Development of Cleanroom Energy Benchmarks and Baselines for Use in Industrial New Construction Energy Efficiency Programs

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ABSTACT

Cleanroom facilities typically require a variety of energy-intensive supporting mechanical systems, making them a natural target for energy efficiency programs. However, before higher-efficiency design and practice can be recognized and encouraged, a baseline of what constitutes typical performance must be set. For new construction projects, the difference between system designs that constitute standard practice and those that can be considered high-performance must be evaluated and understood. Energy evaluation programs such as DOE2 and EnergyPro are widely considered by designers as inadequate for such tasks (Sartor et al. 1999). Further, cleanroom facility operators frequently do not prioritize energy efficiency assessment during early stages of the usually expedited design phase. Therefore, demonstrating the value of energy efficiency through integrated design analysis and program intervention is highly desirable.

Since whole building energy use is process and facility dependant, this system-based approach benchmarking and baselining effort focuses on mechanical system efficiency, which is generally less facility-dependant and which provides an evaluative basis that is more consistently transferable among facilities. The majority of energy savings in cleanroom facilities tend to fall into three subsystems including (1) Recirculation air handling, (2) Make-up air handling, and (3) Chilled water production. In this paper, these subsystems are investigated and baseline efficiency metrics are developed. In addition, these baseline metrics may be easily used by utility energy efficiency program administrators to encourage more efficient design through incentive and education programs or by facility operators to evaluate their own systems for retrofit potential.

Introduction

Cleanroom facilities require a number of energy intensive supporting mechanical systems. However, before higher energy efficiency design and practice can be recognized and encouraged, a baseline of what constitutes typical performance must be set. A Cleanroom Benchmarking and Baselining Project that aims to establish an energy baseline for cleanroom facilities for use in calculating energy savings for improved operation in new or retrofit facilities has been initiated. This information could typically be used by utilities to administer energy efficiency rebate programs such as the Savings By Design (SBD) Program in California to target reduction in large power demand from new cleanroom facilities.

Whole building analysis programs such as DOE2 or Trane Trace are not suited for modeling cleanrooms, which have multiple systems serving a single space, high process loads, and significant process cooling chilled water loops at different temperatures. Since whole building energy use, such as w/sf, is process and facility dependant, the baselining focuses on mechanical support *system* efficiency (system approach), which is generally less dependant on facility particulars and provides a more consistently transferable basis for evaluating different facilities. While cleanroom processes and loading vary widely, they have many supporting mechanical systems in common, such as the recirculation air handling, make-up air handling, and chilled water production.

One significant area of energy use not evaluated by the system approach is the power consumed by the process equipment itself. While there is potential for energy efficiency improvements in this area, there are two reasons it is not investigated further: First, the vast variety and evolution of process equipment makes it very difficult to develop any sort of metric that would hold for more than a single process, or even a single tool type. Second, cleanroom operators and equipment manufacturers tend to be simply not willing to consider any changes to their process that are not production driven. To accommodate the variety of process loads that occur in cleanrooms, each subsystem needs a capacity-independent measure of its energy efficiency. Use of efficiency metrics allows for an comparison between facilities with vastly different process loads and types. Each subsystem will have an efficiency metric consisting of system output (cfm of filtered air, tons of cooling, kbtu/hr of heat, etc.) divided by system input (kW or kbtu/hr of gas).

Methodology

Cleanroom benchmarking data collected through publicly and privately funded projects form the foundation of the baseline efficiencies (Tschudi et al., 2001). This measurement and design data provides insight into the design of currently operating facilities. The baseline cleanrooms are referred to by the Federal Standard 209E classification they were certified under; the data is applicable to the equivalent current ISO standards. Interviews with cleanroom designers, facilities managers and operators ensured that current standard design practices, which may not yet be fully represented in operating facilities, have been properly accounted for.

Initially, a series of algorithms and criteria for calculating energy savings were developed. By taking the system approach, the impact of the actual cleanroom process was greatly reduced. For example, a class 10 facility is expected to require more chilled water than a class 10000 facility, but the chilled water plant design is the same, only the size is different. By disassociating the metrics from the loads, the impact of the cleanroom classification is minimized since the cleanroom system configurations were not seen to vary in type between the classification levels, they only varied in the load. For example, a higher cleanliness rating space typically had a higher chilled water load, but the same type of chilled water plant design, just of a higher capacity. This methodology allows practical comparison of design options for cleanroom classes 10 to 1000 based on the benchmarking database.

Recirculation Air Handling System

The recirculation air handling system provides a constant flow of filtered air in the cleanroom. The level of filtration used varies with the cleanroom class. The recirculation system configuration chosen, the type of filters selected and the air flow volume per filter are three major design parameters that have a large impact on the pressure drop and recirculation fan power.

Fan power consumption is significant in a cleanroom, frequently matching or exceeding the energy consumption of the chillers on an annual basis in northern California mild climate (LBL, 2001). To obtain an accurate measure of a cleanroom's efficiency, the fan energy used must be evaluated in a consistent manner. Sensible cooling is typically provided via the recirculation airstream, usually by a coil in the recirculation air handler, fan coils in plenum space or a separate sensible cooling air handler. Any dedicated sensible cooling fans should be considered part of and included in the recirculation system fan energy. (Note: the cooling load and associated chilled water plant energy use on the sensible cooling units is not considered in this metric. This metric deals solely with the recirculation air and sensible cooling fan power. The efficiency of the cooling will be dealt with in the chilled water plant section.)

There are many methods of improving the recirculation air handler efficiency. During new construction design, typical options to improve energy efficiency include selecting a more efficient system configuration, lowering the air handler face velocity and lowering the overall air velocity in the cleanroom. In existing facilities, implementing an unoccupied turndown or rebalancing the cleanroom to lower recirculation airflow are potential measures.

For this metric, the system fan power and the system airflow are needed. The power of the recirculation system will be defined as the power used by the recirculation fans *and* the sensible cooling system fans. Fan power should be specified on project schedules or equipment submittals in the form of brake horsepower (BHP) and converted to kW (0.746 x BHP = kW). This power will be used as the input power for the metric.

The amount of recirculation airflow volume in a cleanroom is an important design parameter. The recirculation airflow volume also does not vary; the recirculation system operates continuously in a baseline system, with no off-shift setback. Variable speed fans are sometimes utilized in recirculation systems, but the speeds are set during startup and not dynamically altered. Recirculation system are sometimes designed with additional capacity, to allow for future reconfiguration, long-term filter loading, and unanticipated system pressure drop, however, the recirculation airflow will be measured and set to the design airflow during cleanroom certification.

The cleanroom cleanliness rating and the process requirements dictate the amount of recirculation air. The actual flow is typically measured during start up balancing. In more sensitive cleanrooms, typically class 10 or 100, the airflow is periodically checked to verify adequate flow and particulate counts. Balance reports offer the best measure of recirculation system flow if they are available. Recirculation airflow is not varied in standard design, so no weather or process impacts need to be considered. In new construction, the design flow can be determined from design documents. This flow is used along with the power to produce the metric. The recirculation airflow is 100% of the airflow indicated by cleanroom certification or balancing records.

To determine an annual energy usage, the cfm/year of the system needs to be determined. As already discussed, while VFDs are often used in recirculation units, they are manually locked to a single speed during balancing. The recirculation airflow is not varied in standard cleanroom operation, with a change in the airflow usually requiring a significant quantity of rebalancing and cleanroom verification work. No weather adjustments, diversity, turndown or other adjustments are required to determine an accurate annual load for the baseline. Some adjustment is required of proposed system's annual load if a recirculation setback is proposed.

The Recirculation Air metric here will consist of w/cfm where the input information needed will be total brake horse power of all connected recirculation fans, including fan powered hepa filters and sensible coil units divided by the total cfm required or flow. This w/cfm depends on the total system static pressure of the system and the efficiency of the system components. The *typical design baseline is 0.43 w/cfm* (2,325 cfm/kW). The baseline value is derived from measurement of 18 cleanrooms, and interviews of current designers and operators to verify the benchmarked cleanrooms still represent current practice. Multiplying the metric by the airflow and 8760 hours yields an estimated energy consumption for the system in kWh/yr.

Table 1. Recirculation Air System Benchmarking Data

Site	Class	Fan kW	Operating CFM	w/cfm	AirChange/hr
A	10	348	1684080	0.21	395
H	10	27.2	85740	0.32	591
I	10	22.2	73280	0.30	516
M	10	72	221605	0.32	678
P	10	15.4	100600	0.15	368
K	10	380.9	722464	0.53	241
B	100	28.8	203,040	0.14	94
C	100	233.7	516990	0.45	153
D	100	44.4	56660	0.78	133
E	100	92.7	123060	0.75	191
F	100	191.5	208450	0.92	175
G	100	101	486,100	0.21	276
J	100	17.8	180450	0.10	-
O	100	62	148,160	0.42	225
Q	100	34	41325	0.15	215
R	100	47.6	60775	1.15	316
S	1000	-	-	0.56	-
N	10000	50.4	82385	0.61	82

Make-Up Air Handling System

The make-up air system is essential for the operation of a cleanroom. This system works in conjunction with the recirculation air system to create a stable and "clean" environment by maintaining humidity levels and positive space pressurization. Additionally, outside air ventilation required for occupants' health is delivered via the make-up air handling system. In California, semi-conductor cleanrooms are commonly classified as H6 space. Typically, cleanrooms easily exceed the building code minimum H6 ventilation requirement due to the extensive exhaust and space pressurization requirements. The large need for outside air, at least 6.5 times that the ventilation rate required by offices, combined with the tight temperature and humidity requirements (office buildings typically have no humidity control) is one reason cleanrooms are so much more energy intensive than offices.

A make-up air handler is served with various systems, such as chilled water, heating hot water, and purified water and compressed air (where needed for humidification). Due to the need for simplicity, this metric currently focuses on just the fan energy required by the makeup air handlers.

There are two common makeup air system designs that differ in the method of providing redundancy. One design approach installs a fully redundant air handler while the other common approach installs a redundant fan in each unit. All typical designs now utilize VFDs for active pressurization/flow control.

The configuration of the air handlers, filtration, humidification method and ducting all contribute to the power required by the makeup air fans. The air handler pressure drop has the largest impact on the efficiency of the system and can be reduced by a number of design decisions and modest equipment investments. Sizing the makeup air handlers to allow for lower pressure drop operation, specifying deeper and lower pressure drop filtration, minimizing the ducting required, using variable speed drives and efficient control sequences are all strategies to improve the energy performance over the baseline.

Opportunities to increase the efficiency of the make-up take two main forms: minimizing the amount required and maximizing the efficiency of the makeup air system. Minimizing the make-up air use is often difficult due to safety and product yield concerns. To reduce the make-up air requirements typically calls for pushing a tool vendor to provide tools that necessitate lower exhaust airflows (every cfm of exhaust equals a cfm of make-up air load). For most operators, a more practical option is to perform a tracer gas test on every tool to determine if a lower volume of exhaust air can be used and still maintain safe containment. There is strong anecdotal evidence that manufacturer's stated tool exhaust requirements are very conservative and can be reduced through such measurement. The second area, efficiency improvement, is most easily achieved by reducing the total static pressure of the makeup air handler and supply system.

The system's static pressure drop can be reduced through many well known design approaches (using lower face velocity filters, minimizing duct runs, using of turning vanes, etc.). One approach focuses on the air handler pressure drop, modifying the redundant make-up air control sequence to reduce the face velocity in the air handler. The air handler unit contains a significant amount of system's pressure drop in the form of two or three coils and two or more banks of filters. While redundant makeup air handlers are a common necessity in cleanroom design, keeping standby units shut off is not necessary and makes the operating units work harder. Operating the redundant unit in parallel reduces the amount of air that all the units are delivering. Because of the approximately cube relationship between airflow and fan brake horsepower (BHP) (described by the fan affinity 'laws'), the total HP of three units is less than operating only two for the same total flow supplied.

For this metric, an estimate of the airflow and the system power is required. The airflow should be the sum of all the installed makeup air unit design flows. If a redundant unit is installed, its airflow should be included in the sum (the redundant power will also be included to cancel it out). For units with a redundant fan, only the design airflow of the normally operating fan should be included.

The BHP should be determined by summing the brake horsepower (BHP) for all of the makeup air units, including any redundant units, operating at design flow. For units with a redundant fan, only the design power of the normally operating fan should be included. Fan power should be specified on project schedules or equipment submittals in the form of brake horsepower (BHP), which is easily converted to kW (0.746 x BHP = kW). This supplies the power portion of the metric.

Quantifying the annual savings in the design stage requires an estimate of the actual average operating makeup air flow in addition to the metric based on the above information. Benchmarking data is used again to determine the annual load as 75% of the installed make-up air capacity.

The *Makeup Air metric* will be w/cfm. This metric provides a system independent measure of efficiency. The volume of makeup air required by the process, a highly variable parameter between facilities and processes, does not impact the metric. The *typical design baseline is 1.04 w/cfm*. The baseline value is derived from the measurement of 12 cleanrooms. Interviews with 10 current cleanroom designers, owners or operators were done that support that current make-up design practice has not changed significantly since the benchmarked cleanrooms were measured. Multiplying the metric by the annual average makeup flow and 8,760 hours provides an estimate of the system's annual energy usage in kWh that can be used for rebate purposes.

Table 2. Makeup Air System Benchmarking Data

Site	Class	Fan BHP	Operating CFM	Total Installed CFM Capacity	w/cfm
A	10	15	17700	21000	0.85
I	10	5.3	6985	7225	0.76
J	10	247	132700	171000	1.86
K	10	153	70500	90503	2.17
B	100	6.4	9600	17600	0.67
C	100	8.5	13500	13500	0.63
E	100	1.48	2660	n/a	0.56
F	100	4.36	6010	n/a	0.73
G	100	104	82650	130000	1.26
H	100	13.3	8260	n/a	1.61
O	10000	3.9	4100	7000	0.95
L	10000	17.8	19475	30065	0.91

Chilled Water System

Chilled water is required for several critical uses in a cleanroom, including moisture removal in makeup air, sensible cooling, and process cooling systems. Chilled water flows directly or indirectly to every part of the cleanroom and must be maintained to continue production. The power used by the Chilled Water System includes all the systems components – pumps and cooling towers as well as the chiller power.

Chillers

Selecting chillers based on their energy performance at full load is a standard practice. However, to accurately assess the energy efficiency of a chilled water system, the chiller efficiency needs to be considered not at the full load condition, but at the part load conditions at which it will actually be operating. Barring equipment or design failure (undersizing that negatively impacts cleanroom operation), cleanroom chillers do not operate at 100% load for substantial lengths of time.

To estimate the actual chiller efficiency, the typical chiller load is required because chiller efficiency varies considerably with the load. Cleanroom chilled water plants are critical to safe and profitable facility operation. As an essential system, they are oversized as a matter of reliability and to allow for future load changes. Interior process loads in cleanrooms tend to be very high, 10 – 100 times those of office buildings, and it is not uncommon for production facilities to operate around the clock operation (Mills et al., 1996), resulting in continuous process loads ranging from 4 – 26 w/sf. The recirculation air fan system alone can add and additional 19 w/sf of continuous internal load (assuming 300 ACH, 9' ceilings and an efficiency of 2,325 CFM/kW). These high internal loads result in a significant baseline load that is not impacted by outside conditions. The only significant weather-impacted variable in the annual cleanroom load is the amount and the conditioning of the make-up air required (wall conduction is negligible with standard constructions when compared to the internal loads and windows are uncommon in production cleanrooms). The stability of the majority of the cleanroom load in the relatively temperate California climate is used to justify the use of the baseline data to estimate the average chiller operating point. To establish the baseline data, the chiller load was recorded over a 24 to 120 hour period. The average load point is approximately 50%. For baseline purposes, the chiller load point for efficiency should be 50%.

Two parameters are required to determine the chiller's efficiency: the condenser water temperature and the average load. Chiller efficiency can often vary by up to 50% based on chiller load and 20% or more based on condenser water temperature alone; for a meaningful efficiency metric, both the part load condition and the condenser water temperature must be properly defined. A constant condenser water temperature setpoint is often used in cleanrooms. The actual condenser water control method, constant setpoint or reset, should be used to determine the average condenser water temperature supplied to the chiller.

The chiller efficiency will be the efficiency provided by the chiller manufacturer's selection software at 50% load and the condenser water temperature that correlates with 50% load. Note that selection programs default to a condenser water reset (ARI 550/590 standard conditions include condenser water relief); if no reset is actually in the controls sequence to be used, the chiller must be rated without one. The chiller efficiency in kW/ton will be added to the efficiency of the other system components to produce the Chilled Water *Plant* metric.

Pumps

There are three standard pumping systems for chilled water distribution: primary, primary - secondary and primary – secondary – tertiary. These three systems vary significantly in efficiency.

For the purposes of evaluating the efficiency of the system, the pumping scheme chosen is not relevant as long as the power used by **all** the chilled water pumps (primary, secondary, tertiary, boosters, coil circulators, etc.) is accounted for. The pump power can be determined from the mechanical schedule, where the brake horsepower should be listed. The chilled water pumps, including any tertiary or booster pumps that may be included at coils with the air handlers, and condenser water pump horsepower should be summed to determine the pumping power. Frequently, the level of pumping redundancy will be equal to the level of chiller redundancy. For example, there will be a total of three chillers and 6 pumps (3 primary and 3 secondary), with 2 chillers and 4 pumps operating. So, by considering the full installed power and full installed capacity, the redundancy will cancel out when the installed pump power is divided by the installed chiller power. Where pumping redundancy differs substantially from chiller redundancy, adjustments may need to be made to determine a pump kW.

For constant flow pumps, the pump power is taken from the mechanical schedule as BHP and converted to kW. For variable flow pumps, the pump affinity laws can be used to relate the pump power to the pump flow. The flow can be estimated from the chiller load, 50%. Based on the affinity laws, at 50% flow the theoretical pump power will be only 12.5% of the design power. In practice, system control to a constant ΔP and VFD inefficiencies at half speed increase the power at half flow. A value of 25% of the mechanical schedule BHP should be used for variable flow pump power, unless the designer can support a higher turndown efficiency, for example, if the loop ΔP setpoint is actively reduced as demand drops.

The total annual average pumping power is divided by the annual average load to determine the pumping kW/ton.

Cooling Towers

The final, and typically smallest, component of the cooling plant power consumption is the cooling towers. The efficiency of cooling tower systems can vary by a factor of three or more between the most efficient and least efficient tower styles and implementations. For constant speed fans, the tower efficiency can be determined simply as the fan BHP converted to kW divided by the tower's capacity. When variable speed fans are employed, the fan affinity laws indicate an approximately cube relationship between load and fan power. As with the variable speed pumps, an average fan speed of 50% results in a theoretical fan power of 12.5% of the schedule value. To account for actual system efficiency, a value of 25% of the mechanical schedule BHP should be used to determine the cooling tower fan power.

Chilled Water System Metric

The *Chilled Water System metric* will be kW/ton, but it is important to realize that the metric quantifies the efficiency of not only the chiller, but also the pumping and cooling tower system. It is not the same as the chiller kW/ton. It is the Chiller kW/ton + Pumping kW/ton + Tower kW/ton. *The typical design baseline is 0.96 kW/ton*. The baseline value is derived from measurement of 12 cleanrooms, and interviews of current designers and operators to verify the benchmarked cleanrooms still represent current standard design practice. The benchmarking data is also used to create an estimate of the annual average plant load as 48% of the total installed capacity. Multiplying the metric by the annual average plant load and 8,760 hours gives an estimate of the system's annual energy usage in kWh that can be used for rebate purposes.

Table 3. Chilled Water Plant Benchmarking Data

Site	Class	Chiller (kW)	Avg. Load (tons)	Operating Cap. (tons)	Installed Cap. (tons)	CHW Pumps (kW)	CW Pumps (kW)	Cooling Tower (kW)	Efficiency (kW/Ton)
A	10	486	968	1600	2,400	95	40.4	45	0.69
T	10	3064	4000	N/A	N/A	242	303	146	0.94
K	10	1200	2208	3900	3,900	140	N/A	N/A	0.61
D	100	53.1	62.6	80	80	6	N/A	N/A	0.95
E	100	139	166	420	420	45.1	N/A	N/A	1.1
F	100	29.7	37.2	160	160	3.5	N/A	N/A	0.89
C	100	544	489	810	810	37.8	N/A	N/A	1.2
G	100	547	497	1000	1,000	101.3	38.2	105	1.6
H	100	360	896	1600	2,100	58	78	65	0.63
J	100	338.3	530.6	1350	1,350	204	63.7	11	1.2
S	100	1573	2560	N/A	N/A	347	371	29	0.91
V	100	N/A	N/A	N/A	N/A	N/A	N/A	N/A	0.83

N/A - Data not available (not in database or not retained from past studies)

Sample Energy Efficiency Measures

Based on benchmarking data and past experience in the cleanroom industry, a few Energy Efficiency Measures (EEMs) with wide potential were identified as follows:

Low Face Velocity (350-400 fpm)

Traditionally, the size of the air handler is designed using a coil face velocity of 500 fpm. Supposedly based on balancing the first cost with the lifetime energy cost of equipment, this decades-old rule of thumb face velocity was probably never intended for sizing a unit that operates 8,760 hour per year. The reason the face velocity is so important is it has a direct impact on the energy consumption of the air handler.

By selecting a lower face velocity, the pressure drop of the air handling unit, and the proportional energy consumption, is reduced. For example, a reduction in the face velocity reduces the power requirement by the square of the velocity reduction, i.e., a 25% reduction in face velocity yields a 44% reduction in power requirement. Often, a lower face velocity air handler can be utilized with a net *reduction* in the first cost, provided the savings from smaller motors, VFDs and infrastructure are considered (Owen 2000).

Free Cooling for Process Loads

Process cooling typically utilizes an isolated chilled water loop at 60-65°, making it an ideal candidate for free cooling. Process chilled water is usually produced from the plant chilled water using a heat exchanger located near the central plant. There are two options for free cooling, dependent upon whether or not a cooling tower can be dedicated to free cooling.

Option one is to install a precooler heat exchanger in series upstream of the chilled water fed heat exchanger(s). The precooling heat exchanger is cooled using condenser water produced from a cooling tower dedicated to free cooling. Dedicating a tower to free cooling allows the production of low temperature water, lower than some chillers allow in a condenser loop. The free cooling tower can be valved to allow it to function as redundant capacity for the main plant. Option two is to run the main chiller plant loop at a low condenser water temperature, around 55°. This requires careful chiller selection, active head pressure control of the condenser flow to each chiller or the implementation of a condenser water bypass to create a higher temperature condenser loop for the chillers.

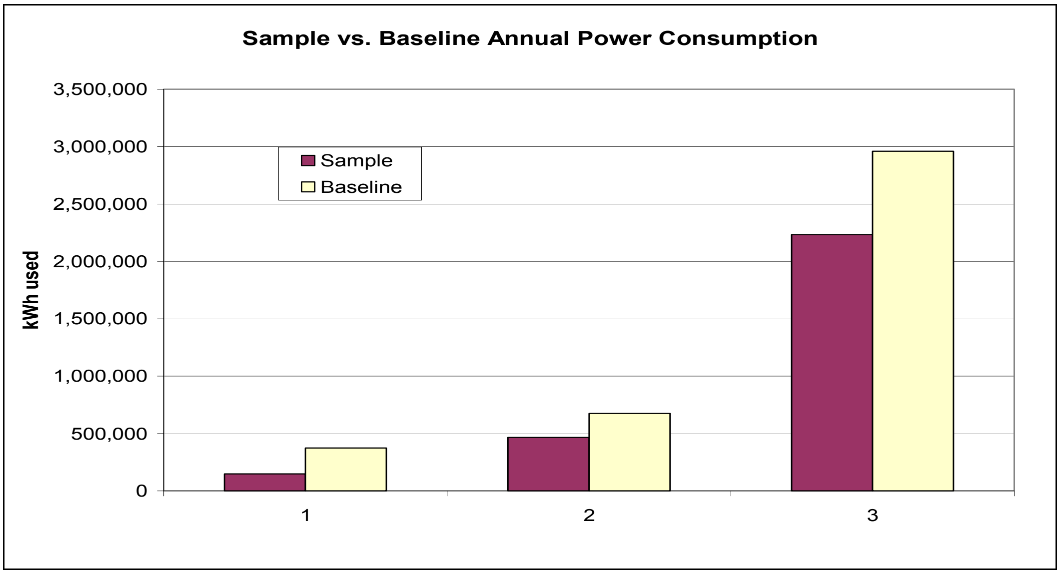
Free cooling chilled water plant efficiencies are not zero. The cooling tower and pumping system power are still considered, with savings of 0.4 – 0.6 kW/ton served by free cooling commonly resulting from the elimination of mechanical cooling operation.

Dual Temp Chilled Water Loops

A large amount of the chilled water usage in cleanrooms, often over half, is used for process chilled water, at 60-65°F, or sensible cooling of recirculation air, at 50-55°F. Chillers run more efficiently when producing higher temperature water, about 1.5% for every degree the chilled water temperature is increased. The smaller difference in temperature between the chilled water and the condenser water, the "lift," reduces the work the compressor must do. However, typical design does not realize any of the savings possible from producing chilled water directly at the higher temperature required by these significant loads. Medium temperature chilled water is traditionally produced using chilled water from the same plant that provides water for humidity control at 39-42°F, an inherently less efficient chilled water temperature than is actually required by much of the load.

The solution is to implement two chilled water loops, one to serve the medium temperature sensible and process cooling loads and one dedicated to the dehumidification loads. A medium temperature chiller loop would bypass the heat exchangers currently used to supply process cooling water and sensible cooling water from the lower temperature, inherently lower efficiency loop. In designs where return water mixing is used to produce sensible cooling water at the air handlers, a medium temperature loop can eliminate the maintenance, reliability, and efficiency costs associated with having numerous small pumps distributed throughout the facility.

A recent SBD rebate project provided new cleanroom construction energy design assistance for a fiber optics customer utilizing the benchmarking and baselining information. The project has produced significant estimated energy savings by focusing on the three subsystems discussed in this paper, including the free cooling EEM described above.



Note: 1 = Variable Primary Pumping, 2 = Free Cooling, 3 = High Efficiency VSD Chiller

Conclusion

While office buildings share very similar loads, cleanrooms are several times more energy intensive facilities that house a wide variety of process tools, and generally include common types of HVAC mechanical supporting systems. By using metrics developed in cleanroom benchmarking that focus on the efficiency of the mechanical supporting systems (Tschudi et al., 2001), it is possible to compare cleanrooms and improve the energy efficiency of these mechanical systems. Benchmarking studies have shown a wide variation in the end uses of the supporting systems, suggesting a good opportunity for incentive programs to encourage lower energy use designs through the determination of baseline performance or values.

It is recommended to continue energy benchmarking measurements of operating cleanrooms to offers a larger body of data for determining a set of more specific baseline standards for more specific classes of cleanrooms. The current baseline information focuses on higher cleanliness level (class 10 – class 100) semiconductor cleanrooms. While it is reasonable to extrapolate this mechanical system data to lower cleanliness cleanrooms and cleanrooms of with identical mechanical support system design in other industries, additional measurement data should be continuously sought to refine the benchmark database.

Since whole building energy use is process and facility dependant, the use of this system-based approach baselining effort will provide a generally less facility-dependant and more evaluative basis that is more consistently transferable among facilities. By using these baseline metrics to determine standard energy efficiencies, incentive or rebate programs administered by utilities can be effectively and efficiently used to encourage sustainable cleanroom energy efficiency design.

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