borne contaminants (e.g. building envelope off gassing). In practice, ventilation requirements for maintaining ideal cleanroom air quality conditions may exceed this baseline.

Recirculation

Due to the stringent cleanliness requirements for cleanroom air, only the minimum outside air (OSA) required for ventilation purposes is introduced to the space. As a result, the majority of the air in a cleanroom environment is recirculated. Baseline practice entails a constant recirculation rate.

Ventilation

The baseline ventilation air system consists of one or more dedicated MUAHs equipped with constant-speed fans. The MUAH is assumed to deliver air at a neutral drybulb temperature of 72F.

Exhaust

The exhaust air volume is assumed to be equal to the ventilation air volume. The exhaust fans are assumed to be constant-speed. The exhaust system is not equipped with any heat recovery devices.

Recirculation System Efficiency Metric

The operative metric for a cleanroom recirculation air delivery system is the total recirculation fan energy in Watts divided by the cleanroom floor in square feet.

This differs from the standard air delivery system metrics of cfm/kW or W/cfm. These metrics are fine when the system needs to deliver a fixed amount of air, but the objective of the cleanroom space is to maintain a certain level of cleanliness. If the cleanroom operator can reduce the recirculation air flow and still maintain the cleanliness rating, this should be reflected in the energy efficiency metric for the cleanroom air recirculation system. The W/sf metric captures this. The baseline values are shown in bold in the table on the next page.

For a given cleanroom cleanliness rating, the efficiency of the air recirculation system is dictated by the recirculation air flow rate, the total static pressure drop (TSP) of the recirculation system, and by the efficiency of the recirculating fans (the fan/drive/motor combination).

Improving any one of these three aspects alone (reduced air volume, reduced pressure drop, or increased fan system efficiency) does not necessarily provide better-than-baseline overall recirculation system efficiency. It is the combination of the three aspects that determines the resulting value of W/sf.

Baseline Cleanroom Air Recirculation System Efficiency

US Class	ISO Class	Air Change Rate (air changes per hour)	Total Static Pressure Drop (in. w.g.)	Fan System Efficiency (%)	Recirculation Fan Energy Density (W/sf)
1	3	800	1.1	51%	**34**
10	4	465	1.6	53%	**27**
100	5	200	2.7	73%	**15**
1,000	6	100	3.1	68%	**9**
10,000	7	50	3.1	58%	**5**
100,000	8	20	3.2	51%	**2**

The values in this table for ISO Classes 4 through 7 are based on a benchmark study of cleanroom efficiency conducted by PG&E in 2003. The values for ISO Classes 3 and 8 are extrapolated. A 10-foot cleanroom ceiling is assumed.

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.

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.

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

<u>RAHU Face Velocity</u>
The baseline RUAH 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. Note that the following values apply only at the peak design airflow.

<u>Recirculation Duct Static Pressure Drop</u>
The baseline duct static pressure drop is 0.10" w.g. per 100 ft. This is also a long-standing design rule of thumb.

Exhaust

Exhaust path pressure drop = 1.7 in. w.g. We use 0.7" for the exhaust stack plus 1.0" 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 the MUAH, RAHU, and exhaust fans.

Fan Filter Units (FFUs)

The baseline 2-ft by 4-ft fan filter unit delivers 2,490 cfm per kW at an external pressure drop of 0.3 in. w.g. This result was obtained by analyzing data in a study conducted by Lawrence Berkeley National Laboratory. The report is available on-line at http://gaia.lbl.gov/btech/papers/62163.pdf.

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

Baseline recirculation fans are constant speed, balanced at startup, and run 24/7/365. There is no setback of recirculation air flow or the space temperature setpoint during unoccupied hours.

Ventilation

If there is no process exhaust requirement, the baseline ventilation rate is constant 24/7/365 at the Title 24 minimum requirement for office space, 0.15 cfm/sf. This is provided by a constant-speed MUAH.

If there is a process exhaust requirement, the baseline ventilation system is variable-volume. The MUAH supply fan is equipped VFD, controlled to maintain a constant supply duct static pressure.

In either case, there is no setback of ventilation air flow or the ventilation air temperature setpoint during unoccupied hours. There is no demand-controlled ventilation sequence.

Exhaust

If there is no process exhaust requirement, the baseline exhaust air flow rate is assumed to be equal to the ventilation air flow rate. The exhaust fan runs at constant speed.

Cleanroom process exhaust must be ejected at high velocity (2,000-3,000 fpm), which requires significant fan energy. Minimizing the pressure drop in a variable exhaust system is difficult as the exit velocity must be constant. The standard approach is to draw in outside, dilution air just prior to the exhaust stack fan. The exhaust fans run at constant speed, continuously moving the peak design exhaust volume.

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 lower-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 25..
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 cleanrooms 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 kW/ton. 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 Cleanrooms
The baseline cooling system for cleanrooms served by less than or equal to 360 tons total installed capacity not including redundancy is an air-cooled chilled water plant serving RAHUs and MUAHs.

Large Cleanrooms
The baseline cooling system for cleanrooms served by more than 360 tons total installed capacity (not including any cooling system redundancy) is a water-cooled chilled water plant serving RAHUs and MUAHs.

Air-Cooled Chilled Water Plant

The baseline air-cooled chilled water plant has:

- No CHW storage.
- Redundancy = N+1 on chillers and CHW pumps.
- 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.

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.
- Safety factor on capacity = 98% design condition * 1.20.
- Chillers are equally-sized.
- Chiller run-time is divided equally among all chillers.
- Baseline chillers are electric (not absorption or adsorption).

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.

System Efficiency Metric

The operative metric for cleanroom cooling systems is kW/ton.

Air-Cooled Chilled Water Plant

The baseline performance of an air-cooled chiller plant has not been addressed yet.

Water-Cooled Chilled Water Plant

If the entire water-cooled 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 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.

Economizing & Heat Recovery

Air Side Economizing

Air-side economizing is not baseline practice for cleanrooms. The filtration requirement usually makes this configuration not cost-effective.

Water-Cooled Chilled Water Plant

The baseline water-cooled chilled water plant does not use water-side economizing (aka "water-side free cooling").

Component Efficiency

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, or "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 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.

Air-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.

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

Minimum Chiller Load Factor
The baseline minimum operable chiller load factor is 20%.

Air-Cooled Chiller Efficiency

The full load efficiency of a baseline air-cooled chiller at ARI Standard 550/590 test procedure conditions equals the minimum full-load efficiency requirement from the 2008 California Non-Residential Title 24 Standards. A portion of the table in which the efficiency requirement appears is provided here.

California Non-Residential Title 24 Standards, 2008: Table 112-D – Water Chilling Packages – Minimum Efficiency Requirements

Equipment Type	*Size Category*	*COP*	*kW/ton*
Air-Cooled, With Condenser, Electrically Operated	< 150 tons	2.80	1.256
	>= 150 tons		

The default chiller curve formulas defined in the CA Title 24 2008 Alternative Calculation Method (ACM) Manual are used to generate part-load chiller efficiency curves at varying outside air dry bulb temperatures. The minimum 2008 Title 24 chiller efficiency shown above is applied as the full load ARI standard condition to these formulas.

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.

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 operable chiller load factor is 20%.

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.

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)*
0.10	2.575	2.339	1.242
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

The RAT is controlled to 72F drybulb.
The SAT floats with the load. The SAT is assumed to never drop below 55 F.

Heating Systems

Cleanrooms are cooling dominated. The only need for heat is in the:

1. MUA system for preheating cold outside air, or reheating after dehumidification.
2. Humidifying system, to create steam.

System Configuration

The following configuration applies to small and large cleanrooms alike.

Preheat & Reheat

In a baseline MUAH, preheat and reheat are provided by HW coils served by a natural gas-fired Title 24-compliant boiler.

Humidification

See next section on Humidity Control Systems.

System Efficiency Metric

Natural Gas-Fired Boiler

For baseline boiler efficiency, we use an annual average of 82%. For detailed calculations, we use the following baseline boiler performance curve:

Load Factor	Efficiency as a percentage of quoted full-fire efficiency
10%	56%
20%	71%
30%	80%
40%	85%
50%	89%
60%	92%
70%	95%
80%	97%
90%	99%
100%	100%

Note: You must multiply the percentages shown here by the boiler's full-fire efficiency to get the actual efficiency.

Economizing & Heat Recovery

There is no air-side heat recovery system in the baseline case.
There is no boiler stack heat recovery in the baseline case.

Component Efficiency

Natural Gas-Fired Boiler

See System Efficiency above.

Humidity Control Systems

The space humidity range in the baseline cleanroom is set equal to the proposed range. If the proposed range is not given, we assume 45% to 55% RH.

System Configuration

Dehumidification

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

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 ventilation supply air never exceeds the proposed upper limit.

Reheat

If dehumidification is required, the baseline MUAH employs the HW reheat coil to raise the temperature of the ventilation supply air back to baseline space drybulb setpoint (72F).

Humidification

A humidifier is considered baseline equipment in a cleanroom MUAH. In a small cleanroom facility, the humidifier is assumed to use natural gas to create steam at the MUAH. In a large cleanroom facility, steam is supplied to the MUAH humidifier by a steam boiler in the central mechanical plant.

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

System Efficiency Metrics

Dehumidification

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

Reheat

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

Humidification

The baseline gas-fired, MUAH-mounted humidifier uses 1,126 BTU (0.0113 therms) 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

There is no steam boiler stack heat recovery in the baseline case.

Component Efficiency

A baseline steam boiler has an annual average thermal efficiency of 76%. The steam injected in to the ventilation supply air stream becomes a load on the cooling system.

Control Sequences

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

Electrical

Electric Motors for Fans and Pumps

Refer to the table below for motor baseline efficiencies, given in 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%).

Process Systems

There are many possible types of process systems to be found in cleanrooms. The list below is only a sampling. The assessment of energy efficiency opportunities in process systems is usually best done by an expert in the respective system.

System	Description
Compressed Dry Air	Air compressors create clean, high-pressure air for multiple purposes.
De-Ionized (DI) Water	Water that has been treated to remove minerals and ions, greatly reducing its electrical conductivity.
Fume Hoods	Enclosed workbenches with a constant velocity air flow from the room in to the workbench space. Equipped with a sash that can be raised and lowered as needed for access to the bench top.
Process Vacuum	Process vacuum pumps create a negative-pressure distribution system needed for certain types of lab equipment. Not to be confused with the "house" vacuum system, used for janitorial purposes.
Reverse Osmosis (RO) Water	Similar to de-ionized water, RO water has had most minerals removed by pumping it through a semi-permeable membrane.

System Configuration

Compressed Dry Air

Compressed air for use in processes must be provided by oil-less compressors, to prevent contamination.

System Efficiency Metrics

Compressed Dry Air

Baselines and energy efficiency opportunities are addressed by the AirMaster software program.

Fume Hoods

Fume hoods generally do not contain fans themselves. The energy use of a fume hood is reflected in the amount of air moving through the hood and the efficiency of the exhaust system that serves it.

Economizing & Heat Recovery

Compressed Dry Air

Baseline compressed air systems do not employ heat recovery.

Control Sequences

Compressed Dry Air

In the baseline system, the supply air pressure setpoint is constant. The air dryers are regenerated with compressed air, on a regular time-based interval.

Fume Hoods and Biosafety Cabinets

Fume hoods can be operated as constant air volume (CAV) or variable air volume (VAV). At this writing, the baseline for academic institutions is VAV and the baseline for industry is CAV. The baseline sash opening face velocity is a constant 100 ft/min, regardless of space occupancy and regardless of whether an occupant is standing in front of the hood.

Biosafety cabinets can also be operated as CAV or VAV. However, biosafety cabinets have a more demanding tolerance (+/-5%) on variations in air flow than fume hoods. The air flow variation in a typical VAV system is on the order of +/-10%, so the necessary control may be difficult to achieve. The baseline for biosafety cabinets is CAV. Reference: http://ateam.lbl.gov/Design-Guide/DGHtm/biologicalsafetycabinets.htm

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.

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.

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.