Energy Consumption Characteristics of Commercial Building HVAC Systems

Volume II: Thermal Distribution, Auxiliary Equipment, and Ventilation

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1. Executive Summary

This report is the second volume of a three-volume set of reports on energy consumption in commercial building HVAC systems in the U.S. The first volume, in process but not yet complete, focuses on energy use for generation of heating and cooling, i.e. in equipment such as boilers and furnaces for heating and chillers and packaged air-conditioning units for cooling. This second volume focuses on *parasitic* energy use, the energy required to distribute heating and cooling within a building, reject to the environment the heat discharged by cooling systems, and move air for ventilation purposes. Parasitic energy use in commercial building HVAC systems accounts for about 1.5 quads of primary energy[1] use annually, about 10% of commercial sector energy use. The third volume in the set will address opportunities for energy savings in commercial building HVAC systems.

The energy use estimates presented in this report have been developed using a rigorous bottom-up approach which has not previously been used to estimate national parasitic energy consumption. Distribution of the commercial building floorspace among building type, system type, and region was based largely on the 1995 Commercial Building Energy Consumption Survey (CBECS95, Reference 3). Models for cooling and heating loads were obtained from Lawrence Berkeley National Laboratory (LBNL) and were based on a set of over 400 prototype building models (Reference 19). Models of HVAC equipment design loads and operating characteristics were developed based on engineering calculations, product literature, discussions with equipment suppliers, and actual building site-measured data collected by Xenergy[2] as part of the XenCAP™ demand-side management (DSM) program. The XenCAP™ data was also used for checking the building energy use models. Energy use estimates were developed for more than 1,500 technology/market segments representing the different building types, regions, system types, and equipment considered in the study.

Figure 1-1 below shows the breakdown of this energy use by equipment type. Most of the energy is associated with fans, either the supply (and return) fans of air-handling units, or the exhaust fans used for ventilation.

Supply fans use so much energy (about 0.75 quad total) because (1) they are used in virtually 100% of system types as defined (note that the evaporator fans of packaged or individual systems as well as fan-coil unit fans are considered in this category), (2) air is an inherently inefficient heat transfer medium, (3) typical air distribution design practice involves considerable pressure drop for filtration, cooling and heating coils, terminal boxes, and diffusers, and (4) many of these fans operate at 100% power during all building occupied periods.

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[1] Conversion of site electricity use to primary energy is based on 11,005 Btu per kWh heat rate which includes transmission and distribution losses.

[2] Xenergy is a well-established energy service company which has done energy auditing work on about 5% of the nation's commercial floorspace. The XenCAP™ database is described in more detail in Appendix 1.

Exhaust fans, while generally representing much less horsepower than supply fans, do use considerable amounts of energy (about 0.5 Quad), since they are nearly all operated at 100% power during all building occupied periods. The contributions of central system auxiliary equipment (condenser water and chilled water pumps, cooling tower fans, and a portion of the condenser fans) are relatively modest because (1) their power input per ton of cooling is very low and (2) central systems represent less than one third of commercial building floorspace. Some of this equipment also has very low utilization values due to its operating characteristics – it is used at full power very infrequently.

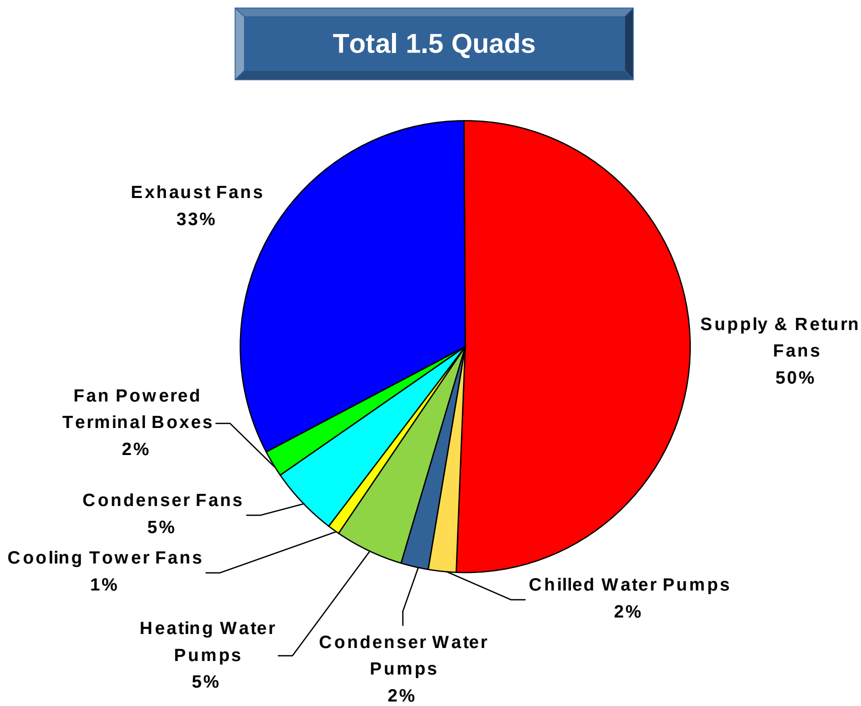


Figure 1-1: Parasitic Primary Energy Use -- Equipment Breakdown

The distribution of parasitic energy use by building type is shown in Figure 1-2 below. The building categories are identical to those used in the 1995 Commercial Building Energy Consumption Survey (CBECS95-Reference 3)[3]. The most energy use is in offices, representing 25% of commercial building fan and pump energy. The Office, Mercantile and Service, and Public Building Categories are large energy users due to their large floorspace in the commercial sector (they each represent at least 7 billion sq. ft.). The health care sector, also a large energy use category, has high energy use intensity (energy use per square foot of floorspace) due to high ventilation rates, high cooling loads, and long hours of occupancy.

[3] The Building Category "Public Buildings" includes CBECS95 categories "Public Assembly", "Public Order and Safety", and "Religious Worship".

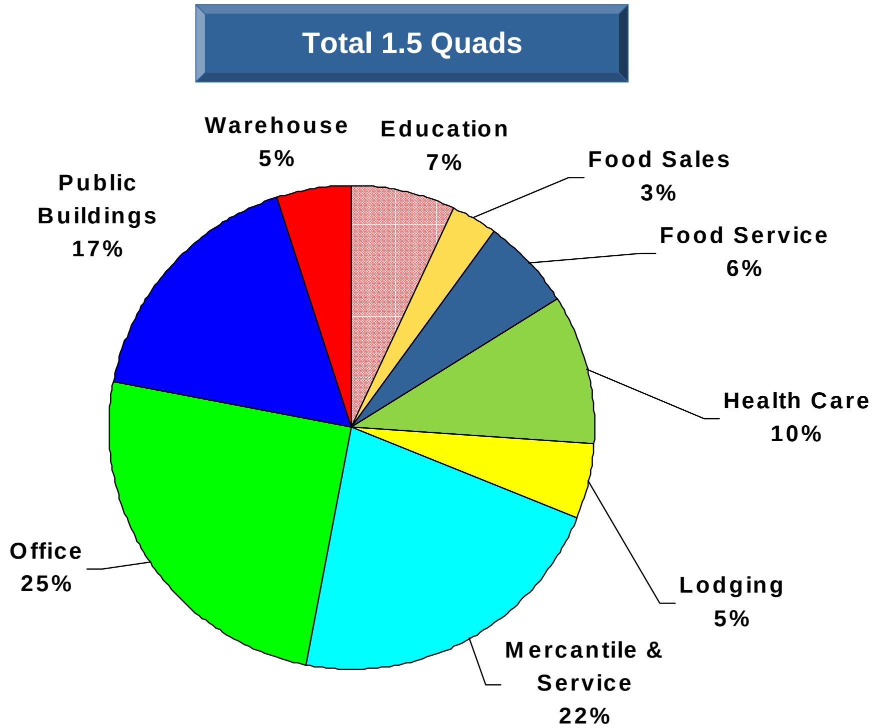


Figure 1-2: Parasitic Primary Energy Use -- Building Type Breakdown

The distribution of HVAC parasitic energy use by geographic region strongly reflects the commercial building floorspace breakdown. The energy use and floorspace distributions by region are show in Figure 1-3 below. The differences in the two distributions are due to the expected differences in energy use intensity resulting from higher cooling loads in warmer regions.

The share of parasitic energy associated with different types of HVAC systems is shown in Figure 1-4 below. The largest percentage of the energy use is with packaged systems[4], since these systems represent the most commercial building floorspace (41%). Central systems, which use chilled water for thermal distribution from a central chiller to air-handling units or fan-coil units, were split into three groups in the study. These systems account for about a third of the commercial building HVAC parasitic energy use. Individual air-conditioners and uncooled buildings have relatively modest fan and pump energy use.

[4] Packaged systems are cooled with single-package rooftop air-conditioning units or with two-package "split systems," which cool with direct-expansion (DX) cooling coils (rather than chilled-water coils), and deliver cooling to the building through ductwork.

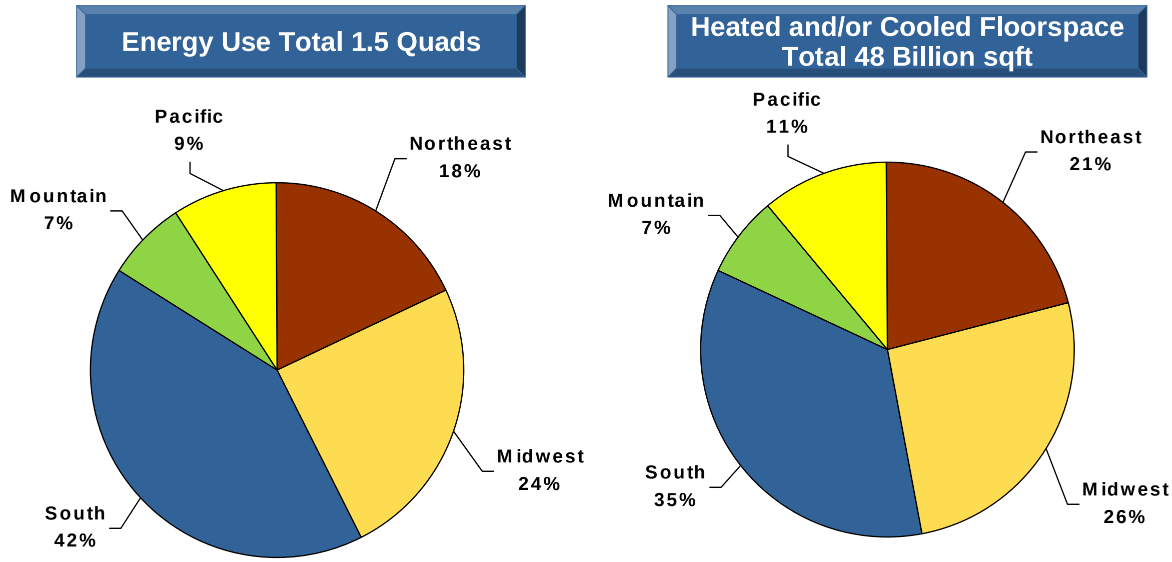


Figure 1-3 : Parasitic Primary Energy Use and Floorspace– Regional Breakdown

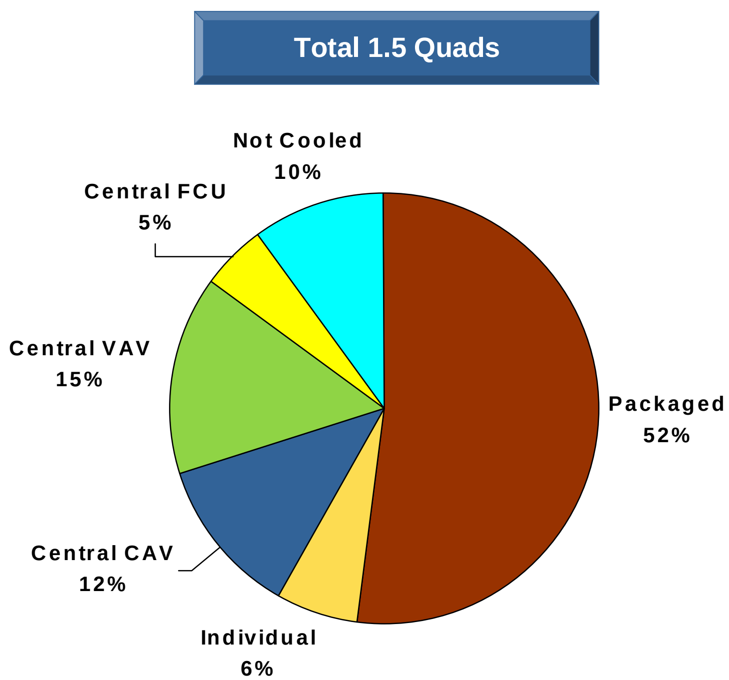


Figure 1-4: Parasitic Primary Energy Use -- System Type Breakdown

Central variable air volume[5] (VAV), central constant air volume (CAV), and constant air volume packaged cooling systems (based on a large New York office application) are compared in Figure 1-5 below. This comparison of prototypical systems shows that the central system with VAV air handling units is typically more efficient than a packaged system. The differences are primarily due to:

- Heat rejection in the central system using a cooling tower, which enhances heat rejection through evaporation of condenser water.
- Use of larger more-efficient refrigerant compressors for the central systems
- Constant-volume operation of the packaged unit supply fan in spite of varying cooling loads

These three factors more than make up for the central system disadvantages of additional heat exchangers and thermal distribution associated with the central chiller. Incorporation of energy-saving features such as VAV, high-efficiency compressors, and evaporative condensing in a packaged unit would eliminate the efficiency advantage of central systems.

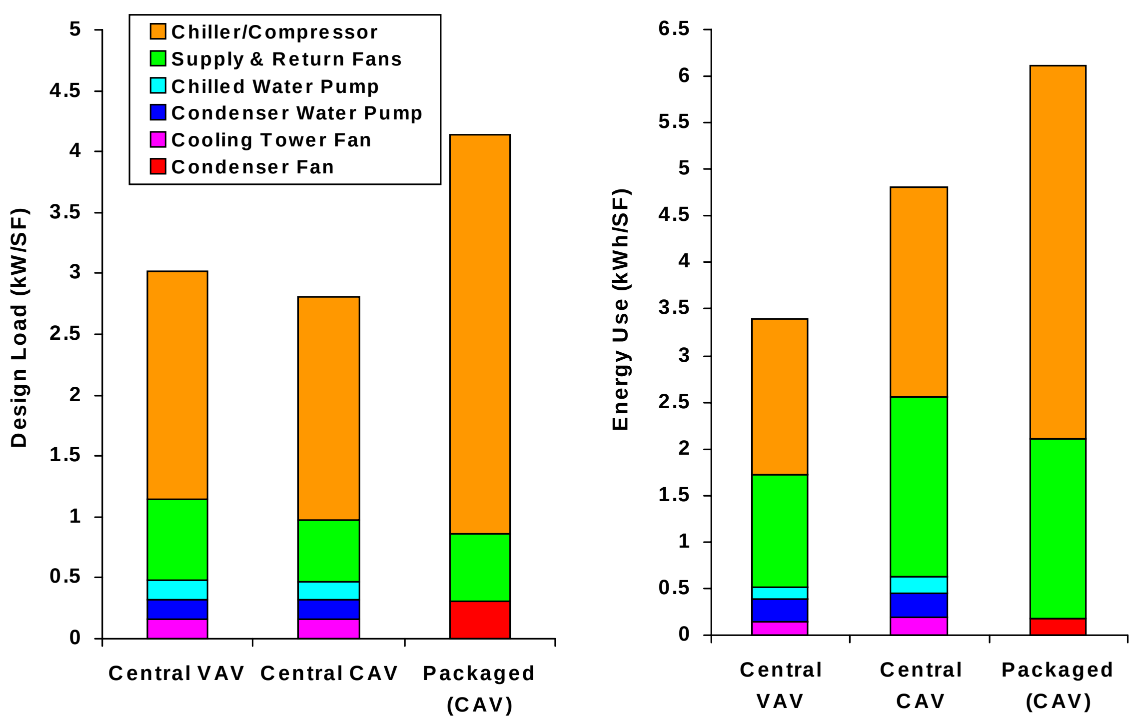


Figure 1-5: Design Load and Energy Use Comparison of Central VAV, Central CAV and Packaged Systems

[5] VAV air handling units vary air flow to supply only as much cooling as is needed

Segmentation of the commercial building stock by buildings and HVAC systems is shown in Figure 1-6 below. The building groups with the most floorspace are mercantile and service, office, education, and public buildings. Distributions of HVAC systems within each building type vary significantly. These differences are largely due to the varying HVAC needs of the buildings (for instance, many of the education buildings are not cooled because schools are closed during the hot summer months).

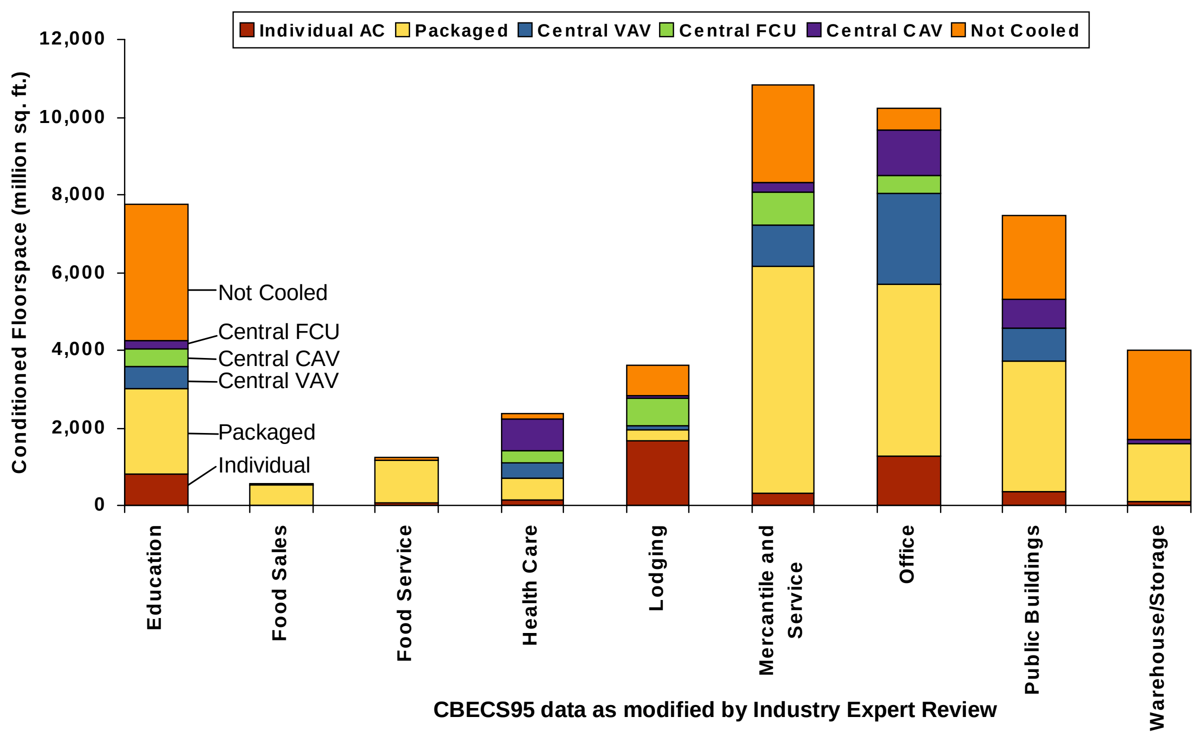


Figure-1-6: Building Stock Segmentation: Building Types and System Types

2. Introduction

2.1 Background

Energy use for heating and air-conditioning accounts for more than 25% of the primary energy consumed in commercial buildings in the U.S. (EIA, Annual Energy Outlook 1998, Reference 1). Parasitic energy use, the energy used to power the fans and pumps which transfer heating and cooling from central heating and cooling plants to conditioned spaces, can represent a significant portion of this energy (from 20% to 60% of HVAC electricity use in a building). There is currently very little information about the national impact of this important part of the energy use "puzzle". A good understanding of the magnitude of parasitic energy use and the system characteristics affecting overall system HVAC energy use is needed.

Significant efforts have been made in the past 20 years to reduce the energy use of chillers and refrigerant compressors. In this period, the typical efficiency of a centrifugal chiller has increased 34%, from a COP of 4.24 to 5.67 (from 0.83 kW/ton to 0.62 kW/ton) (Reference 2). Such improvement to the efficiency of fans and pumps has not occurred. Although variable air volume (VAV) systems have been used for many years, the fan energy savings associated with this system type were limited by the technologies available for air flow reduction. In more recent years, the improved reliability and lower cost of variable speed drives (VSD's) has led to increased potential for fan power savings, but other trends associated with indoor air quality (IAQ) concerns (such as increases in minimum air flow rates, increased filtration, and use of series fan boxes) have limited fan power reductions.

The reduction of fan and pump power is more complicated than the reduction of chiller power because of a greater dependence on system design, testing and balancing, system control, and system operation. As compared with chiller efficiency, responsibility for efficient thermal distribution rests more with the engineer who designs the system, the installing contractor, and the operating staff than with the component manufacturer.

The reduction of fan and pump power is also complicated by the myriad of possible system types, and the many system components which must be taken into account in order to achieve optimum performance. In comparison, chiller performance is much easier to define and quantify.

2.2 Study Approach, Scope, and Statement of Work

This report is the second of three volumes characterizing commercial HVAC energy use:

- Volume 1: Chillers, refrigerant compressors, heating systems — baseline equipment and current energy use.

- Volume 2: Thermal Distribution, Auxiliary Equipment, and Ventilation — baseline equipment and current energy use. This equipment, referred to as "parasitic" in this report, consists primarily of fans and pumps.
- Volume 3: Assessment of energy saving options, identification of barriers to implementation, and development of programmatic options.

A detailed examination of cooling and heating delivery equipment in commercial buildings is covered in this report: system configurations; estimates of energy use; market characterization; trends in system and equipment design.

The study focused on central station HVAC systems which use chilled water for cooling and/or hot water or steam for heating as well as packaged and individual HVAC systems. All parasitics included in the overall HVAC systems have been addressed: condenser water piping systems, cooling towers, chilled water piping systems, central station air handling units, ductwork, terminal units, and exhaust or return air systems which are required for space conditioning.

The study examined a large range of commercial building types, including all of the building categories in the Department of Energy's Commercial Building Energy Consumption Survey (Reference 3). The building stock was further segmented by HVAC system and by geographic region. The tasks comprising the study were as follows:

Task 1: Characterize Equipment contributing to Parasitic HVAC Energy Use and HVAC Distribution System Design Practice

Typical systems for delivery of heating and cooling and for supply/exhaust of air for prototypical commercial buildings were described. System descriptions were segmented by building function, size, and climate. HVAC system components (fans, pumps, cooling towers, etc.) control strategy for these prototypical systems were described.

Task 2: Establish a Baseline for HVAC Parasitic Equipment Energy Consumption

Annual site and primary energy use associated with the parasitic equipment of the prototypical HVAC systems were estimated. Total US commercial sector primary energy use for HVAC parasitics was estimated for the examined prototypical systems and compared with estimates prepared by other investigators.

Task 3: Identification of Trends

Issues and trends affecting parasitic energy use were identified, along with the drivers for these trends (IAQ, system costs, energy costs, controllability, etc.).

Task 4: Market Characterization

The HVAC equipment design and selection process was described. The key decision makers have been identified, the system/equipment supply chain was described, and the most important purchase criteria were discussed.

Task 5: Industry Review

The draft final report was reviewed by 8-10 HVAC industry representatives, including equipment manufacturers, A&E's, and ESCO's/utilities. This allowed practitioners to comment and revise input data, modeling, results, etc.

2.3 Report Organization

This report is organized as follows:

Section 3 describes and illustrates some of the common commercial building HVAC system types, and also the equipment comprising these systems.

Section 4 discusses the market for the thermal distribution and auxiliary equipment covered by the study.

Section 5 lays out the estimate of current fan and pump energy use which was developed in this study, discussing calculation methodology, underlying assumptions, and results.

Conclusions and Recommendations are presented in Section 6.

Section 7 lists the References.

Five Appendices are included in this report.

The first describes the XenCAP™ building energy use database, which was used as input and backup for many of the calculations.

Appendix 2 provides tables with the study's calculated segmentation of commercial building floorspace by building type, region, and HVAC system type.

Appendix 3 provides a detailed explanation of the modeling methodology used to calculate equipment energy use.

Appendix 4 provides data from Reference 3 which is used as a basis for the floorspace segmentation.

Appendix 5 is a summary of interviews with industry experts conducted during the course of the study to help answer some of the important underlying questions regarding fan and pump energy use.

The estimation of energy use by thermal distribution systems and other parasitic loads in commercial buildings is invariably tied to the system type which is under consideration. The systems are comprised of energy-using equipment such as fans and pumps, as well as passive equipment such as ductwork and filters, which nevertheless strongly affect system energy use.

There are a myriad of HVAC system types which have been used for commercial buildings. This study has attempted to provide a reasonable representation of the system types in the US commercial building stock, but by no means is exhaustive in covering all possible variations.

3. Description of Systems and Equipment

This section gives a brief description of the system types under consideration in the study. It follows with description of the important equipment types. This study is mostly focused on central systems, but parasitic energy use in packaged and individual systems is also addressed.

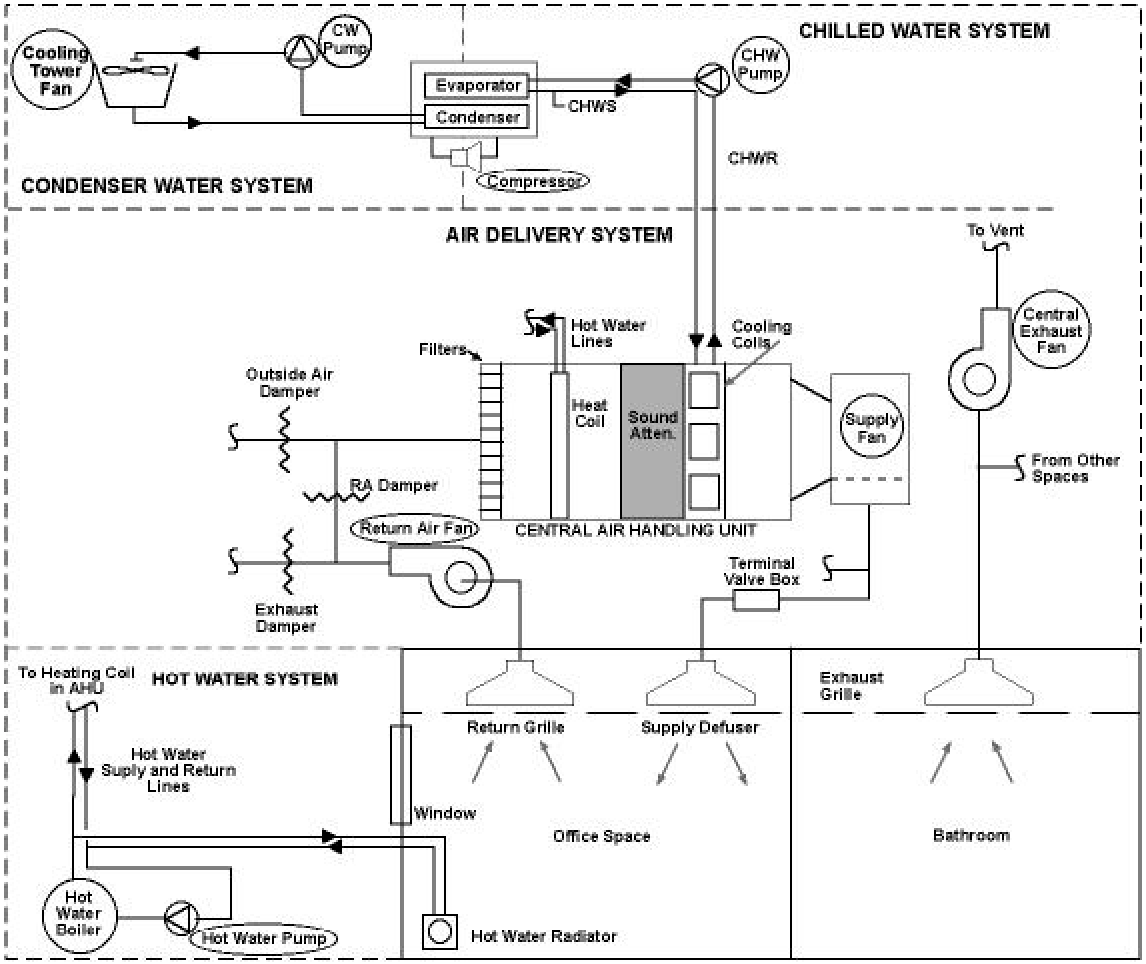
3.1 System Types

HVAC system types in commercial buildings are broken down into four broad categories for the purposes of this study: central, packaged, individual AC and uncooled. Central systems are defined as those in which the cooling is generated in a chiller and distributed to air-handling units or fan-coil units with a chilled water system. Heating in central systems is generated in a boiler and distributed to local fan-coil units, radiators, or baseboard heaters via a steam or hot water system. Packaged systems include rooftop units or split systems which have direct-expansion cooling coils, with heat rejection remote from the cooled space. Individual AC systems involve self-contained packaged cooling units which are mounted in windows or on an external wall such that cooling occurs inside and heat rejection occurs outside. Uncooled buildings of interest are heated but not cooled.

3.1.1 Central

Central systems are defined as any HVAC systems which use chilled water as a cooling medium. This category includes systems with air-cooled chillers as well as systems with cooling towers for heat rejection. Heating in these systems is usually generated in a boiler and is distributed in hot water or steam piping.

A central system serving office space is depicted in Figure 3-1 below. The space which is conditioned by the system is in the lower right part of the figure. The system is broken down into three major subsystems: the air-handling unit, the chilled water plant, and the boiler plant.



Note: Power-using components are circled

Figure 3-1: Schematic of a Central System

The air-handling unit conditions and supplies air to the conditioned space. Air is taken by the unit either from outside or from the space itself through a return air system. The three dampers are controlled to mix the air according to the chosen control strategy. When the enthalpy of outdoor air is lower than that of the return air, it is more economical to use the outdoor air for cooling of the building than to circulate return air (this is called economizing). When the outdoor air is warmer than return air, or when the outdoor temperature is very low, a minimum amount of outdoor air will be mixed with the return air in order to provide fresh air ventilation for removal of indoor contaminants such as carbon dioxide. The air is filtered and conditioned to the desired temperature (the air may require preheating rather than cooling, depending on outdoor conditions). Preheating and cooling are done with heat exchanger coils which are supplied with a heat exchange medium, typically steam or hot water for heating, and chilled water for cooling.

Air flow to the conditioned space may be controlled, as in the case of a variable air volume (VAV) system, with a terminal valve box. The air is finally delivered to the space through a diffuser, whose purpose is to mix the supply air and the room air. The terminal box may or may not have a reheat coil, which provides additional heat when the space does not need to be cooled or needs less cooling than would be delivered by supply air at the terminal box's minimum air quantity setting. Constant air volume (CAV) systems, which are not allowed by energy codes in many applications, do not reduce air delivery rates and are dependent on reheat coils to control the delivered cooling.

Air leaves the conditioned space either through the return system, or through the exhaust system. In many installations, the ceiling plenum space is used as part of the return ducting in order to save the cost of return ductwork.

The chilled water system supplies chilled water for the cooling needs of all the building's air-handling units. The system includes a chilled water pump which circulates the chilled water through the chiller's evaporator section and through the building. The system may have primary and secondary chilled water pumps in order to isolate the chiller(s) from the building: the primary pumps ensure constant chilled water flow through the chiller(s), while the secondary pumps deliver only as much chilled water is needed by the building. The chiller is essentially a packaged vapor compression cooling system which provides cooling to the chilled water and rejects heat to the condenser water. The condenser water pump circulates the condenser water through the chiller's condenser, to the cooling tower, and back. The cooling tower rejects heat to the environment through direct contact of condenser water and cooling air. Some of the condenser water evaporates, which enhances the cooling effect.

The heating water system indicated in Figure 3-1 includes a boiler and a pump for circulating the heating water. The heating water may serve preheat coils in air-handling units, reheat coils, and local radiators. Additional uses for the heating water are for heating of service water and other process needs, depending on the building type. Some central systems have steam boilers rather than hot water boilers because of the need for steam for conditioning needs (humidifiers in air-handling units) or process needs (sterilizers in hospitals, direct-injection heating in laundries and dishwashers, etc.).

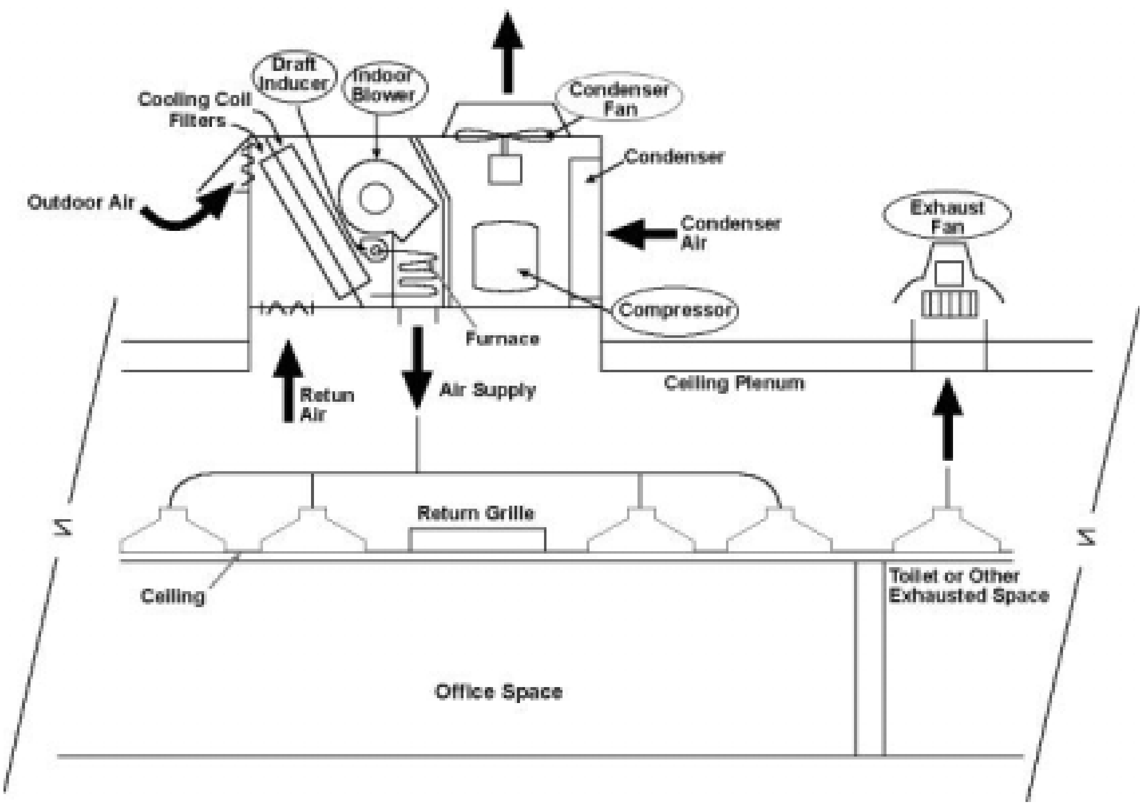
For the purposes of this study, the central system category has been further broken down into the following.

- Central systems with VAV air-handling units
- Central systems with CAV air-handling units
- Central systems with fan-coil units for delivery of cooling (Fan-coil units are small typically unducted cooling units).

3.1.2 Packaged

Packaged systems include both unitary systems such as rooftop units, and split systems. Essentially, these are systems which do not used chilled water as an intermediate cooling medium. The cooling is delivered directly to the supply air in a refrigerant evaporator coil. Packaged units have either a gas furnace or an electric resistive heating coil for heating, or they are designed as heat pumps (in which the refrigeration system pumps heat from the outdoors into the building).

A packaged system serving office space is depicted in Figure 3-2 below.



Note: Power-using components are circled.

Figure 3-2: Schematic of a Packaged System

The figure shows a rooftop unit used for cooling an office. Again, air is circulated from the conditioned space through the unit and back. Rooftop units can use outdoor air for cooling when outdoor temperature is cool enough, using the outdoor and return dampers to mix the air. The air moves through a filter, through the evaporator coil, through the indoor blower, through a furnace coil, and is supplied to the space through ductwork and supply diffusers. The figure shows air being returned through the ceiling plenum. Some air is pulled from the space through exhaust fans.

Cooling for the unit is again provided by a vapor compression cooling circuit. However, cooling is delivered directly to the supply air, and the heat is rejected in a condenser coil directly to the ambient air.

Heating for the rooftop unit in the figure is provided with a furnace. Most small rooftop units use draft inducing fans to move combustion products through the furnace coil. Some larger units use forced draft fans which push combustion air into the furnace. Heat can also be provided by resistance electric heat or by the vapor compression circuit (operating as a heat pump).

In a split system, the two sides of the unit shown in the figure are separated, with refrigerant piped between them. A condensing unit, consisting of the refrigerant compressor, the condensing coil, and the condensing fan, is located externally. The indoor unit, consisting of the evaporator and indoor blower are located near or in the conditioned space. Inclusion of a furnace or provision for intake of outdoor air will depend on proximity of the indoor unit to the outside.

3.1.3 Individual Room Air Conditioning

Individual room air conditioning includes window AC units, packaged terminal air-conditioners (PTAC's), packaged terminal heat pumps (PTHP's), and water-loop heat pumps. Window AC units similar to those used in residences are frequently used in commercial applications. PTAC's or PTHP's are used primarily in hotels and motels. The unit is mounted on an external wall, and a hole in the wall provides access to outdoor air, which is used for ventilation, heat rejection, and heat pumping (for the PTHP).

Water loop heat pumps (also called California heat pumps) are similar to PTHP's except that water piped to the unit takes the place of the outdoor air. This allows more flexibility in placement of the unit, allows pumping of heat from warm to cool parts of the building through the circulated water loop, but requires installation of the water loop system. The water loop requires a cooling tower and a boiler for heat rejection or heat addition when the building thermal loads do not balance.

3.2 Equipment

A fairly exhaustive list of equipment contributing to HVAC parasitic loads is shown in Table 3-1 below. The table also gives typical ranges of the design load intensity in W/sqft of this equipment when used in commercial applications. The study has given less emphasis to some of the equipment types with lower load intensity. The table indicates the equipment types which were included in the fully-segmented baseline energy use analysis of Section 5.

Table 3-1: Parasitic Equipment Types

Equipment Type	Typical Design Load Intensities in Commercial Building Applications (W/sqft)	Comments	Included in Full Baseline Estimate (Y/N)
DISTRIBUTION SYSTEM FANS			
Central System Supply Fans	0.3 – 1.0		Y
Central System Return Fans	0.1 – 0.4		Y
Terminal Box Fans	0.5		Y
Fan-Coil Unit Fans	0.1 0.3	Unducted With some ductwork	Y
Packaged or Split System Indoor Blower	0.6		Y
PUMPS			
Chilled Water Pump	0.1 – 0.3		Y
Condenser Water Pump	0.1 – 0.2		Y
Heating Water Pump	0.1 – 0.2		Y
Condensate Return Pump	0.002 – 0.005		N
Boiler Feed Water Pump	0.02 – 0.05		N
Domestic Hot Water Recirculation Pump	0.002 – 0.005		N
OTHER			
Cooling Tower Fan	0.1– 0.3		Y
Air-Cooled Chiller Condenser Fan	0.6		Y
Pneumatic Controls Compressor	0.03 – 0.06		N
Exhaust Fans	0.05 - 0.3	Strong dependence on building type	Y
Condenser Fans	0.6		Y
Furnace Induced Draft Fan	0.01		N
Furnace Forced Draft Fan	0.005		N
Boiler Burner Fan	0.005 - 0.01		N

Source: ADL estimates based on product literature, discussions with industry representatives, and engineering calculations.

3.2.1 Air-Handling Units

Air handling units are used in central systems to move and condition air which is supplied to the conditioned spaces. A typical air-handling unit is shown in Figure 3-3 below. Mixing dampers are shown at left end of the unit. The outdoor air dampers at the far left are fully open and the return air dampers at the top of the unit are fully closed. To the right of the dampers is the filter section. The fan, showing inlet dampers, the pulley and belt drive system, and the drive motor, is at the right of the unit. Heating and cooling coils are shown just to the left of the fan.

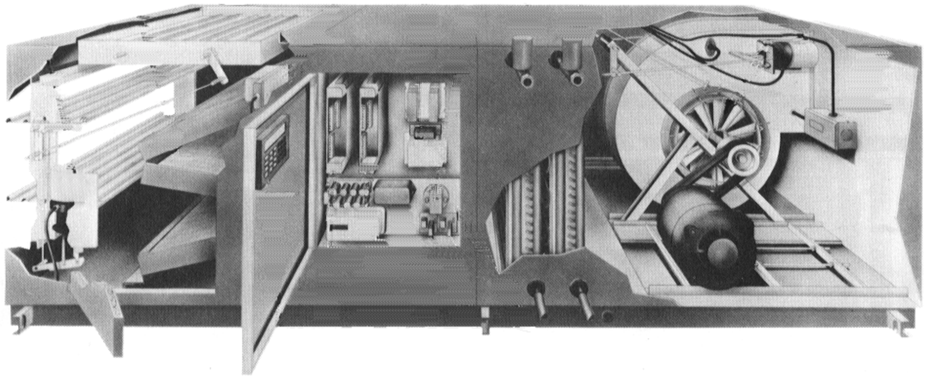


Figure 3-3: A Central System Air Handling Unit

Source: Carrier

Many air-handling units are manufactured in modular sections. Figure 3-4 shows a typical fan section. The small rectangular opening at the right is the fan discharge connection. The figure also shows two common fan wheels. Forward-curved blades provide more static pressure for a given size and wheel speed, but backward-curved and airfoil blades are more efficient. The greater the air flow, the more likely the unit will have backward-curved or airfoil blades.

Figure 3-4: An Air-Handling Unit Fan Section, Showing Two Fan Types

Source: Carrier

Typical filter sections are shown in Figure 3-5 below. The main filter will usually be preceded by a coarser prefilter. Three common main filter types are shown: the angle filter, the roll filter, and the bag filter. In an angle filter, square two-inch thick filters slide into the angled racks. The roll filter automatically rolls from the fresh end to the used end, thus reducing the frequency of manual filter replacement. Bag filters consist of multiple bags of filter material, thus packing much filter surface into a limited volume.

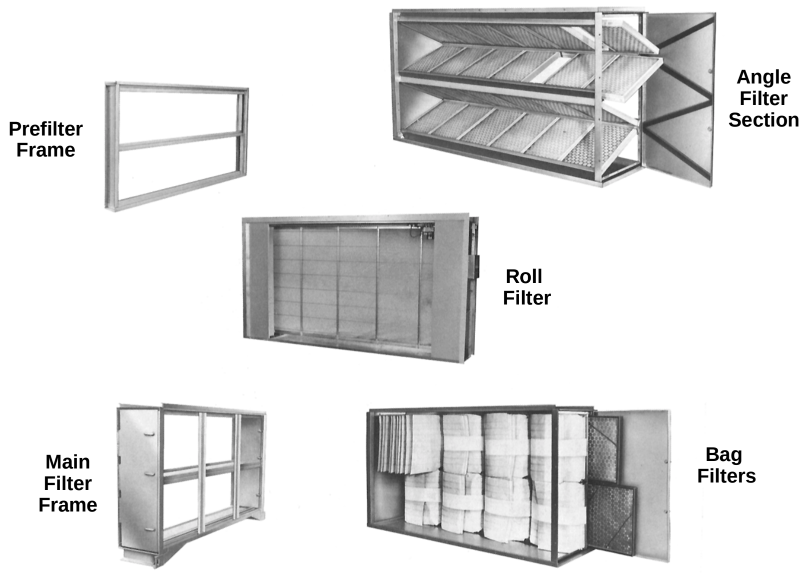


Figure 3-5: Typical Air Handling Unit Filter Sections

3.2.2 Terminal Units

Terminal units provide local control of airflow in a large central air-conditioning system. The majority of terminal units are used for air volume control in variable air volume (VAV) systems. Many also have reheat coils. Most of these are valve boxes, which simply have a sophisticated damper (valve) for control of air flow.

Figure 3-6 shows a typical fan-powered terminal box. Figure 3-7 shows the difference between series and parallel fan-powered terminal boxes. These units allow for air flow in addition to the air supplied by the central fan. In series boxes the central air and ceiling plenum air are mixed before entering the fan — these units typically involve constant operation of the fan. In parallel boxes, where the fan is cycled as a first stage of reheat, the ceiling plenum air is mixed with central air after passing through the fan. Fan boxes are used often in perimeter spaces, where the perimeter heating load requires that more air be delivered to satisfy the load. The first stage of control as space temperature falls is to reduce central air flow. This reduced volume may not be sufficient as temperature falls further and the thermostat calls for heating.

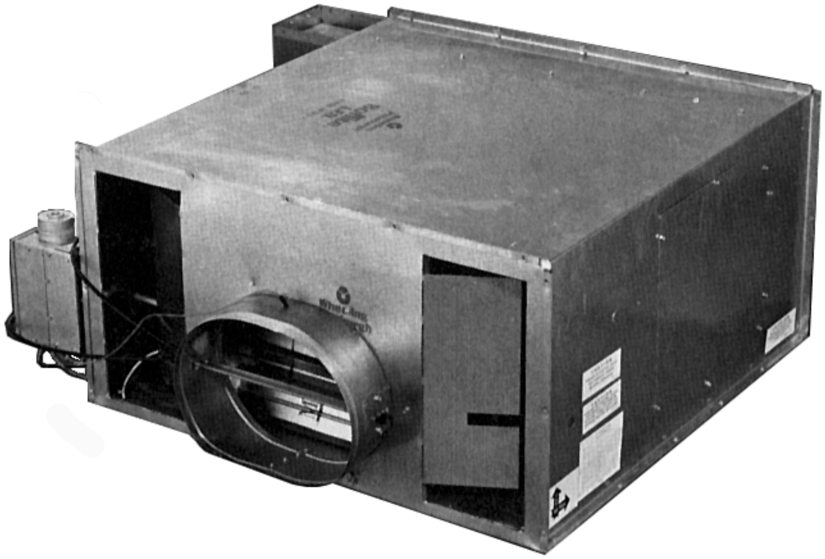


Figure 3-6: A Fan-Powered Terminal Box

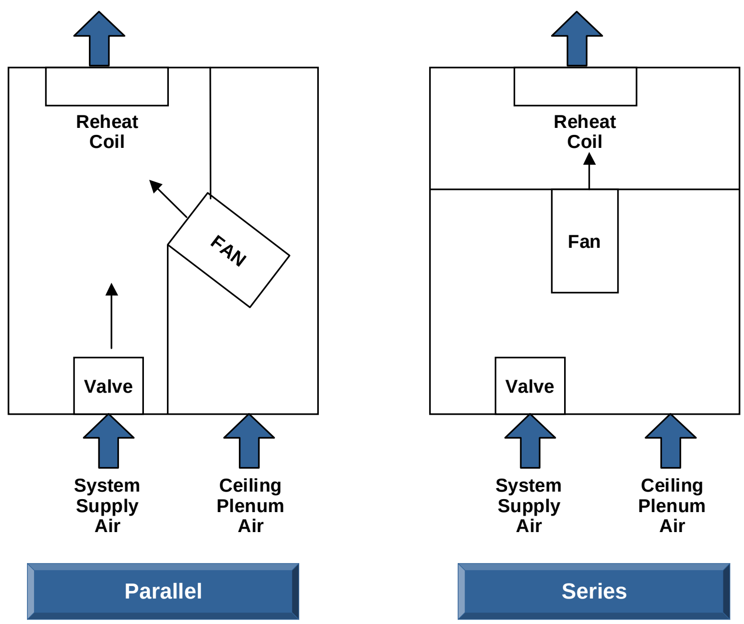


Figure 3-7: Parallel and Series Fan Boxes

Final delivery of air to the space is through diffusers. Figure 3-8 shows a diffuser used for VAV systems. The opening of VAV diffusers varies in order to assure adequate mixing of room and supply air over the range of air flow rates. Figure 3-9 shows a typical ceiling diffuser, which may be used for VAV or CAV systems.

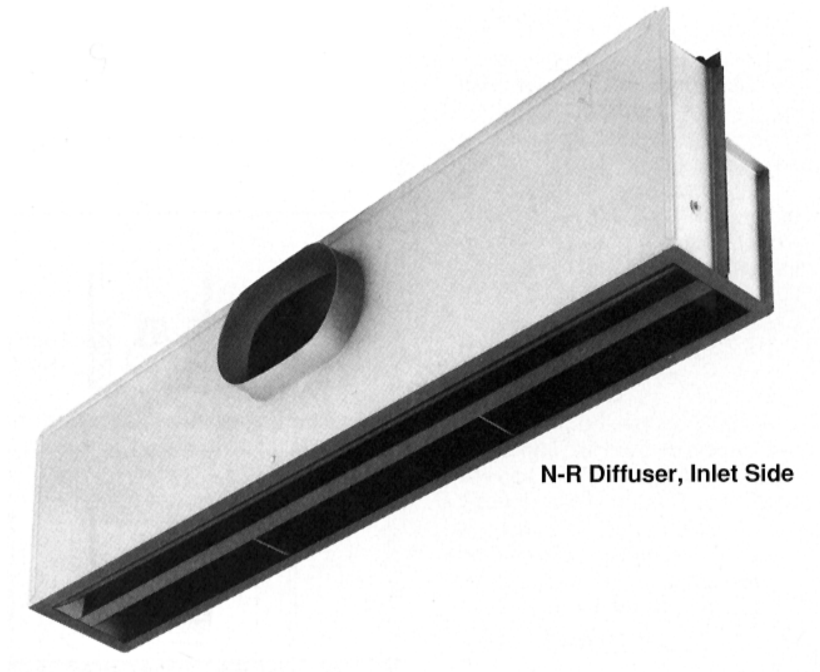


Figure 3-8:   A Variable-Volume Diffuser

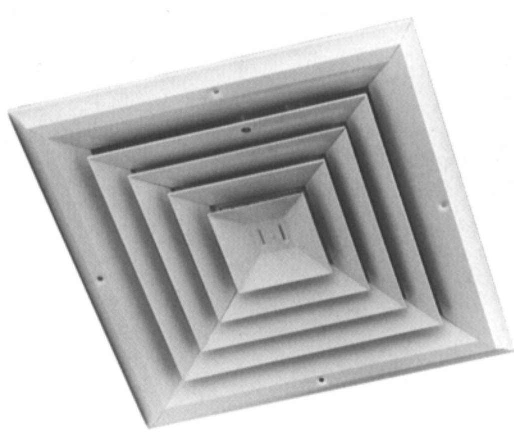


Figure 3-9:   A Typical Ceiling Diffuser

3.2.3 Exhaust Fans

Figure 3-10 below shows a typical roof exhaust fan. This style of fan is used in many applications, especially in flat-roof buildings with a limited number of floors. The roof exhaust fan mounts easily on the roof at the top of an exhaust riser, making additional mechanical room space unnecessary. Another advantage is that the entire exhaust duct system within the building is at negative pressure, eliminating the possibility of contamination of interior spaces by leakage of exhaust air.

In other exhaust applications a variety of exhaust fans are used which are ducted on the inlet and discharge. The most common of these configurations is a single-width single-inlet (SWSI) centrifugal fan.

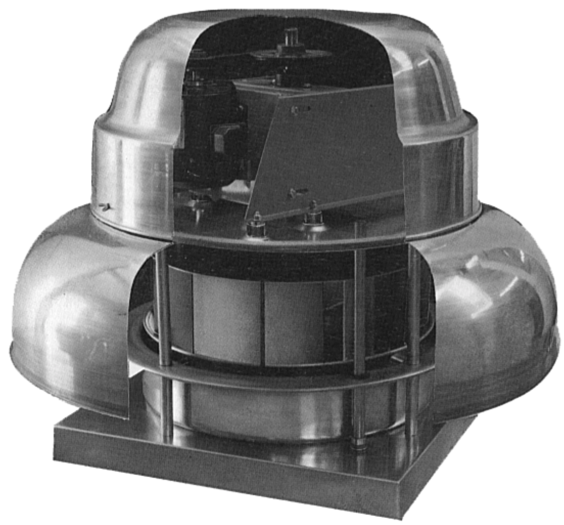


Figure 3-10: A Belt-Drive Rooftop Exhaust Fan

3.2.4 Pumps

Pumps are used in HVAC systems for circulation or transfer of water or water/glycol solutions. Figure 3-11 and Figure 3-12 below show two common HVAC pump types. The split case horizontal pump is used in many larger applications (>1000 gal/min). It has a higher purchase cost than other pumps, but is more efficient and the split case allows inspection and maintenance without disturbing the rotor, motor, or the connecting piping. End-suction pumps are used in smaller applications. Both pumps in the figures show the pump body, the motor, and a mounting frame. Shaft couplings are hidden by the shaft guards. Some smaller end-suction pumps are direct-coupled: the impeller mounts directly on the shaft of a face-mounted motor. A third popular HVAC pump is the in-line centrifugal, in which inlet and discharge piping are in line.

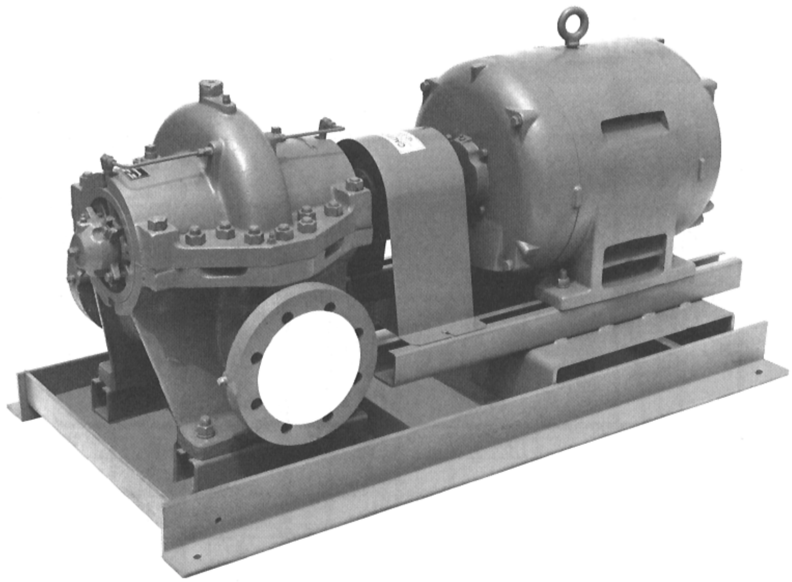


Figure 3-11: A Split-Case Horizontal Pump

Source: Taco

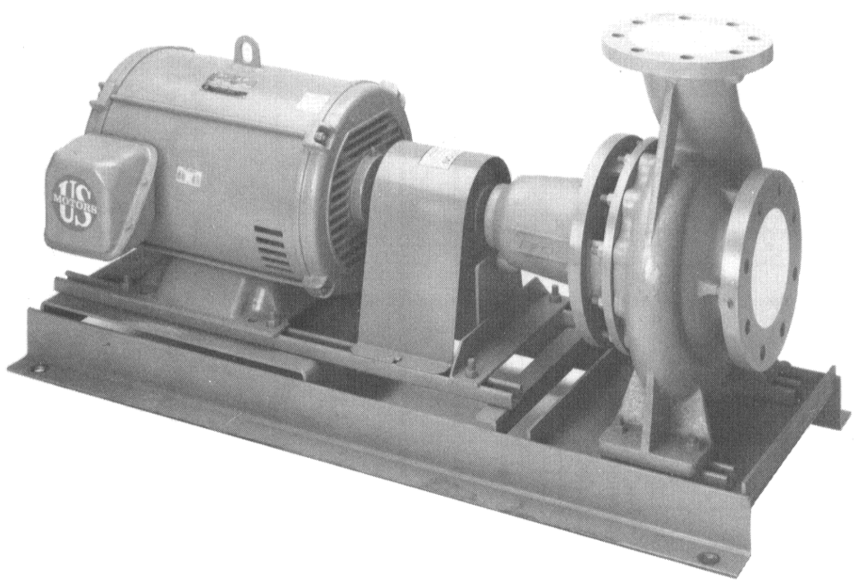


Figure 3-12: An End-Suction Pump

Source: Taco

3.2.5 Cooling Towers

Cooling towers are used in HVAC applications to cool condenser water for rejection of chiller condenser heat. Cooling towers can be classified as open or closed — in open towers, the condenser water is contacted directly by cooling air. Most cooling towers for HVAC duty are open. In closed cooling towers, the condenser water flows in closed piping.

Figure 3-13 below shows a typical cross-draft cooling tower with a propeller fan. Condenser water is distributed over the packing on either side of the tower which forces the water to flow in thin films, thus improving heat and mass transfer. Air is drawn in from the sides and discharges up through the fan grill. Some of the water evaporates during tower operation, thus enhancing cooling of the water. The condenser water system requires a fresh supply of water, which is supplied through a float valve in the tower sump. Some amount of condenser water must also be drained continuously to remove sediment.

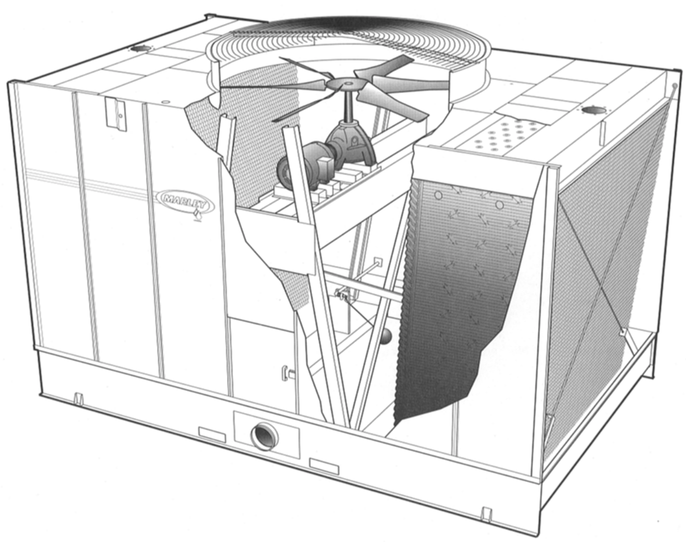


Figure 3-13: A Cooling Tower With A Propeller Fan

Source: Marley

Figure 3-14 shows forced-draft cooling tower fitted with centrifugal fans. This type of tower typically requires more fan power but is generally quieter. The forced draft design is used mostly for smaller applications.

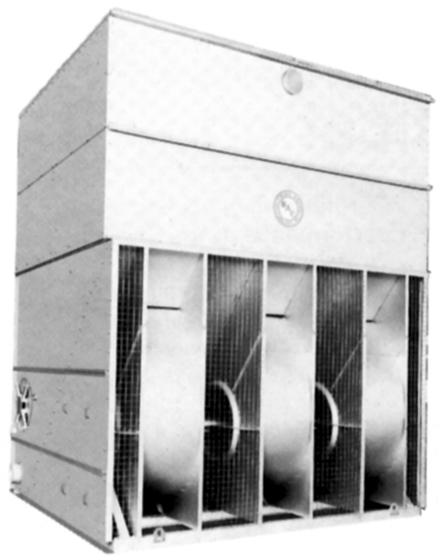


Figure 3-14: A Centrifugal-Fan Cooling Tower

Source: Baltimore Air Coil

3.2.6 Other Equipment

Other HVAC parasitic equipment of interest are:

- Condenser Fans
- Pneumatic Controls Compressors
- Burners
- Forced or Induced Draft fans for combustion air or combustion products

Packaged rooftop units, split-system air-conditioning units, and air-cooled chillers reject heat in air-cooled condensers. The condenser fans used to move the cooling air are generally axial propeller fans which are generally mounted directly onto the shafts of the drive motors.

Although the trend in commercial building HVAC controls is for more direct digital control (DDC), the control systems installed traditionally in commercial buildings were pneumatic controls. These controls involve the use of compressed air at 15 to 25 psig to actuate dampers or valves. Some older pneumatic controls constantly bleed air to maintain control pressures, but all pneumatic systems use air during cycling of controls. Standard reciprocating air compressors are generally used to supply this compressed air.

Burner fans and fans for combustion air or combustion products are used in boilers and furnaces. Figure 3-15 shows a typical commercial oil burner. The motor provides power both for the combustion air, but also for the oil pump. The body of the burner doubles as the fan volute. In the figure, the motor is partially hidden to the left of the fan, and the oil pump is at the right of the fan. Forced draft fans for gas furnaces or boilers are similarly incorporated into the burner assembly. Induced draft fans, used mainly for gas furnaces, are generally separately mounted on the furnace housing, with the motor in the ambient air.

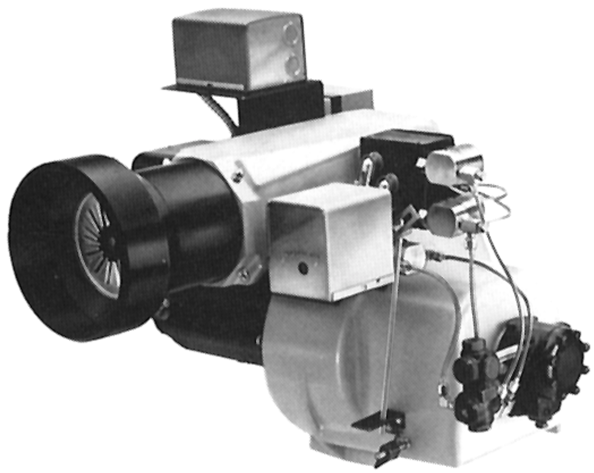


Figure 3-15: An Oil Burner

Source: Beckett

4. Market Description

This section provides background information regarding the HVAC equipment market and some significant trends affecting market structure and HVAC equipment.

4.1 Market Structure

The primary forms of HVAC projects in commercial buildings are as follows.

- *Design-Bid:* The installation is designed by a design engineer and installed by a mechanical contractor. This form of project is the most prevalent for central system HVAC projects.
- *Design-Build:* Both design and construction are handled by a "Design-Build" firm.
- *Limited Design Projects:* In many small projects, especially retrofit or replacement, there is little need for a detailed design package. The contractor who is hired, perhaps based on an informal bidding process, will do the required design work.

For each of these project types, engineers on the building owner's staff may be involved in clarifying the owner's requirements, reviewing designs and construction plans, etc.

The key participants in a design-bid project are shown in Figure 4-1 below. This structure applies to design and construction of the overall HVAC system, including a chiller for central systems. The most important participants are the building owner and the Architecture & Engineering (A&E) firm's design engineer. Decisions regarding thermal distribution equipment represent a step closer to the details of system installation, and for these decisions the role of the engineer becomes more important.

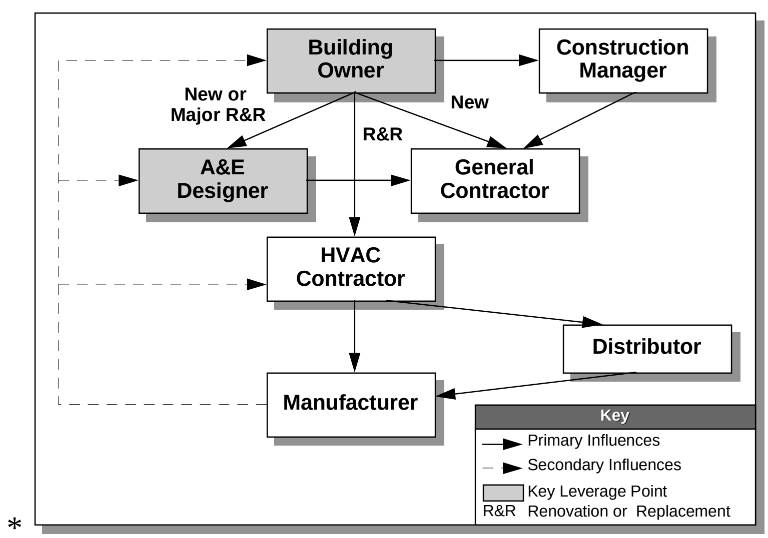


Figure 4-1: Decision Makers in Commercial Sector HVAC Projects

The building owner or developer typically makes the decision to install or replace an AC system, and sometimes is involved in decisions regarding which type of system is installed. In many cases, however, the owner gives guidelines regarding acceptable initial costs and an indication about the relative importance of comfort, aesthetics, energy use, etc.

The design engineer is typically part of an independent consulting engineering firm or a part of an architectural firm. The engineer may select the system type based on economic parameters, and will select system components, specify performance criteria for the equipment, and may also specify manufacturer and model. The engineer is also involved throughout a larger project to resolve technical issues which come up during installation, to inspect the installation, and to direct the system commissioning activities.

The contractor who installs the HVAC system may be the prime contractor working directly for the owner or may be working as a subcontractor for a general contractor. The job is typically awarded after a bidding process. Selection of contractor is typically based on lowest price, but other factors may be considered such as the particular contractor's track record, or approach to the job. The contractor will usually select the equipment vendor as long as performance specifications are met. The basis of this decision is usually cost, but could be influenced by other considerations such as the contractor's relationship with a particular vendor.

As mentioned, in some cases formal design documents are not prepared, in which case the contractor interacts directly with the building owner. In this case the building owner and the contractor are the important decision makers, depending on the HVAC knowledge of the building owner.

In a design-build project, engineers of the design-build company do the design work, specifying equipment types and performance. This type of project speeds up the overall construction process, but puts a lot of control in the one company doing the work. It is used more extensively in specialty market areas, for instance design and construction of refrigerated warehouses.

Specialty companies may also be involved in the installation of larger projects. This group of companies would include controls vendors and testing and balancing specialists.

In some cases decisions regarding installation and operation of equipment in commercial buildings are made by property management firms which manage the buildings rather than the building owners themselves. The building maintenance and operations staff may be part of the management firm rather than of the building owner. In these cases, the property management firm may take the role of the building owner in the decision process.

HVAC equipment can be sold through a distribution network or by factory representatives. For larger, more complex equipment, such as chillers or field-assembled cooling towers, there is a greater likelihood that factory representatives will be involved.

Energy retrofit work represents a distinct part of the market which has evolved over the years in response to the increase in energy costs and energy use awareness which began in the 1970's. This market has arisen because money spent on "wasted" energy in inefficient buildings represents a potential revenue stream. A number of types of companies are involved in accessing this market with one or more of the following contractual arrangements:

- *Third party financing:* Companies which fall into this group put up money for energy retrofit work. They are paid with money which is saved by these retrofit installations.
- *Energy Service Companies (ESCO's):* The ultimate model for ESCO's is that they own HVAC equipment in other parties' buildings and sell HVAC "services" (heating, cooling, etc.) to the building owner. There are varying degrees of actual ESCO-Building Owner relationships approaching this model. However, the basis of the arrangement is that the ESCO takes responsibility for operation of the equipment and profit is made by economical installation and operation of HVAC systems.
- *Performance Contracting:* In a performance contract, the contractor guarantees performance criteria of the equipment which is installed or operated. Monetary penalties are applied to cases where guaranteed performance is not achieved.

4.2 Major Companies

Major companies for some of the key players in the HVAC market are discussed in this section.

4.2.1 Manufacturers

The leading manufacturers for the key central system equipment types covered in this study are listed in Table 4-1 below.

Table 4-1: Major HVAC Equipment Manufacturers

Product	Major Companies	Total Market Size ($Million) 1996
Air-Handling Units[1]	Trane York Carrier McQuay Dunham Bush	775
Central System Terminal Boxes[1]	Titus Trane ETI	144
Fan-Coil Units[1]	IEC Trane McQuay	92
Classroom Unit Ventilator[1]	Trane McQuay	120
Cooling Towers[1]	Baltimore Air Coil Marley Evapco	400[3]
Pumps[2]	Taco Bell & Gosset Paco	250

Sources: 1. BSRIA/Ducker, October 1997 (Reference 4)
2. Based on discussions with manufacturers

Notes: 3. Does not include evaporative condenser and closed-circuit evaporative coolers. Does include many cooling towers used in industrial applications.

4.2.2 Escos

The Energy Service Company (ESCO) market has been evolving dramatically over the last few years, spurred by greater emphasis on total service HVAC companies and utility deregulation. The major companies in the market are some of the controls manufacturers who have expanded their services to include ESCO work (i.e., Honeywell, Johnson Controls, etc.), utility-owned ESCO's, and some of the early ESCO companies (for instance HEC, Inc.) who have expanded and remained competitive in this evolving market. A comprehensive review of the ESCO market is contained in the 1997 Directory of Leading U.S. Energy Service Company Providers (Reference 5).

4.2.3 Architecture and Engineering Firms (A&E's)

The top twenty firms involved in mechanical design engineering for nonresidential buildings are listed in Table 4-2 below.

Table 4-2 : Top 20 A&E's

Rank Based on Adjusted Revenues	Company	Location	Adjusted Revenues (US $MM)
1	Jacobs Engineering Group	Pasadena, CA	354.7
2	Fluor Daniel Inc.	Irvine, CA	240.9
3	BE&K	Birmingham, AL	207.5
4	Raytheon Engineers & Constructors	Lexington, MA	188.1
5	Sverdrup Corp.	Maryland Heights, MO	165.3
6	Dames & Moore	Los Angeles, CA	157.5
7	Lockwood Greene Engineers	Spartanburg, SC	174.4
8	URS Greiner, Inc.	New York, NY	92.8
9	Simons International Corp.	Decatur, GA	81.0
10	Burns and Roe Enterprises	Oradell, NJ	60.0
11	Daniel, Mann, Johnson & Mendenhall	Los Angeles, CA	57.6
12	Holmes & Narver, Inc.	Orange, Ca	51.2
13	Day and Zimmermann, International, Inc.	Philadelphia, PA	50.0
14	Hellmuth, Obata, Kassabaum, Inc.	St. Louis, MO	45.2
15	Bechtel Group, Inc.	San Francisco, CA	42.2
16	Syska & Hennessy Inc.	New York, NY	38.0
17	R.G. Vanderweil Engineers Inc.	Boston, MA	37.0
18	SSOE Inc.	Toledo, OH	34.9
19	Martin Associates Group Inc.	Los Angeles, CA	34.8
20	Parsons Brinkerholl Inc.	New York, NY	34.5

Source: Building Design and Construction Magazine (Reference 6)
Note: Adjusted Revenues based on assumptions regarding percentage of firm's total revenues which are associated with commercial mechanical design

4.2.4 Property Management Firms

The total and commercial floorspace of the largest property management firms serving commercial floorspace is shown in Table 4-3 below. These top ten firms manage 1,150 million sqft of commercial floorspace. Although large, this represents only about 2 percent of the total 58,772 million sqft reported in the 1995 Commercial Building Energy Consumption Survey (CBECS95, Reference 3). The conclusion can be drawn that management of property is a diversified business involving a very large number of companies.

Table 4-3: Major Commercial Property Management Firms

Company	Location	Total Floorspace (million sqft)	Estimated Percent Commercial	Total Commercial (million sqft)
Lasalle Partners	Chicago, IL	202	76	154
Koll Real Estate Services	Newport Beach, CA	176	78	137
Compass Management and Leasing	Atlanta, GA	160	85	136
Trammel Crow Co.	Dallas, TX	257	51	132
Simon Debartolo	Indianapolis, IN	114	100	114
Heitman Properties	Chicago, IL	159	71	113
PM Realty Group	Houston, TX	119	90	108
General Growth Propertied, Inc.	Chicago, IL	97	100	97
Insignia Financial Group, Inc.	Greenville, SC	330	25	83
Lincoln Property Co.	Dallas, TX	180	42	76
Total, Top Ten				1,149

Source: Commercial Property News (Reference 7)

4.3 Trends

HVAC market and equipment trends are discussed in the following sections.

4.3.1 Controls Trends

Three major trends in controls technology which affect HVAC energy use are as follows.

- Move from electric and pneumatic controls to Direct Digital Control (DDC)
- Increase in the use of Energy Management Systems (EMS)
- Proliferation of reliable low-cost Variable-Speed Drives (VSD)

DDC first appeared in the 1970's. As the name implies, it involves direct digital communication between sensors, controllers, and actuators. The market share of DDC has been increasing to the point where almost all new commercial building HVAC systems use DDC controls. This has been aided recently by the advent of more compact reliable DDC actuators, particularly for dampers.

The major impacts of DDC are:

- Flexibility: DDC systems can be controlled by microprocessors or computers, which allows for more flexibility in control algorithms and also allows remote monitoring. This flexibility can come at the price of more complexity.

- Reduced Energy Use: Minimal electricity use as compared with pneumatic controls[6], which require a compressor for control air. This energy impact is in addition to savings associated with improved control of HVAC and other equipment.

An increasing number of commercial buildings are controlled with Energy Management Systems (EMS) or Building Automation Systems (BAS). The latter system extends automation to non-energy functions such as sprinkler system control and monitoring, security, etc. It is estimated that energy savings resulting from installation of an EMS in a typical commercial building is about 5% (Reference 8).

Energy Management Systems use computer-based electronic technology to add "intelligence" to the automatic control of energy using equipment. In commercial facilities, EMS applications can range in sophistication from a simple programmable thermostat with start/stop control over only one or two pieces of equipment, to an extensive network of monitoring stations and hundreds of control points, offering comprehensive management of lighting, heating, cooling, humidity, maintenance, emergency and security alarms, etc. EMSs may perform only "supervisory" functions, or can be programmed to carry out complex localized tasks in response to interactive input from end-users.

Regardless of size, a typical EMS will consist of a computer, software that will allow creation of a specialized energy management program, sensors, and controls. In simpler systems, the "computer" may appear as a wall-mounted box with a digital display. In larger applications, the user interface consists of a PC work station with video display(s) and printer, custom-designed software, and off-site monitoring and control capability.

Energy Management Systems can be used to provide supervisory control over non-electronic control devices, e.g., pneumatic or electric actuators, or can be installed as part of wholesale conversion to direct digital control (DDC). Although conversion to DDC is not required, proper operation of existing controls is necessary for successful use of an EMS. As a result, most EMS installations also involve recalibration and repair or "fix-up" of existing equipment, which must also be considered as integral to the installed cost.

The latest trend in EMS is the increasing move towards compatibility of the systems, controllers, and sensors of competing vendors. In the past, purchase of an EMS locked a building owner into dealing with the same vendor for all system upgrades and modifications. For example, in the case of an addition to a building, the sensors and actuators controlling the new HVAC equipment would all have to be obtained from the same vendor if the new equipment was to be incorporated into the central EMS. As a

[6] Pneumatic controls, used for many years as the standard control system for large buildings and HVAC systems, use 15 to 25 psig air to control dampers, valves, etc.

response to this inflexibility, there has been a move to develop standardized protocols for communication between components of EMS systems. The two “industry standard” protocols which are now being honored by the major controls vendors are BACnet (developed by ASHRAE) and LonWorks (developed by Echelon of Palo Alto, CA). The integration of these protocols into control hardware should help to open competition among controls manufacturers. The result would be an increase in control and monitoring capabilities, which would allow more sophisticated HVAC control strategies to be implemented at reasonable cost.

Air quantity in VAV systems can be controlled with discharge dampers, inlet dampers, or variable speed operation of fans. Although the last of these is the most efficient, there were no reasonable-cost, reliable variable-speed drives (VSD) in the size ranges required for commercial building HVAC during the 1980’s, when VAV systems became the norm for buildings such as offices. The improvement of the technology, driven in part by large expenditure on utility Demand-Side Management (DSM) programs, has resulted in reduction in cost and improved reliability that have made VSD’s the technique of choice for any new VAV installation.

4.3.2 Indoor Air Quality (IAQ)

The design, construction, and operation of ventilation systems are strongly affected by concerns regarding Indoor Air Quality (IAQ). The emphasis on energy savings which came about as a result of the oil embargo of the 1970’s, the emergence of energy savings as a national priority, and the significant increases in the cost of energy in the 1970’s and early 1980’s resulted in building and HVAC design changes which do not always have beneficial effects on building occupants. The reduction of infiltration rates as well as mechanical ventilation rates resulted in a significant decrease in the movement of fresh air to dilute contaminants. Recognition and concern about Sick Building Syndrome (SBS) began in the mid-1980’s. This term refers to buildings with a high level of occupant complaints regarding comfort and perceived and actual health effects caused by poor air quality within the occupied spaces.

The causes of IAQ problems are somewhat complex and varied. There are a large number of potential indoor pollutants which can cause discomfort or ill health. Potential sources of these contaminants are (1) building occupants, (2) construction materials, (3) building operations or equipment (such as copying machines), (4) contaminants brought in from the outside, and (5) contaminants associated with the building HVAC system. The building ventilation system, part of whose role is to dilute and remove these pollutants, can in some cases contribute to or cause IAQ problems, for instance, if a fresh air intake is located near a loading dock (#4 above), or if microbes can grow on wet surfaces within the HVAC system (#5 above).

The concern of engineers or building owners regarding IAQ is heightened by the threat of lawsuits associated with ill effects of poor IAQ. Even when there are no lawsuits, in many cases buildings are thoroughly contaminated, and must be thoroughly cleaned before operations can resume. Problems can in most cases be avoided before they begin if care is taken to ensure proper design, installation, commissioning, and operation of the building HVAC equipment. However, the standard for "proper" procedure to avoid IAQ problems is still evolving.

Current practice for ventilation is embodied in the American Society of Heating, Refrigeration, and Air-Conditioning Engineers (ASHRAE) Standard 62-1989, "Ventilation for Acceptable Indoor Air Quality". This standard has evolved over the years in response to current understanding regarding ventilation needs. The 62-1981 standard called for significantly reduced fresh air ventilation rates in order to achieve energy savings, a change which was reversed with the 62-1989 standard. The standard allows two alternative approaches for determining fresh air ventilation, the "Ventilation Rate Procedure" and the "Air Quality Procedure". The first procedure is a fairly straightforward prescription of ventilation rates in cfm/sqft or cfm per occupant, depending on the space use. The second procedure, which is rarely used, allows reduction of overall fresh air ventilation if procedures to capture and remove contaminants are followed.

Revision to the ASHRAE ventilation standard has been ongoing for a number of years. ASHRAE published a proposed revision to the standard (the revision was designated Standard 62R) in August 1996. Some of the major proposed changes were (1) more emphasis on an air quality approach (the "Analytic Procedure") as opposed to prescriptive ventilation rate guidelines in an attempt to allow for energy saving design, (2) responsibility by building designers and others associated with HVAC system installation and operation for satisfaction of occupants, (3) emphasis on system commissioning to verify proper operation and procedures for periodic system verification, (4) no allowance for smoking in buildings which comply with the standard, and (5) use of language which allows the standard to be used as an enforceable code. Consensus on the 62R document was not achieved, and the proposed revision has since been withdrawn. Standard 62-1989 was moved to "continuous maintenance" status in June 1997. This means that revision to the standard will occur in the form of addenda, rather than an entire revision. At least fifteen addenda have been proposed. Four have been approved by ASHRAE, but they are currently being appealed. In spite of this long process of revision to the standard, it is clear that IAQ problems which have surfaced in a number of buildings over the last two decades have heightened the importance of provision of adequate ventilation as a part of HVAC system design. The possibility that HVAC system energy use will increase as a result does exist, but this potential increase can be mitigated by energy-efficient design. Energy savings strategies would fall into three broad categories: (1) system design (i.e., use of separate fresh air ventilating units), (2) control, and (3) energy recovery.

4.3.3 Market Structure Trends

The major recent market trends are:

- Shift from new construction to replacement work
- Rise of innovation and value-added as an alternative to strict focus on price
- Consolidation of wholesalers and contractors
- Rise of total service firms and ESCO's
- The move of utilities into contracting and ESCO work

The HVAC industry has undergone a fundamental change from about a decade ago. Previously the HVAC market was largely in new construction, where contractors and manufacturers were concentrating on design engineers and building owners to sell their products. Two major changes are the rise of the replacement market and the drive for innovation.

As the age of buildings increases, a growing percentage of the HVAC market share has gravitated toward replacement and maintenance rather than new construction. This causes some additional problems for manufacturers in that with a new installation, you are dealing with a clean slate. The manufacturer mainly works with the owner of the project and possibly the engineering firm that designed it. This tends to make the path to the market more straightforward because there are a very limited number of people involved in the decision process. With the concentration being shifted toward the replacement market, a whole new set of variables is thrown into the equation. When a manufacturer or contractor goes in to complete work in an existing building, the current tenants as well as the owner and possibly the engineering firm will have input to the final decision. This complicates matters greatly. This complication has led the manufacturers to concentrate on improving their strategic alliances and partnerships to strengthen their positions in the market.

The innovation push in the market supports the fact that price is no longer the main focus for competition. Innovation in product design and delivery is replacing strict price competition. New manufacturing processes are being developed and implemented to increase the quality and speed-to-market of the new product coming out. A few examples of these new processes are Total Quality Management (TQM) which concentrates on introducing quality control measures at all parts of the manufacturing process, and Demand Flow Manufacturing (DFM) which is focused on the product flow through the manufacturing process. Both of these new processes focus on getting a quality unit to the market faster than the competition. Customers are willing to pay a little more if they can get their unit when they want it and not when the manufacturer is ready to deliver it.

Not only is there a fundamental shift in the direction the HVAC market is going, but there is also a fundamental shift in how the new products are getting to the market. In the past, most manufacturers sold to their contractors through their own sales offices and through their distributors. In the past decade, light commercial HVAC equipment has begun to move through large retail outlets such as Home Depot, Sears, Home Base, and Circuit City. These stores, because of their ability to buy in quantity, are able to offer contractors lower prices than they could normally get from their regular independent wholesalers. This has led to a continuous decline of the independent wholesalers, but has contributed to the rise of the national wholesalers like Pameco, Sid Harvey, Baker Brothers, and United Refrigeration who have the capital clout to compete directly with the large retail chains. Consolidation has also been a strong trend among contractors. Many articles in the trade press over the last few years have highlighted this move by so-called "consolidators" to buy up independent contractors nationwide to create huge contractor chains. These large companies have immense buying power and can approach the market in a more streamlined standardized fashion, with an aim to increase market share and boost profits.

A fourth trend that is influencing the HVAC market is the rise of <u>total</u> service firms. What this statement of "total" implies is companies, sometimes contractors, sometimes HVAC manufacturers themselves, are creating companies that offer a full range of HVAC equipment as well as the knowledge, ability, and willingness to install and operate the heating and cooling systems for building owners. Current owners are not interested in becoming HVAC service experts. They simply want cooling in the summer, heating in the winter and lighting year-round. They do not want to concern themselves with the maintenance and repair of their HVAC equipment. This has opened a new market for contractors and manufacturers alike. Honeywell, traditionally just a manufacturer of HVAC system and component controls, is probably the best example of this. They have started to offer complete HVAC system monitoring and repair services that eliminate the need for the building owner to become an expert in the field of HVAC. If there is a problem in a building that Honeywell is monitoring, they can fix it either by changing setpoints through the computer line connected to the building or by calling one of their local service representatives to go the monitored site and correct the problem. Most of this action occurs usually before the owner even realizes he has a problem. They also are able to change conditions based on immediate feedback from the customer. If someone in the monitored building calls and complains about the temperature in their office, Honeywell is able to immediately change those specific conditions. The owner does not need to be an expert.

One thing that building owners would rather do is they would much rather pay one bill for all their HVAC and lighting needs. This has given rise to a new market for companies called ESCO's, or Energy Services Companies. This market is still in the development stage, but the concept is intriguing. The ESCO would offer all the services of heating, cooling, and lighting that a building owner needs and the building owner would pay only one bill. A very good example of this is in the city of Denver, Colorado where a two-block shopping mall called Denver Pavilions is being constructed without any on-site cooling. The mall will be relying on a district cooling system owned and operated by Public Service Company of Colorado.

Much of this change is occurring coincidentally with electric and gas utility deregulation. This change in these industries allows utilities more flexibility to compete in non-power markets, while creating intense price competition. Many utilities have responded by branching out into contracting and other service-oriented markets for building owners. The utilities have a leg up on other service providers, since they already have their "foot in the door". Claims of unfair competition by contractor groups, such as the Air Conditioning Contractors of America (ACCA) have led to restriction of utilities' freedoms in some states' deregulation legislation.

Finally, some manufacturers have begun buying independent equipment sales offices and total service firms in order to preserve their path to market without being squeezed by the downward price pressures exerted by the emerging consolidated contractors and wholesalers.

4.3.4 System/Equipment Trends

Complementary to the market changes that are occurring within the HVAC industry, manufacturers are being pressured now, more than ever to create environmentally friendly (non-CFC) HVAC equipment that is a generation ahead of the existing equipment in regards to performance and efficiencies. Also, since the shift in the market is concentrating more on the existing buildings, equipment must now also be more compact. There is no longer a clean sheet of paper to design on. There are size restrictions required for the new designs before they are even conceived. For instance, to replace an existing water-cooled chiller that is located in the basement of a building, it is not possible to remove the roof of the building to drop in a large chiller. The new equipment must be designed such that it will fit inside the maintenance elevator and will be easy to install once it is brought to the site.

Also, with respect to the rise of the global market, land and space is at a premium. HVAC equipment must be efficiently sized to accommodate the lack of room in other areas of the world that still require cooling. This puts an added difficulty on the design of equipment that primarily resides outside, such as air-cooled chillers. The physical and operating envelope size of these air-cooled chillers must be kept to the minimum required to produce the efficiencies demanded by the world market. This has produced a revolution in thinking: in the past, most design engineers were mostly concerned with designing equipment for the domestic market, however, in spite of the recent stall in the world economy, it is predicted that most market growth will come from outside the United States.

Another trend that is being noticed in the HVAC market is the movement from the traditional water-cooled chillers to air-cooled chillers. A number of reasons are behind this shift, with the lower first cost and lower maintenance of the air-cooled chillers topping the list. For budget conscious owners, the air-cooled system is obviously the cheaper choice due to the fact that no cooling tower is required and there are no additional condenser water pumps required. Also, since building owners and their maintenance staff do not have a lot of expertise in HVAC systems and maintenance, the air-cooled system is again their best option because of the lowered maintenance of the overall system. This fact is especially strong in schools where the typical school district has no professional HVAC maintenance staff. The staff usually consists of a janitor who has absolutely no experience in servicing HVAC equipment.

There has been little changes in fan and pump design in recent years. The greatest impacts on thermal distribution and auxiliary equipment technology have been made by introduction of variable speed drives. For the most part, fan and pump impellers and housings themselves have changed little over the years. The possibility exists for efficiency improvement through increased use of airfoil-shaped and more complex fan blades, use of alternative materials (such as plastics), and housing shape modifications, and some developments along these lines are being pursued. However, most system development and design makes use of conventional fan and pump technology. In any case, system design and proper component selection generally makes a greater impact on overall energy use than fan and/or pump design itself.

5. Baseline Energy Use Estimate

This section describes the estimate of HVAC parasitic energy use, which was developed during this study. A fairly comprehensive description of the approach to the estimate is presented. Results are presented in Section 5.6.

5.1 Overview

The goals of the baseline energy use estimate are

- To provide an accurate estimate of the energy used by fans and pumps which are used to distribute heating or cooling in the US commercial building sector.
- To provide a physical understanding of the factors which contribute to energy use by the thermal distribution equipment.
- To provide a baseline estimate of current national energy use which can be used for calculation of the national energy savings impact of various Research, Development, and Demonstration (RD&D) options for reducing energy usage. The estimate is based on calendar year 1995.

The energy use estimate developed in this study is a "bottom-up" estimate, which means that it is based on building floorspace, and estimates of annual energy use intensity (EUI), (kWh/sqft) for typical building systems. Many of the important parasitic equipment types are discussed in Section 3. The estimate is also an "as-designed" estimate, which means that equipment is assumed to operate properly according to design intentions. For instance, modeling of chilled water systems does not allow for operation with reduced chilled water temperature to account for inadequate air flow in air handling units. Such operation can occur in the field, but its prevalence and impact cannot be adequately predicted.

Because the study takes an "as-designed" approach to energy estimates, the estimates are considered a conservative approximation of actual conditions. Unintended operation can result in increase or decrease in energy use. The magnitude of the uncertainty associated with the unintended operation is difficult to predict, but might be as much as 20% of overall estimates.

The baseline estimate starts with a segmentation of the US commercial building stock floorspace by building type, system type, and region. The segmentation is based on the 1995 Commercial Building Energy Consumption Survey (CBECS95), (Reference 3) data, and is discussed in Section 5.2 below. Building conditioning load estimates developed by Lawrence Berkeley National Laboratory (LBNL) were used as the basis for the energy use calculations. This set of load estimates, based on building models described in Reference 19, is the best and most complete space conditioning load database anywhere available for the commercial sector. Energy use estimates for the HVAC parasitic equipment was calculated based on the building load data and models of HVAC system and component operation developed in this study (these models are described in Appendix 3).

The fundamental equations for the baseline estimate are listed in Table 5-1 below. They are shown graphically in Figure 5-1.

Table 5-1: Baseline Energy Use Estimate Equations

÷ For a given Building Type/ Climate/System Type segment	Component Annual EUI (kWh/sqft)	=	Component Design Load Intensity (DLI) (W/sqft)	*	Component Effective Full-Load Hours (hrs)	÷	1000W kW
	Total Building EUI (kWh/sqft)	= Σ	Component EUI (kWh/sqft)				
	Total Segment Energy Use (kWh)	=	Total Building EUI (kWh/sqft)	*	Total Segment Floor Area (sqft)		
For a given group of segments (ie all hospitals or the entire commercial sector)	Total Group Energy Use (kWh)	= Σ	Total Segment Energy Use (kWh)				
	Average Group EUI (kWh/sqft)	=	Total Group Energy Use (kWh)	÷	Total Group Floor Area (sqft)		

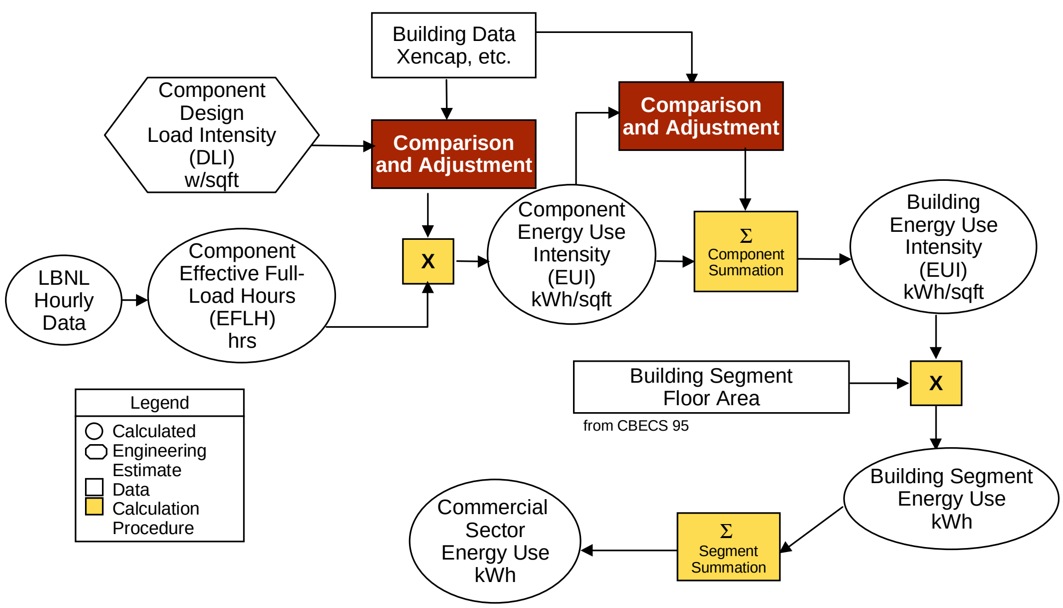


Figure 5-1: Baseline Energy Use Estimate Equation Flow Chart

Estimates of component design load intensities (DLI), (W/sqft) for the components of typical systems used for each of the building segments have been developed. These estimates are based on standard design practice and specifications for the equipment. For instance, supply fan DLI is based on typical values of cfm per square foot, required supply fan pressure, typical fan efficiency, and typical motor efficiency, as shown below for the New York Large Office with a central VAV system.

$$
\begin{aligned}
\textit{Supply Fan DLI} = {} & \left(0.86\,\frac{cfm}{sqft}\right) x \left(4\ \textit{in wc fan total pressure rise}\right) \div 8.5\,\frac{cfm * in\ wc}{W} \div \\
& \left(69\%\ \textit{fan efficiency}\right) \div \left(97\%\ \textit{drive efficiency}\right) \div \left(90\%\ \textit{motor efficiency}\right) \\
= {} & 0.67\ W/sqft
\end{aligned}
$$

The engineering estimates of DLI values are compared with actual building data from the XenCAP™ database (this database of commercial buildings is described in Appendix 1).

The EUI for a particular system component is equal to the DLI times the effective full load hours (EFLH) of operation in a year. In many cases, the EFLH is simply equal to the total number of hours of operation. However, some fans and pumps cycle depending on building conditions, and some fans and pumps operate with variable flow. System modeling was done to determine EFLH for system components with varying load or with varying percentage “on” time. This is described in more detail in Section 5.4.

The estimates of EUI are compared to the XenCAP™ data and the estimates of others (References 10, 11, 12, 13, 14, and 15).

Component EUI’s are combined to give building EUI's. The building EUI's are multiplied by the floorspace of a given building stock segment to give energy use for that segment. Summation over segments gives total energy use for a group of segments or for the entire commercial sector. Component EUI's themselves can also be multiplied by floorspace to give estimates of total energy use for a given component type. The final sectorwide estimates of parastic energy use are presented in Section 5.6.

5.2 Building Stock Segmentation

This section describes the segmentation of building floor area and includes the following
- Discussion of the important segmentation variables
- Description of the geographic segmentation
- Description of the segmentation methodology
- Discussion of external review of the results by industry experts
- Graphical display of the results

The segmentation focused first on cooling systems, since parasitic loads associated with cooling systems, or cooling/heating systems are significantly greater than those associated with heating-only systems, especially for the larger buildings with central systems, which are emphasized in this study. The dominance of the cooling parasitic loads is illustrated in the following example of a building with a central chiller, VAV units for cooling, and baseboard perimeter heating. The major cooling parasitic design loads might be 0.6W/sqft for the supply fans, 0.2 W/sqft for the return fans, 0.2 W/sqft for the chilled water pump, 0.2 W/sqft for the condenser water pump, and 0.2 W/sqft for the cooling tower fan for a total of 1.4 W/sqft. The parasitic load associated exclusively with heating would be approximately 0.1W/sqft for the heating water pump.

5.2.1 Segmentation Variables

The segmentation variables used in the study were:
- Climate or geographic region
- Building type
- System type

There is no study or survey which gives an adequate breakdown of the U.S. commercial building stock by all of these variables. The CBECS95 data represents the most complete survey which can be used for such a segmentation, and this database has been used as a basis for the segmentation used in this study.

Simplification of the segmentation process was necessary due to the limitations of the data. Simplifying assumptions are as follows.
- The building type distribution of floorspace does not vary significantly with region (see Figure 5-2 below)
- System type distribution does not vary significantly with region (see Figure 5-3 below)

The following plot (Figure 5-2) shows building type distributions of floorspace for the chosen geographic regions based on data from CBECS95. The plot shows that, although there is some variation in the distributions as we move from region to region, the basic shape of the distributions remain similar.

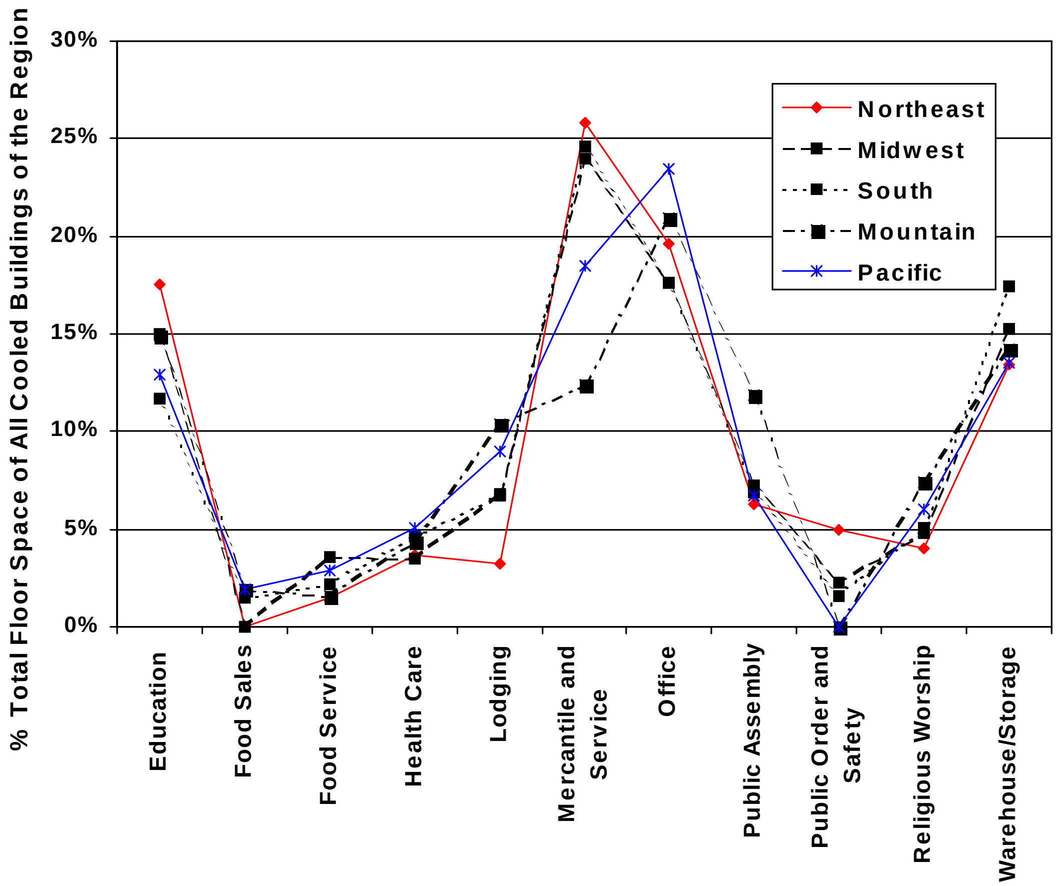
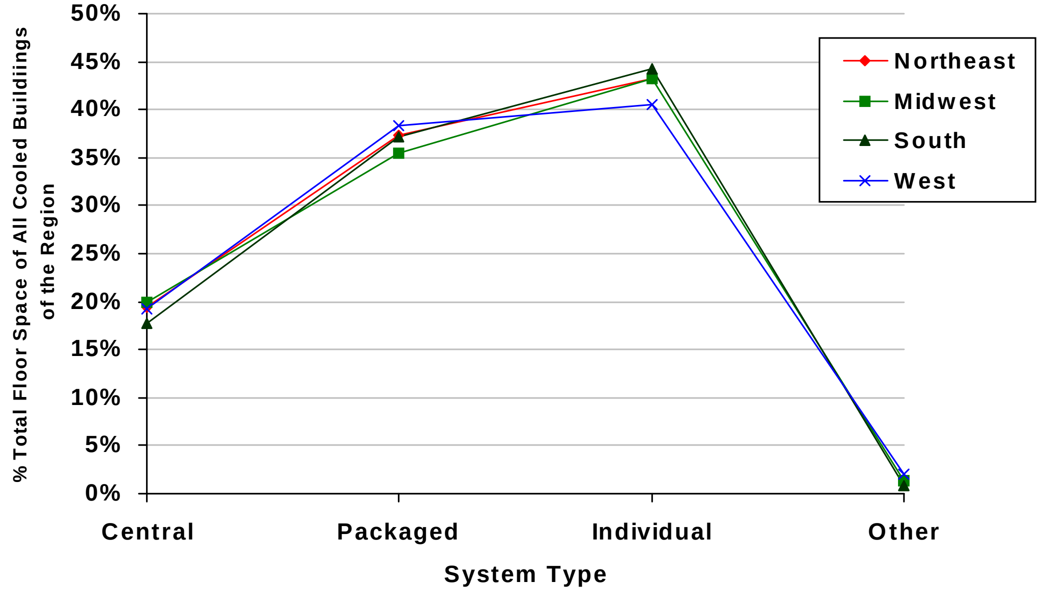


Figure 5-2: Regional Variation of Building Type Distribution

Figure 5-3 below shows CBECS95 (Table BC-36) data for system type distributions by region. The data clearly show that system type distribution is not strongly affected by region. System types are defined in Table 5-2 below.



Note: Swamp Coolers combined with Individual for this comparison

Figure 5-3: System Type Distributions

The chosen segmentation approach is three dimensional, based on building type, system type, and geographic region. The distributions of floorspace by building type and system type are assumed to be constant when moving from region to region. However, system size and operational characteristics depends on the climate of the different regions. This certainly represents a simplification of the commercial building stock, but it is intended to be a reasonable approximation for development of national parasitic energy estimation. Table 5-2 below shows the considered ranges of the segmentation variables.

Table 5-2: Segmentation Variables

Variable	Categories	Descriptions
Building Type (CBECS95 categories except as noted)	Education	See Reference 3 (CBECS95) for further definition of building types
	Food Sales	
	Food Service	
	Health Care	
	Lodging	
	Mercantlie & Service	
	Office	
	Public Buildings	Includes CBECS95 Categories Public Assembly, Public Order and Safety, and Religious Worship
	Warehouse/Storage	
System Type	Individual or Room AC	Window AC, Packaged Terminal AC, Packaged Terminal Heat Pumps
	Packaged	Unitary, Split Systems, Residential-Type Central AC, Residential-Type Heat Pumps
	Central VAV	Variable Air Volume Systems Served by Central Chillers
	Central CAV	Constant Air Volume Systems Served by Central Chillers, Includes Multizon and Dual-Duct Constant Volume
	Central FCU's	Fan-Coil Unit Systems Served by Central Chillers
	Not Cooled	
Region (according to CBECS95)	Northeast	
	Midwest	
	South	
	Mountain	
	Pacific	

[1]The West Census region was split to better represent the very different weather patterns in the Mountain and Pacific regions.

5.2.2 Geographic/Climate Segmentation

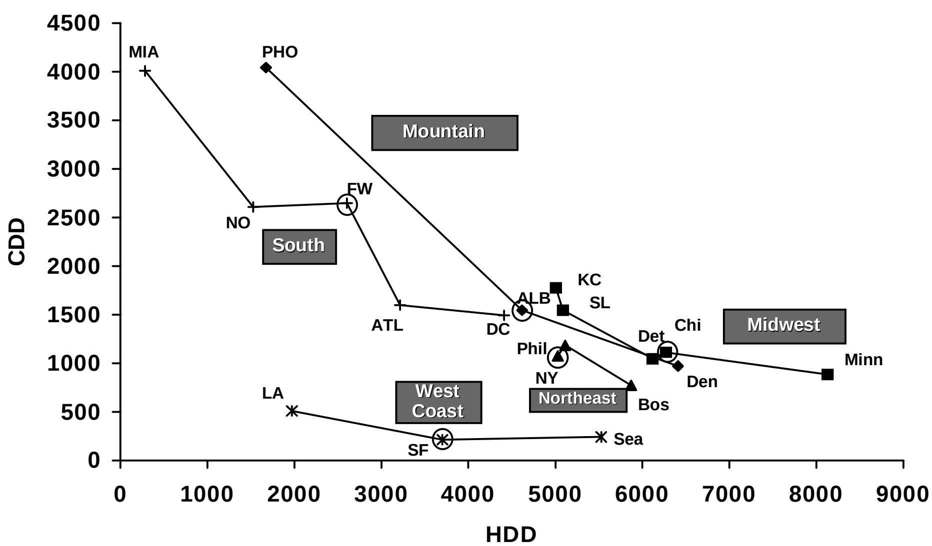
As shown in Table 5-2 above, five regional categories are used in this study. The primary objectives for selection of regions were (1) sufficient number of regions to give a reasonable representation of US climate variation, (2) the number of regions should not be excessive, (3) consistency with CBECS95 regions, and (4) one city per region for representative weather data.

The representative cities for the five regions are listed in Table 5-3 below.

Table 5-3: Segmentation Regions and Representative Cities

Region	City
Northeast	New York
Midwest	Chicago
South	Fort Worth
Mountain	Albuquerque
Pacific	San Francisco

The five regions are intended to reflect the range of US climate variation. The plots shown in Figure 5-4, Figure 5-5, and Figure 5-6 show the characteristics for representative cities from these regions of cooling degree day (CDD) vs. heating degree day (HDD), insulation vs. HDD, and latent cooling vs. HDD plots. The plots show that the five chosen regions and cities represent fairly distinct climates.



Legend			
Alb	Albuquerque	Mia	Miami
Atl	Atlanta	Minn	Minneapolis
Bos	Boston	NO	NewOrleans
Chi	Chicago	NY	New York
DC	Washington DC	Phil	Philadelphia
Den	Denver	Pho	Phoenix
Det	Detroit	Sea	Seattle
FW	Fort Worth	SF	San Fransisco
KC	Kansas City	SL	St. Louis
LA	Los Angeles		

Figure 5-4: Regional Distribution--Cooling Degree Days vs. Heating Degree Days

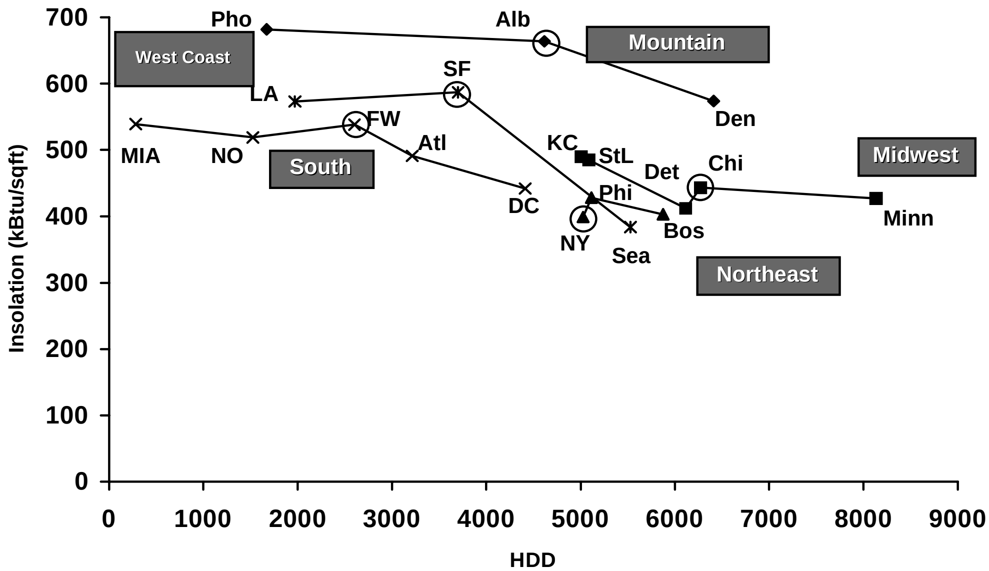


Figure 5-5: Regional Distribution--Insolation vs. Heating Degree Days

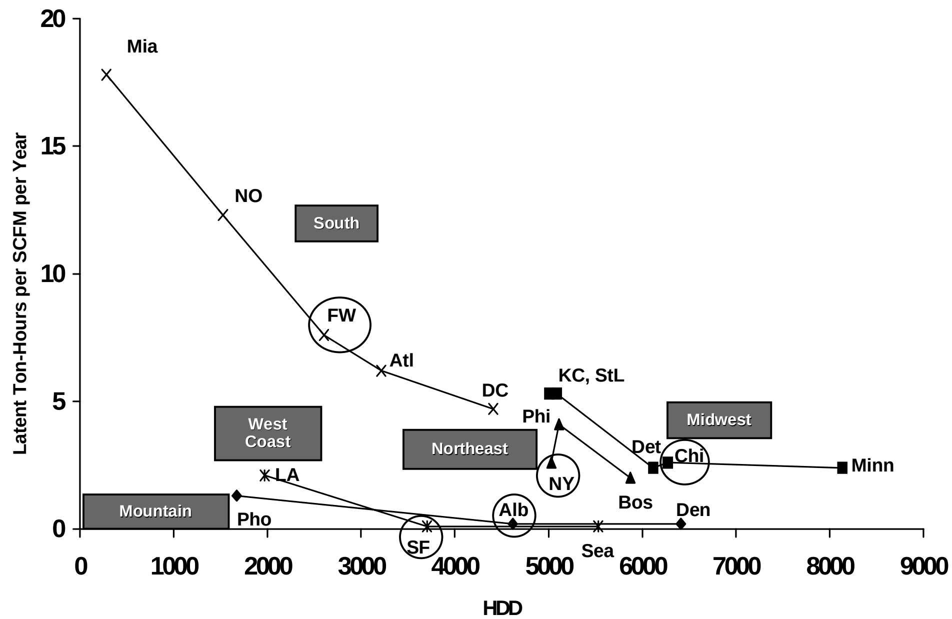
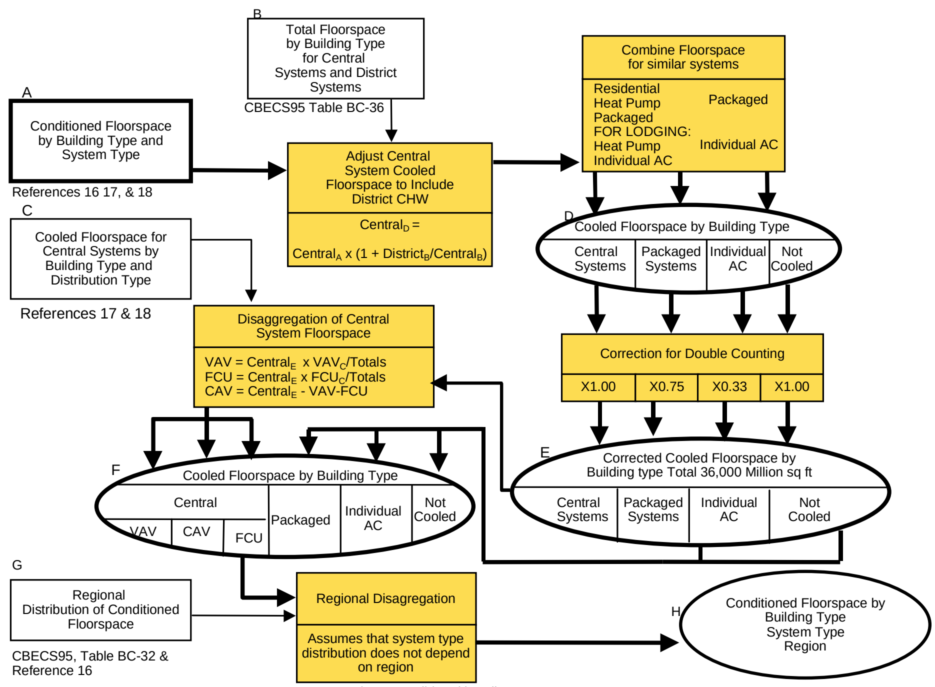


Figure 5-6: Regional Distribution - Latent Cooling vs. Heating Degree Days

The plots also show the selection of representative cities for the regions. The cities are intended to be average for their regions. System modeling uses weather data for these representative cities.

5.2.3 Segmentation Methodology

This section describes the procedure used to assign floorspace to the selected building stock segments. The procedure is illustrated in Figure 5-7 below.



Source 1: CBECS95 data consolidated by Allan Swenson

Figure 5-7: Building Stock Segmentation

The segmentation is based initially on estimates of conditioned floorspace Box (A) (about 48 billion sqft) provided in References 16, 17, and 18 and tabulated in Appendix 4. The references provide a breakdown of cooled floorspace by building type and by cooling system type (central, packaged, individual AC, heat pump, and residential-central) and a breakdown of heated floorspace by building type. The heated-but-not-cooled floorspace is set equal to the difference between heated and cooled floorspace[7]. Adjustments to these floorspace numbers are as follows.

- District cooling floorspace is added to central system floorspace. The ratio of district cooling floorspace to central cooling floorspace (Box B) is estimated from CBECS95 Table BC-36 for the applicable building types (Education, Health Care, Lodging, Mercantile & Service, Office, Public Buildings, and Warehouse). These ratios are applied to the cooled floorspace estimates for central cooling to get estimates of buildings with chilled water cooling for each building type category.
- Similar system types are combined to allow for simplified building stock characterization. All residential-central AC floorspace and heat pump floorspace is combined with the packaged unit floorspace to give an overall estimate for packaged system floorspace.

---

[7] This assumes that cooled-but-not-heated floorspace is insignificant.

- For the lodging building category, the heat pump floorspace is assumed to be associated with packaged terminal heat pumps rather than with packaged ducted heat pumps. Hence, for this building category, the heat pump floorspace is combined with the individual AC floorspace.
- The floorspace estimates (Box D) need to be reduced because of the inherent double-counting applicable in the Reference 17 and 18 data. The CBECS95 survey allows overlap of cooling system types serving a building's cooled floorspace. Hence summation of floorspace associated with each cooling system gives a sum (46.6 billion sqft) which is larger than the total cooled floorspace (36 billion sqft) in commercial buildings. The double counting is taken into account by reducing cooled floorspace for each system type so that the total cooled floorspace equals the total 36 billion sqft. The reduction is applied by multiplying segment floorspace by the factors shown in Table 5-4 below. Justification for the factor selection is the relative importance of each of the system types in cases where overlap of cooling systems occurs. Note that for uncooled floorspace there is no double counting, so the adjustment factor is 1.00.

Table 5-4: Double-Counting Adjustment Factors

System Type	Adjustment Factor
Central	1.00
Individual AC	0.33
Packaged	0.75
Not Cooled	1.00

- The central system floorspace is disaggregated by distribution type: constant air volume (CAV) air handling units, variable air volume (VAV) air handling units, and fan-coil units (FCU). This disaggregation is based on Reference 17. Again, an adjustment must be made for double-counting inherent with this data. However, it is assumed that double counting applies equally to each of these distribution types. Hence, the ratios of distribution types in the reference is applied to the central system cooled floorspace.
- The floor areas for each building type/system type segment are further disaggregated by region. This is done based on the assumption that the building type/system type distribution does not vary significantly from region to region. The overall regional distribution of *conditioned* floorspace is as shown in Table 5-5 below (Box G). Information sources from the CBECS95 survey are indicated in the table.

Table 5-5: Regional Distribution of Conditioned Floorspace

Region	Conditioned Floorspace (million sqft)	Percent of Total Conditioned Floorspace	Information Source
Northeast	9,919	20.6%	CBECS95, Table BC-32 (Reference 3)
Midwest	12,382	25.8%	CBECS95, Table BC-32 (Reference 3)
South	16,667	34.7%	CBECS95, Table BC-32 (Reference 3)
Mountain	3,272	6.8%	Swenson Fax 10/8/97, Table 7 (Reference 16)
Pacific	5,824	12.1%	Swenson Fax 10/8/97, Table 7 (Reference 16)
Total	**48,064**	**100%**	

5.2.4 External Review of Segmentation Data

The building type/system type distribution of cooled floorspace for the major building types of interest for large central systems (Education, Health Care, Lodging, Mercantile & Service, and Office) was reviewed by three industry experts. Major points made in this review and changes made to reflect the comments are shown in Appendix 2. The segmentation estimates presented in this Section and in Appendix 2 are already adjusted to reflect the reviewers' comments.

The segmentation approach described herein has as its basis the CBECS95 commercial building survey. Inherent assumptions in the segmentation development are (1) the building type distribution is not a strong function of region, (2) the system type distribution is not a strong function of region, and (3) double counting is eliminated using the factors of Table 5-4. Assumptions (1) and (2) are supported by the data illustrated in Figures 5-2 and 5-3 respectively. Assumption (3) is made based on a logical judgement regarding system importance in cases of system overlap. In any case, the industry review of the segmentation provides an overall endorsement of the results, with some suggestions for changes. The segmentation calculation has made use of the best available data, has followed a carefully thought-out approach, and has been adjusted as recommended by industry expert review.

5.2.5 Segmentation Results

The building stock segmentation is tabulated in detail in Appendix 4. Graphical representation of the segmentation is shown in Figure 5-8 and Figure 5-9 below.

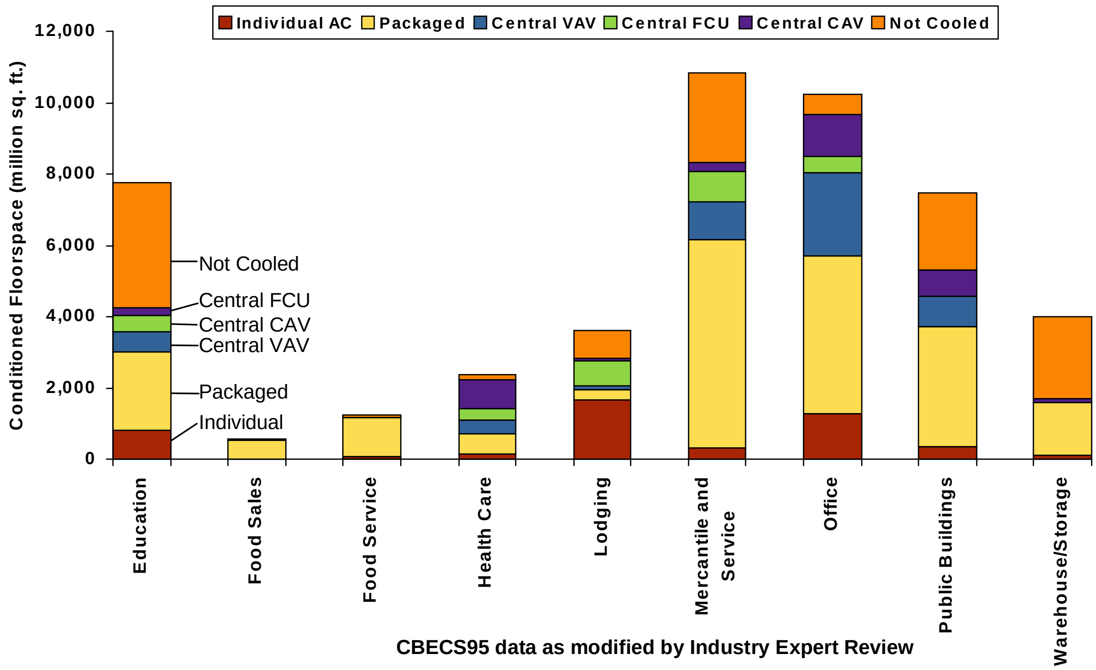


Figure 5-8: Building Stock Segmentation: Building Types and System Types

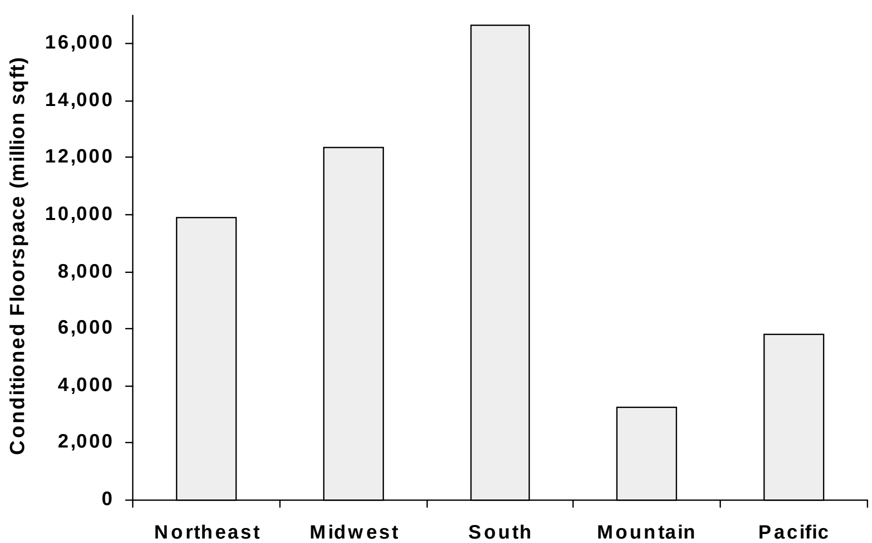


Figure 5-9: Regional Distribution

5.2.6 Segmentation Refinements

This section discusses procedures adopted in order to incorporate additional complexities in the baseline energy use analysis without adding structure to the building segmentation scheme. The building floor area of a segment is further subdivided, and averages for the overall segment are calculated and reported. This averaging process typically affects a limited number of segments.

The situations taken into account with this segment extension procedure are described below.

Chillers: Water-Cooled vs. Air Cooled: Water-cooled chillers require condenser water (CW) pumps and cooling towers to reject heat. Air-cooled chillers are generally smaller. They typically have reciprocating rather than centrifugal or screw compressors, and they reject heat in air cooled condensers which use significant fan power.

VAV Terminal Boxes: Valve Boxes vs. Fan Boxes: Energy use for VAV system terminal boxes varies significantly depending on the fan arrangement. Series boxes are designed to operate during all building occupied hours. Parallel fan boxes are controlled as a first stage of reheat. Valve boxes require no power for operation of fans. Bypass boxes dump unneeded air into the ceiling plenum to be returned to the air-handling unit, which saves cooling and reheat but not central fan power.

Water-loop (California) heat pumps: These heat pumps reject and take heat from a water loop. The water is circulated throughout the building, allowing heat to be moved from areas that don't need it to those that do. Excess heat can be rejected in a cooling tower and needed heat can be added with a boiler.

These added complexities to the analysis are summarized in Table 5-6 and 5-7 below.

Table 5-6: Segmentation Refinements

Refinement	Segments Affected	Components Affected	Distribution (by floor area)
VAV Valve Series Fan Parallel Fan	Central VAV in Offices	Terminal Units	Percentages of VAV Floorspace Valve 50% Series 30% Parallel 20%
Chillers Air-Cooled Water-Cooled	All Central	CW Pump Tower Fan Condenser Fan	Depends on Building Type (see Table 5-7 below)
Water-Loop Heat Pumps (WLHP)	Lodging and Office Individual AC	CW Pump Tower Fan	Percentages of total building type floorspace WLHP's put into Individual AC segment. Office: Window AC: 3% WLHP: 10% Lodging: PTAC, PTHP: 44% WLHP: 15%

Table 5-7: Chiller Distribution

	Percent of Floorspace Served by Chiller Type	
Building Type	**Water-Cooled**	**Air-Cooled**
Education	40%	60%
Health Care	45%	55%
Lodging	70%	30%
Mercantile and Service	70%	30%
Office	50%	50%
Public Buildings	55%	45%
Warehouse/Storage	0%	100%

*ADL estimates based on industry interviews

5.3 Building Thermal Loads

Accurate estimates of building thermal loads are required in the calculations for effective full load hours for equipment which cycles or varies its capacity in response to space conditioning needs. The building thermal loads have been estimated by LBNL based on building models reported in Reference 19. The loads were calculated hourly using DOE2. This is the most complete database of commercial building HVAC load data which is available.

The building thermal loads represent the heating, cooling, and latent cooling loads which must be offset by the HVAC system in order to maintain setpoint conditions. As such, they include the effects of (1) internal heat gain, (2) heat transmission through the building shell, (3) solar load either directly through windows or transmitted through the building shell, (4) internal water vapor generation, (5) infiltration, and (6) building thermal mass. These loads do not include the added loads associated with fresh air ventilation, overcooling and necessary reheat, duct thermal losses, duct air losses, fan heat, etc.

The DOE2 output is arranged into 8,760 rows representing the hourly data. In addition to thermal loads, the data files contain weather data (dry bulb and humidity ratio), time data, fan electric loads, coil loads (heating, cooling, latent cooling), and thermal and electric loads for plant equipment (pumps, cooling tower, etc.). The additional information was used (1) as an additional background source for parasitic load data, and (2) as a check for the system modeling.

The hourly DOE2 loads were divided for spreadsheet calculations according to building operational status (occupied/unoccupied) and by outdoor dry bulb temperature. Averages of the space load variables were determined for use in the calculations.

5.4 Building System Modeling

Building system models were developed in order to provide a logical, accurate, and transparent representation of system energy use. Inputs for these models are the LBNL building load data, XenCAP™ data, and assumptions regarding system configurations and component descriptions. The outputs are the equipment design load intensity (DLI, W/sqft), and effective full-load hours (EFLH). The annual energy use intensity (EUI), (kWh/sqft) is the product of DLI and EFLH. Detailed description of the model equations is presented in Appendix 3.

Detailed building modeling was done for the office building for all regions and system types, and for the Northeast region for all building and system types. An extrapolation was used for estimates involving other building types and regions.

5.4.1 Design Input Power Loads

Aggregate design load intensity (DLI, W/sqft) values were estimated for each pertinent system component type for each of the building stock segments. The ranges for the DLI values for the major equipment types are summarized in Table 3-1 in Section 3.2. DLI values were estimated based on typical equipment characteristics for the given building application. Sources of input data for these calculations were the LBNL zone thermal loads, typical product literature values, interviews with industry experts, and engineering calculations.

The DLI estimates were cross-checked with XenCAP™ data and modified if necessary. The DLI estimates are described in more detail in Appendix 3.

5.4.2 Effective Full Load Hours

Effective Full Load Hours (EFLH) is a term used to describe the effective time that a particular component has been consuming energy, at full load, over an entire year. As an example, assume the supply and return fans in an air handler are running at full load for 12 hours a day and at half load for the remaining 12 hours. Over the course of a year, the fans would have an EFLH of 6,570 hours (18 hours per day, calculated by summing 12 hours full load plus half of 12 hours for the 50% load).

Estimates of effective full load hours (EFLH) depend on the type of operation of the component in question. The three basic types of operation considered are (1) schedule—the component operates according to a set schedule, (2) Cycling and (3) Variable. Classification of components into these categories is illustrated in Table 5-8 below.

Table 5-8: Effective Full-Load Hour Calculation Types

System Types	Components							
	Supply and Return Fans	Terminal Box Fan	CHW Pump	CW Pump	HW Pump	Cooling Tower Fan	Condenser Fan	Exhaust Fan
Central VAV	V	C	C/V	C[1]	C[2]	C/V		S
Central CAV	S		C/V	C[1]	C[2]	C/V		S
Central FCU	S/C		C/V	C[1]	C[2]	C/V		S
Packaged	S/C						C	S
Individual AC	C						C	S

Legend: S: Schedule
C: Cycling Operation
V: Variable Operation
[1]Operates when cooling is required
[2]Operates when heating is required

For components which operate on a schedule, EFLH is simply equal to the annual on-time, which is typically equal to the building's occupied hours.

The operation of variable or cycling equipment is modeled to determine the EFLH. The analysis starts with hourly building loads developed by LBNL for the prototypical buildings. The hourly building load data are organized into dry bulb temperature groups of 5ºF range for occupied and unoccupied hours. Also extracted from the LBNL data are the weather conditions, specifically mean coincident wet bulb temperature, mean coincident humidity ratio, and hours for each temperature group. Modeling of equipment operation is discussed in detail in Appendix 3.

5.5 Extrapolation of Values

The building modeling spreadsheet analysis was carried out rigorously for 84 building/region/system combinations: (1) all regions and systems for office buildings and (2) all building types and systems for the Northeast region. Estimates of DLI, EFLH, and EUI values for the remaining building/region/system combinations were developed by extrapolation according to the relations below.

$$DLI(Building, Region, System) = DLI(Building, Northeast, System) * RATIO$$

$$RATIO = \frac{DLI(Office, Region, System)}{DLI(Office, Northeast, System)}$$

To demonstrate the accuracy of the extrapolation ratios, various "spot-check" calculations were conducted for each component in the buildings with different building and system types as listed below in Table 5-9.

Table 5-9: EUI Extrapolation Data Comparison Choices

Region	City	Building Type	System Type
South	Fort Worth	Education	VAV
		Warehouse	Packaged
Midwest	Chicago	Health Care	FCU
		Large Retail	CAV
Mountain	Albuquerque	Food Service	Packaged
		Small Retail	Packaged
Pacific	San Francisco	Food Sales	Not Cooled
		Small Hotel	Individual

The direct calculations of the EUI's for each of the building and system types were then compared, graphically, to the extrapolated values as shown in Figure 5-10, for equipment loads (kWh/ft ^{2}), and in Figure 5-11, for coil and building loads (kWh/ft^2). The solid line in both figures represents an exact match between the extrapolated value and the direct calculated value.

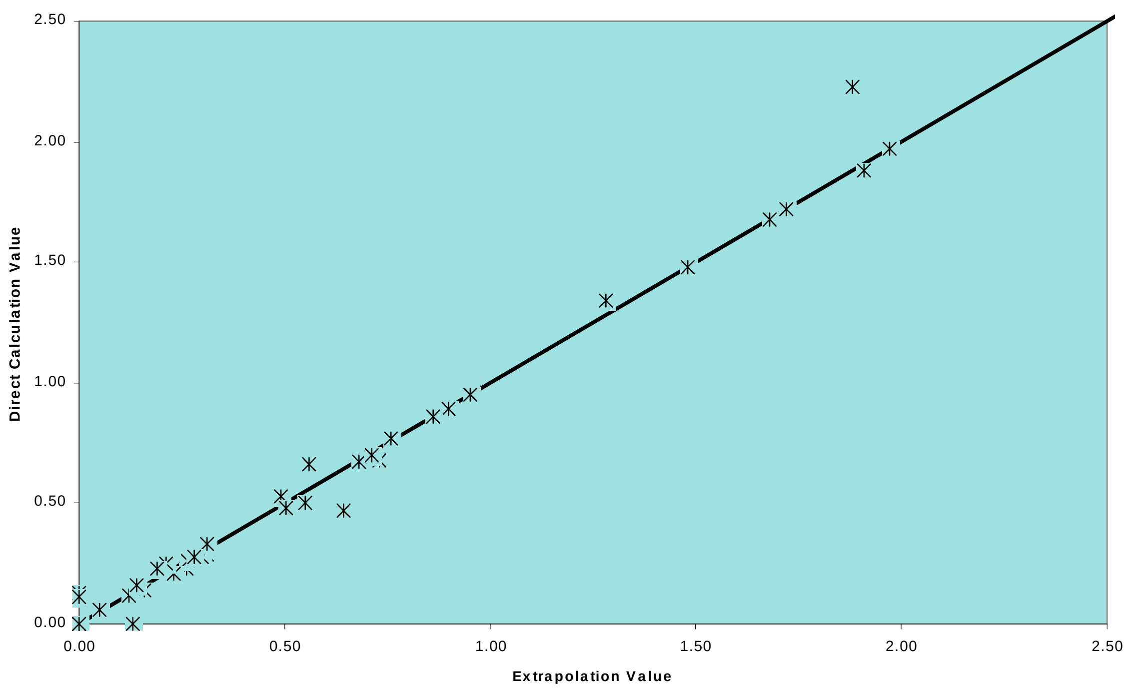


Figure 5-10: EUI Extrapolation Data Comparison (Equipment Loads kWh/SF)

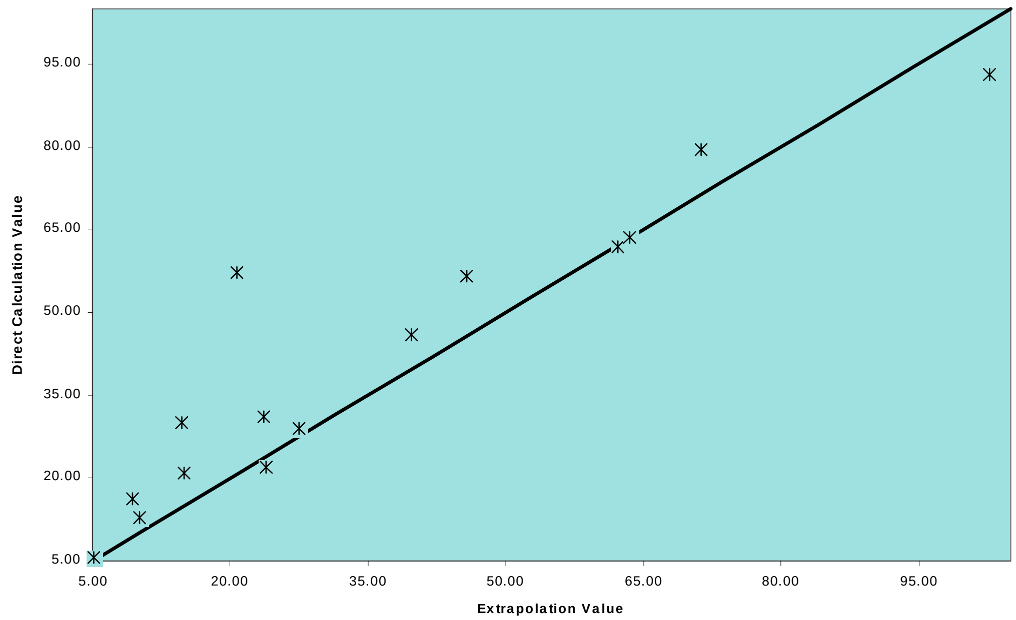


Figure 5-11: EUI Extrapolation Data Comparison (Coil & Building Loads kBtuh/SF)

As can be seen from the above graphs, the extrapolation method used to estimate the remaining values for all the building and system types outside the Northeast, was relatively accurate. Figure 5-11 does show some data points that were above the target line, showing that the extrapolation approach has resulted in a conservative estimate of energy use.

5.6 Energy Use Results

Total 1995 national commercial building HVAC parasitic energy use is estimated to be 1.5 quads of primary energy (a heat rate including generation, transmission, and distribution losses of 11,005 Btu/kWh has been assumed in conversion to primary energy). The breakdown of this energy by equipment, building type, geographic region, and system type are shown in the figures of this section. As can be seen from Figure 5-12, the largest users of this parasitic energy are the supply and return fans[8] and the exhaust fans. Together, these two system components comprise about 83% of the total parasitic load.

[8] Most of the supply and return fan energy is associated with supply fans, since return fans are used in a minority of cooling systems.

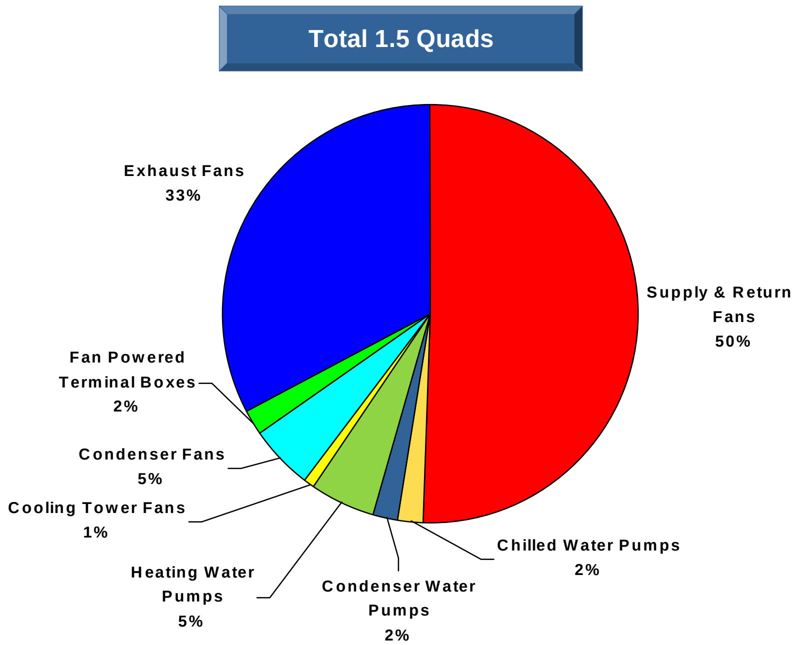


Figure 5-12: Parasitic Primary Energy Use – Equipment Breakdown

Supply fans use so much energy (about 0.75 Quad Total) because (1) they are used in virtually 100% of system types as defined (note that the evaporator fans of packaged or individual systems as well as fan-coil unit fans are considered in this category), (2) air is an inherently inefficient heat transfer medium, (3) typical air distribution design practice involves considerable pressure drop for filtration, cooling and heating coils, terminal boxes, and diffusers, and (4) many of these fans operate at 100% power during all building occupied periods.

Exhaust fans, while generally representing much less horsepower than supply fans, do use considerable amounts of energy (about 0.5 quad), since they are nearly all operated at 100% power during all building occupied periods. The contributions of central system auxiliary equipment (condenser water and chilled water pumps, cooling tower fans, and a portion of the condenser fans) are relatively modest because (1) their power input per ton of cooling is very low and (2) central systems represent less than one third of commercial building floorspace. Some of this equipment also has very low EFLH values due to its operating characteristics – it is used at full power very infrequently.

As observed in Figure 5-13 below, the building type that consumes the most parasitic energy is office (comprising about 25% of the total parasitic load). Energy Use Intensity for all HVAC parasitics equipment is shown for the building categories in Figure 5-14 below. Figure 5-13 also indicates that the smallest users of parasitic energy are Food Sales, Food Service, Lodging, and Warehouse. Although the Warehouse sector comprises a large share of the total floorspace in the commercial building sector, it is also the least cooled. Therefore, it is also one of the lowest consumers of parasitic energy. The percentage of parasitic energy associated with lodging may be surprisingly small, but the majority of hotels and motels utilize small, individual room AC PTAC's (Packaged Terminal Air Conditioners) as their cooling and heating source. These small, individual air conditioners typically have just one small fan motor.

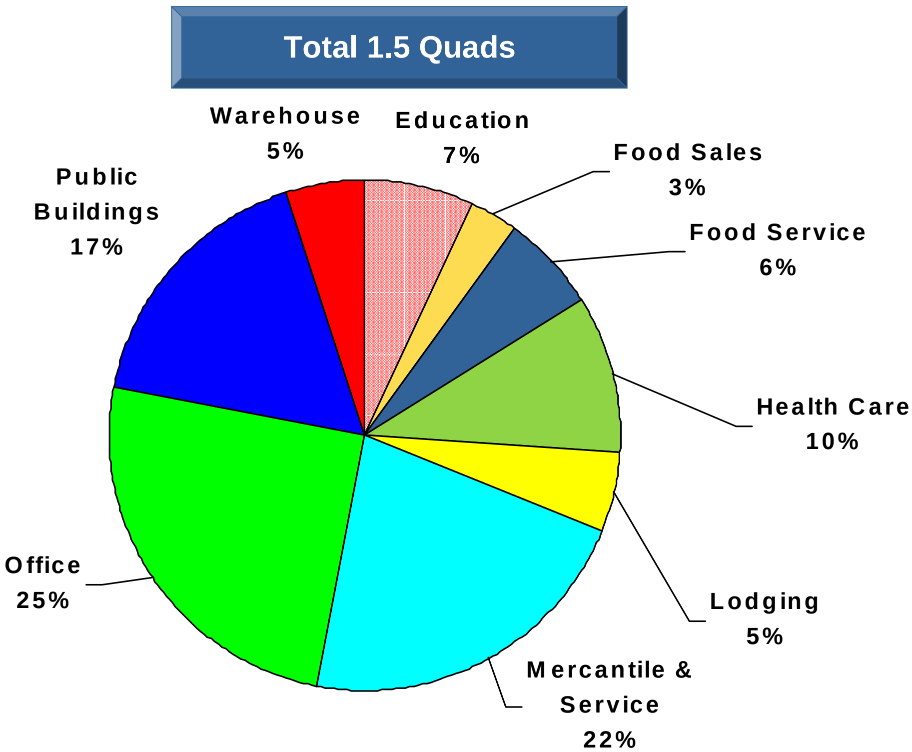


Figure 5-13: Parasitic Primary Energy Use - Building Type Breakdown

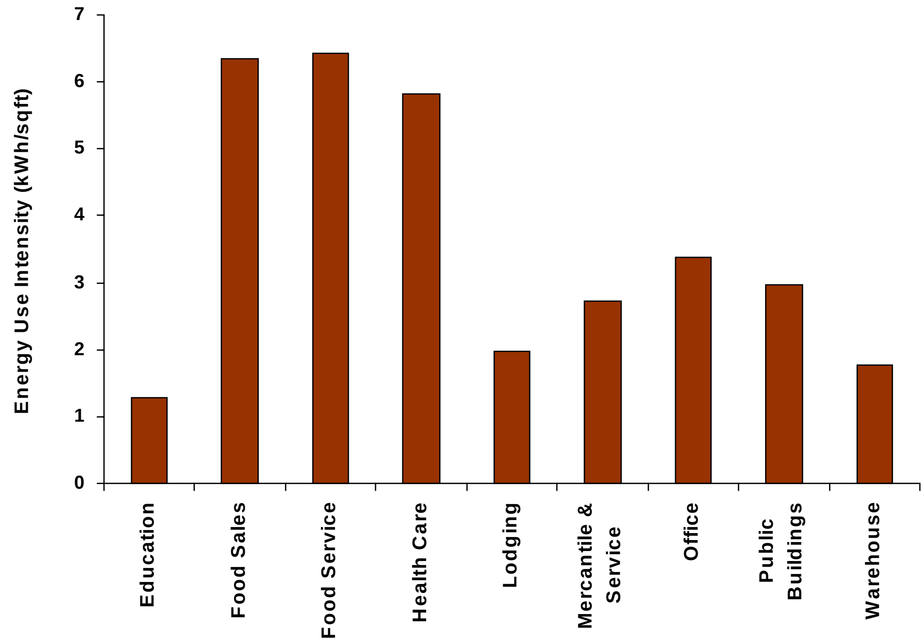


Figure 5-14: Parasitic Site Energy Use Intensity by Building Type

The Office and Mercantile & Service building types, which together account for nearly half of the HVAC parasitics energy use, are examined further in Table 5-10 below.

Table 5-10: Office and Mercantile & Service HVAC Parasitics Primary Energy Use Breakdowns (TBtu)

	Office	Mercantile & Service
Equipment Breakdown		
Supply & Return Air Fans	162	186
Exhaust Fans	138	95
Terminal Box	23	—
Condenser Fan	15	17
Cooling Tower Fan	6	3
Heating Water Pump	11	6
Condenser Water Pump	8	5
Chilled Water Pump	10	4
System Breakdown		
Central CAV	57	9
Central VAV	111	34
Central FCU	15	21
Packaged	162	215
Individual	21	6
Not Cooled	7	33

The distribution of HVAC parasitic energy use by geographic region strongly reflects the commercial building floorspace breakdown. The energy use and floorspace distributions by region are show in Figure 5-15 below. The differences in the two distributions are due to the expected differences in energy use intensity resulting from higher cooling loads in warmer regions.

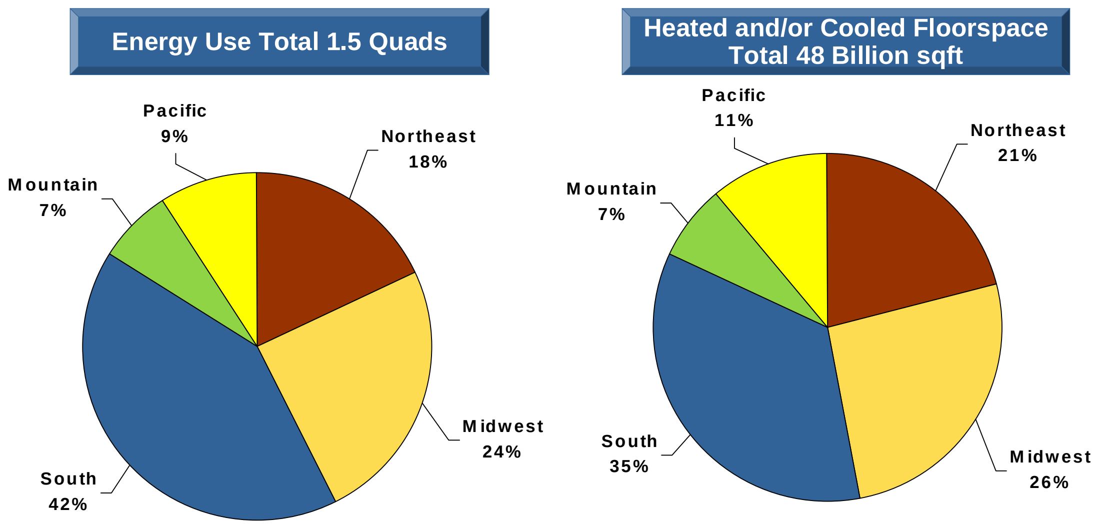


Figure 5-15: Parasitic Primary Energy Use and Floorspace - Geographic Region Breakdown

The system type breakout shown in Figure 5-16 below shows that packaged systems represent the largest amount of parasitic energy use. This is primarily because there is much more floorspace associated with packaged systems than with the other system types.

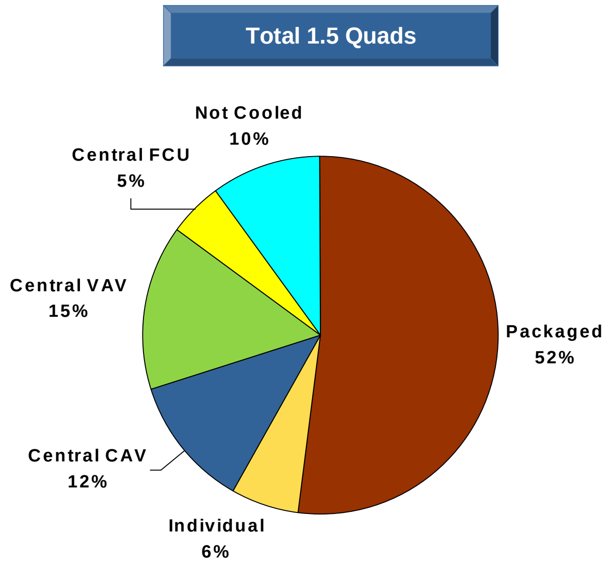
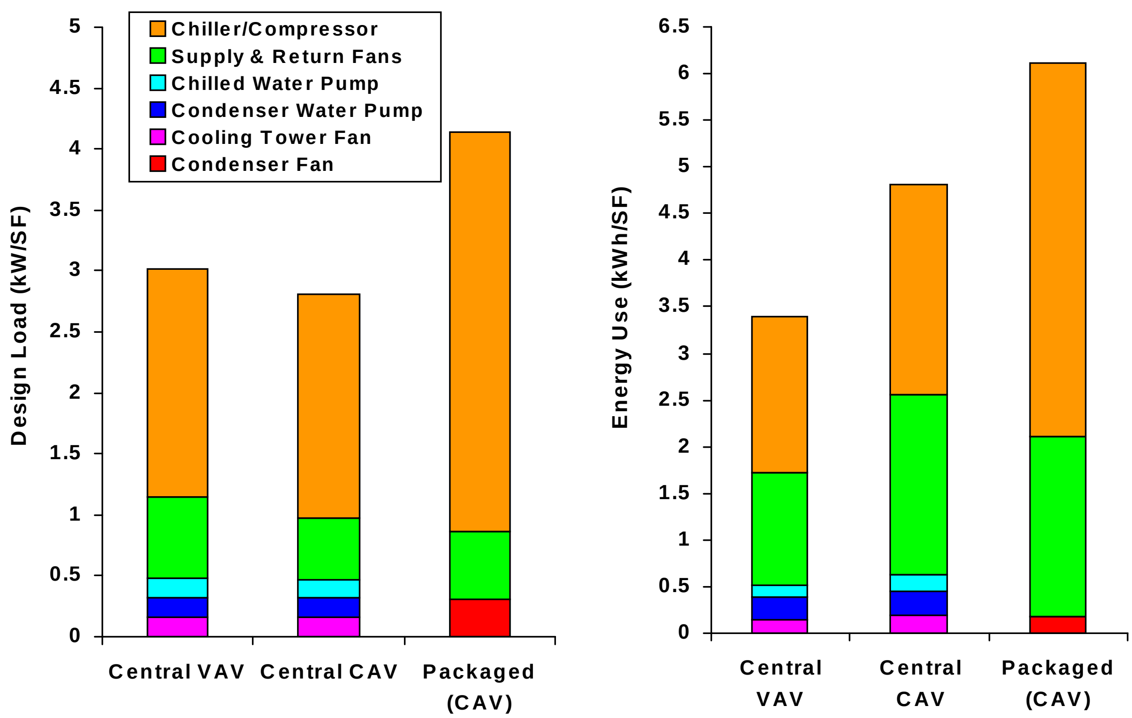


Figure 5-16: Parasitic Energy Use - System Type Breakdown

Efficiency of central and packaged systems is compared in Figure 5-17 below for the office building type. This comparison of prototypical systems in prototypical buildings shows that the central system with VAV has better design condition efficiency and also has better part-load performance than a packaged system. The differences are primarily due to:

- Heat rejection in the central system using a cooling tower, which enhances heat rejection through evaporation of condenser water.
- Use of larger more-efficient refrigerant compressors for the central systems
- Constant-volume operation of the packaged unit and the Central CAV supply fans in spite of varying cooling loads. This accounts for the fact that supply fan energy use is higher for these two systems, even though design fan input power is higher for the VAV system.
- Chilled water pump energy is higher for the CAV than the VAV system due to the higher annual cooling.
- As expected, the packaged system *parasitic* energy use is lower than for the CAV system, since less equipment is required and thermal distribution distance is typically less.

These five factors more than make up for the central system disadvantages of additional heat exchangers and thermal distribution associated with the central chiller. However, it should be noted that packaged systems can be designed for variable-volume operation, be fitted with higher-efficiency components, and utilize evaporative condensers, which would practically eliminate the efficiency advantage of a central system.

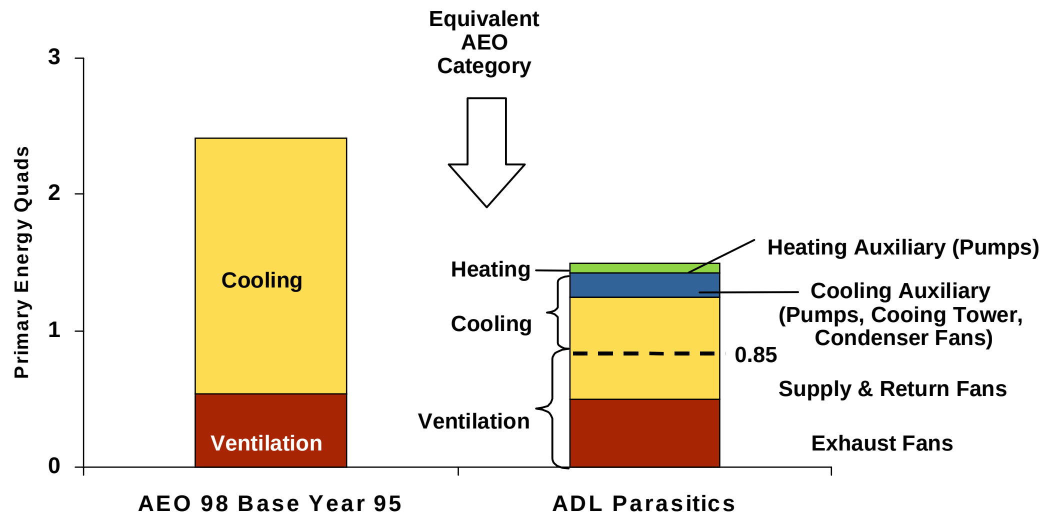


Note: Refrigerant Compressor Efficiency assumed to operate with constant efficiency for simplification. Typical compressor efficiencies for prototypical systems have been assumed.

Figure 5-17: Design Load and Energy Use Comparison of Central VAV, Central CAV and Packaged Systems

5.7 Comparison to Other Studies

The overall results of this study are compared to the AEO98 (Reference 1) estimates in Figure 5-18 below. Comparison is complicated by the differences in categories. There is an obvious mapping of exhaust fans to “ventilation”, of cooling auxiliary equipment to “Cooling”, and of heating water pumps to “Heating”. However, the supply and return fan energy could be in any of these three categories. The figure shows the comparison assuming half of the supply and return fan energy is considered “Ventilation”. The results of this study show somewhat higher energy use than AEO98.



*Assuming 50% of Supply/Return Fan Power is "Ventilation"

Figure 5-18: Comparison of This Study's Results to AEO 98

The table below, Table 5-11, references the DLI and EUI numbers compiled from a variety of sources. The numbers represent XenCAP™ data described in Appendix 1, and data from References 11, 13, 14, and 15. The data comparisons show that agreement between studies is not consistent at this level of detail. However, the comparisons do show that the ADL estimates are within ranges of estimates made by others.

Table 5-11: Data Comparisons for DLI and EUI Values

	THIS STUDY			MEASUREMENTS					ANALYTICAL STUDIES		
References	N/A	N/A	N/A	N/A	N/A	15	15	15	19	11	13
	ADL Parasitics Estim. National DLI (W/SF)	ADL Parasitics Estim. National EUI (kWh/SF)	ADL Parasitics Estim. Pacific EUI (kWh/SF) ** Used for comparison to Reference 13	XenCAP - DLI (W/SF)	XenCAP - Annual EUI (kWh/SF) AHU	ELCAP - Washington State Annual EUI (kWh/SF) *Vent/Aux*	ELCAP - Washington State Annual EUI (kWh/SF) *Cooling*	ELCAP - Wash. State DLI (W/SF) *Total HVAC*	LBL - Annual Energy Use Intensity (kWh/SF)	PNNL Offices - March 1992 Annual Energy Use Intensity (kWh/SF)	California Study -- Annual Energy Use Intensity (kWh/SF)
Footnotes				1	1			2	1	1	
Overall Total	0.95	2.77	2.15	0.45	2.83						
Education, tot	0.52	1.28	0.99	0.37	1.91				0.75		1.12
Educ. - School											0.68
Educ. - College											1.85
Food Sales	1.06	6.35	4.24	0.33	2.22	2.83	2.17	1.48	5.50		1.97
Food Service	1.52	6.43	4.22	0.45	2.35	5.54	5.79	2.79	4.40		5.63
Health Care	1.47	5.58	4.47	0.47	3.99				3.90		2.49
Lodging	0.52	1.86	1.67	0.19	1.43				1.90		1.06
Merc & Serv, tot	0.89	2.68	1.95	0.35	1.78						1.40
Merc & Serv, Lg									3.00		1.40
Merc & Serv, Sm						1.19	1.22	0.76			1.40
Office, tot	1.33	3.32	2.61	0.50	2.84						2.48
Office, Lg									5.50	3.30	2.91
Office, Sm						3.77	1.96	1.95		2.40	1.00
Public Assembly	1.22	2.98	2.10								
Warehouse	0.40	1.77	1.51	0.32	1.58	0.52		0.22			0.25
Footnotes											
1	Includes only Air-Handling Units and Exhaust Fans										
2	Summer Load, may not be peak load										

Figure 5-19 compares the distribution by building of energy use of all fans and pumps reported in the LBNL Study "Efficient Thermal Energy Distribution in Commercial Buildings" (Reference 13) with the calculations of this report for the Pacific region. Except for differences in the Mercantile/Service and Office categories, the results compare very well.

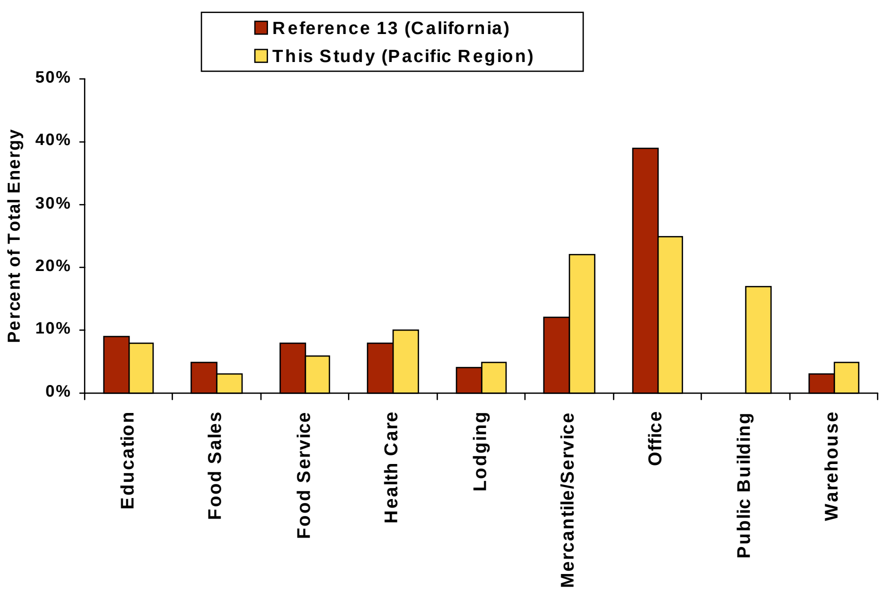


Figure 5-19: Thermal Distribution Energy Use Breakdown by Building: Comparison to Reference 13

6. Conclusions and Recommendations

A rigorous bottoms-up analysis was done to estimate energy use in commercial building HVAC parasitic equipment. This equipment includes the fans and pumps used for thermal distribution and ventilation, as well as auxiliary equipment such as cooling towers and condenser pumps. A rigorous segmentation of the commercial building stock was done based on the CBECS95 survey of commercial buildings. The building stock was segmented according to building type, region, and HVAC system type. Energy use analysis focussed on those equipment types of most significance: Supply and Return Fans, Exhaust Fans, Condenser Fans, Cooling Towers, Chilled Water Pumps, Condenser Water Pumps, Heating Water Pumps, Terminal Box Fans, and fans of Fan-Coil Units, Room Air-Conditioners, etc. Energy use estimation was based on commercial building HVAC load models developed by LBNL—this is the most thorough database available for this type of information. Component equipment energy use was estimated based on a rigorous set of system operating models which reflects typical system operating practice and equipment energy use characteristics.

Interim estimates and final results of the study were compared with a number of data sources. The XenCAP™ commercial building energy use database, representing about 2,000 buildings of varied geography, building type, and age, was used extensively throughout the analysis as an input to design load estimates and a check for annual energy use estimates. Final results compared well with a number of estimates and measurements of commercial building fan and pump energy use. Extensive review of the interim results and the draft final report by industry experts serves to further strengthen the methodology and conclusions.

Summary results of the study are as follows.

The parasitic equipment (pumps and fans) used in commercial building HVAC systems for thermal distribution and ventilation represent a considerable amount of total HVAC energy use: about 1.5 quads annual national primary energy use for parasitics as compared with 1.87 quads for space cooling and 1.85 for space heating (Reference 1). The major users of this parasitic energy are fans associated with the air handling units and exhaust fans. While some of the supply fans, especially large VAV units, are fairly efficient at design load and are controlled to vary flow efficiently, many small-size fans have low or modest efficiencies, especially when installed in tightly packaged air conditioning systems. Energy use of fans is significantly affected by system design practice, installation procedures, whether the system is properly commissioned, and whether the system receives proper maintenance. While the national impact of some of these factors cannot readily be determined, it is clear that A&E firms, installers, and users have a significant impact on system energy use.

The energy use associated with chilled water pumps, condenser water pumps, cooling tower fans, condenser fans, and heating water pumps, while not insignificant, is dwarfed by that of supply, return, and exhaust fans.

The upcoming second phase of this study will focus on opportunities for energy savings. However, a few recommendations do become clear at this stage:

1) An investigation of the impact of departures from as-designed energy performance of HVAC systems is in order. Quantification of this issue will help significantly in guiding future energy reduction efforts.

2) Research and development of high-efficiency fans is an area that has a dramatic potential to impact national energy use. Peak efficiencies achieved in centrifugal compressors approach 80%. It is reasonable to assume that such efficiencies could be achieved in HVAC fans. Our interviews with industry representatives suggests that little is currently being done to boost fan design-load efficiencies. More focus has been on part load efficiency achievable with variable volume operation. However, many smaller systems and exhaust fans do not operate with variable volume. Furthermore, these smaller fans are typically not as efficient.

   Trade-offs exist between cost and efficiency. Fans in smaller packaged units must be compact. Typical blade design is forward-curved, which provides for good pressure rise for a given diameter and speed. However, the introduction of low-cost, higher-speed airfoil fan blades could improve the energy performance while minimizing cost impact.

3) Development of lower-cost variable-speed drives, especially in smaller sizes, would increase the proliferation of variable-speed air-conditioning. In many market sectors, installation cost is still one of the most important issues, and the cost of these drives is prohibitive. Further research into lower-cost power electronics would help to reduce these costs.

4) High-efficiency motors are an option that would reduce fan and pump power in all applications. While many large-hp motors are relatively efficient, reduction of the cost premium of high-efficiency motor technology could make a dramatic impact. For instance, 5% average reduction of motor power is worth 100 TBtu of primary energy in commercial HVAC parasitic applications.

5) Further study of potential energy saving options is necessary. The impact of advanced cooling techniques that don’t rely on air as the primary thermal transport fluid may be significant. Lower-cost ways to efficiently satisfy varying cooling

loads while also satisfying ventilation needs in commercial buildings need to be identified and discussed within the HVAC design community.

The next phase of this study will investigate these issues further.

7. References

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21. Technology Forecast Updates — Ventilation Technologies in the NEMS Commercial Model, prepared for DAC & EIA by ADL, August 1996.

Appendix 1: XenCAP™ Energy Use Data

The energy uses results presented in this report are estimates based on a variety of information sources. A critical step in development of the estimates has been comparison with the XenCAP™ data described in this appendix. The favorable comparison of our estimates to this field-collected building data serves to strengthen the credibility of the final results.

This appendix describes the collection and reduction of the XenCAP™ building data, illustrates its broad representation of the national building stock, and provides a summary of the main findings.

Source of the Data

An existing database of site measured building information was used in this study of parasitic HVAC loads. This building database was built from data collected under several different DSM programs conducted by electric utilities in the period from 1986 to 1995. There were two primary motives for the collection of this data. First of all, the utilities used the data to perform energy audits on the facilities. The results of the energy audits were provided to the facility owners in the form of a written report which contained an analysis of the energy consumption by end use and recommendations for conserving energy. The second motive for collecting the data was to develop a database of information on how different types of facilities use energy as an indication of market potential for demand side management initiatives. The utilities involved are listed in the following table. The primary rationale for selection of the utility data sets for this study was to obtain a diverse geographical representation.

Table A1-1: XenCAP™ Utility Companies Surveyed

Utility Name	Time frame of Data Collection	DOE Climate Zone[1]
Georgia Power Company	1993-1994	Zone 4/5
Anaheim Public Utilities Dept.	1993-1994	Zone 4/5
Kansas City Power and Light	1993	Zone 3
Northern States Power	1988-1989	Zone 1
Orange and Rockland Utilities	1986 and 1991-1994	Zone 2
Ohio Edison	1993-1996	Zone 2
Omaha Public Power District	1993-1995	Zone 2
Wisconsin Electric Power Company	1988-1992	Zone 1
Missouri Public Service	1992	Zone 3
City of Pasadena Water and Power Dept.	1991-1993	Zone 4/5
Central Maine Power	1993	Zone 1
Public Service Electric and Gas	1990-1994	Zone 3

[1]See Table A1-2

Table A1-2: U.S. DOE Climate Zones

DOE Climate Zone	Heating Degree Days (HDD)	Cooling Degree Days (CDD)
Zone 1	>7,000	<2,000
Zone 2	5,500 – 7,000	<2,000
Zone 3	4,000 – 5,499	<2,000
Zone 4	<4,000	<2,000
Zone 5	<4,000	>2,000

Data Collection

All of these programs utilized the XenCAP™ energy analysis software program for collecting and analyzing the data. XenCAP™ requires detailed building data including: occupancy, operating hours, historical energy use, an inventory of all energy using equipment (lighting, HVAC, process, etc.), and a description of the building shell. This data is typically obtained by trained energy auditors. The auditors visit the subject building and collect information through: interviews with facility managers, review of blueprints and equipment records, and by physically observing and recording nameplate information from all equipment in the facility. Historical energy use data is normally obtained from the utility sponsoring the program. The auditor uses a formset to collect the data required by XenCAP™. For a typical piece of equipment, the auditor must enter the following information into the formset:

- System or equipment type
- Capacity (e.g., horsepower or CFM)
- Operating hours
- Control scheme
- Age
- Efficiency

The exact data required varies depending on the type of equipment. The auditor also selects energy efficiency measures appropriate for the equipment and enters this information into the formset.

The XenCAP™ Analysis

Once all of the building data has been entered into the XenCAP™ input record, the data is analyzed. XenCAP™ calculates the annual energy consumption of each piece of equipment included in the input record. The energy consumption is summed by end-use (e.g., lighting, cooling, heating, ventilation, etc.). The total energy consumption of all end-uses is then compared with the actual energy consumption of the building from the energy bills. XenCAP™ then automatically reconciles the calculated energy consumption with the actual energy consumption by adjusting various modeling parameters. The adjusted model is checked by a quality control engineer and manual adjustments are made if necessary. Typical adjustments that are made by the XenCAP™ software and the QC engineer involve changes to building heat loss factors, internal heat gain factors, lighting operating schedules, heating and cooling equipment efficiencies, and others. The final product is a model of the building that results in energy consumption matching the actual energy bills. The original data collected by the energy auditor, along with the results of the analysis are stored in a database with all other audits performed under that particular DSM program. This is the data used in the HVAC parasitic load study.

Database Structure

The XenCAP™ database is organized by end-use. Each end-use has a separate data table. The primary end-uses are heating, air-conditioning, domestic hot water, cooking, refrigeration, exterior lighting, interior lighting, industrial process equipment, and ventilation. Equipment must be inventoried under the appropriate end use. In some cases a single piece of equipment is included in several end-use tables. For example, a packaged rooftop unit may serve heating, cooling, and ventilation end-uses. In this case, the rooftop unit would be included in all three of these tables. The information in the heating table is used to calculate the heating end-use, the information in the cooling table is used to compute the cooling end-use, and so forth. These three occurrences of the rooftop unit are in no way linked. For example, it can not always be determined which fan on the ventilation table is associated with a given compressor on the cooling table. Likewise, pumps are not linked to the chillers or boilers which they serve.

Fans are included in the ventilation table. Unfortunately, there is no end-use for pumps. Pumps typically serve heating and or cooling end-uses. They are not inventoried on the heating or cooling tables, however. Pumps are inventoried on the motors table, where they are grouped with all motors, serving various end-uses.

Processing the Database for the Parasitic Load Study

Because XenCAP™ was designed for performing energy audits and not HVAC parasitic load studies, some preprocessing of the data was necessary to obtain a dataset that would be useful for this study. The following data calculations were performed in order to prepare the database for the parasitics study.

1. Buildings that do not have chillers or boilers were filtered out.
2. The cooled square footage of the building was calculated. This involved summing the area of all zones that are cooled.
3. The total area (sqft) served by all ventilation equipment within a building was calculated. This calculation involved summing the area of all zones that are either heated or cooled.
4. The area served (sqft) by individual chillers was calculated. This was determined by summing the area of all zones served by a chiller. If more than one chiller serves a zone, then the total zone area was distributed among the chillers based on their relative sizes.

5. The building total installed kW of cooling equipment was calculated from rated capacity data, nameplate COP, and fraction of original nameplate COP. The COP and fraction of nameplate COP are default values in the XenCAP™ program and are determined by age and type of equipment. The fraction of nameplate COP parameter adjusts COP to account for chiller aging.
6. The area (sqft) served by individual fans was calculated. This was determined by summing the area of all zones served by a fan. If more than one air handler serves a zone then the total zone area was distributed among the air handlers based on their relative sizes in CFM.
7. Building cooling energy use (from database) was disaggregated among all cooling equipment serving the building. Disaggregation was based on relative sizes of cooling units in terms of peak kW demand. This calculation was needed because XenCAP™ does not provide cooling energy use for an individual piece of equipment.
8. Fan input power and fan energy use were calculated for air handling equipment serving zones that are also served by chillers.
9. Pump motor peak demand and annual energy usage were calculated.

The results of these calculations were stored in the following set of data tables:

- building data
- boilers
- chillers
- distribution system
- motors (pumps)

Relevance of the XenCAP™ database

It is not unreasonable to assume that the buildings included in the XenCAP™ database represent "typical" situations in commercial buildings. While no statistical sampling was performed in selecting the buildings for this study, we are using a large data set formed from a somewhat arbitrary collection of geographically scattered databases. The DSM programs which were responsible for obtaining the data had a variety of objectives. In most cases, the buildings were selected to obtain a statistical sample of the utility's customer base. In a few, buildings that had already performed energy conservation upgrades were targeted. Others involved audit programs in which the building owners had to make a request to receive an energy audit and thereby be included in the database. Because of this diversity, as well as the sampling techniques used in selecting many of the facilities, the buildings in this database should provide a fairly good representation of the general building population with respect to the HVAC systems, as well as other end users.

Results

The data provided in the distribution system and motors tables can be used to look at the electrical demand and energy consumption of fans and pumps. The data can be sorted and summarized by geographic region, facility type, and facility age. The types of fan systems in use can be determined for different facility types and ages. The database also gives insight into the energy consumption of the various fan system types on a watt/sqft basis.

Distribution of Buildings

The XenCAP™ data used for this study includes 1,978 commercial buildings representing 246 million square feet. The following figures provide a description of the range of buildings included in the database.

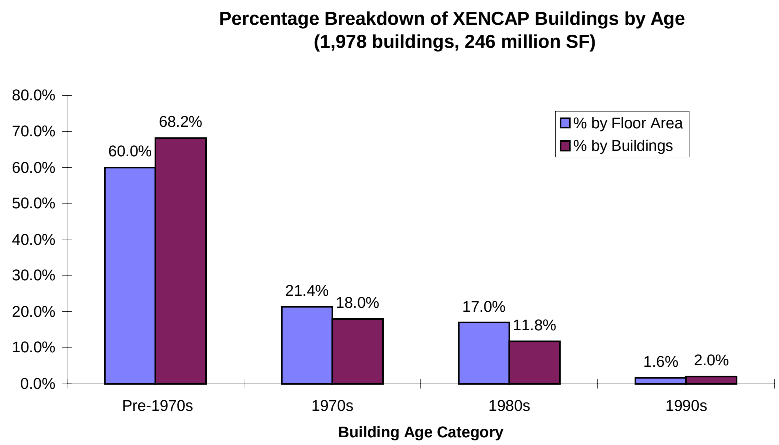


Figure A1-1: XenCAP™ Database Building Distribution by Building Age

Figure A1-1 shows the distribution of the XenCAP™ buildings by building age. Figure A1-2 shows the building distribution among the DOE climate zones.

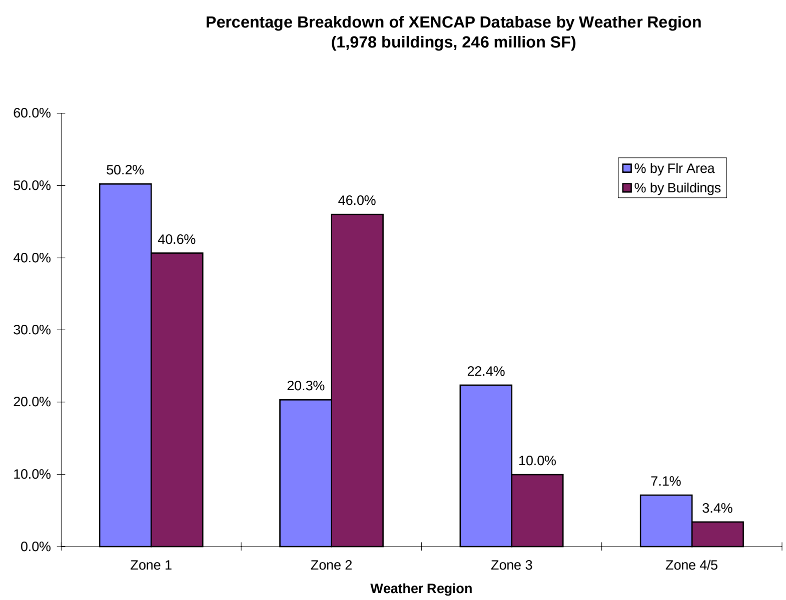


Figure A1-2:XenCAP™ Database Building Distribution by DOE Climate Zone

Figure A1-3 below shows the building distribution by building type.

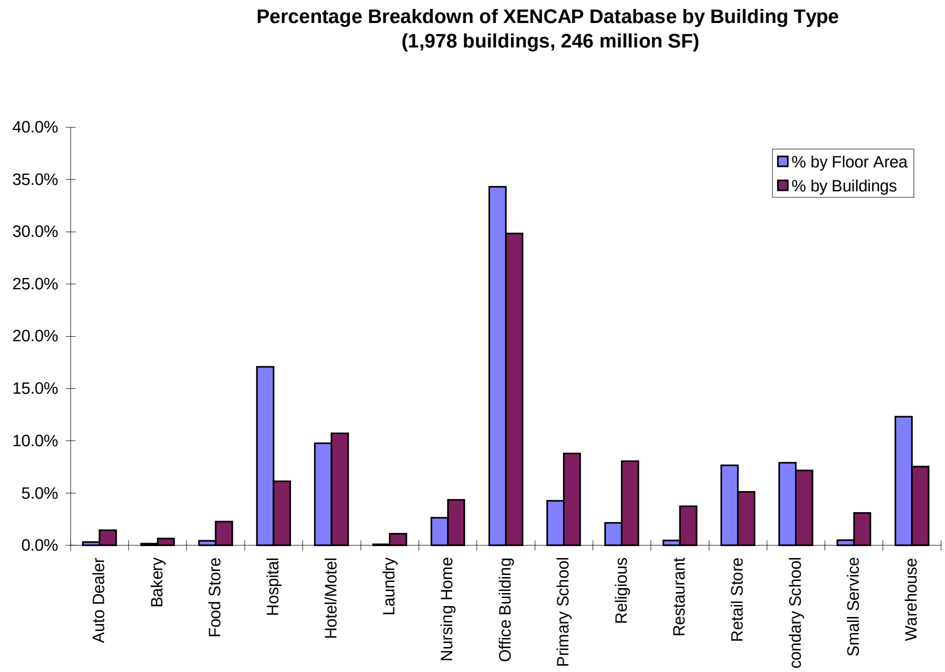


Figure A1-3:XenCAP™ Database Building Distribution by Building Type

Energy Use Data Summary

Average Design Load Intensity (DLI), (w/sqft) and Annual Energy Use Intensity (EUI), (kWh/sqft) breakdowns by equipment type are presented for the XenCAP™ buildings in Figures A1-4 and A1-5 below.

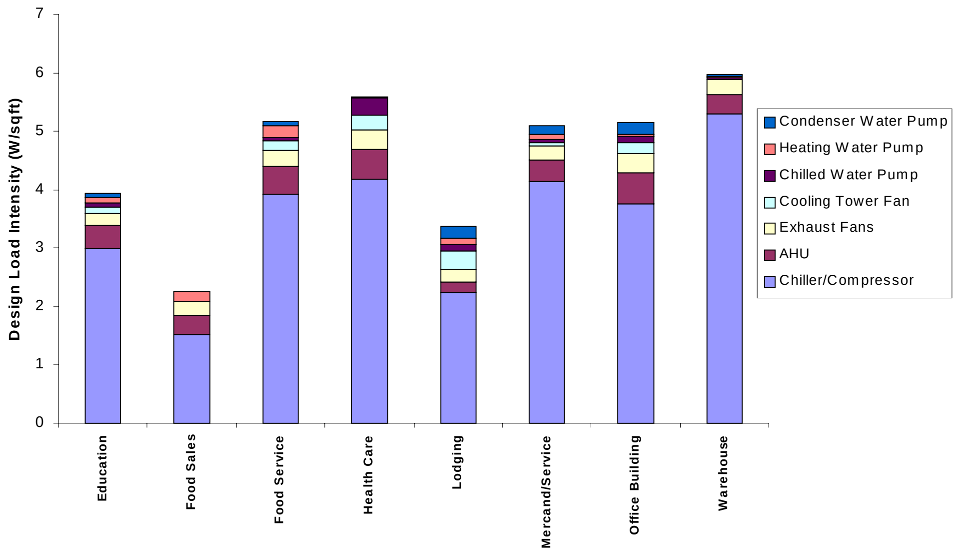


Figure A1-4: XenCAP™ Building DLI (w/sqft) Summary

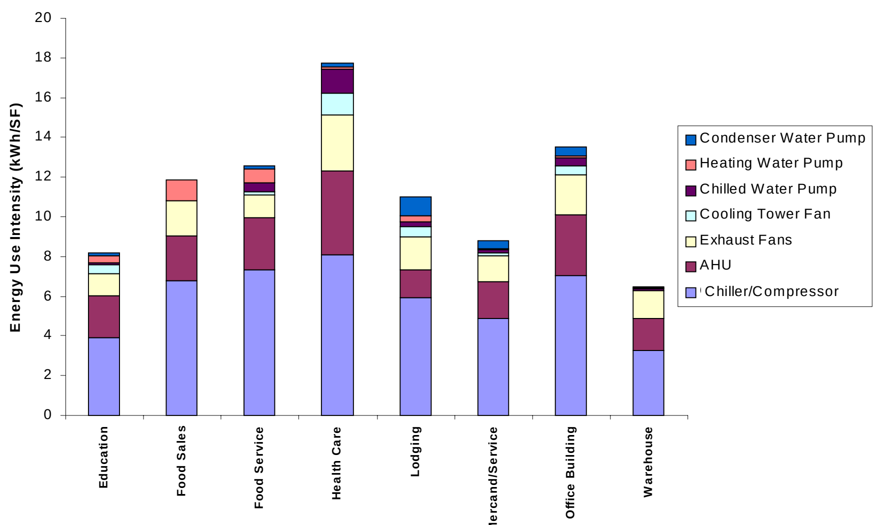
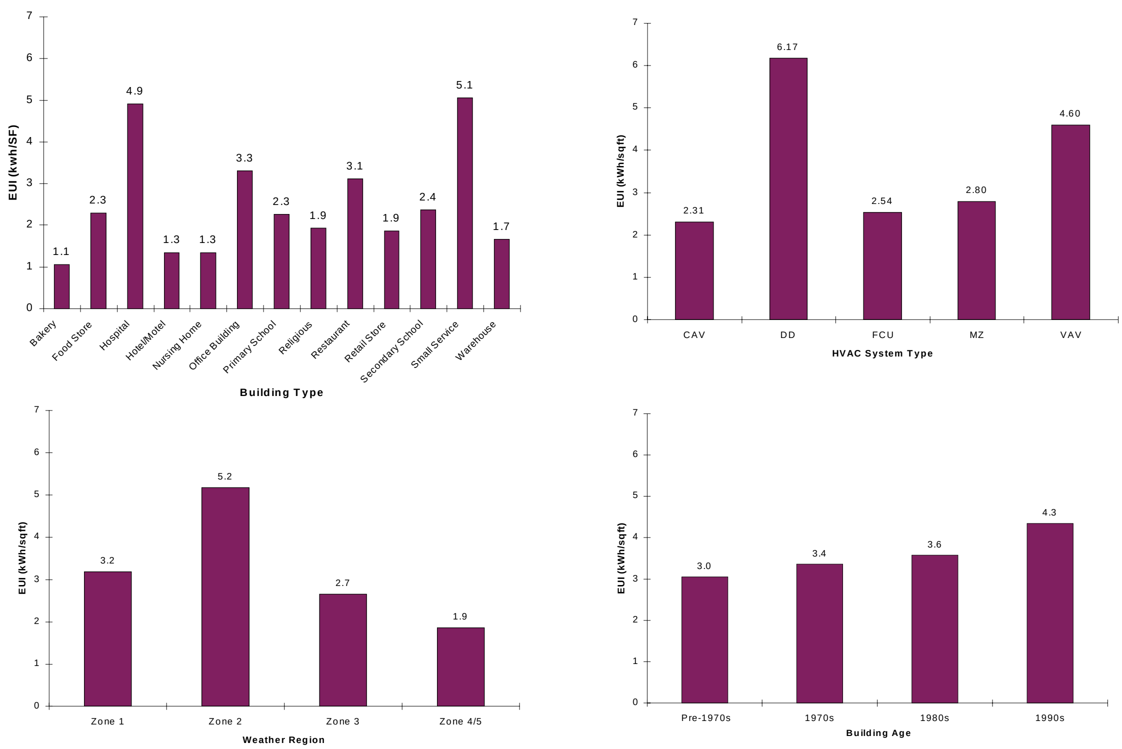


Figure A1-5: XenCAP™ Building EUI (kWh/sqft) Summary

Four breakdowns of the XenCAP™ fan electircity use data is shown in Figure A1-6 below. Although the number of buildings in the database is not high enough to draw conclusions from those plots with high statistical confidence, the plots do show some interesting patterns with respect to fan electricity use. First, the most important variables determining fan energy use are building type and system type. Further, the plot based on building age suggests that fan energy intensity is increasing, a trend which was not fully corroborated by the interviews presented in Appendix 5.



FCU: Fan Coil Units
MAU: Make Up Air Units
CAV: Constant Air Volume
VAV: Variable Air Volume

MZ: Multizone
DD: Dual Duct
Zones: DOE Climate Zones (Zone 1 is coolest)

Figure A1-6: XenCAP™ Fan Power Data

The building stock segmentation developed in this study is represented by the building/system distribution and the regional distribution presented in the following tables.

Table A2-12: Conditioned Floorspace Segmentation: Building Type and System Type (million sqft)

	Education	Food Sales	Food Service	Health Care	Lodging	Mercantile and Service	Office	Public Buildings	Warehouse/ Storage	Totals
Individual AC	805	0	83	134	1,669	333	1,257	371	119	4,771
Packaged	2,204	534	1,100	557	283	5,820	4,450	3,337	1,482	19,767
Central VAV	551	0	0	401	85	1,081	2,322	847	0	5,287
Central FCU	466	0	0	334	707	831	484	0	0	2,822
Central CAV	212	0	0	802	85	249	1,161	741	102	3,352
Not Cooled	3,522	20	64	159	779	2,507	561	2,168	2,285	12,065
Totals	7,760	554	1,247	2,387	3,608	10,821	10,231	7,464	3,988	48,064

Table A2-13: Floorspace Segmentation: Geographic Region (million sqft)

Northeast	Midwest	South	Mountain	Pacific	Total
9,919	12,382	16,667	3,272	5,824	48,064

Sources: CBECS 95 (Reference 3); References 16, 17, 18; ADL estimates

The industry review of the floorspace segmentation is presented in Table A2-3 and A2-4 below. Table A2-3 shows distributions of *cooled* floorspace for the buildings types discussed with the reviewers. The arrows indicate the adjustments made as a result of this industry review. Table A2-4 summarizes the reviewers' comments.

Table A2-3: Industry Review of Segmentation Estimates: "Before" and "After" Cooled Floorspace Distributions

Building Type	System Type				
	Central			Packaged Systems	Individual Room AC*
	With VAV Systems	With CAV Systems	With Fan Coil Units		
Education	13%	16%→5%	11%	51%→52%	8%→19%
Health Care	24%→18%	9%→36%	15%	45%→25%	6%
Lodging	9%→3%	9%→3%	12% →25%	43%→10%	27% →59%
Mercantile & Service	8%→13%	8%→3%	0% →10%	79%→70%	4%
Office	22%→24%	10%→12%	5%	61%→46%	3% →13%

*PTAC, PTHP, window units, and Water-Loop Heat Pumps

Table A2-4: Industry Review of the Preliminary Building Stock Segmentation

	Expert 1	Expert 2	Expert 3
Education	• Few Central systems • About half of floorspace not cooled • Many Unit Ventilators	• Numbers OK for secondary schools of smaller size • Primary will typically have smaller packaged systems	• There is very little central in education • About 50% of floorspace will have through-the-wall unit ventilators
Health Care	• Much more CAV (40%) • Less VAV and packaged	• Increase CAV • Reduce Packaged	• CAV is too low – more like 30% • Packaged too high – more like 20%
Lodging	• Majority of space should be PTAC's or FCU • Very few VAV, CAV, packaged	• Increase individual AC (should be much higher than 27%) • Reduce packaged	• 9% each for CAV & VAV is high • Water source heat pump is the predominant system for high rise hotels built since ~1980 • 43% for packaged is high
Mercantile & Service	• Less CAV • More VAV & FCU		
Office	• Suggest some minor adjustments	• More VAV & CAV • Less packaged • Central vs. packaged based on # of floors	• Speculative Office space has used water source heat pumps since early 1980's
General Comment		• Regional variation is significant	

Note: Percentages represent portion of cooled floorspace rather than portion of conditioned (heated and/or cooled) floorspace.

This Appendix describes in detail the approach to estimation of equipment loads and energy use, particularly for cycling and variable equipment.

Building system models were developed in order to provide a logical, accurate, and transparent representation of system energy use. Inputs for these models are the LBNL building load data, XenCAP™ data, and assumptions regarding system configurations and component descriptions. The outputs are the equipment design load intensity (DLI, W/sqft), and effective full-load hours (EFLH). The annual energy use intensity (EUI), (kWh/sqft) is the product of DLI and EFLH. Aggregate design load intensity (DLI, W/sqft) values were estimated for each pertinent system component type for each of the building stock segments. DLI values were estimated based on typical equipment characteristics for the given building application. Sources of input data for these calculations were the zone thermal loads, typical product literature values, interviews with industry experts, and engineering calculations. The DLI estimates were cross-checked with XenCAP™ data and modified if necessary. Estimates of effective full load hours (EFLH) depend on the type of operation of the component in question. The three basic types of operation considered are (1) schedule—the component operates according to a set schedule, (2) Cycling and (3) Variable. The operation of variable or cycling equipment is modeled to determine the EFLH. The analysis starts with hourly building loads developed by LBNL for the prototypical buildings. The hourly building load data are organized into dry bulb temperature groups of 5ºF range for occupied and unoccupied hours. Also extracted from the LBNL data are the weather conditions, specifically mean coincident wet bulb temperature, mean coincident humidity ratio, and hours for each temperature group.

System and equipment state is determined for occupied and unoccupied status at each applicable dry bulb temperature. Hours at each temperature are then used to calculate annual energy use. The modeling starts with building thermal load data developed by LBNL. From thermal loads, the operation of the air-handling unit is determined. Supply airflow and conditions and ventilation (outdoor) air quantity are used to calculate coil loads. The coil load determines operation of the chiller and auxiliaries such as the condenser water pump, the chilled water pump, and the cooling tower fan (or the condenser fan for an air-cooled system). The component modeling approach is described in some detail and it is assumed that the reader has some technical knowledge of HVAC equipment.

Air-Handling Unit Operation

Air Quantity

Design temperature and humidity ratio conditions are established for indoor air, air-handling unit supply air, and outdoor air. Design sensible and latent loads are set equal to the maximum loads calculated from the 5^{o}F dry bulb groups derived from the LBNL analysis. The design airflow rate required to satisfy the sensible and latent loads are determined, allowing for some oversizing.

$$A^{sens}(design) = \frac{Q^{sens}_{space}(design) + 3.413 \times P_{term.box}(design)}{1.08 \times \left(T^{DB}_{space} - T^{DB}_{supply}\right)} \times (1 + OSF)$$

$$A^{lat}(design) = \frac{Q^{lat}_{space}(design)}{4.5 \times \left(HR_{space} - HR_{supply}\right) \times \left(1061 + 0.444 \times T^{DB}_{space}\right)} \times (1 + OSF)$$

In these equations A represents supply air flow in cfm/sqft, and Q represents thermal load in Btu/hr/sqft, P represents power in W/sqft, T^{DB} are dry bulb temperatures, HR are humidity ratios, and OSF is a non-dimensional oversizing factor set equal in most cases to 10%. The terminal box power will be zero at design conditions for parallel fan boxes and valve boxes. The LBNL thermal load data does not split total thermal load into sensible and latent components. To simplify, the sensible load is in all cases set equal to the LBNL total load. The design air flow A(design) is therefore set equal to A^{sens}(design).

Off-design air flow rate in VAV systems for a given temperature group depends on the sensible load, but may be limited to a fixed minimum, for instance 25% to 50% of design flow. An air flow ratio AFR is determined for each condition such that air flow is equal to design air flow times AFR:

$$AFR\left(T^{DB}_{OA}\right) = max\left(AFR_{min}, \frac{Q^{sens}_{space}\left(T^{DB}_{OA}\right) + 3.413 \times P_{term.box}}{Q^{sens}_{space}(design) + 3.413 \times P_{term.box}(design)} \right)$$

The value $Q^{Sens}_{Space}(T^{DB}_{OA})$ is the building thermal load derived from the LBNL data for the outdoor drybulb temperature T^{DB}_{OA}. Note that for fans with cycling operation, the factor AFR is the on-time fraction.

In the following circumstance, airflow ratio will depend on the heating load rather than the cooling load. This occurs if all of the following conditions are met: (1) the heating load is larger than the cooling load, (2) the central system air handling unit is used for heating, and (3) the system's preheat coil (rather than reheat coils) are used for heating. In this case airflow depends on heating load:

$$AFR\left(T^{DB}_{OA}\right) = max\left(AFR_{min}, \frac{Q^{heat}_{space}}{A(design) \times 1.08 \times \left(T^{DB}_{heating} - T^{DB}_{space}\right)} \right)$$

where $T^{DB}_{heating}$ is the design supply temperature in heating mode and Q^{heat}_{space} is the heating load in Btu/hr-sqft.

Fan Box Power

VAV systems may have a parallel or series fan box. Power for series fan boxes, which are always "on" during building occupied hours, is during these times equal to the design input power. Valve boxes, which have no fans, have zero input power.

For parallel fan boxes, which cycle "on" as a first stage of reheat, the value of $P_{term.box}$ will vary with conditions. A simplified approach to estimating percentage "on" time of these fan boxes, PBF, is shown in Figure A3-1 below. A preliminary AFR' is calculated, which ignores the fan box power contribution. The parallel box factor PBF is assumed to be 100% when AFR' is equal to $AFR_{PBF=100\%}$. At design conditions, PBF is assumed to be equal to zero. Between these extremes the relationship is linear.

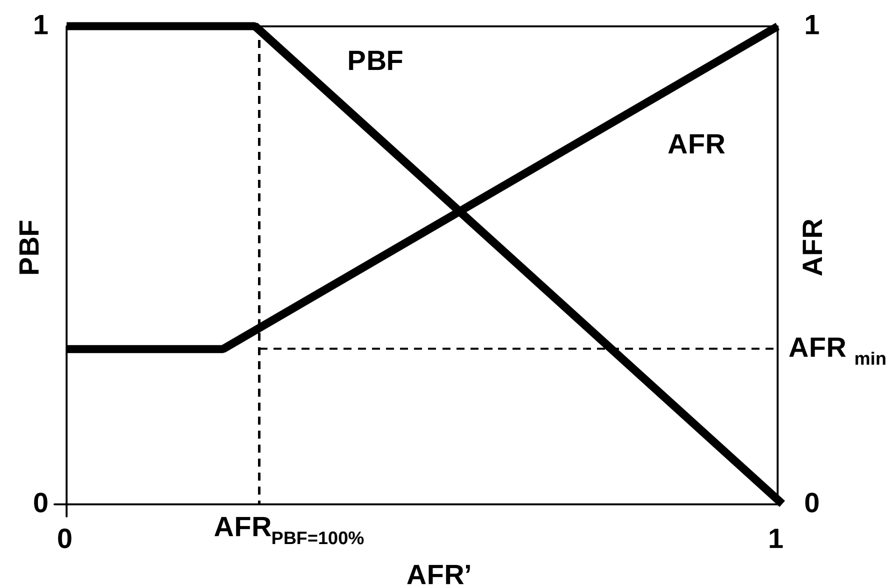


Figure A3-1: Parallel Fan Box Percentage "on" Time

Supply and Return Fan Power

Power input for the central supply and return fans are considered together as central fan power equal to the sum of power associated with both fans. Differences in fan flow rates, efficiency, and part load behavior are not addressed directly, but the fan power is divided between supply and return fans according to the supply fan power ratio SFPR, which represents the portion of central fan power associated with the supply fan. Design power for central fans is calculated as follows:

$$DLI_{fan} = \frac{0.118 \times A(design) \times DP_{fan}}{EFF_{fan}}$$

where DLI_{fan} is the fan design load in Watts/sqft, A(design) is the design air flow in cfm/sqft, DP_{fan} is the pressure rise of supply and return fans in inwc and EFF_{fan} is the composite efficiency of the central fan system including motor and drive losses.

Off-design fan power depends on the percent of design flow. Typical part load curves for different fan types and flow modulation strategies are discussed in Reference 20. A modified power law relationship between flow and power is assumed. This relationship, defined by the zero flow power ratio ZFPR and the exponent n is illustrated in Figure A3-2 below.

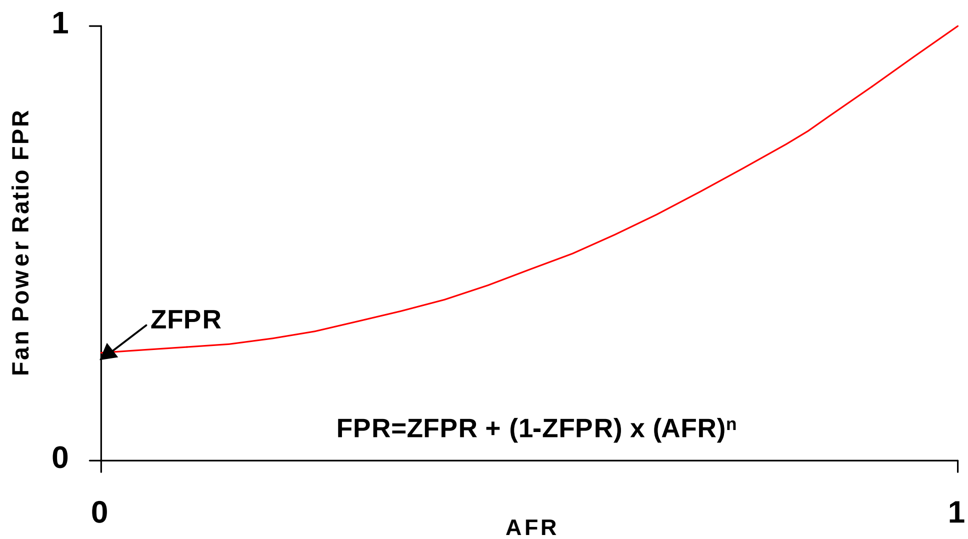


Figure A3-2: Fan Power Curve

Fan input power is equal to FPR times design condition input power.

Note that for cycling operation "ZFPR" is zero and "n" is zero.

Reheat and Local Heat

Once the supply airflow rates are determined, the reheat load can be calculated. Reheat loads consist of two components. The first is compensation for overcooling by the air-handling unit. The second is supply of heating load, if reheat coils are used to supply perimeter heating.

$$Q_{RH} = Q_{RH}^{overcool} + Q_{RH}^{heating}$$

$$Q_{RH}^{overcool} = AFR \times A(design) \times 1.08 \times \left(T_{space}^{DB} - T_{supply}^{DB}\right) - Q_{space}^{sens} - 3.414 \times P_{term.box}$$

$$Q_{RH}^{heating} = Q_{space}^{heat}$$

If local heating units (such as baseboard heating) are used to supply the perimeter heating load, the second contribution to reheat is zero.[9]

Unoccupied Operation

During unoccupied times, the air-handling unit may be cycled to satisfy setback temperatures. It is assumed that the unit will be operated with a fixed reduced airflow rate during these times, and that terminal box fans will not operate. The percentage "on" time required to satisfy the load is calculated. Cycling may be required for cooling or for heating. Cycling will be for heating if all of the following three conditions are met.

- Outdoor temperature is lower than the setback heating temperature.
- The heating load is larger than the cooling load.
- The air-handling unit rather than local heating units supply perimeter heating.

If these conditions are not met, cycling will be determined based on the cooling load. For unoccupied cycling for cooling the reheat is inactive. The system capacities CAP in Btu/hr-sqft for unoccupied operation for heating and cooling are calculated as follows.

$$CAP_{setback}^{heating} = A(design) \times AFR_{unoccupied} \times 1.08 \times \left(T_{\text{supply},unoccupied,heating}^{DB} - T_{space,unoccupied,heating}^{DB}\right)$$

$$CAP_{setback}^{heating} = A(design) \times AFR_{unoccupied} \times 1.08 \times \left(T_{\text{supply},unoccupied,heating}^{DB} - T_{\text{supply},unoccupied,cooling}^{DB}\right)$$

The supply air temperature will typically be 120ºF for heating. For cooling, the supply temperature used for occupied operation is used. The "on" time ratio OTR for cycling operation is calculated as follows.

- If the unit is being cycled for heating:

$$OTR = Q_{space}^{heat} \Big/ CAP_{setback}^{heating}$$

- Otherwise:

$$OTR = Q_{space}^{sens} \Big/ CAP_{setback}^{cooling}$$

Fan power for cycling operation is simply the fan power for the reduced flow rate times OTR.

Mixing

[9] Note that reheat, which is necessary due to diversity of cooling loads, is not included in the model. The calculation assumes that the ratio of sensible cooling load to design sensible cooling load does not vary in the space.

Operation of mixing dampers takes into consideration the minimum outdoor air percentage[10] and use of outdoor air for cooling when possible (economizing). The desired mixing temperature is equal to the coil discharge temperature. Typical fan-coil arrangement is assumed to be draw-through. Hence, the cooling coil discharge temperature is slightly lower than the supply temperature to account for the supply fan heat:

$$T_{coil}^{DB} = T_{supply}^{DB} - \frac{(3.413 \times P_{AHU})}{1.08 \times AFR \times A(design)}$$

The mixed air temperature is determined as follows.

- If the unit does not economize, if the return temperature is less than the outdoor air temperature[11], or if mix temperature would be less than coil temperature at the minimum outdoor air setting, then the mix ratio is equal to the minimum percentage outdoor air.
- If the unit does economize and the outdoor temperature is between the supply temperature and the return temperature, the dampers will deliver 100% outdoor air, and the mix temperature is equal to outdoor temperature.
- If the unit does economize and the outdoor air temperature is less than the coil temperature, the mix temperature will be equal to supply temperature, except if this would result in delivery of less than the minimum outside air quantity. In case of the latter, the minimum air quantity is delivered.

If the minimum outdoor air quantity is being delivered, the mix temperature is calculated as

$$T_{mix}^{DB} = T_{OA}^{DB} \times OAR_{min} + T_{Retrun}^{DB} \times (1 - OAR_{min})$$

If the dampers can mix to obtain the desired supply temperature, the outdoor air ratio is calculated as

$$OAR = \frac{\left(T_{coil}^{DB} - T_{return}^{DB}\right)}{\left(T_{outdoor}^{DB} - T_{return}^{DB}\right)}$$

Operation of the mixing function is illustrated in Figure A3-3 below for a system with a 52.5°F coil discharge temperature, a 75°F return temperature, and a 30% minimum outdoor air ratio. This chart is for illustrative purposes only.

[10] The minimum outdoor air is calculated as a percentage of delivered air rather than as a percentage of design air flow. The added complexity of increasing minimum outdoor air percentage as total airflow is reduced (in order to maintain constant outdoor air quantity) is not built into the model.

[11] Note that economizer control is assumed to be based on dry bulb temperature rather than enthalpy, in order to simplify the analysis.

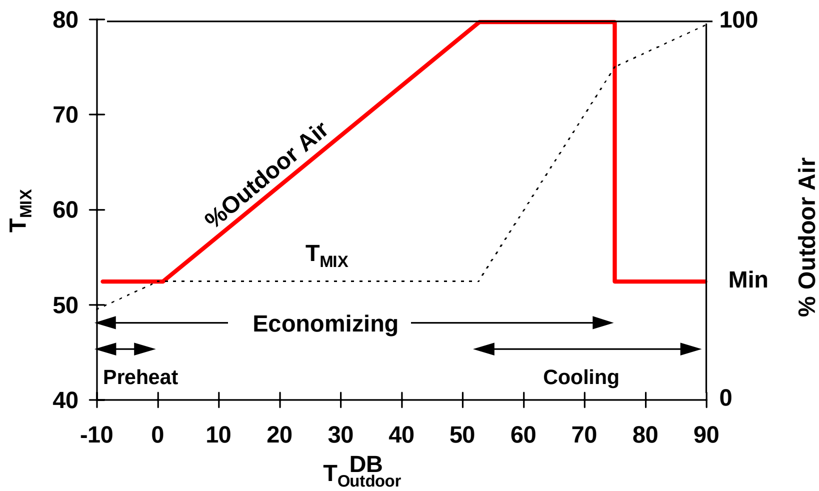


Figure A3-3: Air Handling Unit Air Mixing

The mixed-air humidity ratio H_{mix} is calculated as follows.

$HR_{mix} = HR_{OA} \times OAR + HR_{Return}\,(1\text{-}OAR)$

During building occupied periods, the return temperature is equal to the design space temperature plus added heat associated with the return fan:

$$T^{DB}_{return} = T^{DB}_{spaced} + \frac{(1 - SFPR) \times FPR \times DLI_{fan} \times 3.413}{AFR \times A(design) \times 1.08}$$

During unoccupied periods, the return temperature depends on whether the heating load or cooling load is larger. If the cooling load is larger, the average return temperature is equal to the cooling setback setpoint. If the heating load is larger, the average return temperature is equal to the heating setback setpoint.

Cooling Coil Load

The cooling coil load has sensible and latent portions. These loads are calculated as follows[12].

$$Q_{COOL} = Q^{sensible}_{COOL} + Q^{latent}_{COOL}$$

$$Q^{sensible}_{COOL} = AFR \times A(design) \times 1.08 \times \left(T^{DB}_{mix} - T^{DB}_{coil}\right)$$

$$Q^{latent}_{COOL} = AFR \times A(design) \times 4.5 \times \left[HR_{mix} \times \left(1061 + 0.444 \times T^{DB}_{mix}\right) - HR_{coil} \times \left(1061 + 0.444 \times T^{DB}_{coil}\right)\right]$$

[12] Although the coil temperature is less than the air-handling unit supply temperature to account for fan heat, the humidity ratio of the coil discharge is not adjusted downward in the same fashion. In actual practice the coil is controlled based on coil discharge rather than fan discharge. The downward adjustment of coil temperature is done to insure that the sensible heat gain represented by the fan power is taken into account.

The coil discharge humidity ratio HR_{coil} is either equal to the mixed-air humidity ratio HR_{mix} (when the coil discharge temperature is above the mixed-air dewpoint), or it is calculated assuming saturated conditions. If either cooling coil load component is non-positive, it is set equal to zero.

If the unit is cycling to maintain an unoccupied setback temperature, the loads are also multiplied by the unit on-time ratio, OTR.

Preheat Load

If there is need for preheating, a preheat coil load is calculated:

$$Q_{PREHEAT} = AFR \times A(design) \times 1.08 \times \left(T_{coil}^{DB} - T_{mix}^{DB}\right)$$

If the unit is cycling to maintain an unoccupied setback temperature, the load is also multiplied by the unit on-time ratio, OTR.

Humidification Load

If the humidity ratio of the supply air is less than a minimum space humidity ratio, a humidification load is calculated:

$$Q_{HUMID} = AFR \times A(design) \times 4.5 \times \left[HR_{min} \times \left(1061 + 0.444 x T_{coil}^{DB}\right) - HR_{coil} \times \left(1061 + 0.444 \times T_{coil}^{DB}\right)\right]$$

Humidification is assumed to be inoperative during unoccupied times.

Pumps: Chilled-Water, Condenser-Water, Heating-Water

Chilled water pumping is complicated by (1) the used of primary and secondary pumps on many large chilled water systems and (2) the strategy used to respond to reductions in the amount of chilled water flow required by the system's cooling coils.

Within central building systems, there are two basic ways for distributing the chilled water from the chiller to the building valve boxes. One design is to have one set of pumps handle the entire system, typically referred to as the single pump system. The second is to have two sets of pumps; one set to pump the water within the chiller loop (referred to as the primary loop), and the second set to distribute the cooled water to the building system (referred to as the secondary loop). The latter of the two systems has inherent energy usage advantages if the secondary loop also has a variable speed pump, since secondary pump flow can be reduced to meet building demand without reducing water flow in the evaporator.

Chilled water pressure drops are divided into contributions from (1) the chiller, (2) the distribution piping, and (3) minimum valve and coil pressure drop. Only contribution (2) will vary — the distribution piping pressure drop is assumed proportional to the square of the chilled water flow. For single pump systems the pump flow rate is fixed, so power ratio PR_{CHW} (fraction of design input power) is calculated:

$$PR_{CHW} = \frac{\Delta P_{chiller} + \Delta P_{piping} + \Delta P_{coil}}{\Delta P_{chiller} + \Delta P_{piping}(design) + \Delta P_{coil}}$$

$$\Delta P_{piping} = \Delta P_{piping}(design) \times (WFR)^2$$

WFR is the system water flow ratio, which is proportional to coil loads. For primary/secondary systems the secondary pump flow is reduced. Hence chilled water pumping system power ratio is calculated:

$$PR_{CHW} = \frac{\Delta P_{chiller} + WFR \times (\Delta P_{piping} + \Delta P_{coil})}{\Delta P_{chiller} + \Delta P_{piping}(design) + \Delta P_{coil}}$$

The power ratio is zero when there is no cooling load.

Condenser water pumps circulate condenser water through the chiller condenser and to the cooling tower. Most operate with constant water flow rate, hence pumping power is constant. EFLH depends only on hours of operation, i.e. when there is a cooling load.

Heating water pump operation is similar to that of chilled water pumps. However, the importance of variable-volume heating water pumps is not sufficient enough to warrant separate consideration. Heating water pumps are modeled as constant-power during times of operation. This includes times when reheat is required, or in buildings which use heating water for heating of service water and other loads not associated with space conditioning.

Cooling Tower Fan

The operation of a cooling tower is interdependent on the operation of the chiller it serves. Condenser water temperature returned to the chiller can be varied depending on cooling tower fan speed. Traditionally, cooling towers were operated at full speed during all times of chiller operation. This was because the chiller performance degradation resulting from an increase in condenser water temperature would offset any gain associated with reduced airflow in the cooling tower. However, today’s chillers have significantly better efficiencies, at design conditions and at part load. The increase in chiller power is not as great. Hence, the cooling tower model allows for variable airflow rate.

The delivered condenser water temperature depends on the cooling load, the ambient wet bulb temperature, and the tower airflow ratio AFR_{tower}. The difference between condenser water temperature and the ambient wet bulb temperature is assumed to vary linearly from zero airflow to design airflow. Curve fits of the ΔT vs. AFR_{tower}

relationship have been determined for a typical cooling tower for both full and 10% load conditions. The variation of ΔT with the load at a fixed airflow is assumed to be linear for the analysis. The fan power variation is modeled with a power law and characterized by the exponent N_{tower}. These relationships are illustrated in Figure A3-4 below.

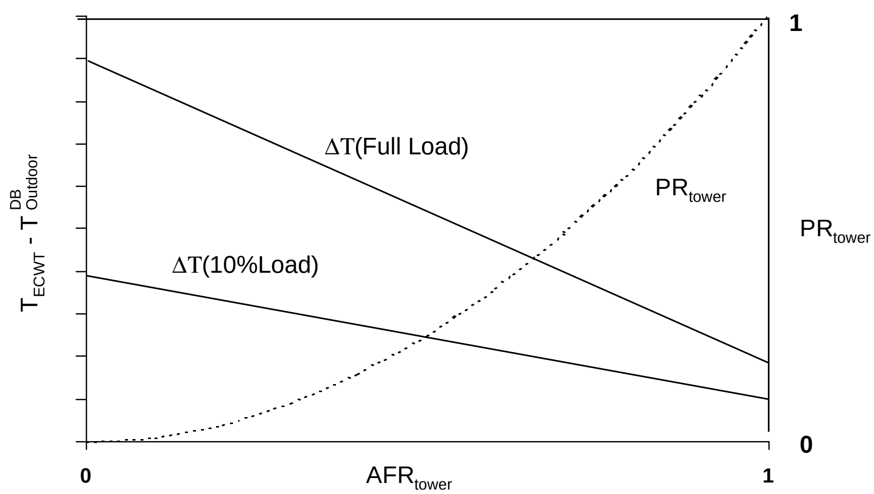


Figure A3-4: Cooling Tower Fan Operation

Control of the cooling tower fan is based on a minimum condenser water temperature $T_{ECWT,\ min}$. The airflow rate is set to the value required to achieve this temperature. When outdoor wet bulb temperature is high enough, the airflow ratio will be equal to one, and the condenser water temperature will be higher than $T_{ECWT,min}$. Likewise, when the wet bulb temperature is low enough, airflow will be zero and condenser temperature will be lower than $T_{ECWT,\ min}$.

Condenser Fans

There are two types of condenser fans under consideration in this study: those used for air-cooled chillers and packaged units which are cycled based on condenser pressure, and those in individual AC units, which operate continuously during compressor operation.

For condenser fans in individual AC units, the EFLH for the fan is equal to the on time of the refrigerant compressor. For the purposes of this study, the compressor percent "on" time is equal to the cooling load divided by the unit design capacity. This will be an overestimate of "on" time because of the increase in capacity for outdoor temperatures lower than the design temperature.

Condensers in most packaged units of greater than 5-ton size and for all air-cooled chillers have more than one fan. The fans are staged and cycled to maintain condenser pressure within a desired range. The actual cooling capacity will vary less with outdoor temperature as for the individual AC units discussed above. However, the EFLH for the condenser fans will be less than compressor "on" time. The compressor "on" time is multiplied by a power reduction factor PRF representing reduction in required condenser air flow and fan power when the outdoor air temperature is lower than the design outdoor air temperature. A 120ºF condenser temperature and a 100ºF equipment design outdoor temperature are assumed. Air flow and condenser fan power are inversely proportional to the actual condenser-to-outdoor temperature difference:

$$PRF = \frac{20^\circ F}{\left(120^\circ F - T_{outdoor}^{DB}\right)}$$

When the outdoor temperature exceeds 100ºF, PRF is not allowed to exceed 1.

Appendix 4: Background Segmentation Data

The data in this appendix, based on the CBECS95 survey, were used as the basis for the study's segmentation calculations.

Table A4-1: Heated, Cooled, and Total Floorspace

Source: Allan Swenson Fax 10/8/97 (Reference 16)

	Heated Floorspace (million square feet)	Cooled Floorspace	Total Floorspace (million square feet)
Northeast	9,919	5,936	11,883
New England	2,697	1,432	3,140
Middle Atlantic	7,222	4,504	8,743
Midwest	12,382	7,997	14,323
East North Central	8,219	5,032	9,655
West North Central	4,163	2,965	4,668
South	16,667	14,716	20,830
South Atlantic	7,621	6,776	9,475
East South Central	3,953	3,292	4,917
West South Central	5,093	4,648	6,438
West	9,096	7,352	11,736
Mountain	3,272	2,574	3,855
Pacific	5,824	4,778	7,881
Totals	**48,064**	**36,001**	**58,772**

Table A4-2: Cooled Floor Areas: Raw Data
Sources: 1. Alan Swenson Fax, 10/14/97, Table 4 (Reference 17)
2. Alan Swenson Fax, 10/16/97, Table 11 (Reference 18)

	Building/System Breakdown			Disaggregation for Central (Source 1)		
Building Type	**System Type**	**Cooled Floorspace (million sqft)**	**Source**	**FCU (million sqft)**	**VAV (million sqft)**	**Ducted (million sqft)**
Education	Residential Type	542	2			
	Heat Pump	481	2			
	Individual AC	1090	1			
	Central	1304	2	427	506	1112
	Packaged	1984	2			
Food Sales	Residential Type	149	2			
	Heat Pump		2			
	Individual Ac		1			
	Central		2			
	Packaged	312	2			
Food Service	Residential Type	299	2			
	Heat Pump		2			
	Individual Ac	181	1			
	Central		2			
	Packaged	724	2			
Health Care	Residential Type	547	2			
	Heat Pump	300	2			
	Individual Ac	627	1			
	Central	1288	2	569	906	1236
	Packaged	1221	2			
Lodging	Residential Type	397	2			
	Heat Pump	721	2			
	Individual Ac	1389	1			
	Central	781	2	411	316	626
	Packaged	1101	2			
Mercantile and Service	Residential Type	1206	2			
	Heat Pump	936	2			
	Individual Ac	856	1			
	Central	1190	2		558	1120
	Packaged	5330	2			
Office	Residential Type	1478	2			
	Heat Pump	2034	2			
	Individual Ac	924	1			
	Central	3382	2	489	2177	3191
	Packaged	5178	2			
Public Assembly, Public Order and Safety Religious Worship	Residential Type	1267	2			
	Heat Pump	634	2			
	Individual Ac	1105	1			
	Central	1141	2		575	1068
	Packaged	2428	2			
Warehouse/Storage	Residential Type	417	2			
	Heat Pump	216	2			
	Individual Ac	324	1			
	Central	90	2			89
	Packaged	1071	2			

Table A4-3: Cooling Equipment
Source: CBECS95 Table BC-36

			Cooling Equipment (more than one may apply)							
Principal Building Activity	Total Floorspace of All Buildings	Total Floorspace of all Cooled Buildings	Residential-Type Central Air Conditioners	Heat Pumps	Individual Air Conditioners	District Chilled Water	Central Chillers	Packaged Air Conditioning Units	Swamp Coolers	Other
Education	7,740	6,741	865	615	2,869	653	1,715	2,942	222	Q
Food Sales	642	612	173	Q	Q	Q	Q	362	Q	Q
Food Service	1,353	1,310	381	Q	247	Q	Q	815	Q	Q
Health Care	2,333	2,323	579	327	749	403	1,370	1,291	Q	Q
Lodging	3,618	3,193	473	827	1,629	Q	873	1,348	354	Q
Mercantile and Service	12,728	11,086	1,835	1,164	1,761	Q	1,389	6,762	523	Q
Office	10,478	10,360	1,663	2,229	1,179	568	3,683	5,847	257	301
Public Assembly	3,948	3,394	552	426	764	372	872	1,669	Q	Q
Public Order and Safety	1,271	856	193	Q	383	Q	287	420	Q	Q
Religious Worship	2,792	2,414	800	356	576	Q	Q	971	Q	Q
Warehouse and Storage	8,481	5,991	1,561	623	1,835	Q	247	3,445	Q	Q
Other	1,004	921	Q	Q	Q	Q	281	414	Q	Q
Vacant	2,384	732	Q	Q	192	Q	Q	342	Q	Q

Q: Data not reported because it is based on too few survey response

Appendix 5: Industry Expert Interview Summaries

A comprehensive series of interviews was conducted during the course of this study to test assumptions regarding HVAC parasitic energy use. This section provides a summary of these interviews. Interviewees are identified by their businesses according to the following groups.

ABBREVIATIONS

AE	Architectural & Engineering Firm
BO	Building Owner (National hotel & retail account chain engineers)
CV	Control Vendor
EPCU	Energy Service Provider Company, Performance Contractor, Utility
M	Manufacturer
TA	Trade Association

ID	Agree	Disagree	Comments
	Question 1: Energy Use, both instantaneous and annual, for Thermal Distribution and Cooling/Heating Plant auxiliaries is on the same order as that used by chillers and packaged system compressors.		
M1		√	Chillers have compressors which use a lot of energy.
AE1	√		Fan and pump energy use is on the same order of cooling.
AE2			
AE3	√		Energy use is on same order annually, but not on a peak basis.
AE4	√		
AE5		√	
EPCU1		√	Fan/pump lower by 25% based on simulation, desegregation & end use monitoring
EPCU2	√		Compressor Operation is becoming more efficient, where motor operations are stagnant.
M2		√	
CV1	√		
CV2		√	Must look at totalized horsepower of fans and pumps.
TA1			Fans usually run on a continuous duty basis.
EPCU3		√	The total for cooling towers, fans and pumps is about half that of chillers.
AE6		√	2/3 central plant, 1/3 fans and pumps.
M3		√	Depends on whether it is air cooled or evaporative. Power consumption for air cooled is much higher.
TOTAL	5	8	

Question 1 Detailed Comments:

- Rule-of-thumb total energy use in buildings (Office bldg. = 60,000 Btu/(sf-yr) [17.58 kwh/(sf-yr); Hospitals = 300,000 Btu/(sf-yr) [87.9 kwh/(sf-yr)] (AE2)
- Agrees that fan & pump energy use is on the same order (probably larger than) as cooling on an annual basis, but not on a peak basis. This is especially true in mild climates like California where the pumps & fans have much greater runtimes. On a peak basis he had calculated chiller demand at .6 kW/ton & fan demand at .3 kW/ton. (AE3)
- For small packaged equipment I work with fan power approximately equal to 25%, compressor approx. equal to 75%, e.g. per ton: 12000 BTUs/yr./10SEER = 1200 watts total. Indoor fan appox. equal to 0.4 W/cfm = 0.4x400 =160 watts. Add outdoor fan and fan equals approx. 25%. [Answer to follow up question:] Yes I am speaking with packaged systems in mind. We don't do a lot of work with larger systems. Only packaged systems 99% of what we deal with is in the 5 to 20 ton range. A lot of our jobs are service stations and low-income housing. (AE4)
- A cooling tower is about 0.1kW/Ton and so is fan energy. Pumps are a little less. But chillers still are about 0.6 kW/ton. So the total for cooling towers, fans, and pumps is about half that of chillers. (EPCU3)

	Question 2: The percentage of energy used by fans and pumps in commercial buildings is increasing.		
ID	Agree	Disagree	Reason
BO1		√	Energy efficiency of split case compressors causes them to use less energy.
M1	√		Due to trend towards IAQ and a reduced duct area.
AE1	√		Due to IAQ and mandate increase in filter efficiency. No fan or pump improvements recently.
AE2	√		Due to increase of fan powered boxes and the typical electric terminal reheat which is used.
AE3	√		Due to increase of fan powered boxes an to better chiller efficiency. Fan energy efficiency is difficult to regulate.
AE4	√		Due to better chiller/compressor efficiency, better filtration which increases system static pressure, and little to no improvement of fan or pump hardware used in installation.
EPCU1		√	Due to reduced duct area.
EPCU2	√		Due to better chiller efficiency and the lack of efficiency improvements in fan/pump hardware.
M2		√	Because of better chiller/compressor efficiency and better filtration which increased the static pressure.
CV1	√		Due to better chiller efficiency and the lack of efficiency improvements in fan/pump hardware
CV2		√	If the building is retrofitted then the usage will go down.
TA1	√		More Buildings have more fans which are using more energy.
EPCU3	√		Chiller efficiency is improving.
AE6	√		Due to better chiller/ compressor efficiency and better filtration which increases static pressure
M3		√	Lower fan and pump requirements are becoming more common.
TOTAL	10	5	

Question 2 Detailed Comments:

- Agree. Agrees that energy use by fans & pumps is increasing. "Rapid system improvements have developed in chiller efficiency while there has been little or no improvement in pump or fan efficiency (over the last 5 years). Also, changes in ASHRAE 62-89 have demanded better IAQ which mandates increased filter efficiency." (AE1)
- Agreed that fan & pump energy use has been increasing, especially over the last 5 - 8 years, mostly due to the increased use of fan-powered boxes. Also noted that typically electric terminal reheat is used in the fan box due to cheaper first cost) Straight VAV doesn't work well in office buildings, so the trend has been to use more fan-powered boxes to get back to a "constant volume" type of operation (AE2)
- Agrees that energy use by fans & pumps is increasing, due mostly to better chiller efficiency (driven by standards & codes) Also, fan energy efficiency is very difficult if not impossible to regulate due to the system effects and greater variety of operating environments. Also, said that there is definitely a higher incidence of fan-powered boxes as of the last 5 years. (AE3)
- Filtration: Also agreed that better filtration is being used. Office buildings that used to use 30% filters are moving to 65-85% ones. Applications that used to use low-efficiency furnace filters are currently using 30% filters. Although this does not necessarily mean significantly higher static pressure is being seen. (AE3)
- Duct sizes: He thinks duct sizes are getting bigger in buildings due to noise concerns, although this upsizing is not especially cost-effective and can actually increase noise due to the use of tighter transitions which increase turbulence. Some people size ducts the same for VAV systems as they did for CV systems which can't be right. (AE3)
- Agree. Better chiller/compressor efficiency, Use of more and/or better filtration which increases system static pressure, little or no efficiency improvement in fan and pump hardware used for most installations. [Answers to follow-up questions: What is the efficiency of indoor blowers typically used in small package units? What changes are occurring in packaged unit filters: what type of filter used to be used, and what type is being used now?:] Indoor blowers are 400 w/1000 cfm, roughly and I vaguely remember condenser fans being 200W/1000 cfm. Condenser fans are typically propeller fans and do not require any ductwork. In regard to filters, we've used up to 4 inch deep filters (the thicker the filter the more efficient the system), but I think that 1 inch deep filters are pretty standard. There's a Carrier unit that has a 2 inch deep filter so maybe there is a trend toward more efficient filters. (AE4)
- Disagreed: Large heat exchangers with a lower fan and or pump requirement are becoming more common in evaporative systems. Energy usage due to air cooled exchangers is increasing below 300 ton size.

	Question 3: There is more room for improvement in saving fan energy with packaged units than with central station air handlers.		
	Agree	Disagree	Reason
AE1		√	Trend towards high efficiency motors for all fans and pumps. Energy loss related to packaged units is minimal
AE3	√		Room for improvement due to the tighter packaging and type of fan used.
AE4	√		Variable speed drives not used as frequently on smaller equipment.
AE5	√		Smaller packaged units use small inefficient fans. Retrofit with one main central system ad scatter small fans about the building
EPCU1	√		Due to less efficient smaller fans presently used , tighter packaging and greater air leakage potential.
AE6	√		Due to less efficient smaller fans presently used, tighter packaging and greater air leakage potential.
EPCU2	√		Due to less efficient smaller fans and greater air leakage potential.
CV1	√		Due to less efficient fans that are presently used.
TA1		√	Fan performance doesn't' lend its to that sort of evaluation.
M2		√	Typically less efficient fans are used in packaging units
M3		√	More flexibility to adjust sizing of evaporative equipment to reduce use with chiller/cooling tower systems.
TOTAL	7	4	

Question 3 Detailed Comments:

- Agreed that there is somewhat greater room for improving packaged equipment efficiency, due to the tighter packaging and the type of fan used. FC fans tend to be used at air volumes < 20,000 CFM, since they are quieter and AF's are not available in this size range. (AE3)

	Question 4: What are the typical paths for selection of thermal distribution and plant auxiliary systems and equipment?
M1	Engineer will make the final decision.
AE1	60% Engineer, 10% Design-Build, 30% Owner
AE2	Either the engineer or the builder will select the equipment.
AE3	Builder or initially the engineer and the contractor makes the final selection
AE4	40% Engineer/A&E, 40% Design/Build Firm, 20% Owner
AE5	80% Plan-Spec, 15% Design Build, 5% Owner driven
AE6	0% Plan Spec, 10% Design Build, 0% Owner driven, 90% Other (Engineer selected)
EPCU1	70% Plan-Spec, 25% Design Build, 5% Owner driven
EPCU2	30% Plan-Spec, 50% Design Build, 15% Owner driven, 5% Other
CV2	Engineer makes the initial selection based on input from owner & contractor
TA1	Design Build Companies
EPCU3	Engineer provides the initial selection
CV1	45% Plan Spec, 35% Design Build, 20% Owner driven
M4	60% Engineer, 30 % Design Build, 10 % Owner driven

Question 4 Detailed Comments:

- Fan/pump selection: typically either the engineer or the builder will select the equipment. The builder will select in cases where he is trying to meet a cost target and knows he can do it with a specific product. In some rare instances "educated" owners, such as Hewlett-Packard, who have their own engineering staff will have guidelines. But most office buildings do not have equipment guidelines, just cost ones. (AE2)
- RGV specifies only high efficiency motors, rarely does an owner provide efficiency guidelines. (AE1)
- Owners seldom give direction as to equipment type, they are more interested in low cost. Two most common selection paths are Design-Build & Plan-Spec. In the Design-Build route, first the decision is made to go with a packaged or a built-up system, then the specific product is chosen based on cost quotes. In the Plan-Spec route, the engineer makes an initial selection which then goes out to bid, and the contractor ultimately selects the equipment provided it is to specification (AE3)
- Usually the engineer provides initial selection based on first cost and operating cost guidelines requested by the owner and the contractor and A&E work together on the final selection. (CV2)

	Question 5: Who has the key role in driving equipment selection decisions toward efficiency?
M1	A&E's and Engineers
AE1	A&E's and Engineers
AE2	A&E's and Engineers within the Owner's Constraints
AE3	A&E's and Engineers within the Owner's Constraints
AE4	Government Efficiency Regulation (NEACA)
AE5	A&E's and Engineers
AE6	A&E's and Engineers
EPCU1	A&E's and Engineers
EPCU2	A&E's and Engineers
CV2	Consulting Engineers
TA1	Building Owners
EPCU3	Standards
CV1	A&E's and Engineers
M3	Building Owners

Question 5 Detailed Comments:

- The engineer has the KEY role in driving equipment selection towards higher efficiency. But it typically comes down to whether the owner wants a Cadillac or a Chevy, and the engineer will choose the most efficient product that is within the owners cost constraints (AE2)
- A builder will typically look mostly at first cost when selecting equipment (AE2)
- Roles: The engineer makes the actual equipment selection but is usually influenced by the priorities of the owner for the specific job (i.e. "energy efficient" vs. "low cost" project) (AE3)
- Standards play the key role in driving equipment selection decisions toward efficiency. Especially California standards. (EPCU3)
- Enforcement may become dominant with new ASHRAE SSPC90
- "The engineers set up a base system which is affordable by owner. Rebate program attempts to beat this efficiency while reducing for owner cost (first)" (AE5)

	Question 6: How are IAQ (Indoor Air Quality) concerns affecting fan energy usage?
BO1	Requirements make it harder for building owners to meet fresh air demands. Increase in fresh air, increase in fan size. Insignificant pressure drop due to duct cleaning. No inc. in fan energy due to better humidity control
M1	They are important. Increase requirement will help fan sales. Potential to increase fan efficiency through the use of efficient motors and drives.
AE1	Filter pressure drops are inc.. Negligible pressure drop due to duct cleaning
AE2	Concerns are not affecting fan usage
AE3	Concerns are affecting energy use through higher min. air volumes, use of reheats on VAV boxes, use of fan powered boxes.
AE4	Filter pressure drops are inc.. There are pressure drop due to duct cleaning. Increased fresh air increases the fan size.
AE6	Air movement and fan energy are increasing, more fans are on projects. Filter pressure drops are increasing, increased fresh air increases the fan size, and systems with better humidity control use more fan energy.
EPCU1	OA% increasing = Higher heating/cooling energy. Filter pressure drops are increasing and more frequent duct cleaning reduces this pressure drop. Humidity controls adds heating/cooling energy requirements but doesn't increase fan energy appreciably
EPCU2	Use of more fans with IAQ. Fresh air increases the fan size, more duct cleaning reduces the pressure drop, and systems with better humidity control uses more fan energy.
M2	Believes that increased fresh air more often may increase the time in which the fans run. Duct Cleaning reduces the pressure drop, but not greatly.
CV2	IAQ concerns are increasing fan and pump energy use a little bit. There are pressure drop due to duct cleaning. Increased fresh air increases the fan size. Systems with better humidity controls use more fan energy because it is a humidifier.
TA1	More fans means more energy. Negligible pressure drop due to duct cleaning. Systems with better humidity controls do not use more fan energy.
CV1	Believes that increase in fresh air will increase the fan size. Filter pressure drops are increasing, more frequent duct cleaning does reduce the pressure drop, and systems with better humidity control will use more fan energy.

Question 6 Detailed Comments:

- He does not think IAQ concerns are affecting fan energy use. The overall supply CFM is based on heat gain or loss in office buildings, and air change requirements in hospitals. (AE3)
- IAQ: He says IAQ concerns are very much affecting energy use due to: 1. higher minimum air volumes, 2. use of reheat on VAV boxes, 3. use of fan-powered boxes. He said that increased fresh air requirements does not usually increase fan sizes unless you have a dedicated OA unit. Does not think filter pressure drops are increasing substantially. (e.g. pressure drop on a 65% filter ranges from .25" when clean to .75" at change out, whereas 30% filters also start around .25" clean, but need to get changed out at lower static pressures) He also said he believes filters are being changed more often nowadays, than in the past. (AE3)
- Definite increase in fan energy use. Increased use of fans. Fewer fans allowed to remain broken (e.g. bathroom exhaust). Government efficiency regulations are sorely needed for fans. It is not a level playing field at present. (Answers to Follow-Up Questions:) As far as it not being a level playing field, I'm mainly referring to stand-alone fans: exhaust fans, make-up air, fan coils. Efficiencies may vary from manufacturer to manufacturer and sometimes significantly. The designer also has an effect on how much power the fan uses. (AE4)

Q: Does increased fresh air also increase fan sizes? Not directly, but can indirectly in the case where the AHU size needs to be increased due to the need to increase the cooling coil to handle a higher OA %, and therefore a larger fan is needed in the new configuration. One large fan is more efficient than two smaller fans (AE1)

Question 7: How are the following trends affecting fan and pump energy use?	Increased use if Variable Speed Drives.	Increased use of Series Fan Boxes.	Increased use of smaller ducts to boost building utilization.	Greater incidence of building commissioning procedures,
BO1	S			
M1	G		G	
AE1	G		N	N
AE2	G	S	G	N
AE3	G		N	N
AE4	S		N	S
AE5				
EPCU1	G	S	S	N
EPCU2	S	S	G	N
M2	S	S	N	S
CV1	G	S	S	S
M4				
CV2	G			G
TA1	S	G	G	N
EPCU3	G			S
AE6				
M3	S			
TOTALS				
G = Greatly	8	2	4	1
S= Somewhat	6	5	2	4
N= Not at all	0	0	4	6

Question 7 Detailed Comments:

- Increased use of variable speed drives greatly affects fan energy use because as you slow down the fan you use more horsepower. He also thinks that IAQ issues greatly affect fan and pump energy use. (M2)
- He thinks that use of VSD's and increased use of fan-powered boxes have the greatest effect on fan and pump energy use. Smaller ducts are typically used only in low-temperature supply air systems, which have not caught on in the Northeast. VSD's are very strongly encouraged today, on both the cooling and heating side. Building commissioning: Agrees that there is a greater incidence of it, but does not think it is being done better. Many design engineers have never done commissioning. Most firms treat it as an additional service. Sometimes an outside "commissioning firm" is called in, which doesn't necessarily improve the situation. In general he states that building commissioning is not currently very effective. Also, many people are not fully qualified to do air balancing, and therefore do not do a thorough and effective job. (AE2)
- Trends: Ducts are getting bigger not smaller due to noise concerns. Building commissioning does not currently have much of an effect on fan & pump energy use because it is only done in about 5% of the installations. It would be helpful if it was done more, but the process is not ingrained in the industry. (AE3)
- AE6: Commissioning is still lacking
- EPCU1: Commissioning is still lacking.

	Question 8: What percent of retrofits are initiated by DSM programs or ESCO's?	Comments
AE1	10%	
AE2	50%	
AE4	10%	This is a guess
AE5	20%	
EPCU1	80%	
EPCU2	80%	
CV1	70%	
AE6	20%	

Question 8 Detailed Comments:

- He says that about 50% of retrofits are due to energy saving desires, while the rest are due to the need to replace old equipment, though these needs can often coincide. The utility/ESCO programs give the extra incentive to go ahead with the retrofit, in many cases. (AE2)

	Question 9: What percent of these include modifying fan and pump and fan systems (i.e. through new higher efficiency equipment or variable speed drives)?	Comments
AE1	15%	
AE2	50%	
AE4	80%	A Guess
AE5	80%	
EPCU1	70%	
EPCU2	10% actual implementation	90% of projects look at this option
CV1	70%	
AE6	80%	

Question 9 Detailed Comments:

- He says only 15% of retrofits include fan & pump system mods. Most often are chiller and/or lighting retrofits. For example, MASS Electric reviews the study performed by the engineering firm and usually decides to only pay for the chiller or lighting portions of the job. (AE1)
- More than half of the retrofits include VSD’s. Though he also said that in many cases, the building owner will deal directly with the chiller manufacturer, and only replace the chiller, without considering the rest of the system. (AE2)

	Question 10: When a retrofit manages to save significant amounts of fan and pump energy, what are the major factors?
BO1	It is because more efficient fans and pumps weren't specified at the time that the system was built.
M1	Efficient fans and pumps were not specified.
AE1	Efficient fans and pumps were not specified, system was not operating as designed and also, better modern control system (DDC) provide efficient control sequence.
AE2	Due to overdesigning fans and pumps.
AE3	Use of VSD.
AE4	Efficient fans and pumps were not available or specified. The original system was not designed efficiently.
AE5	The original system was not designed efficiently. The systems hasn't been operating as it was designed
EPCU1	Efficient fans and pumps were not available. The systems hasn't been operating as it was designed
EPCU2	Efficient fans and pumps were not available. The original system was not designed efficiently. The systems hasn't been operating as it was designed.
CV1	Efficient fans and pumps were not available or specified. The systems hasn't been operating as it was designed and it hasn't adapted to building operation changes.
CV2	Efficient fans and pumps were not available or specified. The original system was not designed efficiently. The systems hasn't been operating as it was designed and it hasn't adapted to building operation changes.
TA1	Efficient fans and pumps were not available when the system was first built and the system hasn't adapted to building operating changes.
EPCU3	Efficient fans and pumps were not available or specified. The original system was not designed efficiently. The systems hasn't been operating as it was designed and it hasn't adapted to building operation changes.
AE6	Efficient fans and pumps were not available or specified. The original system was not designed efficiently.
M3	System components, such as cooling towers, selected from lower operating cost due to power consumption, with higher capital cost due to larger tower size.

Question 10 Detailed Comments:

- The reason retrofits save energy is often due to overdesigned fans & pumps. For example a pump may have been selected based on 120ft of head, but the system only has 60ft, so the difference is made up in the balancing valve. This is like hitting the brake and the gas at the same time. In the case of fans, many times the fan is kept in place, but the motor is changed or a VSD is added. (AE2)
- Retrofits: Use of VSD's are the most prevalent reason why retrofits save fan and pump energy use. Physical changes to the system are very difficult to do in a retrofit situation. Also, in some instances newer VAV boxes with reduced pressure drop are used. One additional retrofit action is to remove sound traps. This reduces pressure drop and thus can actually reduce overall noise of the system due to the effect on the fan. (AE3)

Additional Questions for A&E's

1. For all commercial buildings using each of the following central systems please give the incidence of use as a rough percent.

Central System Type	New Construction	Existing Building Stock
VAV		
CAV with Reheat		
Fan Coil Units		
Multizone		
Dual Duct		
Induction Units		
Other		

- In office buildings in the northeast VAV is the most prevalent central system type. CAV with reheat is not used at all (not allowed by code, he thinks). Fan coil units are too expensive. Dual duct is also not used much in office buildings. (AE2)
- CAV & multizone systems are against the law in most states. Use of fan coil units is driven mostly by application (not used in office buildings, used mostly in hotels) The word VAV is incomplete, we need to specify a heating system type. (AE3)

2. How often are the following VAV control mechanisms used in existing buildings:
    - Inlet Guide Vanes
    - Variable-Speed Drives
    - Other

- See questionnaire for details (AE1)
- Indicates that VSDs are used the majority of the time for VAV control, although in some rare instances inlet guide vanes can be used if the owner requests them or through a value-engineering exercise. Also, present in some older buildings is a Parker system or bypass terminals, which both allow the supply fan to run constant volume but bypass primary air into the RA plenum to vary the air to the space (AE1)
- VSD's are used most often for VAV systems (AE2)
- Inlet guide vanes were used up until about 5 years ago. Now almost all VAV installations use VSD's (especially with rebates available). For pumps, VSD's are used less often, since pump systems are not typically variable volume and VSD's usually only make sense for larger size systems. (AE3)

3. How often are the following VAV control mechanisms used in new buildings:
    - Inlet Guide Vanes
    - Variable-Speed Drives
    - Other

- Inlet guide vanes are not currently used (AE2)
- Very few fan-powered boxes are used in California. Fan boxes are very popular in Texas, Georgia, the Southeast & the Northwest and in general, in places that use electric heat. Previously they were used in perimeter zones but have since expanded to the interior zones as well. There is a perception of improved comfort & IAQ. (AE3)
- Though fan boxes can be used to effectively transfer air from over-ventilated areas to under-ventilated ones, this is seldom done. This is also due to the fact that internal loads (lighting, PC's) are decreasing causing load calcs to show low air volumes and designers get nervous and specify fan boxes. (AE3)

4. For buildings with VAV systems, how often are each of the following terminal boxes used in <u>existing</u> buildings:
    - Series Fan Boxes
    - Parallel Fan Boxes
    - Valve-Only Boxes

- (4) Indicates that valve-only boxes tend to comprise as much as 60 - 70% of the terminal units in new and existing systems. Also says that use of fan-powered boxes is on the way down in favor of VSD's at the AHU which can turn down to 25%. (AE1)
- Some new systems do use controllable-pitch fans, but these are noisy and require room for sound attenuation so they are not typically used in retrofits (AE2)
- Very few fan-powered boxes are used in California. Fan boxes are very popular in Texas, Georgia, the Southeast & the Northwest and in general, in places that use electric heat. Previously they were used in perimeter zones but have since expanded to the interior zones as well. There is a perception of improved comfort & IAQ. (AE3)
- Though fan boxes can be used to effectively transfer air from over-ventilated areas to under-ventilated ones, this is seldom done. This is also due to the fact that internal loads (lighting, PC's) are decreasing causing load calcs to show low air volumes and designers get nervous and specify fan boxes. (AE3)

5. For buildings with VAV systems, how often are each of the following VAV terminal boxes used in <u>new</u> buildings:
    - Series Fan Boxes
    - Parallel Fan Boxes
    - Valve-Only Boxes

- Fan-powered boxes are used almost all the time on the perimeter, and may be used in the interior (AE2)

- Series fan boxes are used very rarely. He gave a typical distribution of 50% parallel fan boxes and 50% valve-only boxes in a VAV system. (AE2)

6. In buildings with VAV systems, what is the typical distribution of terminal box types:

Series Fan Boxes ___%Parallel Fan Boxes ___%Valve-only Boxes ___%

7. With respect to Cooling Tower Operation, is better overall performance achieved with the fan at full speed?

Is this standard operating practice?

- Cooling Towers: Agrees that better overall performance is achieved with tower at full speed, but that the fan usually operates at part load because towers are sized for design days. (AE1)
- Cooling Towers: In the past, when chillers were less efficient, it made sense to run CT's at full speed. Now the question is more complex, and is based on cooling load and wet bulb temperature. ASHRAE Std 90 is attempting to include cooling tower efficiency albeit currently at the status quo of the industry, to get it included. Also they tried to regulate away from the use of centrifugal fans in CT's since propeller fans are more efficient, but this was rejected. Centrifugal fans are used by some manufacturers because they are quieter. (AE3)

Additional Questions for Manufacturers

For the manufacturer's key product type, obtain answers to the following questions related to equipment efficiency.
Key Product Type:
(Centrifugal fan, axial fan, chilled water pump, hot water pump, cooling tower fan, cooling tower)

- Key product type for Power Line Fan Co. is aluminum centrifugal (M1)
- Parent company is Air Master Fan Co. which makes air circulators (M1)

1. How is the efficiency of this product reported (Definition, units)?

- Fans are rated at the operation point (M1)
- I don't believe efficiency is reported for any of these. (AE4)
- There are no current standards. New ASHRAE SSPC90 will introduce kW/TON, or kW in Fan/ KW transferred. (M3)

2. What is the current "efficiency" range of this product?

- 75-80% efficient at the top of the fan curve(LM1)
- Not available as best I know. (AE4)
- A very wide variation between and within product types and application conditions. (M3)

3. How does the efficiency vary with equipment size?

- Efficiency doesn't vary with size(M1)
- Not available. (AE4)
- COP improves as equipment size increases, in general (M3)

4. Is efficiency dependent more on hardware or on system design and application?

- Efficiency is very dependent on application and system design (M1)
- Hardware primarily. (AE4)

- They are very interdependent. System design affects thermal duty very much. Efficiency is decreased with more difficult duty. (M3)

5. Has there been any significant efficiency change recently?

- No, there has been no significant change in efficiency lately. (M1)
- No. (AE4)
- More low power product opinions, same or similar equipment at a lower capacity, lower power.(M3)

6. Is there significant room for improvement?

- No, there is not much room for improvement with fans. At least, that is, with fan configuration. Any efficiency improvements would come from motor or drive improvements. (M1)
- Yes. (AE4)
- Yes, typical applications are high power, low cost. (M3)

7. What constrains this improvement potential? (product cost, technical risk, market apathy)

- The technology itself constrains the improvement potential. It's a very mature industry. (M1)
- No regulation/requirements. (AE4)
- The constrains are cost/benefit, ability and inclination of evaluator to weigh power costs to us. The capital cost. (M3)

8. What research is needed to further improve equipment efficiencies?

- With R&D there's always a concern about the trade off between cost and ease of manufacturing [of systems with improvements in efficiency] (M1)
- Research to show a wide disparity in efficiencies of fans and pumps. (AE4)
- System optimization software needs to be researched (M3)

9. Is this currently research being performed? By who?

- Almost everyone is doing some sort of R&D (M1)
- Not that I'm aware of. (AE4)

- No, there is proprietary software by chiller vendors. (M3)

Additional Questions for ESCO's

1. What levels of parasitic (e.g. fans & pumps) power reductions have been achieved?

- Very small overall. (AE4)

2. What types of retrofits have contributed most to overall energy use reduction? (e.g. chiller replacement, air handler replacement, new pumps, new drives)

- Chiller replacements. Motor replacements. Variable speed drives. (AE4)
- Lighting retrofits. (EPCU3)

3. To what extent has energy use reduction relied on the use of:

new more-efficient equipment	Greatly	Somewhat	Not at all
system configuration changes	Greatly	Somewhat	Not at all
operational changes	Greatly	Somewhat	Not at all
more aggressive preventative maintenance	Greatly	Somewhat	Not at all
other:	Greatly	Somewhat	Not at all

- Energy use reduction has relied on new more efficient equipment greatly, system configurations somewhat, operational changes somewhat, and more aggressive preventative maintenance somewhat. (AE4)
- All of the above have affected it greatly. (EPCU3)

4. What are the greatest constraints on reduction of fan and pump energy use?

- Absence of government regulation. Fan and pumps have little incentive to improve efficiency. Little technical change. Technology remains driven by installed cost. (AE4)
- Existing ducting and diffusers. (EPCU3)