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Efficient Thermal Energy Distribution in Commercial Buildings

Final Report
to
California Institute for Energy Efficiency

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EXECUTIVE SUMMARY

According to California Energy Commission (CEC, 1998a), California commercial buildings use approximately 86,593 GWh or 35% of statewide electricity consumption, and about 1,945 million therms or 15% of statewide gas consumption. Space conditioning in commercial buildings accounts for approximately 18% of their electricity consumption, and 42% of their natural gas consumption. An additional 10% of commercial-building electricity consumption is used for building ventilation, i.e., fans and pumps.

This project focuses on thermal energy distribution systems in commercial buildings, namely the fans, pumps, ducts and pipes used to transport heating and cooling, as well as ventilation air. The tasks of the report include: 1) to characterize the stocks of commercial buildings, thermal distribution systems and their energy consumption in California; 2) to conduct an industry survey of thermal energy distribution design practice; 3) to perform energy analyses of several distribution-system; and 4) to preliminarily identify savings opportunities and some of the efforts required to realize those opportunities.

To optimize the use of limited resources, the energy analyses and opportunity assessments were directed at a subset of the issues and technologies deemed as important by the stock characterization and survey results. These include thermal loss reduction for rooftop-packaged systems, fan energy reduction in thermally perfect systems and thermally imperfect systems. The energy analyses performed focus on: 1) improving the performance of small rooftop package units, 2) fan energy reduction in medium-to-large buildings by use of variable-air-volume systems in new and retrofit cases, and by expanded use of hydronic systems in new construction, and 3) the impacts of duct thermal losses on fan energy in larger commercial buildings.

The study identifies significant energy-saving opportunities for space conditioning and thermal energy distribution. These include, but not limited to, 1) reducing thermal losses induced by air leakage through system components (i.e., duct, equipment), 2) decreasing thermal losses induced by heat conduction, convection, and radiation, and 3) improving equipment efficiency and system design and control strategies.

The report consists of the following sections: 1) Introduction, 2) HVAC system descriptions, 3) Stock characterization, 4) Design practice surveys, 5) Energy analyses and assessments for the improvements to thermal energy distribution systems, 6) Energy-savings opportunity assessment for statewide thermal energy distribution systems, 7) Summary and conclusions, followed by the acknowledgement, references, and appendices (a-d).

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1. INTRODUCTION

Due to the large number of options for thermal energy distribution in commercial buildings, new ideas for energy-saving opportunities continually arise. The goals of this project were to identify large energy savings opportunities in the thermal energy distribution (TED) systems used in commercial buildings in California, and to outline the research needed to achieve the identified savings. The objectives of the effort were the following: 1) identify the prevalence of each major type of thermal distribution system in existing and new commercial construction, 2) estimate the typical energy performance and load impacts of these systems, 3) use available energy analysis methods and performance data, as well as a survey of design engineers, to identify the significant factors (i.e., inadequate design tools, construction practice, operating strategies) contributing to poor distribution system performance, and 4) identify the research or technology transfer needed to achieve the identified savings potential. A team of researchers from three institutions was assembled to accomplish these objectives, including the University of California at Berkeley (UCB), Lawrence Berkeley National Laboratory (LBNL) and Humboldt State University (HSU).

For the purposes of this project, the term "thermal distribution system" refers to all equipment and pathways between the source of heating or cooling to the point of use (i.e., delivery of air to rooms, delivery of water to coils). The typical distribution media include air, water, steam, and refrigerant systems. Of these, air systems are by far the most popular, and have the largest number of variations.

Our technical approach included four tasks: 1) a characterization of the stock of thermal distribution systems in California commercial buildings based on existing literature (UCB), 2) an industry survey of design practice related to thermal distribution systems (UCB), 3) monitoring and energy analyses of selected thermal distribution systems (LBNL, HSU), and 4) an assessment of the opportunities available for saving energy and peak demand in commercial-building thermal distribution systems, and identification of appropriate research (LBNL, UCB). The technologies initially identified as having energy and peak-demand savings potential included: a) improved commissioning and maintenance practices as well as improved controls for air distribution systems, b) integration of design and energy simulation tools for large commercial buildings (for both air and water systems), c) retrofits of small commercial building distribution systems, focusing on thermal issues (i.e., similar to residential), d) more efficient fans, motors and pumps, as well as reduced frictional losses in ductwork, and e) various forms of hydronic distribution systems as replacements or supplements to air distribution systems (i.e., flexible hydronic piping for localized thermal distribution systems, radiant-panel heating and cooling with hydronic distribution, distributed water-to-air heat pumps tied to a hydronic distribution system, and reduction of pumping power by means of reducing fluid friction).

In order to determine the potential savings due to some of these strategies, we first describe the HVAC systems and characterize the thermal energy distribution systems in the California commercial building stock. Then we analyze the results of the design practice surveys and conduct energy analyses for improvement of thermal energy distribution systems. Combining the results from stock characterizations, survey, and analyses, we then determine or estimate the potential energy-savings opportunities in a separate section titled "the energy savings opportunity assessments – thermal energy distribution system.

2. HVAC SYSTEM DESCRIPTIONS

Commercial HVAC systems can be classified according to many different sets of criteria, including system components and categories. For the purposes of this study, we will consider two basic HVAC system components: the heating and/or cooling source and the thermal energy distribution system. System categories are generally defined based on the location of the heating and/or cooling sources. For the purpose of this study, they are grouped into central and local system categories.

Central systems are those which have heating and/or cooling equipment located in a single mechanical space (or in a few locations in a very large building). Heating or cooling generated by this equipment is then distributed to the occupied spaces using air or water as a heat transfer medium. Local (or distributed) systems consist of heating and/or cooling equipment, such as smaller packaged roof-top units, which are distributed throughout the building and serve different areas of the building.

Generally speaking, central systems tend to cost more than distributed systems, take up more space and require more ductwork and/or piping within the distribution system. Distributed systems tend to be less expensive and use less efficient components, though their lower inefficiency is due more to market forces than to inherent design limitations. Even though they tend to be less efficient, local systems use less distribution energy than central systems since the local systems' heating and/or cooling source is closer to the point of use. A previous LBNL study (Akbari et al. 1993) using DOE-2 simulations indicates that fan energy per unit floor area in large office buildings is four times greater for a central system than for a distributed packaged system.

2.1 CENTRAL SYSTEMS

Central systems are normally discussed in terms of the way in which they transfer heating and/or cooling energy to the occupied space (see Table 1). The advantages of All-Air systems and of All-Water systems are listed in Table 2 and Table 3, respectively. Following are descriptions of various types of central systems.

Table 1. Energy Transfer Methods

System Type	Heating and Cooling Energy Transferred by:
All-Air systems	Air through ductwork
All-Water systems	Water through piping
Air/Water systems	Combination of air through ductwork and water through piping

Table 2. All-Air System Advantages

All-Air Systems
No piping for water or condensation in occupied spaces
Centralized mechanical equipment room concentrates maintenance in unoccupied space
Economizer operation allows use of outdoor air for cooling
Good humidity control
Air filtration

Table 3. All-Water Systems Advantages

All-Water Systems
Take up less space
Better opportunity for individual control
Lower distribution energy consumption
Better control of ventilation as a separate system

2.1.1 All-Air Systems

All-Air systems include single duct constant-air-volume (CAV), single duct variable air volume (VAV), dual duct and multizone.

Single Duct CAV. Single duct CAV air systems are the most commonly found HVAC systems in commercial buildings in California. These systems deliver a fixed quantity of air to the conditioned space and maintain desired conditions by varying the temperature of the supply air. While not as prevalent, single duct constant volume systems may also have reheat coils at the individual terminal units, which are used to provide additional heating for the supply air when needed.

Single Duct Variable Air Volume. Variable air volume (VAV) systems take advantage of the fact that buildings rarely operate under the extreme conditions for which their mechanical systems were designed. Space temperature is maintained by varying the quantity of supply air, generally at a fixed temperature. Reheat coils at the individual terminal units can provide heating of the supply air when needed. In the ideal situation, fan power is proportional to the cube of the volume of airflow moving through the fan at a certain operating pressure. In VAV systems, although the static pressure of the main duct section is usually constant, together with changes in dynamic pressures, the total operating pressures in the main duct section usually would change overtime. Therefore, fan power is not exactly proportional to the cube of flow rates overtime. However, a reduction in supply air volume through the main supply fan and duct can still result in a significant reduction in fan energy consumption overtime.

Dual Duct. Rarely used in new designs, dual duct systems supply both heated and cooled air to each zone. A thermostatically controlled damper adjusts the ratio of warm and cool air to obtain the desired supply temperature for the space. While these systems provide good zone control due to the availability of simultaneous heating and cooling in separate zones, they are quite energy inefficient since excess heating and/or cooling is almost always taking place at the main plant.

Multizone. Multizone systems use separate ducts for each of the conditioned zones served by a single multi-zone system. Air is conditioned to the appropriate temperature for each zone by mixing warm and cool air at the central plant. These systems provide good zonal control at the expense of energy efficiency. The major difference between dual duct and multizone systems is where the mixing of air-streams occurs.

2.1.2 Water Systems

In water systems, chilled or hot water is delivered to each zone and distributed to the space through terminal units (fan coils or induction units) or radiant panels. Water systems generally have much lower distribution energy consumption due to the significantly lower friction losses per unit of energy transfer. Compared to all-air systems, water systems (piping vs. duct) typically require less building space, a smaller air system and little duct space to provide ventilation and/or local distribution. On the other hand, they require more maintenance in occupied space.

Two-Pipe. Two-pipe systems use a supply and return piping network to distribute chilled water to the zones. The two-pipe changeover system allows hot water to be circulated during the heating season. The system cannot supply both heating and cooling simultaneously, and when both are required, the system usually is operated in cooling mode and zonal space heaters are employed.

Four-Pipe. Four-pipe systems circulate both hot water and chilled water to terminal units (fan coils or induction units) in each zone. Although more expensive than two-pipe systems to install, they have simpler operation, and can provide both heating and cooling during the entire year.

Water-Loop Heat Pump. This system uses a two-pipe loop to distribute water to water-to-air heat pumps located in each zone. The heat pumps can either heat or cool as required by the zonal loads. The big advantage of these systems is their high efficiencies, since heat extracted from a zone, which required cooling, could be transferred through the loop to a zone, which requires heating. A boiler and a cooling tower are used to keep the water loop within the proper operating temperature range.

2.2 LOCALIZED SYSTEMS

Localized systems include packaged units (rooftop units) as well as split systems (i.e., an air handler unit with heat exchangers at or near the corresponding zone with a remotely installed outside compressor unit or heat pump).

3. STOCK CHARACTERIZATION

3.1 OVERVIEW

According to California Energy Commission (CEC, 1998a), commercial buildings in 1997 use 86,593 GWh or 35% of the total electricity consumption, and 1,945 million therms (57,000 GWh) or 15% of the total gas consumption in the state of California. For comparison, residential buildings use 73,759 GWh or about 30% of the statewide electricity consumption, and 4,810 million therms (141,000 GWh) or about 38% of the statewide gas consumption in California.

Distribution systems in this study are defined as all equipment and pathways between the source of heating, cooling, or ventilation to the point of use. Distribution systems directly account for approximately 10% of commercial building electricity consumption. Thermal distribution systems have an impact on the heating and cooling energy, which makes up an additional 18% of the building electricity consumption and 42% of the commercial building gas consumption (See Figure 1 and Figure 2). This section characterizes the distribution systems in commercial buildings in the state of California and identifies those systems that might prove to be good energy conservation resources. Sources used to develop this characterization and identification include, among others, the California Energy Commission's Energy Demand (CEC 1991a, 1991b), the DOE/EIA Commercial Building Energy Consumption Survey (EIA, 1991), and individual utility surveys.

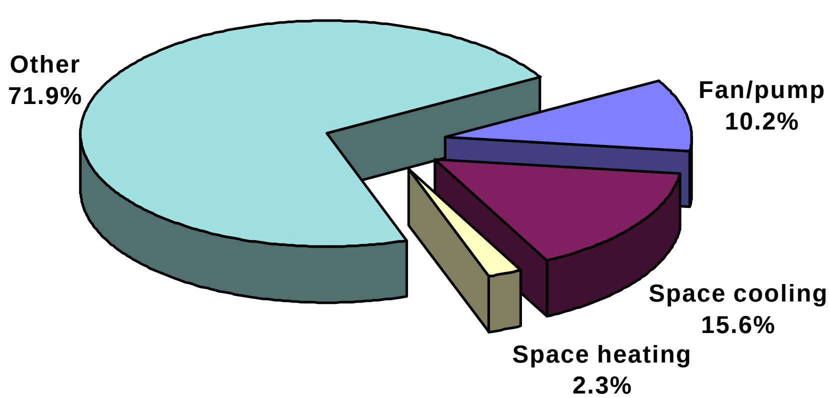

Figure 1. 1997 Commercial Building Electricity End-Use

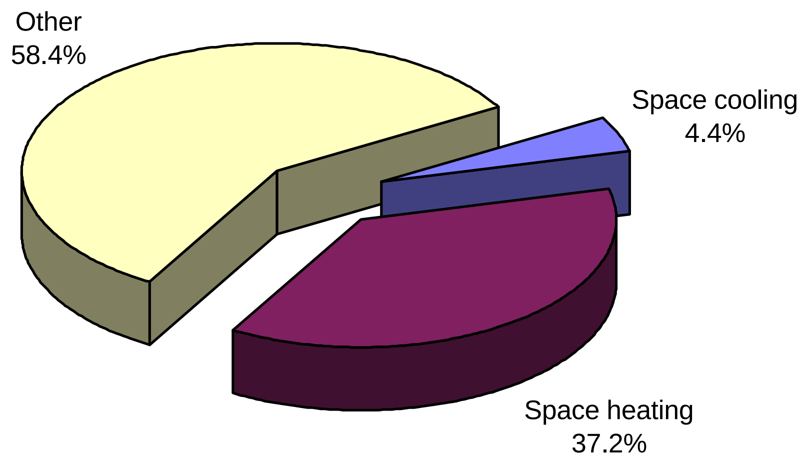

Figure 2. 1997 Commercial Building Gas consumption

3.2 BUILDING TYPES

The buildings classification system used in this report is the same as that used by the California Energy Commission (CEC) in their Commercial Sector Forecasting Model (CEC 1991b). Table 4 shows the projected floor area and annual new additions for each of the eleven CEC building categories for the year 1994, based on the CEC Energy Demand 1991-2011 (CEC 1991a). For comparison, California residential stock floor area data are provided, based on the 1990 U.S. Census (1990, number of houses) and the 1993 Residential Energy Consumption Survey (RECS) (average RECS Western region floor area, EIA/DOE 1995). Projections of residential annual additions were however not provided in these two reports.

Table 4. CEC Floor Space Projections for 1994 (in 10^6 ft^2)

Building Type	Stock (10^6 ft^2)	Annual Additions (10^6 ft^2)
Small Offices (< 30,000 ft^2)	303	9.8
Large Offices (> 30,000 ft^2)	1,058	35.6
Retail Stores	764	23.1
Restaurants	123	2.8
Food and Liquor Stores	213	6.7
Warehouses	948	32.1
Schools	419	6.1
Colleges and Trade Schools	253	3.1
Health Care	274	7.3
Hotels and Motels	290	13.5
Miscellaneous	774	21.1
Total Commercial	5,418	161.3
Total Residential (1990)6,7	14,827	N/A

The CEC floorplan projection shows an annual growth rate of approximately 3% for most commercial building types. Educational buildings are projected to have a lower growth rate (1.2% and 1.5%), while hotels are projected to grow approximately 4.7%. One point worth noting is that the CEC projections for additional floor area between 1989 and 2003 is 18%, whereas their projected increase in energy use is only 12%, which suggests that new buildings and systems should be 33% more efficient on a per square foot basis.

3.2.1 U.S. Building Stock

Although the focus of this work is on evaluating the commercial building stock in California, some useful information was found in sources that contain data on nationwide building characteristics.

The 1989 Commercial Building Characteristics (EIA 1991) report, published by the Energy Information Agency and based on the Commercial Buildings Energy Consumption Survey (CBECS) data, provides some general information about heating and cooling systems. The data are provided by census region as well as by building type. The total CBECS Western census region floor area is given as 11,620 million square feet as compared to a total of 63,184 million square feet for all US commercial buildings (Table 5). The CEC estimation of California commercial building floor area, in 1989, is 4,773 million square feet (approximately 41% of the floor area listed for the CBECS Western census region).

Table 5 shows that on the basis of building numbers, approximately 86% and 70% of commercial buildings in the Western census region have heating and cooling, respectively. On the basis of building floor area, the data show that approximately 92% and 80% of commercial buildings in the Western census region have heating and cooling, respectively. These percentages are very similar to the ones on the national level.

Approximately 97% office buildings have heating and/or cooling systems, while almost 99% of the floor areas have heating and/or cooling systems in the US. The un-conditioned building floor accounts for higher portion in retail sector (3% un-heated and 13% un-cooled).

Table 5. 1989 CBECS US and the Western Region Building System Characteristics

Building category		Building Stocks (in 1,000)				Building Floor Areas (in 10^6 ft^2)			
		US	West	Office	Retail	US	West	Office	Retail
Buildings, or Floor Areas		4,528	851	679	1,278	63,184	11,620	11,802	12,365
Heated Buildings, or Floor Areas		3,865	732	660	1,216	57,764	10,638	11,678	12,037
Cooled Buildings, or Floor Areas		3,184	576	656	929	51,761	9,319	11,636	10,803
Conditioning type	% conditioned	Percentage of Building Stocks				Percentage of Building Floor Areas			
Heated	0% (no heating)	15%	14%	3%	5%	9%	8%	1%	3%
	1-50%	14%	15%	5%	18%	15%	16%	4%	15%
	51%-99%	11%	13%	15%	15%	14%	18%	26%	16%
	100%	60%	58%	76%	62%	63%	58%	69%	66%
Cooled	0% (no cooling)	30%	32%	3%	27%	18%	20%	1%	13%
	1-50%	23%	20%	12%	32%	28%	20%	8%	31%
	51%-99%	13%	13%	22%	14%	21%	20%	38%	21%
	100%	34%	35%	63%	27%	33%	40%	53%	35%

The CBECS data for commercial buildings with cooling and heating equipment is summarized in Table 6. The table shows that one-third of the commercial buildings with *heating* in the West census region contain room space heaters, including portable heaters, hanging units heaters, heating panels, electric baseboards, wood stoves and fireplaces. On the national level, we observed a higher percentage (36%) of the buildings using space heaters. On the basis of building floor area, both fractions increased slightly (35% for the Western census region and 39% for US). In effect, when electric space heaters are used for a building's supplemental or primary heating source, the building electrical system is acting as the thermal energy distribution system for these heaters. While this work will not be examining electrical distribution efficiency, it is worth pointing out that the use of electric resistance heaters can be very inefficient with respect to source energy consumption, and better air and water distribution systems may be able to reduce the use of the device. On the basis of building floor area, approximately 73% of the commercial buildings in the Western census region have air ducts for heating while for the US commercial buildings, the percentage was slightly less than two-third. On the basis of building floor area, 78% of office buildings have air ducts for heating and 58% of retail buildings have air ducts for heating.

Table 6 also shows that 61% of the commercial buildings with *cooling* in the West census region have packaged cooling units, 76% have ducts, 21% have heat pumps, 7% have central chillers and 5% have fan-coil units as part of their cooling systems. Further examination of the data shows that the use of fan-coil units is very dependent on building size. Figure 3 shows the incidence of ducts, central chillers, and fan-coils by building size. While ducts are slightly more common in larger buildings, central chillers and fan-coils are used far more often in larger buildings. This is reflected in the floor area data, which shows that fan-coils are used in 23% of the cooled floor area in the West region.

Table 6. 1989 CBECS Western Region – Heating/Cooling Systems Characteristics*

Conditioning type	Equipment Type	Percentage of Building Stocks				Percentage of Building Floor Areas			
		US	**West**	**Office**	**Retail**	**US**	**West**	**Office**	**Retail**
Heating	Boilers	18%	14%	18%	13%	34%	27%	34%	18%
	Furnaces	42%	36%	42%	48%	27%	22%	17%	42%
	Space Heaters	36%	34%	26%	42%	39%	35%	31%	43%
	Packaged Units	22%	29%	27%	19%	27%	33%	25%	39%
	Heat Pumps	12%	17%	18%	8%	14%	20%	18%	13%
	Air Ducts	51%	58%	66%	43%	65%	73%	78%	58%
	Heating/Reheat Coils	6%	6%	10%	3%	27%	28%	44%	13%
	Fan Coils	5%	5%	3%	2%	20%	21%	22%	5%
	Baseboards/Radiators	13%	10%	13%	9%	27%	18%	30%	12%
	Other	1%	2%	1%	0%	3%	3%	3%	0%
	Total Buildings (in 1,000) or Floor Areas (in 10^6 ft^2) with Heating	**3,865**	**732**	**660**	**1,216**	**57,764**	**10,638**	**11,678**	**12,037**
Cooling	Central Chillers	6%	7%	8%	4%	27%	29%	46%	12%
	Room AC	34%	24%	20%	38%	37%	22%	23%	31%
	Packaged Units	62%	61%	70%	60%	67%	66%	66%	77%
	Heat Pumps	14%	21%	18%	10%	15%	19%	18%	12%
	Air Ducts	54%	61%	64%	46%	66%	74%	77%	61%
	Fan Coils	3%	5%	4%	1%	21%	29%	33%	6%
	Other	3%	11%	N/A	3%	3%	N/A	N/A	2%
	Total Buildings (in 1,000) or Floor Areas (in 10^6 ft^2) with Cooling	**3,184**	**576**	**656**	**929**	**51,761**	**9,319**	**11,636**	**10,803**

* Many buildings have more than one system in the building, which is why the column totals add up to more than 100%. For an assessment of the prevalence of combinations of HVAC systems based on the 1989 CBECS, see Sezgen and Koomey 1998.

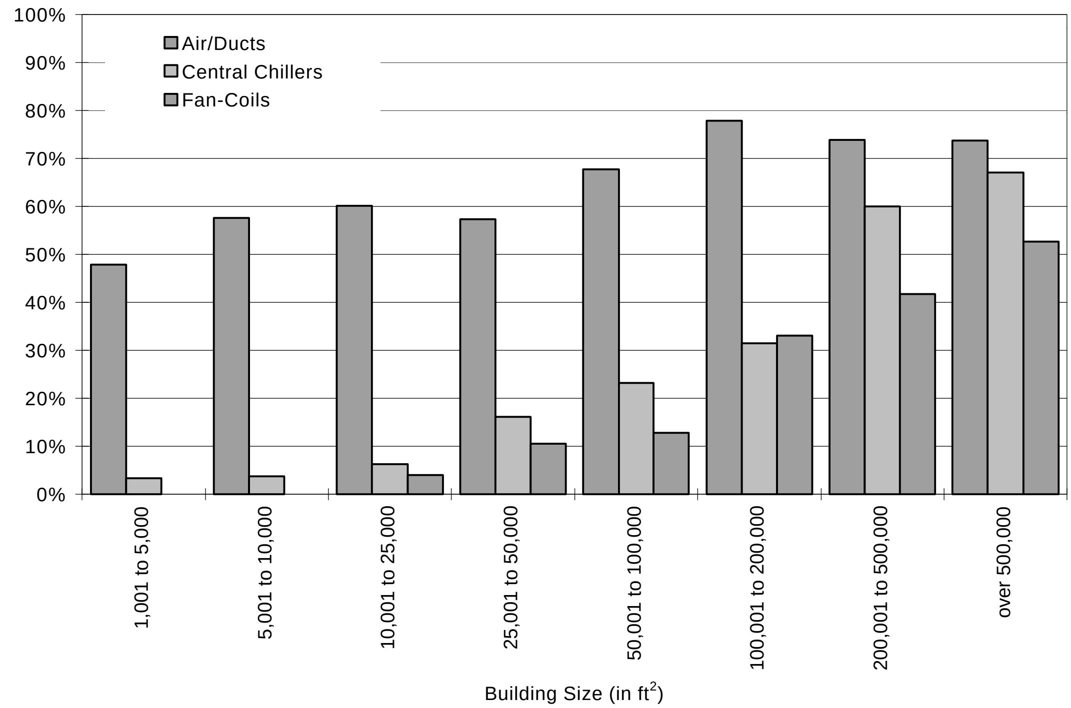

Figure 3. Occurrence of Cooling Equipment vs. Building Size (CBECS 1989 Data)

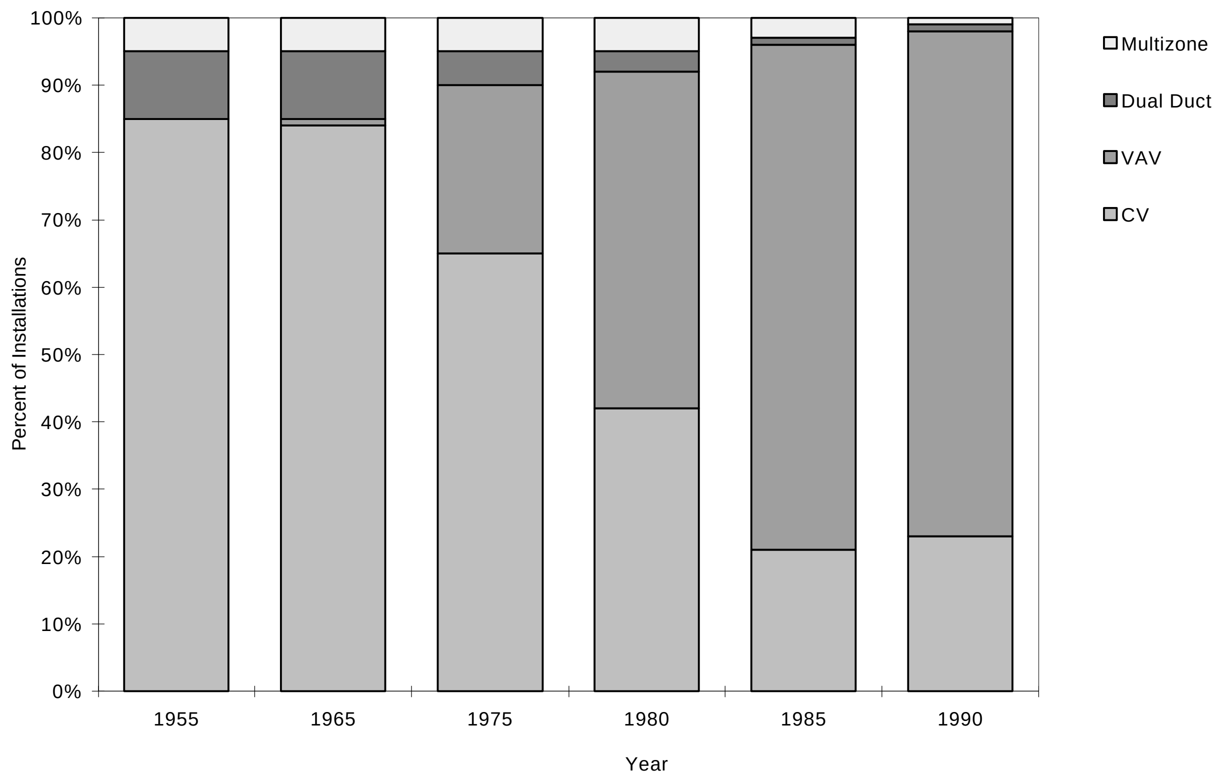

Figure 4. Central Air Distribution Systems - New Construction Trends for the U.S. (Pietsch 1991)

More information regarding distribution systems was found in an EPRI Report on Water-Loop Heat Pumps (Pietsch 1991). Figure 4 shows the growth of VAV systems relative to CAV systems in new central system installations nationwide over the past 30 years.

3.2.2 California Building Stock

Our characterization of thermal distribution systems in California commercial building is based largely on previous surveys carried out by or for California utilities and/or for the California Energy Commission. It is important to note that these surveys were carried out for the purpose of understanding energy end-use in commercial buildings in individual service territories. They were not intended to obtain detailed information on thermal energy distribution systems specifically, although some of the surveys do provide some information on these systems. The basis for the California stock characterization is a group of commercial energy use surveys (CEUS) carried out for four California utilities: Pacific Gas and Electric (PG&E), Southern California Edison (SCE), San Diego Gas and Electric (SDG&E) and Sacramento Municipal Utility District (SMUD).

The SMUD survey instrument provided the best information about distribution systems, allowing multiple system descriptions for each building zone. Table 7 shows the distribution choices defined in the SMUD survey instrument. Since most buildings have more than one system or system type, this capability of defining each system and the zone which it serves is very valuable. Unfortunately, the SMUD survey data set is the only data with this level of system detail.

Table 7. SMUD Survey - Distribution System Types

Radiator	4 pipe, Induction Unit
Hydronic Baseboard	4 pipe, Radiant Panel
2-pipe, Fan Coil	AIR Single Duct, Variable Volume
2 pipe, Induction Unit	AIR Single Duct, Constant Volume
2 pipe, Radiant Panel	AIR Dual Duct, Variable Volume
3-pipe, Fan Coil	AIR Dual Duct, Constant Volume
3 pipe, Induction Unit	AIR Multizone
3 pipe, Radiant Panel	AIR Variable Volume, Variable Temperature
4-pipe, Fan Coil	Radiant Non-Fan System

Table 8 summarizes the description, size and methodology of the four utility surveys (shaded) as well as that of the other sources consulted for the characterization portion of this project. The surveys vary in the level of detail collected about HVAC systems. The most useful of the surveys for our purposes are the three on-site surveys, which have some detailed information about distribution systems.

In order to compare the different surveys and use them to generate statewide estimates, a common format was developed to represent the survey data. Information was not available from each survey for every entry in the common format, and in some cases interpretation was required to determine what the data from a given survey actually represented. In some surveys, distribution system information was only collected for central chilled water systems but not for packaged units that could be configured as either VAV or CAV systems for single or multiple zones. Because packaged units are quite common, this omission in a given survey was significant.

Another confusion arises with respect to multizone systems. It is quite possible that some affirmative responses to multizone systems are meant to reflect that the system has more than a single zone, rather than a true multizone system with a separate duct and temperature sensor for each zone. This type of problem can be resolved in future studies by improving the survey instrument format and providing clear explanations of the choices.

Despite the shortcomings of these surveys, the information they provide is useful. In order to use the data to generate a statewide characterization of distribution systems, we have weighted the results of each survey by the CEC floor area estimations for 1994 for each utility service territory. If data are not available for a given distribution system and building combination in a specific utility service territory, the breakdown is determined by assuming that it is the same as that of the statewide average for the same combination. Table 9 through Table 14 shows the statewide estimates resulting from these procedures.

Table 8. Stock Characterization Resources

Title	Date	Prepared By	Prepare d For	Source of Information
Integrated Estimation of Commercial Sector End-Use Load Shapes and Energy Use Intensities in PG&E Service Area	June 1993	Lawrence Berkeley National Laboratory	PG&E CEC CIEE	-PG&E on-site survey data of 855 buildings by ADM Associates -1988 PG&E commercial sector end-use mail survey
PG&E Company 1988 Commercial Energy Use Survey Major Findings (Final Report)	Oct. 1990	Synergic Resources Corporation	PG&E	-1988 PG&E commercial sector end-use mail and telephone survey of 5,973 buildings. Developed by BR Assoc. fielded by Freeman, Sullivan & Co.
1990 Commercial Energy Use Survey (CEUS) Volume I: Findings and Research Methods	June. 1991	QEI, Inc.	SDG&E	314 on-site interviews from Nov. 1990 to Jan 1991
Commercial Energy Use Survey For 1988 Volume 1 of 2 (Summary)	Nov. 1989	ADM Associates	SDG&E	4,428 respondents: app. 2,200 by mail, 2,100 on-site
Market Research and Evaluation 1988 Commercial Energy Use Survey -Supplementary Tables -Report and Weighted Codebook	Dec. 1988	Decision Sciences Research Assoc., Inc.	SCE	4,800 respondents: app. 4,500 by mail, 300 by telephone
Commercial Energy Use Survey in the SMUD Service Territory (Final Report)	Nov. 1990	ADM Associates	CEC SMUD	23,256 single accounts and 5,180 multiple accounts
Energy Edge Impact Evaluation (Middle Review)	May 1992	Lawrence Berkeley Lab	BPA	28 monitored commercial buildings
*Commercial Buildings Energy Consumption and Expenditures 1989	April 1992	DOE/EIA	public	Random samples collected from commercial buildings across the United States
Commercial Buildings Characteristics 1989	June 1991	DOE/EIA	public	Random samples collected from commercial buildings across the United States
California Energy Demand 1991-2011 (PG&E Service Area)	June 1991	CEC	public	Electricity consumption and peak demand forecasts for PG&E service area.
Water-Loop Heat Pump Systems: Assessment Study Update (Final Report)	Oct. 1991	J.A. Pietsch	EPRI	Report on WLHP systems.
The State of the Art: Space Cooling and Heating	Aug. 1992	COMPETITE K		A comprehensive report on space cooling and air handling
California Energy Demand 1991-2011 Volume I: Revised Electricity Demand Forecasts Final	Dec. 1991	CEC	public	Revised demand forecast for most customer sectors in California

* In this revised version, we still use the 1989 data because they are similar to the 1995 data.

3.3 ENERGY CONSUMPTION

3.3.1 HVAC-Related Electricity Consumption

Based upon the forecast for 1994 (CEC 1991a), Table 9 summarizes, for each building type, the total floor area, and row percentages of the corresponding building electricity consumption for heating, cooling, and fans and pumps, respectively. Table 9 also shows the percentage of total California commercial electricity consumption by building types (i.e., the electricity consumption for heating in small offices divided by the total electricity consumption of all commercial building types).

Among the total building electricity consumption for heating, hotels and motels account for 46% of the use due to the predominant use of unit heaters and coolers, while large offices account for another 20%. Among the total building electricity consumption for cooling, large office buildings use the largest portion (34%), followed by health/hospital buildings (15%), retail buildings (8%), small offices (7%), and hotels (7%). Among the total building electricity consumption for fans and pumps, large office buildings again use the largest portion (36%), followed by retail buildings (12%), health/hospital buildings (8%), and restaurants (8%).

Heating electricity consumption, per building type, ranges from zero to 46% of building electricity consumption. Cooling electricity consumption, per building type, ranges from 1% to 34% of the total building electricity consumption. Fan and pump energy consumption, per building type, ranges from 3% to 17% of the total building electricity consumption.

On average, annual heating electricity consumption is 2% of the total California commercial buildings' electricity consumption, which is 82,832 GWh annually. Cooling energy consumption is 16% of the total California commercial buildings' electricity consumption, while fans and pumps use 10% of the total California commercial buildings' electrical consumption.

Table 9. Statewide Estimates - HVAC-Related Electricity Consumption (for 1994)

CEC Forecast for 1994	Small Office	Large Office	Retail	Restaurant	Foodstore	Warehouse	School	Univ/ College	Health/ Hospital	Hotel/ Motel	Miscellaneous	**Row Total (Million Square Feet)**	
Floor Space Stock Projection (Million Square Feet)	303	1,057	764	123	213	948	419	253	273	290	774	5,417	
Electricity Consumption Row Percentage for Each Building Type												**Row Total %**	**Row Total (GWh)**
Heating	0%	20%	1%	3%	2%	1%	1%	8%	4%	46%	14%	100%	**2,068**
Cooling	7%	34%	8%	3%	1%	1%	3%	5%	15%	7%	15%	100%	**13,583**
Fans/Pumping	4%	36%	12%	8%	5%	3%	3%	5%	8%	4%	13%	100%	**8,626**
Percentage of Total Building Electricity Consumption												**Row Total (%)**	**Row Total (GWh)**
Heating	0%	0%	0%	0%	0%	0%	0%	0%	0%	1%	0%	2%	**82,832**
Cooling	1%	6%	1%	1%	0%	0%	0%	1%	2%	1%	3%	16%	
Fans/Pumping	0%	4%	1%	1%	1%	0%	0%	1%	1%	0%	1%	10%	

The vast majority of the direct energy consumption for fans and pumps results in heating of the conditioned air and therefore contributes to the internal cooling load. Usibelli (1985) found the following breakdown for peak cooling loads for office buildings in California: 31% for lighting, 13% for people, 14% for air transport, and 6% for equipment. External loads account for only 36% of the peak-cooling load. The continuous ventilation (either by mechanical or natural) is required by Title 24 (CEC, 1998b) to provide outdoor air during occupied hours. It is very likely that commercial-building fans remain on in order to provide ventilation in practice, which would increase the overall contribution of fan energy to space conditioning energy use. Thus, reducing fan energy has the dual benefits of direct fan energy savings and indirect energy savings through reducing cooling loads.

3.3.2 Distribution Systems

Derived from surveys on four utilities in California, Table 10 shows the statewide estimates on the occurrence frequency of various distribution systems by floor area, from 1988 through 1993. Column percentages are given for the distribution systems within each building type. The total percentages given at the right column indicate the overall occurrence frequency for the particular systems.

The most common distribution system across different building types is the single duct, constant-air-volume system (46%, on average), followed by multizone systems (11%). Due to the way the various surveys described multizone systems, they may be over-represented here. This could result in a higher percentage of multizone systems reported than actually occurring. VAV systems are significant only in office buildings and hotel buildings, and even then, much less common than CAV systems. Water systems are most common in health facilities and hotels and motels, though still in relatively small numbers.

Table 10. Statewide Estimates - Distribution Systems by Existing Floor Area*

Column Percentage of Distribution Systems	Small Office	Large Office	Retail	Restaurant	Foodstore	Warehouse	School	Univ/ College	Health/ Hospital	Hotel/ Motel	Miscellaneous	**Overall**
Air												
Single Duct												
Constant Volume/Single Zone	61%	36%	54%	70%	49%	47%	31%	80%	35%	27%	49%	**46%**
Variable Volume	6%	19%	3%	0%	0%	0%	2%	2%	2%	7%	2%	**4%**
Dual Duct	4%	5%	0%	0%	0%	0%	0%	1%	6%	0%	0%	**2%**
Multi Zone	11%	26%	18%	6%	9%	1%	6%	2%	33%	3%	6%	**11%**
Water												
Fan Coil	6%	6%	1%	2%	0%	0%	5%	5%	8%	12%	4%	**5%**
2 pipe	1%	1%	0%	1%	0%	1%	0%	0%	7%	2%	1%	**1%**
4 pipe	1%	1%	0%	0%	0%	1%	1%	2%	4%	4%	0%	**1%**
Radiator	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	**0%**
Induction	0%	1%	0%	0%	0%	0%	0%	0%	1%	0%	0%	**0%**
Other / None / Not Listed												
Other	0%	0%	0%	0%	1%	1%	0%	0%	0%	0%	0%	**0%**
None/Not Listed	10%	5%	24%	20%	42%	51%	55%	9%	3%	45%	37%	**29%**

*Based on utility survey data of 1988 through 1993, absolute row totals not available.

The data suggest that the distribution of central HVAC systems in California may be quite different from the rest of the nation. An EPRI study (Pietsch 1991) suggested a significant national trend towards VAV systems in new construction (75% of all central systems in the last ten years), while our stock characterization indicates only 19% VAV systems in large existing California office buildings, which are usually the predominant user of central systems. 67% of all central systems in existing large commercial buildings nationwide are, according to the same EPRI study, equipped with hydronic thermal distribution systems. The California data shows that an average of only 7% of the thermal distribution systems is the hydronic system (7% to 20% per building type).

3.3.3 Cooling Systems

Table 11 summarizes the statewide estimates for cooling systems distribution by commercial building floor area. Column percentages indicate the percentage of cooling systems within each building type. The total row percentages given at the right indicate the overall occurrence frequency for the particular cooling systems for all floor areas.

Central heat pumps, chillers and direct expansion cooling systems each represent 6% to 7% of the total cooling systems per floor area. Unit coolers represent 9% of the cooling systems per floor area. Overall from the available data, 70% of the cooling systems are described as "unknown," "other" or "none."

Table 11. Statewide Estimates - Cooling Systems Distribution by Existing Floor Area*

Column Percentage of Cooling Systems	Small Office	Large Office	Retail	Restaurant	Foodstore	Warehouse	School	Univ/ College	Health/ Hospital	Hotel/ Motel	Miscellaneous	**Overall**
Main Cooling Systems												
Central Air Conditioning												
Heat Pump	7%	7%	6%	4%	3%	9%	2%	0%	8%	4%	1%	**7%**
Chiller	7%	7%	2%	1%	2%	3%	10%	26%	15%	11%	3%	**6%**
Direct Expansion	7%	7%	9%	11%	11%	11%	2%	0%	5%	5%	3%	**7%**
Evaporative Cooler	0%	0%	0%	0%	1%	0%	0%	0%	0%	0%	0%	**0%**
Unknown	41%	39%	33%	36%	22%	32%	38%	6%	39%	0%	0%	**30%**
Unit Cooling (no distribution system)	12%	11%	13%	4%	1%	3%	2%	2%	5%	32%	8%	**9%**
Other	0%	0%	3%	7%	7%	6%	9%	0%	6%	34%	23%	**5%**
Unknown	21%	22%	19%	17%	17%	25%	15%	26%	22%	14%	60%	**20%**
None	4%	4%	14%	19%	35%	11%	21%	40%	0%	0%	0%	**15%**
Secondary Cooling Systems												
Room Air Conditioners	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	**0%**
Package Units	0%	0%	0%	0%	0%	2%	1%	0%	0%	0%	0%	**0%**
Heat Pump	1%	1%	0%	0%	2%	2%	0%	0%	0%	1%	1%	**1%**
Central Air	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	**0%**
Fan	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	**0%**
Evaporative Cooler	0%	0%	0%	5%	2%	1%	0%	0%	0%	0%	1%	**2%**
Direct Expansion Unit	2%	2%	1%	0%	1%	2%	1%	0%	0%	1%	1%	**1%**
Chiller	1%	1%	0%	0%	0%	0%	0%	0%	0%	0%	0%	**0%**

*Based on utility surveys 1988 through 1993, absolute row totals not available.

3.3.4 Heating Systems

Similar to Table 11, Table 12 summarizes the statewide estimates for heating system distribution by commercial building floor area. Column percentages indicate the percentage of heating systems within each building type. The total row percentages given at the right indicate the overall occurrence frequency for the particular heating systems for all buildings.

Overall, forced air heaters (32%), packaged units (29%) and unit heaters (21%) are the most prevalent followed by boilers (hot water or steam, 13%) and air or water source heat pumps (2%), among the total California commercial building floor area. Particularly, forced air heaters predominate in small offices (36%), large offices (36%) and universities and colleges (34%). Unit heaters predominate in hotels and motels (64%) and miscellaneous buildings (21%). Packaged units predominate in restaurants (39%), food stores (38%), warehouses (38%), health and hospital buildings (34%) and retail buildings (31%).

Table 12. Statewide Estimates - Heating Systems by Existing Floor Area*

Column Percentage of Heating Systems	Small Office	Large Office	Retail	Restaurant	Foodstore	Warehouse	School	Univ/ College	Health/ Hospital	Hotel/ Motel	Miscellaneous	**Overall**
Main Heating Systems												
Central Heating												
Forced Air Furnace	36%	36%	29%	26%	27%	22%	28%	34%	21%	15%	19%	**32%**
Boiler(Hot Water/Steam)	11%	11%	7%	12%	7%	20%	29%	30%	26%	11%	12%	**13%**
Heat Pump(air or water source)	2%	2%	2%	1%	2%	5%	2%	0%	2%	3%	1%	**2%**
Unit Heater(no dist. system)	20%	20%	27%	19%	20%	17%	4%	3%	11%	64%	21%	**21%**
Packaged	28%	28%	31%	39%	38%	33%	22%	10%	34%	6%	16%	**29%**
Other	1%	1%	3%	2%	2%	2%	1%	0%	5%	0%	17%	**3%**
Unknown	1%	1%	1%	0%	1%	1%	15%	18%	0%	0%	2%	**1%**
None	0%	0%	0%	0%	3%	1%	0%	5%	0%	0%	12%	**0%**
Secondary Heating Systems												
Space/unit Heater	0%	0%	1%	0%	0%	1%	0%	0%	0%	0%	1%	**0%**
Heat Pump	1%	1%	0%	1%	2%	2%	0%	0%	2%	1%	1%	**1%**
Boiler	0%	0%	0%	0%	0%	0%	0%	0%	0%	1%	0%	**0%**
Floor/Wall Heater	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	**0%**
Furnace	0%	0%	0%	0%	1%	2%	0%	0%	0%	1%	1%	**0%**
Electric Coil/Resistance	1%	1%	1%	0%	2%	5%	1%	0%	1%	1%	0%	**1%**

*Based on utility survey data 1988 through 1993, absolute row totals not available.

3.3.5 Conservation Measures

Table 13 summarizes the statewide estimates of energy conservation measures undertaken in the California commercial buildings. Column percentages are given as a percentage of floor area within each building type. The total row percentage given at the right represents the overall saturation of the specific conservation measure for all floor area.

The energy conservation measures listed are divided into two groups: general HVAC measures and distribution system measures. In terms of general HVAC measures, installation of time clocks are the common measures undertaken in most building types. Economizers are most common in restaurants and health and hospital buildings (16% and 20%, respectively). In terms of distribution measures, insulating ductwork is the most common in most building types (15% overall, 6% to 31% per building type), followed by insulating piping (7% overall, up to 18% in Hotel/Motel) and high efficiency motors (6% overall, 3% to 19% per building type).

The most prevalent energy conservation measures in the buildings include insulating ductwork (15% of all buildings), time clocks for heating (12%) and cooling (11%), economizers (7%), insulating piping (7%) and high efficiency motors (6%). Other energy conservation measures are undertaken in lesser frequency, including energy management systems (1% of all buildings) and cold / ice storage (1%). Overall, HVAC measures are implemented in 34% of the building floor area, and are most common in school buildings (93%) and university buildings (84%). Distribution measures are implemented in 28% of the building floor area, and are relatively common in health/hospital buildings (60%) and university buildings (50%).

Table 13. Statewide Estimates - Conservation Measures by Existing Floor Area*

Conservation Measures	Small Office	Large Office	Retail	Restaurant	Foodstore	Warehouse	School	Univ/ College	Health/ Hospital	Hotel/ Motel	Miscellaneous	**Total**
HVAC												
Time Clock	1%	1%	0%	0%	0%	1%	1%	0%	1%	0%	0%	**0%**
Air Conditioning	15%	15%	12%	7%	14%	12%	24%	23%	12%	2%	5%	**11%**
Heating	15%	15%	9%	11%	14%	8%	59%	35%	15%	2%	7%	**12%**
Regular System Maintenance	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	**0%**
Economizers	8%	8%	5%	16%	6%	4%	7%	24%	20%	2%	5%	**7%**
Precoolers on A/C units	0%	0%	0%	0%	0%	0%	0%	0%	1%	0%	0%	**0%**
Energy Management System	1%	2%	1%	0%	1%	0%	1%	0%	0%	0%	0%	**1%**
Chiller Optimizer	0%	0%	0%	0%	0%	0%	0%	0%	1%	0%	0%	**0%**
Cold/Ice Storage	0%	0%	2%	0%	0%	0%	0%	1%	0%	0%	0%	**1%**
Optimium Start/Stop	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	**0%**
VAV Conversion	0%	0%	0%	0%	0%	0%	0%	0%	0%	1%	0%	**0%**
Other HVAC	1%	1%	1%	0%	1%	0%	1%	0%	0%	0%	0%	**0%**
HAVC Subtotal	**42%**	**42%**	**32%**	**36%**	**35%**	**24%**	**93%**	**84%**	**51%**	**8%**	**17%**	**34%**
Distribution												
Insulated Ducts	13%	13%	13%	26%	14%	24%	16%	18%	31%	17%	6%	**15%**
Insulated Pipes	7%	7%	3%	12%	0%	4%	18%	15%	10%	18%	7%	**7%**
Var. Speed Fans	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	**0%**
Var. Speed Pumps	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	**0%**
High Efficiency Motors	6%	5%	6%	7%	10%	9%	10%	18%	19%	3%	7%	**6%**
Duty Cycling of Fans or Pumps	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	0%	**0%**
Distribution Subtotal	**26%**	**26%**	**22%**	**44%**	**24%**	**37%**	**44%**	**50%**	**60%**	**38%**	**20%**	**28%**

*Based on utility surveys 1988 through 1993, absolute row totals not available.

3.3.6 Fan and Pump Energy Consumption

Figure 5 shows the 1994 CEC forecast of fan and pump electrical energy consumption by building type. It also indicates that offices and retail buildings account for over half of the projected fan and pump distribution system energy consumption for the state (51%). Therefore, these building types should be a primary target for future efforts in reducing fan and pump energy use.

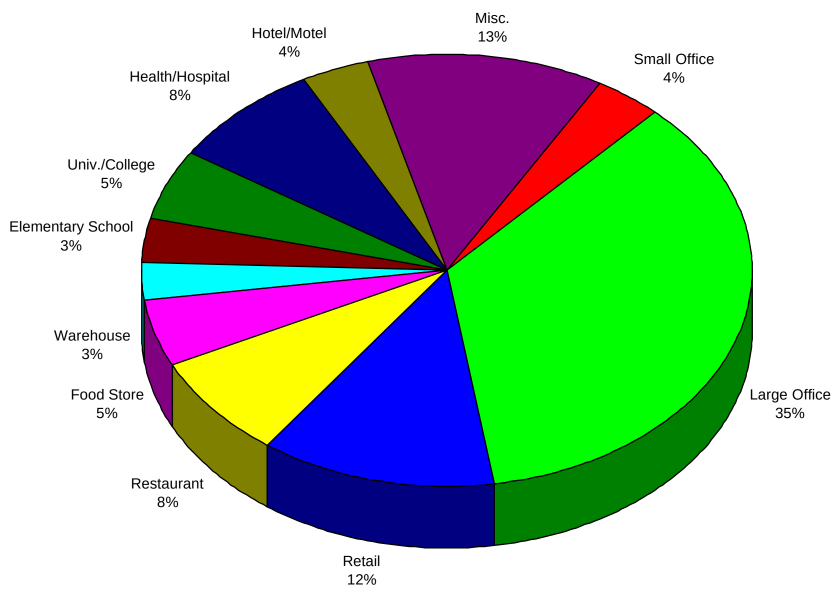

Figure 5. 1994 Fan/Pump Energy Distribution

3.3.7 Summary

The CEC (1998) estimated that, in 1997, approximately 15,500 GWh of electricity and 24,000 GWh (810 million therms) of gas were used in space conditioning for statewide commercial buildings. Fans and pumps use additional 8,832 GWh of electricity in the same buildings (Table 14).

For the purposes of this report, thermal distribution is defined as the ductwork or piping that is used to transport heating or cooling energy from the heating or cooling unit(s) to the spaces being conditioned, as well as duct systems used to distribute ventilation air. In order to explore the extent of energy savings opportunities associated with improving thermal energy distribution in buildings, it is necessary to determine the fraction of this space conditioning energy that is passed through thermal distribution systems, as well as the fraction of the total fan and pump energy used to move air and water through these systems. Borrowing the electricity consumption proportion for fan and pump across different building types as shown in Figure 5 and utility survey results (Table 10), Table 14 lists the electricity energy use for each distribution system per building type.

Table 14. Statewide Estimates – 1997 Fan and Pump Energy Consumption by Building Type

Statewide Projections1997 Fan/pump Electricity Use	Small Office	Large Office	Retail	Restrnt	Foodstore	Ware- house	School	Univ/ College	Health/ Hospital	Hotel/ Motel	Misc.	**Total**
Fans/Pumps (GWh)	353	3,091	1,060	707	442	265	265	442	707	353	1,148	**8,832**
Distribution System												
Air												
Single Duct												
Constant Volume/Single Zone	214	1,107	567	497	216	125	82	355	247	95	568	4,063
Variable Volume	23	589	32	0	0	0	5	7	17	24	20	378
Dual Duct	14	158	2	0	0	0	1	2	42	0	1	140
Multi Zone	39	814	191	42	38	2	16	9	233	11	69	1,003
Water												
Fan Coil	21	183	10	17	0	0	12	23	58	41	48	478
2 pipe	3	37	0	8	0	2	1	0	52	6	9	119
4 pipe	3	29	2	1	0	2	3	7	30	14	5	8
Radiator	1	6	0	0	0	0	0	0	3	1	1	7
Induction	0	17	0	0	0	0	0	0	5	0	0	8
Other/Not listed	35	155	251	142	188	137	144	40	21	160	426	2,633

Space conditioning performed with unitary heaters in warehouses, according to our definition, does not involve thermal distribution as there is no ductwork or piping (there is a fan that blows the heating into the space, however the fan is an integral part of the unitary heater). Similarly, many of the fans in a restaurant are used to vent cooking fumes and heat from the kitchens rather than to transport heat or cooling to the space. While the operation of exhaust fans impacts the amount of heating, cooling and ventilation provided to a given space, these fans will not be considered as thermal distribution systems as they do not deliver heating or cooling energy to that space.

Using CEC forecast data in conjunction with the utility survey information, we can estimate the direct fan and pump energy use associated with different distribution systems. As data were not available to quantify the percentage of total fan and pump energy used for exhaust fans or pumps used for purposes other than the distribution systems, we have assumed that the total commercial building fan and pump energy is used in distribution systems. Although the energy consumption directly related to a given distribution system is obviously dependent on the system type, for the purposes of this analysis we are assuming that fan and pump energy use is independent of system type. In other words, the distribution system energy consumption per system is assumed to be the same for a ducted CAV system, a ducted VAV system, an all-water, hydronic system and a four-pipe system with distributed ducted fan coils. In reality, the energy consumption for these various system types differs widely between system types. The reason for this assumption is that we do not have sufficient utility survey data to support a more detailed analysis, and it would be misleading to suggest that we can discern differences in distribution system energy use at the level of detail of the survey data. In order to examine this issue further, DOE-2 analysis of various distribution systems and their corresponding fan and pump energy usage in a prototypical building will be discussed later in this report.

To estimate the energy consumption of each distribution system type, the CEC projected fan and pump forecast data are multiplied by the occurrence of each system type in each utility planning area for which we have distribution system data (Table 10). If data are not available for a given combination of a distribution system type with a building type for a specific utility service territory, the breakdown is determined by assuming that it is the same as the statewide average for the same combination. Table 14 shows the results of this calculation. Overall, the combination of the single duct CAV and multi-zone air distribution systems has the highest accounts for more than fifty percent of the fan and pump energy consumption for the commercial building sector statewide.

3.4 ENERGY USE IMPACTS AND ENERGY CONSERVATION OPPORTUNITIES

Applying the above distinctions to the stock characterization data, we can further quantify the energy use impacts directly related to thermal distribution systems in commercial buildings. This energy use is divided into two categories: 1) space conditioning energy that passes through thermal distribution systems and thus can be lost through duct leakage and conduction losses somewhere between entering the distribution system and being delivered to the conditioned space; and 2) fan and pump energy used directly in the thermal distribution process. Heat gains from the distribution fan or pump contribute to increases in cooling loads.

In order to assess the savings potentials of different HVAC systems, we need to determine the systems' penetration for each building type. The utility survey data can be used to estimate the distribution of building types and HVAC systems in California.

Table 15. Distribution of HVAC Systems for different Building Types in Percent*

Building Type	**System Type**					
	Air Packaged	Air Central	Hydronic Central	Ductless Unit	Other /None	Row Total
Small Offices	66%	1%	1%	3%	29%	100%
Large Offices	51%	38%	1%	8%	2%	100%
Retail	62%	6%	6%	9%	17%	100%
Restaurants	74%	1%	1%	8%	16%	100%
Food Stores	67%	2%	0%	9%	22%	100%
Warehouses	47%	2%	1%	13%	37%	100%
Schools	44%	27%	1%	18%	10%	100%
Colleges/Universities	34%	4%	1%	6%	55%	100%
Hospitals	40%	32%	8%	5%	15%	100%
Hotel/Motel	47%	10%	3%	37%	3%	100%
Miscellaneous	55%	1%	1%	15%	28%	100%

Based on utility surveys 1988 through 1993, total not available

Derived from the CEC forecast data and utility survey data presented in Tables 10 through 12 Table 15 is created to show distribution of HVAC systems on the basis of building floor-area. When data were not available for a given system type / building type combination from specific utility survey, the distribution of equipment was assumed based on the statewide floor-area-weighted average for that combination.

Based on floor area, air packaged units are the most prevalent in all building types (34% to 74%). Large office buildings, schools, hospitals and hotels and motels are often equipped with central systems (10% to 38%). Units without thermal distribution systems are common in hotels and motels (37%) and some schools (18%) or warehouse (13%). Hydronic distribution systems are installed sparingly: in only 6% of retail buildings and 8% of hospitals, and almost none in other building types are equipped with fan coils.

The estimated energy consumption of a given system type in a given building type is calculated using 1) the energy consumption for heating, cooling and fans and pumps for that building type (Table 9); 2) the percentage of the total building type floor area having that system type (Table 15); 3) the location of system ductwork (rooftop, ceiling plenum or inside the conditioned building envelope); and 4) a cycling factor reflecting the percentage of systems with fans that cycle on-and-off based on thermal conditions rather than running continuously.

The percentage of systems per building type, cycling factors and the duct location (as a percentage of the total ductwork per building type) are given in Table 16 for air packaged systems, and in Table 17 for air central systems. The ductwork location distribution percentages and cycling factors given are also assumed based on our estimation of the building stock (Huang 1989) and HVAC system design, installation and current operating practices. The cycling factors reflect the percentage of air package units in a given building type that cycle on-and-off based on thermal conditions. The remaining systems are assumed to run continuously during occupied hours for the spaces normally used by occupants. The continuous ventilation (either by mechanical or natural) is required by Title 24 (CEC, 1998b) to provide outdoor air during occupied hours.

Table 16. Packaged Air Unit - System Distribution Assumptions

AIR PACKAGED SYSTEMS							
Building Type	% Air Packaged Systems	Duct Location (%)*				Percent Cycling**	
		Roof Top	Top Clg. Plenum	Inside Building	Total	Cooling	Heating
Small Office	66%	10%	40%	50%	100%	40%	60%
Large Office	51%	5%	5%	90%	100%	5%	10%
Retail	62%	10%	40%	50%	100%	40%	60%
Restaurant	74%	10%	70%	20%	100%	40%	60%
Food Store	67%	10%	70%	20%	100%	40%	60%
Warehouse	47%	10%	70%	20%	100%	80%	80%
Schools	44%	5%	25%	70%	100%	33%	50%
Univ/College	34%	5%	25%	70%	100%	10%	20%
Health/Hospitals	40%	5%	5%	90%	100%	10%	20%
Hotel/Motel	47%	10%	40%	50%	100%	10%	20%
Miscellaneous	55%	10%	70%	20%	100%	33%	50%
Total	n/a	n/a	n/a	n/a	n/a	n/a	n/a

*Percent of total ductwork located on the rooftop, within the top floor ceiling plenum (adjacent to the roof deck), and inside the conditioned building space. Distribution of ductwork locations is based on estimates of current practice and commercial building characteristics (i.e, number of stories, system applications, etc.).

**Percent of systems where distribution fans cycle on and off. The remainder of the systems have fans running constantly.

Table 17. Central Air System - System Distribution Assumptions

AIR CENTRAL SYSTEMS							
Building Type	% Air Central Systems	Duct Location (%)*				Percent Cycling**	
		Roof Top	Top Clg. Plenum	Inside Building	Total	Cooling	Heating
Small Office	1%	5%	10%	85%	100%	40%	60%
Large Office	38%	3%	5%	92%	100%	5%	10%
Retail	6%	5%	10%	85%	100%	40%	60%
Restaurant	1%	5%	10%	85%	100%	40%	60%
Food Store	2%	10%	10%	80%	100%	40%	60%
Warehouse	2%	10%	30%	60%	100%	80%	80%
Schools	27%	5%	10%	85%	100%	33%	50%
Univ/College	4%	5%	10%	85%	100%	10%	20%
Health/Hospitals	32%	3%	5%	92%	100%	10%	20%
Hotel/Motel	10%	5%	10%	85%	100%	10%	20%
Miscellaneous	1%	5%	10%	85%	100%	33%	50%
Total	n/a	n/a	n/a	n/a	n/a	n/a	n/a

*Percent of total ductwork located on the rooftop, within the top floor ceiling plenum (adjacent to the roof deck), and inside the conditioned building space. Distribution of ductwork locations is based on estimates of current practice and commercial building characteristics (i.e, number of stories, system applications, etc.).

**Percent of systems where distribution fans cycle on and off. The remainder of the systems have fans running constantly.

Integrating the above-mentioned system distribution breakdowns (Table 15, Table 16, and Table 17) with the heating, cooling and fan and pump electricity energy consumption breakdowns from the 1994 CEC data (Table 9), we can estimate the system/building type-specific electricity energy consumption associated with commercial building heating, ventilation and air conditioning. The results of these calculations are provided in Table 18 (electricity), Table 19 (natural gas) for air packaged systems, and in Table 20 (electricity), Table 21 (natural gas) for air central systems.

Using the “Air Packaged Systems” electricity consumption as an example, Table 18 breaks out the energy consumption into: 1) “Total,” the total energy consumption for all air packaged systems in each building type; 2) “Rooftop,” “Ceiling Plenum,” and “Inside Building,” the energy consumption distributed based on the percentage of each duct location type per building type; and 3) “Cycling Total,” the portion of the total energy consumption of systems which cycle on-and-off (not including fan’s continuous operation for ventilation).

Based on this breakdown, rooftop and ceiling plenum installed ductwork corresponds to approximately 40% of the electricity consumption for air packaged systems (heating, cooling and fans/pumps combined), and approximately 9% of the electricity consumption for air central systems (heating, cooling and fans/pumps combined). Similarly, rooftop and ceiling plenum installed ductwork corresponds to approximately 42% of the natural gas consumption for air-packaged systems (heating and cooling combined) and approximately 10% of the natural gas consumption for air central systems. Ductworks installed inside the conditioned building envelope correspond to the respective remainders of the systems’ energy consumption.

Table 18. Air Packaged Systems - Electricity Consumption (1994 CEC Forecast)

AIR PACKAGED SYSTEMS															
Building Type	Cooling Energy (GWh)					Electric Heating (GWh)					Fan and Pump Energy (GWh)				
	Total	By Duct Location			Cycling Total	Total	By Duct Location			Cycling Total	Total	By Duct Location			Cycling Total
		Roof Top	Top Clg. Plenum	Inside Building			Roof Top	Top Clg. Top	Top Clg. Plenum			Roof Top	Top Clg. Top	Top Clg. Top	
Small Office	612	61	245	306	245	3	0	1	2	2	200	20	80	100	80
Large Office	2,331	117	117	2,098	117	208	10	10	187	21	1,568	78	78	1,411	78
Retail	685	69	274	343	274	15	2	6	8	9	660	66	264	330	264
Restaurant	331	33	231	66	132	39	4	28	8	24	508	51	356	102	203
Food Store	114	11	80	23	46	25	2	17	5	15	281	28	197	56	113
Warehouse	85	8	59	17	68	10	1	7	2	8	113	11	79	23	90
Schools	176	9	44	123	58	13	1	3	9	6	127	6	32	89	42
Univ/College	218	11	55	153	22	54	3	14	38	11	157	8	39	110	16
Health/Hospitals	822	41	41	740	82	35	2	2	32	7	273	14	14	246	27
Hotel/Motel	473	47	189	236	47	450	45	180	225	90	145	14	58	72	14
Miscellaneous	1,138	114	797	228	376	155	15	108	31	77	596	60	417	119	197
Total	6,987	521	2,132	4,333	1,467	1,008	85	377	546	270	4,628	357	1,614	2,658	1,125

Table 19. Air Packaged Systems - Gas Consumption (1994 CEC Forecast)

AIR PACKAGED SYSTEMS										
Building Type	Gas Heating (Billion BTU)					Gas Cooling (Billion BTU)				
	Total	By Duct Location			Cycling Total	Total	By Duct Location			Cycling Total
		Roof Top	Top Clg. Top	Top Clg. Top			Roof Top	Top Clg. Top	Top Clg. Top	
Small Office	52	5	21	26	31	688	69	275	344	275
Large Office	8,769	438	438	7,893	877	1,575	79	79	1,418	79
Retail	240	24	96	120	144	418	42	167	209	167
Restaurant	426	43	298	85	255	267	27	187	53	107
Food Store	752	75	526	150	451	63	6	44	13	25
Warehouse	1,160	116	812	232	928	109	11	76	22	87
Schools	2,015	101	504	1,411	1,008	109	5	27	76	36
Univ/College	4,559	228	1,140	3,191	912	116	6	29	81	12
Health/Hospitals	1,576	79	79	1,418	315	208	10	10	187	21
Hotel/Motel	2,061	206	825	1,031	412	135	13	54	67	13
Miscellaneous	7,341	734	5,139	1,468	3,671	1,492	149	1,045	298	492
Total	28,951	2,049	9,877	17,025	9,004	5,180	418	1,993	2,769	1,314

Table 20. Air Central Systems - Electricity Consumption (1994 CEC Forecast)

AIR CENTRAL SYSTEMS															
	Cooling Energy (GWh)					Electric Heating (GWh)					Fan and Pump Energy (GWh)				
		By Duct Location					By Duct Location					By Duct Location			
Building Type	Total	Roof Top	Top Clg. Plenum	Inside Building	Cycling Total	Total	Roof Top	Top Clg. Plenum	Inside Building	Cycling Total	Total	Roof Top	Top Clg. Plenum	Inside Building	Cycling Total
Small Office	6	0	1	5	*2*	0	0	0	0	*0*	2	0	0	2	*1*
Large Office	1,749	52	87	1,609	*87*	156	5	8	144	*16*	1,176	35	59	1,082	*59*
Retail	71	4	7	61	*29*	2	0	0	1	*1*	69	3	7	58	*27*
Restaurant	4	0	0	4	*2*	1	0	0	0	*0*	7	0	1	6	*3*
Food Store	3	0	0	3	*1*	1	0	0	1	*0*	8	1	1	6	*3*
Warehouse	4	0	1	2	*3*	0	0	0	0	*0*	5	0	1	3	*4*
Schools	106	5	11	90	*35*	8	0	1	6	*4*	77	4	8	65	*25*
Univ/College	28	1	3	23	*3*	7	0	1	6	*1*	20	1	2	17	*2*
Health/Hospitals	646	19	32	594	*65*	28	1	1	26	*6*	215	6	11	197	*21*
Hotel/Motel	101	5	10	85	*10*	96	5	10	81	*19*	31	2	3	26	*3*
Miscellaneous	30	2	3	26	*10*	4	0	0	4	*2*	16	1	2	13	*5*
Total	2,748	90	156	2,502	*247*	301	11	21	269	*50*	1,624	54	94	1,476	*154*

Table 21. Air Central Systems - Gas Consumption (1994 CEC Forecast)

AIR CENTRAL SYSTEMS										
	Gas Heating (Billion BTU)					Gas Cooling (Billion BTU)				
		By Duct Location					By Duct Location			
Building Type	Total	Roof Top	Top Clg. Plenum	Inside Building	Cycling Total	Total	Roof Top	Top Clg. Plenum	Inside Building	Cycling Total
Small Office	1	0	0	0	*0*	7	0	1	6	*3*
Large Office	6,578	197	329	6,051	*658*	1,181	35	59	1,087	*59*
Retail	25	1	2	21	*15*	43	2	4	37	*17*
Restaurant	6	0	1	5	*3*	4	0	0	3	*1*
Food Store	22	2	2	17	*13*	2	0	0	1	*1*
Warehouse	49	5	15	29	*39*	5	0	1	3	*4*
Schools	1,217	61	122	1,034	*608*	66	3	7	56	*22*
Univ/College	577	29	58	490	*115*	15	1	1	12	*1*
Health/Hospitals	1,238	37	62	1,139	*248*	163	5	8	150	*16*
Hotel/Motel	438	22	44	372	*88*	29	1	3	24	*3*
Miscellaneous	196	10	20	166	*98*	40	2	4	34	*13*
Total	10,344	364	653	9,326	*1,885*	1,554	51	89	1,414	*140*

4. DESIGN PRACTICE SURVEYS

In an effort to assess the current state of practice for the design, installation, retrofitting and operation of thermal energy distribution systems, a mail survey of California HVAC professionals was conducted. In addition, in-depth interviews were conducted with two California HVAC engineer/designers.

4.1 MAIL SURVEY

An eleven-page survey instrument (Appendix A) was mailed to 280 HVAC designers, installers, testers, balancers, and operators. As systems are often not built or operated exactly as designed, the survey distribution was intended for not only HVAC designers, but also other HVAC engineers. A single survey format was used for all recipients and questions were posed such that individuals with different backgrounds could answer them according to their specific experience.

The mailing list was assembled from the business yellow pages, ASHRAE Chapter member suggestions, and other individuals suggested by project team members and CIEE staff. Some of the business yellow pages headings used included HVAC engineers, HVAC Contractors, Energy Management Consultants and Air Balancing.

Appendix B contains a summary of the 51 responses received (18% response rate). The majority of respondents (72%) has experience with HVAC design and typically does both new construction and retrofit work. Figure 6 shows the percent of responses of “Always” or “Often” when respondents were asked to indicate the types of distribution systems they specify, install or operate. Variable air volume systems are by far the most common systems. Interestingly, CAV systems are identified almost as often as variable volume systems. This is in contrast to the data presented in the EPRI study (Pietsch 1991), which indicated a relatively larger increase in VAV systems in large office buildings than central systems over the past 30 years.

The survey asked for the importance of cost factors in selecting a thermal energy distribution system (see Question 5, Appendix B). 41% of the respondents think that first cost of a system is “important,” and 48% of the respondents think that first cost of a system is “very important.” 41% of the respondents think that operating cost of a system is “important,” and 43% of the respondents think that operating cost of a system is “very important.”

Hydronic Systems. Although 72% claim that energy performance is “very important” to them professionally (see Question 12, Appendix B), only 13% of the respondents believe that the use of hydronic thermal distribution systems is a “very important” measure to improve the energy efficiency of thermal energy distribution systems, and 39% of the respondents believe that the use of hydronic thermal distribution systems is a “important” measure (Question 4, Appendix B). Only about one-third of respondents indicated they “often” or “always” specify, install, or operate hydronic thermal distribution systems (Question 3).

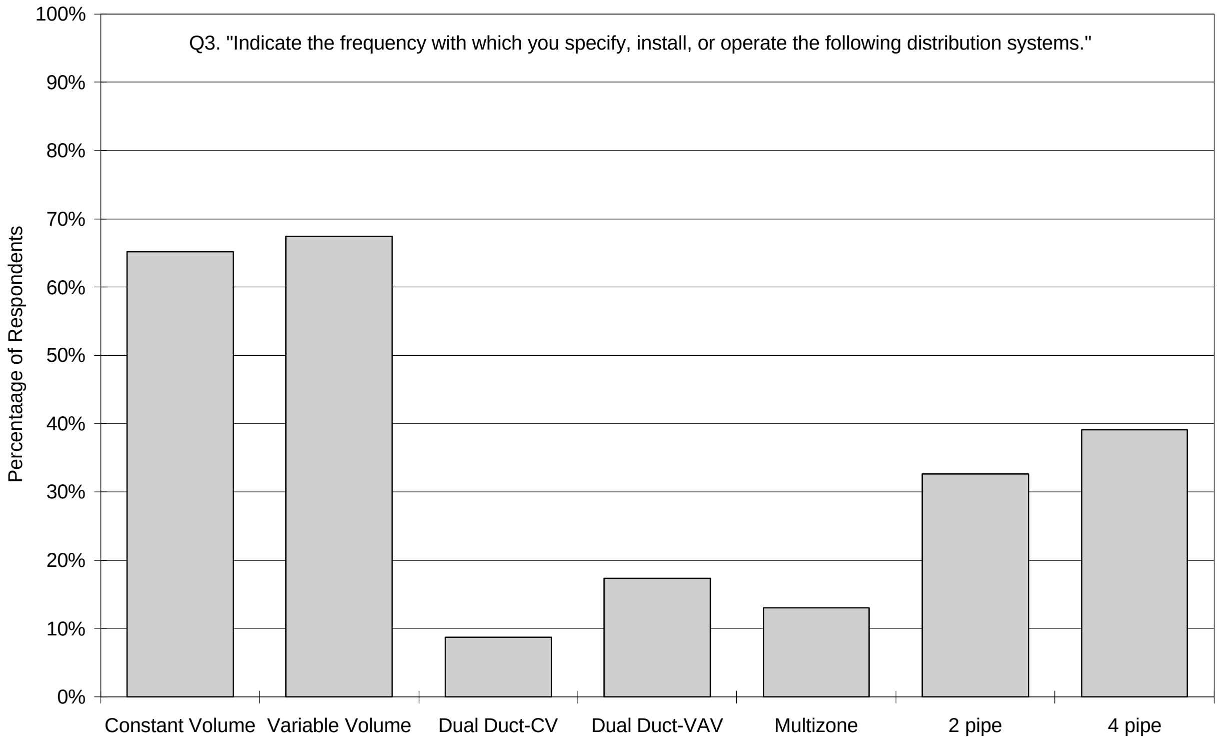

Figure 6. Types of Distribution Systems "Often" or "Always" Specified, Installed or Operated (Q.3)

There are obviously some barriers that prevent a larger market penetration of hydronic thermal distribution. Although there are already several types of hydronic systems in most buildings (i.e., water supply for bathrooms, sprinkler systems, etc.), several of the survey participants expressed their fear of water damage due to leaky pipes. Next on the list of disadvantages are high first cost and high maintenance cost.

Respondents were asked to identify measures to improve distribution system energy performance, to rank factors influencing system selection and to identify barriers preventing greater use of energy-efficient duct systems. The following presents the results.

Measures to Improve Distribution System Energy Performance. System maintenance was the most commonly identified measure used to improve distribution system energy performance. Airside economizers, commissioning, high-efficiency motors, and variable speed fans were the next most commonly identified measures.

Factors Influencing System Selection. System reliability and occupant comfort were most often selected as "very important" with respect to factors influencing system selection. The top three factors influencing the operation of thermal energy distribution systems were, in order of most commonly identified, occupant comfort, operating cost and overall system reliability. Lacks of maintenance and poor initial design were most often identified as poor energy performance for thermal energy distribution systems. Several respondents indicated duct leakage and poor system balancing.

Barriers Preventing Greater Use of Energy-Efficient Duct Systems. Participants were asked to identify three barriers that might be preventing greater use of more energy-efficient distribution systems. The top three responses here were money, money and money. First cost, maintenance budgets, design fees, low energy costs, and lacks of utility incentives were all identified. Overall, survey comments indicate that although the majority of participants are interested and probably capable of designing more energy-efficient systems, the issue of first-cost versus life-cycle cost is likely driving the market towards less efficient designs.

4.2 IN-DEPTH INTERVIEWS

In addition to the mail survey, we conducted two more detailed interviews with HVAC designers. These interviews included a broader range of questions than the mailed survey and were conducted with one engineer who specialized in energy efficient retrofits and the other who had broad experience in large commercial buildings. The purpose of the interviews was to discuss ideas they might have regarding thermal energy distribution systems and to get feedback on proposed research topics for the Phase II of this work.

4.2.1 Interview #1 - Principal, Consulting Engineering Company (Sacramento Area)

In the first interview, we met with the principal of a Sacramento Area consulting engineering company. His firm does engineering design work and facilities engineering for a variety of building types and locations in California. They do considerable retrofit work and, because of that, he encounters many problems with existing systems.

The majority of the systems he sees have problems with air distribution. Most of these problems are in room air distribution. From his perspective, diffuser selection and placement are extremely important and are rarely given the attention they deserve. Many systems deliver more air than required, but may need to do so to provide comfort because of poor diffuser designs. His implication was that improved diffuser designs would allow lower airflow in systems, resulting in lower distribution energy consumption.

His office uses DOE-2, Trane Trace, and Carrier's HAP software for design work. Of these, DOE- 2 is the most sophisticated tool and he likes to use it when he can justify the time required to use it. Trace is used more often, both in his office and generally in the profession, because it is simpler to use, but it does not provide nearly the level of detail as DOE-2 with respect to reporting and energy analysis. HAP is a very simple tool that is quick and easy to use but provides even less information. He has used various duct design tools, but very often still uses hand calculations and intuition.

He indicated that overall he felt Title-24 had resulted in big improvements in mechanical systems. Engineers have generally always tried to design efficient systems, but Title-24 has given them some clout with developers to pay the incremental costs for improved energy efficiency. He did believe that Title-24 places restrictions on creative engineers who could probably do a better job if they did not have to implement all of the prescriptive code requirements. He was unable to give specific examples at the time of the interview, but would be willing to provide some examples.

Commenting on the proposed research topics:

Design Tools. He feels that there is a lot of room for improvements in the existing design tools. Integrating CAD systems with load and energy calculations would be a significant improvement.

Packaged Rooftop Units. There is no way to compete with the low-cost of packaged systems in a very cost-competitive market. He expressed a need for manufacturers to design packaged units for dryer climates such as California. Most packaged units are sized for more humid climates, where the cooling capacity is larger relative to the fan capacity. Because of this, according to his opinion, the packaged units in California are often with excessive cooling capacity.

Increased Hydronic Distribution. He would like to see a study comparing first-costs and life-cycle costs of hydronic vs. air vs. electric distribution. He has serious reservations about radiant cooling systems that provide only ventilation air because he believes that proper air distribution cannot be achieved.

Fan Energy Reduction. He sees good potential here. Talking again about good diffuser design, he stated that air quantities could be reduced if careful diffuser selection was used to achieve good room air distribution. This was particularly true for VAV retrofits, where low airflow can cause dumping in many diffusers, resulting in cold spots and hot spots in the room.

4.2.2 Interview #2 - Principal, Consulting Engineering Company (San Francisco)

The second interview was with a principal of a San Francisco consulting engineering company. He has a broad background in mechanical design, with particular experience in high-end office buildings.

One of his main observations was that there is too great a focus on component efficiency, such as chiller energy efficiency ratio (EER), and not enough on overall system efficiency. It is not enough to just individually specify efficient components. Systems must be designed in the way that their components can work together efficiently. Most of the work they do is based on a fixed fee and he felt that this is typical in practice. The current market is highly competitive, driving design fees down and forcing engineers to spend less time on design.

Another important issue for him is the fact that energy performance is evaluated on design rather than on measured data. As a design engineer, he has limited responsibility for ensuring that the installed system realizes the design intent. He thought that while qualified and competent testers and balancers often do a reasonable job, there are a significant number of less competent testers and balancers that charged lower fees and are less thorough in their work.

He was somewhat skeptical of hydronic distribution systems, primarily due to ventilation issues and maintenance. The trend in large buildings is to use air-handling units on each floor, and provide lots of air. Dilution is perceived to be an effective method of providing good indoor air quality. Providing minimum airflow for ventilation only may lead to lower indoor air quality. VAV systems already have a reputation of causing the perception of staleness at low airflow. Fan coils also introduce maintenance problems, primarily because the maintenance occurs in the occupied space. Induction units are not a good choice due to their inefficiency. He has had no experience with radiant cooling systems, and was unsure about the economics and the ventilation issues corresponding to hydronic systems in general.

The current trend towards floor-by-floor air handling has lowered overall system pressure. Typical floor-by-floor based systems he designs have a 3-inch water pressure drop as compared to the 5-in. water pressure drops in central systems from the late 1980's. He uses very traditional duct design methods, based on 1800-2000 fpm duct air velocity in main trunks and 600-700 fpm in occupied zones. He did not feel that air leakage from ducts was a significant problem since SMACNA specifies fitting qualities and maximum leakage. He assumes a 1 ° F-temperature rise in the ducts when designing a system, and feels that it is a very reasonable assumption. Though they use AutoCAD for drafting in his office, they design their ductwork manually.

4.3 TITLE 24 ISSUES

Our discussions with practicing engineers uncovered some issues associated with the California State Energy Efficiency Standards. The most important issue is that the performance method of compliance does not give credit for reducing fan power. The prescriptive performance method would allow up to 0.8 W/cfm for fan power of CAV systems and up to 1.25 W/cfm for fan power of VAV systems. However, by carefully designing the distribution system and choosing efficient components, energy use by distribution systems can be far less than 0.8 W/cfm. When using the performance method, the standard-design building is simulated using the lesser of the prescriptive value or the proposed design value. This would result in no incentive for the designer to improve fan/ductwork efficiency.

Besides, the performance compliance method also contains a bias against hydronic distribution systems. Since hydronic systems use less distribution energy use than air systems, this advantage of hydronic systems is negated.

In general, the compliance tools do not adequately and accurately address distribution energy consumption in commercial buildings. The difference between a carefully detailed distribution system and a badly detailed system are not modeled, yet this can result in significant differences in building electricity-energy consumption.

5. ENERGY ANALYSES AND ASSESSMENTS FOR THE IMPROVEMENTS TO THERMAL ENERGY DISTRIBUTION SYSTEMS

The issue of reducing space conditioning energy use by improving thermal distribution systems in commercial buildings is significantly more complex than the equivalent issue for residential duct systems. As the costs of performing extensive energy analyses of all of the various thermal distribution system options and issues for commercial buildings are prohibitive, our energy analyses are selective and limited in scope. The decision of which issues to examine was made based upon: 1) stock characterization of energy use per system type, 2) identification of potential savings pathways, and 3) the degree of existing knowledge related to a particular issue.

Energy analyses of varying complexity and sophistication have been performed for three distribution-system issues: 1) thermal loss reduction for rooftop packaged units, 2) direct means for reducing fan energy (including the relative energy consumption of hydronic versus all-air distribution systems), and 3) fan-power impacts of supply-duct thermal losses to suspended-ceiling return plenums.

5.1 CASE STUDIES: THERMAL LOSS IN ROOFTOP PACKAGED SYSTEMS

5.1.1 Thermal Losses from Distribution-system to the Environment

In general, since most commercial-building distribution systems are, in principle, located within the conditioned space (e.g., supply ducts in return plenums), thermal losses to the environment are expected to be smaller from distribution-systems of commercial buildings as compared to those of residential buildings.

However, small commercial buildings, such as small single-story retail stores, low-rise offices, can have thermal losses similar to those encountered in residences. The old small-commercial buildings commonly contain a fair number of ducts located outside the building shell. The vast majority of small single-story commercial buildings, as well as most top stories of large commercial buildings, has the ducts located in a suspended ceiling just below the roof deck. The relative communication between these ceiling plenums and the outdoors or conditioned spaces is variable, ranging from well-ventilated attic-like structures to substantially airtight well-insulated roof decks. To study these effects, in particular the direct impacts of thermal losses from ducts, case studies were performed in two small retail stores in California.

5.1.2 Case Study of Rooftop Packaged Units

Three rooftop package units in two retail stores in a Marin County strip mall were submitted to short-term diagnostic measurements and monitoring. Two-to-three days of diagnostic measurements were performed, followed by one-to-two weeks of monitoring. Both stores had relatively new 5-ton rooftop packaged air-conditioner/furnaces, all with ductwork running on the roof and in the dropped ceiling areas. One store had a single rooftop packaged unit while the other store had two units. Detailed descriptions of these systems, the stores in which they were located, and the results of the field measurements are presented in Appendix C.

The studies included measurements of 1) distribution-system total leakage area; 2) distribution-system leakage to outside; 3) the relative leakage on the return and supply sides of the fan; 4) air flows through all diffusers and registers; 5) building envelope leakage; 6) fan energy and compressor energy consumption; 7) duct and plenum pressures during normal operation and 8) distribution system temperatures (all diffusers and plenums) during normal cyclic operation.

The defining characteristic of the three distribution systems tested was their similarity to residential HVAC equipment. The units and fans cycled on and off based upon a simple thermostat, and the distribution fan energy represented less than 15% of the energy consumed by the complete unit (i.e., fan and compressors). Other characteristics that are very similar to residential systems included the pressure rise across the fan (0.5 iwg, or 125 Pa), the pressure drops through the ducts, and the conduction losses on the supply side.

5.1.2.1. HVAC Leakage

The small commercial duct systems measured had leakage levels similar to or higher than that of residential ducts. Specific duct leakage area for these three systems ranged from 1 to 3 cm^2/m^2 of floor area at 4 Pascal, as compared to 1 cm^2/m^2 of floor area for residential systems (Modera et al. 1991).

For all three systems measured, a large fraction of the air leakage was to outside. Since the outside ducts had no visible leaks and all joints were heavily taped and covered with mastic, the likely location of this leakage is at the packaged unit box. Effective leakage areas (ELA) of ducts to outside ranged from 210 to 247 cm^2 at 25 Pascal (Appendix C). The pressure differentials between outside and return or supply duct were found to vary between 15 and 90 Pa, however the sum of these pressure differentials was constant, as the fan generally produced a pressure rise of 125 Pa.

Assuming an outside temperature of 35^{o}C, a supply plenum temperature of 10^{o}C, and a room temperature of 25^{o}C, the typical leakage levels to outside together with an average pressure differential of 60 Pa translate to a thermal energy loss of 3 to 7.5 kW (10 to 25 kBtu/h) depending upon whether the leaks are on the return or supply side of the fan. For the 5-ton units studied, these losses should result in an increase of system on time of 17% to 43% under the assumed temperature conditions.

5.1.2.2. Radiative / Conductive Gain

The case study found that approximately 35% to 65% of the total duct lengths were located outside the conditioned space for the three systems. The magnitude of the conduction losses from these ducts was measured by monitoring temperatures along duct and registers. Two means of reducing these losses were examined.

For one long duct section of the South system in Store A (4 m [13-ft] of insulated [1”] exterior ductwork), the conduction losses were 20% of the equipment output to that duct. Other registers for the South system in Store A had smaller losses to outside. The North system in Store A had very little exterior ductwork, less than 1.2 m (4 ft) of exterior ductwork per run, and had negligible conduction losses to the outside. In Store B, one register was attached to an uninsulated 2-m (6-ft) duct that had 29% conduction losses to the outside. All of the other registers in Store B branched off of a 5.6-m (18-ft) exterior duct that had 8% conduction losses. At the time of these measurements, surface temperatures were as high as 65^{o}C on the sunlit surfaces of the ducts, whereas the roof-side surfaces were 35^{o}C, and the outdoor air temperature was 33^{o}C.

The magnitude of the measured conduction losses suggests that reducing the radiative gain of the packaged unit box and the exterior ducts be worth investigating. One method to consider is adding more insulation to the ducts or to the packaged unit box. The ducts had 1” thick black foam internal sound insulation. The packaged units had 3/4” of internal insulation. Increasing this to 2” and assuming the surface temperature of the ducts and box didn’t change would save 54 watts on the surface of the box and 200 watts on the surfaces of the ducts. These numbers are made assuming an internal surface area of 1.8 m^2 for the box and 13 m^2 for the ducts, and that the high temperatures exist on 50% of the duct area. Another option is to apply white paint to the surfaces of the box and the ducts. The surface temperature would be reduced to within 5^{o}C of the air temperature and there would be energy savings of 70 and 250 watts for the box and ducts, respectively. A reduction in load of 320 watts would reduce the on time of a 5-ton system by 2%.

5.1.2.3. Air Flow

All of the units tested with airflow rates ranging from 550 to 700 cfm per ton. These exceeded the 400 cfm/ton value, which is usually a fan-flow limit (by the rule of thumb) for achieving good residential air conditioner cooling coil performance; however, it is worth noting that this guideline is for wet coils which is not often the case in California.

In general, excessive restrictions to fan flow would significantly lower the equipment efficiency. Poor duct system design and/or installation can reduce airflow across the heat exchangers, resulting in lower heating and cooling equipment efficiencies. These problems are likely to be more pronounced in smaller lower-cost buildings where: 1) the mechanical budgets are minimal, 2) there is rarely an engineer involved, 3) most decisions are first-cost based, and 4) commissioning is generally not performed.

5.1.2.4. Fan Overrun

Another thermal issue observed in the field study was fan overrun. As California climates are relatively dry, the cooling coils are often dry. Fan overrun refers to continuous fan operation even after refrigerant-induced cooling is called off. In this case, fan overrun can transfer an additional amount of net cooling to the space without inducing latent heat. In the case study, the amount of energy recovered during fan overrun is calculated based on the difference between room air and supply air temperatures at the cooing coil.

Fan overrun behavior was studied on the hottest day of the monitoring period. The average daily outdoor temperature was 25.8°C (78°F), while the average outdoor temperature was 35.4°C (96°F) during the period of cooling operations (10:00 AM to 6:00 PM).

The South system in Store A, having a capacity sized close to the load, cycled three times during the morning hours (10:00 AM to 12:00 PM) and was basically on constantly from 12:00 until 6:00 PM with only one fan overrun. On average, 520 kJ of energy were recovered per overrun cycle for the South system, which resulted in a total of 2080 kJ cooling for the day.

The North system in Store A cycled approximately three times per hour from 10:00 AM to 6:00 PM. Energy recovered during each overrun period was lower than that of the South system due to operation pattern and heat-takeback effect, which occurred when the supply air temperature was greater than the room air temperature. For the North system, energy recovery takeback occurred when the outside temperature is over 32°C [90°F]. On average, taking the takeback effect into account, 80 kJ of energy was recovered per fan-overrun for the North System, which resulted in a total of 1910 kJ for the day.

The two systems in Store A had cooling fan overruns ranging from 105 to 160 seconds per cycle. Based on this analysis, the South system fan overrun was sized correctly for the actual temperature conditions and airflow rates. While the overrun period duration were approximately the same for both systems, the North system was penalized by the takeback factor due to longer off periods between cycles.

This analysis of fan overrun assumes that all energy remaining in the cooling coil and exterior ducts is lost when the fan turns off. It does not include the reality that the cooling coil may still be cool when the next cycle starts, and that thermosiphon flows could pull some of the stored energy into the conditioned space. A detailed analysis of a similar situation for residential duct systems suggested that somewhat less than 50% of the energy stored in the duct system would be recovered into the conditioned space without fan overrun.

We observed no fan-overrun with the Store B system.

5.1.2.5. Envelope Leakage

Envelope leakage was one quantity measured in this case study that differed significantly from the residential studies. The building envelopes have ELAs of 10 to 12 cm^2/m^2 of floor area at 4 Pa, significantly larger than found in the residential envelopes. A field study of California residences found a mean of 3.9 cm^2/m^2 of floor area for post-1979 houses and 6.1 cm^2/m^2 for pre-1980 houses (Modera et al. 1991). Visual inspection of the two case-study buildings showed that little care had been taken to keep the envelope tight. In Store A, a false “roof” was constructed above the windows, which left gaps of up to 0.5 inch. In both stores, it was possible to see light from outside coming in at the eaves. Both stores also had oversized roof penetrations for the exhaust vent ducts. The roof, ceiling and windows all seemed properly sealed, so correcting problems with eaves and exhaust vents would reduce the envelope leakage considerably. Although this is not a distribution-system problem, addressing envelope leakage could be an important addition to DSM audit/retrofit packages for small commercial buildings.

5.1.2.6. Issues and Possible Solutions

These case studies found problems with several aspects of two rather typical light commercial installations. The most significant problem is that large leakage areas of ductwork were exposed to the outside. The leaks found in these systems have the potential of increasing the on-time system operation by 17% to 43%, depending on whether the leaks are on the supply or return side of the system. Leaks of this nature are a serious problem, which are not always easy to identify and/or fix.

Poor insulation on the ducts and the packaged unit coupled with surfaces with high absorbtivity yield high surface temperatures on those exposed to sunshine. This radiation gain potentially contributes to an increased on time of 2%. Repainting the surfaces with a low-absorbtivity paint and adding more insulation to exposed ducts and rooftop units are straightforward repairs. The high ELA found in the building envelope potentially affects thermal distribution systems, particularly in buildings with unducted ceiling returns.

5.2 DOE-2 SIMULATIONS: TRANSPORT (FANS AND PUMPS) POWER REDUCTION

Space cooling is a key contributor to peak electricity demand and, as shown in Figure 1, consumes about eight times as much electricity as space heating in California commercial buildings. Fans and pumps are also important contributors to the electricity demand and consumption. They consumed five times as much electricity as space heating.

Two types of analyses based upon DOE-2 were conducted to investigate fan energy savings: 1) thermally perfect systems, within which the energy put into the thermal distribution system is 100% delivered to the zones; and 2) thermally imperfect systems, within which the energy put into the thermal distribution system is lost along the way through conduction and duct system leakage. The DOE-2 analysis for thermally perfect systems compares fan and pump energy consumption for different types of HVAC systems. The thermally imperfect systems analysis is discussed later and is used to look at the impacts of thermal losses/gains through duct walls, and duct leakage at various locations on HVAC system energy consumption and demand.

5.2.1 Prototypical Building

A prototypical office building (Huang et al. 1991) was used to investigate the characteristics of several thermal distribution systems. This same building has already been used for an assessment of cold air distribution systems (Bauman et al. 1992).

The prototypical building has three floors of 5,570 m^2 (60,000 ft^2) each with steel and spandrel glass walls (U = 0.76 W/m^2K, or R-7.5). Thirty percent of the wall area is double-paned glass, having a transmittance of 62%. The roof is steel and metal decking under tar and gravel and is insulated to U = 0.36 W/m^2K (R-15.8). The interior floors are carpeted 10-cm (4”) concrete slabs and interior walls are steel studs with gypsum board.

For simulation purposes, each floor is divided into a single core zone and four perimeter zones. The perimeter zones are all 4.6 m (15 ft) deep and the core is comprised of interior offices, hallways, elevator shafts, and lavatories. There are 0.9-m (3-ft) high utility plenums between floors and below the roof with bases of lay-in acoustic tiles, resulting in a 2.6-m (8.5-ft) floor-to-ceiling height. Thermal transfer takes place between core and perimeter zones and between zones and plenums above.

Scheduled use of the building is the standard five day workweek with hours of 8 a.m. to 5 p.m. Limited use is assumed during the weekends and evenings. Full occupancy is assumed on workdays with one occupant for every 18.6 m^2 (200 ft^2) of floor area. The lighting load at full occupancy is 18 W/m^2 (1.67 W/ft^2) and equipment loads are 8.6 W/m^2 (0.8 W/ft^2). In addition, maximum domestic hot water use of 21 kW and elevators that draw 57 kW when in full use is assumed. Infiltration is assumed to be 0.35 air change per hour (ACH) in the perimeter zones and 0.25 ACH in the core zones when the building is unoccupied.

5.2.2 Thermally Perfect Systems

Data from DOE-2 simulation results compare the fan and pump power consumption related to three different thermally perfect HVAC system types: 1) base case, variable air volume (VAV), 2) 4-pipe fan coil and, 3) packaged constant-air-volume (CAV).

There are five individual systems in each of the three cases. Of the five individual systems, one serves the core zone and the remaining four systems serve each of the four perimeter zones (one for all North zones, one for all East Zones, etc.). For the comparison of the thermally perfect systems, none of the systems are equipped with airside economizers, which results in a tendency to over-predict the savings potential of hydronic systems. The outside air supply for ventilation purposes is based on ASHRAE Standard 62-89, which specifies 10 l/s (20 cfm) per person for office space (ASHRAE, 1989)[1]. This leads to 3,000 l/s (6,000 cfm) outside air supply. All systems provide outside air for the time the building is occupied. None of the systems have any leakage or conduction losses from the ducts.

VAV (Base Case) System. The VAV (base case) system is represented by a set of five conventional VAV-systems working with 13°C (55 ° F) supply air temperature. In order to provide the amount of energy needed to condition the building, the VAV systems recirculates the amount of air required above the minimum outside airflow. The total amount of air delivered to a given space varies depending on the amount of heating or cooling and ventilation required for that space. Cooling is provided by chilled water to the air handler. Heating is provided, when needed, by hot water to the terminal reheat coils. Air returns through separated plenum spaces via return air fans to the individual systems. Friction losses are 500 Pa (2 iwg) for supply and 380 Pa (1.5 iwg) for return air.

Four-Pipe Fan-Coil System. The fan coil system is able to provide heating and cooling to a number of individually controlled zones simultaneously, depending on each zone's requirements. Each fan coil unit is equipped with separate coils for heating and cooling. Coils are connected to two separate piping systems, one circulating cooled fluid and one circulating heated fluid during the operational hours. Each unit provides a fixed quantity of outside air for ventilation along with recirculated conditioned air, conditioned by the corresponding heating or cooling coils. Friction losses are 57 Pa (0.23 iwg) for the fan coil unit. The calculations assume negligible duct runs,

[1]The new ASHRAE Standard 62-1999 (ASHRAE 1999) prescribes the same supply rate of acceptable the outdoor air required for acceptable indoor air quality. 10 l/s is chosen to control office space to dilute human bioeffluents and other contaminants with an adequate margin of safety and to account for health variations among people and varied activity levels.

causing no additional friction losses. The cooled fluid and the heated fluid loops have pressure drops of 90 kPa (360 iwg) each.

Packaged Constant-Air-Volume System (CAV). The CAV systems (five in total) cool the air by direct expansion of a refrigerant and heat the air with gas. The system provides individual ducting from the air-handling unit to each of the individual spaces being served. Outside air and recirculated air are mixed together before entering the air-handling unit where the air is conditioned to meet the temperature requirements of the zone. The return air is not ducted, but individual zone return plenums are separated from each other. The friction losses for the supply part of the system are 565 Pa (2.27 iwg). The return part of the system has a pressure drop of 174 Pa (0.7 iwg).

5.2.3 Modeling Results - Thermally Perfect Systems

Table 22 shows the fan/pump and cooling energy consumption for the thermally perfect duct systems (i.e., no leakage or conduction losses) for the prototypical building located in San Jose and Fresno.

Of the three systems, the hydronic fan-coil case has the lowest fan/pump electricity consumption at both locations. To be more specific, Table 23 presents the relative fan/pump energy consumption based on the CAV system. Fan and pump energy savings for the fan coil system are about 80% when compared with the CAV system, and are approximately 60% when compared with the VAV system. Fan energy consumption for the VAV system is approximately 47% of the fan energy consumption for the CAV system.

The cooling energy consumption is highest for the CAV system. The cooling energy use for the VAV and hydronic systems is roughly 50% of that of the CAV system.

Overall, fan and pump energy consumption is approximately 20% of the total fan, pump and cooling energy consumption for the hydronic system, 35% for the VAV system and 40% for the CAV systems (see Table 24). For the two climates, hydronic systems use 74% of the fan and cooling energy used by VAV systems and 40% of the energy used by CAV systems. VAV systems save 48% in fan and cooling energy consumption when compared to CAV systems.

Compared with the VAV case, the CAV case uses between 70% and 90% more electricity to cool the building. During occupied hours, the CAV systems provide the same amount of air to the zones at all times and the heating/cooling ability of the supplied air is modulated by changing the supply temperature. As the CAV systems constantly provide the same amount of air, the system fans run at a constant speed and therefore transfer a constant amount of cooling/heat to the spaces. The additional heat supplied to the air alone does not fully explain the high cooling power consumption of the CAV systems. The large difference in electrical consumption between VAV and CAV systems is also due to different chiller COPs. While direct expansion systems (used for the constant-volume simulations) typically have a COP of 2.5 to 3.0, chilled water systems (used for the VAV and hydronic simulations) reach COP values of 5.0.

Table 22 HVAC System Energy Use (Prototypical Building)

System Type	Fan/Pump Energy Use [MWh]		Cooling Energy Use [MWh]	
	San Jose	Fresno	San Jose	Fresno
VAV	155	181	288	333
4-Pipe Fan Coil	63	74	265	307
Packaged CAV	349	369	494	625

Table 23 Fan/Pump Energy Consumption Relative to the Packaged CAV

System Type	San Jose	Fresno
VAV	0.44	0.49
4-Pipe Fan Coil	0.18	0.20
Packaged CAV	1.00	1.00

Table 24 Ratio of Fan/Pump Electricity Consumption to the Total of Space Cooling and Fan/Pump Electricity Consumption for HVAC Systems (Prototypical Building)

System Type	San Jose	Fresno
VAV	0.35	0.35
4-Pipe Fan Coil	0.19	0.19
Packaged CAV	0.41	0.37

5.2.4 Non-Thermal Imperfections

As documented in the CEC forecast data (Table 10), approximately half of California commercial buildings have CAV systems, with fan energy being the bulk of the distribution energy consumption. The potential for saving fan energy is perhaps well understood, but only a small portion of the savings has been realized. Anecdotal evidence suggests that many commercial buildings are not operating at or even near their optimum fan energy efficiency. In spite of this, few building owners or operators have the tools and/or the experience to address this problem. The energy efficiency of an air distribution system is affected by several factors, three most important of which are air volume, system pressure, and efficiency of fan/motor.

Excessive Air Volume. Given a certain duct system and an operating pressure, the fan energy required transport air is theoretically proportional to the cube of the airflow rate. Although this relationship does not strictly hold in a real system (e.g., a VAV system where dampers could move and could change duct pressures), large savings are possible through reduction of airflow. Many HVAC systems are designed with large factors of safety, and may distribute much more air than needed to meet the buildings' heating, cooling or ventilation needs. VAV systems attempt to address this issue during non-peak operation. A reduction in total air volume through the fan will yield both direct fan energy savings and indirect heating and cooling energy savings due to reductions in fan heat, reheat, and conditioning excessive outside air.

Many buildings have systems that operate well beyond the hours necessary for providing thermal conditions and ventilation for its occupants. A study of the operation of seven Minnesota office buildings (Herzog et al., 1990) found that the supply fans ran from 15% to 100% longer than their designed schedule due to a range of problems the operators were unaware of. In most of the cases, identifying the problem through simple tracking of the actual operation and bringing the problem to the attention of the operator was all that was needed to correct the problem.

Excessively High Pressure. Some systems are designed with unnecessarily high-pressure drops. The major system components affecting pressure include filters, coils, heat exchangers, duct fittings, dampers and diffusers. Pressure drops increase as filters and coils become dirty, making regular maintenance important for these components. Reducing the face velocity at which the air passes through filters and coils is an effective way of further reducing the pressure drops. Changing damper positions in the system can also provide significant savings.

Poor Fan Efficiency. Most HVAC systems use centrifugal fans, which have peak mechanical efficiencies in the range of 60% to 80%. The actual efficiency of the fan and the quantity of air it delivers are affected by the static pressure of the system. Proper fan sizing is important to achieve optimum fan efficiency. For existing buildings, replacing existing motors with more efficient motors is often cost effective. Proper motor sizing is equally important.

5.3 THERMALLY-IMPERFECT SYSTEMS

Duct performance of commercial buildings has been neglected in the field, in part due to the lack of moving parts, which would eventually alert maintenance personnel of problems. Once a building has been commissioned and accepted by the owner, it is quite likely that the duct system will essentially be forgotten. Very often, there are design inadequacies, assembly errors, and improper material choices that can easily be overlooked during the commissioning of a building. One simple reason for this is that inspectors, in many cases, are not able to gain access to building areas where flaws have been concealed by the interior finish work. Often, a careful examination and performance testing of the delivery system would indicate serious shortcomings.

More difficult to uncover are the leaks which quietly and slowly develop over time and go unnoticed. Deterioration of thermal barriers frequently occurs where the outer wrap is exposed to the weather. In rooftop areas that do not have routine inspections, whole sections of deteriorated or missing exterior insulation may not be replaced on rooftop ductwork. Due to building shifts, vibration, and remodeling, as well as improper assembly, physical leaks can develop over time which do not cause sudden or obvious changes in system performance.

Duct leaks may be found in one or more of several air delivery systems within the same building, and both thermal and physical leaks may coexist. In larger multistory buildings, the ductwork is almost always within the building shell. Single story commercial retail outlets such as shopping malls are frequently equipped with rooftop HVAC units and exposed ducting. There will therefore be a different system performance penalty as a result of a thermal or physical leak that leads directly to the ambient as opposed to one which leaks within the building shell.

As noted in the previous discussion, both thermal losses by conduction and air leakage from air delivery systems can impact space conditioning energy use. For the common commercial-building design in which the supply ducts are located within the ceiling plenum space, the largest impact of these losses is on fan energy use. An examination of this conventional wisdom, that thermal losses to ceiling return plenums do not have significant energy implications, has been performed. The basic premise behind the analysis performed is that energy lost to a ceiling return plenum from the supply ducts increases the amount of air that must be pushed through the fan towards the conditioned spaces to meet their loads, and thus increases fan energy consumption. This short-circuiting not only directly impacts fan-power and run-time, but it also increases the cooling load due to the extra heat generated by the fan.

The analysis performed is based on a number of assumptions, including the following:

(a) Approximately 25% of the thermal energy entering a supply duct are lost due to leakage and conduction before it reaches the diffuser.

(b) Convective transport through the return plenum dominates conduction through the drop ceiling to the conditioned zones.

(c) There is no penalty associated with simultaneous heating and cooling losses to the return plenum.

(d) The pressure drops through the distribution system and the resulting pressure differential seen by the fan are proportional to the square of the flow rate through the ducts.

Based upon these assumptions, the first step is to estimate the increase of required fan power to compensate thermal losses in the distribution system. A 25% thermal loss implies that 1.33 times the flow is needed to meet the load in the zone given that the supply air temperature does not change. This translates into 33.3% more airflow through the fan. For the same duct system, the overall pressure differential seen by the fan should increase with the velocity (or flow rate) squared. Thus, the 33.3% increase in flow, created by the 25% loss, would increase the fan pressure by $(1.33)^2$, which translates to a 78% larger pressure differential. Fan power scales with the product of the pressure differential and the flow, namely the cube of flow. Thus, a 25% thermal loss would in theory result in as much as 2.37 times of original fan energy required to overcome the cooling load. Therefore, by eliminating the 25% thermal loss would have reduced the fan power required by 58%.

In addition, the extra fan energy computed above creates an additional cooling load. Assuming that the fan represents 14% of the total cooling load seen by the air-conditioning system (Akbari et al. 1993) and that this is already reflected in the 25% thermal loss in supply-duct, then the 58% fan power reduction induced by eliminating the thermal losses from ducts corresponds to a reduction of building cooling load by 8% that would have been created by the bigger fan. As derived from Figure 1, fans and pumps consumes approximately 40% of the total electricity energy used for cooling and ventilation in California commercial buildings. If we further assume fans used approximately 40% of the total electricity energy used for cooling and ventilation, and 60% for cooling, then eliminating the thermal losses would result in a total of 28% energy savings for space HVAC cooling and ventilation.

The above analyses are by no means conclusive, however each of the assumptions made is testable, and some of these have already been examined. For the first assumption of the 25% thermal loss, field measurements on a conditioned-space uninsulated duct system in a house indicated total thermal losses (leakage plus conduction) of 30-40% (Jump et al. 1993). The uninsulated system tested in the residence was also typically installed in small commercial buildings. The strip-mall packaged-unit measurements yielded conduction losses of 4-14% for insulated ducts within the conditioned drop-ceiling return plenum in this study. In addition, a diagnostic survey of four air distribution systems indicated supply air leakage levels of 30-40% of the fan flow (Jansky et al. 1993 & 1994).

The second assumption assumes almost no conduction from return plenums to the “conditioned space” through the dropped ceiling, while such conduction will generally reduce the impact of duct thermal losses. In the extreme case, if all of the cooling losses from supply ducts were conducted to the space, clearly no excess fan power would be required. However, conservatively assuming that a typical recirculation system operates at 5 air changes per hour, and that the ceiling plenum is one third the height of the room, then the return flow represents approximately 15 air changes per hour in the plenum. This air exchange rate can be translated into an effective conductance, which can be compared with the overall conductance of the ceiling. Such a comparison suggests that for an R-3 ceiling ($U = 1.9\ W/m^2K$), approximately 70% of the cooling or heating lost from the supplies will be returned to the fan, the remainder being conducted through the ceiling. Modified DOE-2 simulations as discussed in Section 5.3.4 verify this.

The impact of the third assumption (no penalty with simultaneous heating and cooling losses to the return plenum) is that the energy lost from cooling ducts could possibly increase the heating load in perimeter zones, and that the losses from the heating ducts would increase the cooling load in the core zones. As will be presented and discussed based on the DOE-2 simulations, deviation from this assumption is not a large effect, although this could somewhat increase the energy impacts of the supply-duct thermal losses.

Concerning the fourth assumption (fan power scales with the cube of the flow rate in ideal situation), the effect can most clearly be seen with a reduction in supply air volume. Namely, if the thermal losses from a duct system were reduced or eliminated, much less flow would be required to meet the zonal loads. Despite of great potentials, dampers in VAV system would have to close down during operation, thus changing the system operating pressures. This would cause fan energy reduction due to lower flow, but not following the “cube-law.” To realize the cube-law savings, we need to change the setpoint of the static operating-pressure regulator at the fan, which is limited by the minimum operating pressure requirement of the VAV boxes. For CAV systems, it is clear that an existing CAV system would not automatically adjust its airflow to accommodate a change in required airflow to the zone. Rather it would usually reset the supply air temperature, and thus no fan-power savings would be realized simply by retrofitting the ducts for CAV systems. For such CAV systems, the fan-power and energy savings could be realized by better design in new construction and systems, or by changing the fan speeds or replacing the fans in existing buildings.

To further investigate the validity of this intuitive analysis, modified DOE-2 simulations of the prototypical office building were performed. These simulations were set up specifically to analyze the fan power impact of thermally imperfect duct systems.

5.3.1 DOE-2 Simulations (Thermally Imperfect Systems)

A series of full-year DOE-2 simulations were performed for a modified base case (VAV) system using the same three story office building and climates as for the traditional fan power analyses (thermally perfect systems) presented earlier.

For the thermally imperfect systems analysis, the base case configuration uses five VAV systems with airside economizers. (Economizers were not used in the thermally perfect systems analysis.) The VAV systems were also altered to simulate various duct leakage and conductive exchange situations. Some variations required extensive repartitioning of the building interior. By adjusting the equipment capacities to assure that all zone temperatures remained within their thermostat throttling ranges, all heating and cooling loads were satisfied for all of the various duct system configurations.

A second base case using a single whole-building HVAC system was established to explore the effects of air leakage. The building shell, internal activities, and operating schedules were the same as for the base case prototype (thermally imperfect VAV system). However this base case system was modeled as a CAV system and, as such, has different energy usage than the five-system VAV base case where each system serves a set of zones (core, north, south, east, west) which have quite similar loads over time.

5.3.2 Description of Faulty Duct Parameters

The impacts of thermal conduction and leakage from ducts were examined separately. Thermal conduction from the ducts was modeled in DOE-2 as a 2.8 ° C (5 ° F) temperature rise in cooling ducts between the supply plenum to the terminal registers. The supply-air leakage fraction was chosen to be 30 percent from supply fan to terminals. The air leakage through distribution systems was simulated in two ways: 1) airtight throughout main ducts, and due to improper assembly, 30 percent of the air escapes just before the diffuser, and 2) air leaking uniformly along the entire ductwork from the supply fan to the terminals. The latter is more likely to be observed and has a smaller impact on the required fan power. These leaks were treated both as losses directly to ambient and as losses to the return plenum.

DOE-2 system input instructions allow simulation of both thermal and physical losses only to the building exterior. However, the manner in which these calculations are done is not consistent with normal control system reactions. The response to increased loads due to a thermal leak by DOE-2 is to adjust the supply air temperature at the coil, and to maintain the same fan flow. This means that, in the case of an increased cooling demand by zones served with a leaky duct, the coil temperature would drop to provide additional cooling. This would require a lower chiller suction temperature thus decrease its coefficient of performance (COP), while fan loads would not change.

5.3.3 Simulation Strategy

As DOE-2 is not set up to simulate thermal losses from ducts within the building shell in a straightforward manner, the following approaches were used to simulate air leaks and conduction losses from ducts in return plenums. To compensate for supply ducts that leak 30% of their air flow to the return plenum, the VAV system was oversized by 43% and the excess airflow was injected, in the form of a sensible heat loss, into the zone meant to represent the return plenum. The effect of the escaped supply air upon the loads of zones, which are in thermal contact with the return plenum, but not served by the leaking system, is modeled by their thermal contact with the well-mixed return plenum zone. Two sets of internally leaking systems were simulated. The core system was allowed to leak while the perimeter systems were airtight, and vice-versa. Hand calculations using two sets of hourly zone and plenum air temperatures agreed with the simulated heat transfer between the return plenum and the various zones. These two supply-duct leakage configurations were also simulated for the case of leakage directly to the outside environment.

To simulate the fan energy response to a 2.8 ° C (5 ° F) temperature rise of the cooling air passing through the duct system, a supply air temperature at the cooling coil was specified to be 2.8 ° C (5 ° F) higher than the desired setpoint. The additional flow required to satisfy zone loads was obtained by subtraction and agreed with hand calculations. In a real system, the coil would supply air at the setpoint. A fictitious process load was created in the conditioned zones that would increase the required airflow to that calculated for the 2.8 ° C (5 ° F) higher temperature. To conserve energy, an identical process load of opposite sign was simultaneously placed in the return plenum zone. Both loads only exist when the system operates. The ducts were considered to be airtight and fan pressures were adjusted according to the fan laws. This was first done for the core system, with the impact of the escaped cooling capacity examined for the perimeter zones as well as the core zone. To verify the simulated effects, the perimeter zones were decoupled from thermal contact with the plenum by reconfiguring the return plenum zone such that each VAV system had its own return plenum. The case of thermal conduction directly to the outside was also simulated for both the coupled-return-plenum configuration and the decoupled configuration by removing the process load from the return air plenum.

5.3.4 Modeling Results - Fan-Power / Thermal Loss Interactions

The results of all of the DOE-2 simulations (11 cases each in Fresno and San Jose) are summarized in tabular form in Appendix D, and the highlights of these results are presented below. For the purposes of this analysis, all simulation results are compared with a base-case configuration for which the air distribution system is assumed to have no imperfections in its performance, and an air-side economizer is used for all configurations whenever ambient temperatures are less than 21 ° C (70 ° F). The first set of comparisons is presented in Table 25, which summarizes the impacts of air leakage from the perimeter duct systems. The impacts of air leakage or thermal conduction gains associated with the core-system ductwork are summarized in Table 26.

The fan flow results in Table 25 show a 43% increase in flow through the perimeter zone fan due to 30% leakage from the supply ducts, and more importantly, show the large increases in fan power (peak demand and energy consumption) associated with leakage. Much higher pressures required to increase airflow through the fan induce large increases in energy consumption. The DOE-2 simulations were performed using two different assumptions: 1) duct leakage distributed evenly throughout the duct system; 2) all leakage occurred at the diffusers.

Table 25 illustrates smaller fan energy impact is expected when the leaks are uniformly distributed. Concentrating all of the leaks at the diffuser results in higher velocities throughout the duct system, thereby increasing the friction and resulting in higher fan pressure and more power consumption.

The percentage increase in maximum fan energy, which should be similar to the increase in peak electricity demand, is essentially equal to the increase in fan energy consumption, as would be expected for a fixed percentage leakage rate. However, it is worth noting that air leakage fraction is not likely to be constant, but rather to vary with the fan flow raised to the power 0.2 or 0.3. This results from the fact that duct leaks have flow pressure exponents between 0.6 and 0.65, while fan flow have flow pressure exponent of 0.5. This implies larger impact of duct leakage on peak fan power than its impact on fan energy use for a VAV system.

Table 25 Impacts of Air Leaks from Perimeter-System Supply Ducts in Fresno (San Jose)

Parameter	Perimeter System Base Case (VAV) No Leakage Fresno (San Jose)	Percentage Change From Base Case [%] 30% Air Leakage Supply Ducts to Return Plenum	
		Uniform Leakage Coupled Return Fresno (San Jose)	Leakage at Diffusers Coupled Return Fresno (San Jose)
Perimeter Fan Air Flow [cfm]	65,500 (57,400)	43% (43%)	43% (43%)
Perimeter Fan Consumption [MWh]	115 (101)	111% (113%)	173% (175%)
Perimeter Fan Peak Draw [kW]	39 (35)	113% (109%)	174% (171%)
Annual Perimeter Cooling Load [MBtu]	1,890 (805)	8% (-14%)	14% (-7%)
Annual Core Cooling Load [MBtu]	2,280 (1,225)	-6% (-4%)	-6% (-3%)

As the cooling energy in the ducts is lost directly to ambient, air leaks through supply duct to the ambient obviously have the largest impact on thermal loads and cooling energy use. Leakage to the return plenum by the perimeter system can either increase or decrease the cooling loads for the perimeter systems, depending on the climate. Since economizer control is based on the mixing temperature (return and outside air), the cooler return temperature induced by leakage tends to increase economizer use when outside is relatively cooler, i.e., San Jose. This may reduce the cooling load for the perimeter system. However if outside air is not cool, i.e., in Fresno, the cooling load for core system still increased since the economizer would not operate as often due to higher ambient temperature, even though the cool supply air leaking into the plenum contributed to reducing mixing air. Leakage to the return plenum by the perimeter system will usually decrease the cooling loads for the core system.

Table 26 shows the comparisons of core fan energy and demand impacts resulted from 30% air leakage, 25% thermal conduction loss. Uniform leakage along supply duct is assumed in Table 26. Consistent with the perimeter zone results, the fan flow results for core-system supply ducts in

Table 26 show a 43% increase in fan flow due to the 30% leakage from the supply ducts. Assuming that a temperature rise of 2.8 ° C (5 ° F) from supply plenum to terminals (registers) corresponds to a 25% of thermal loss by conduction, and that increased flow would not change the conduction loss, an increase of 33% in supply air flow is expected to compensate for the conduction loss.

Since the magnitude increase in fan power is much larger than the increases in fan flow itself induced by air leakage or thermal conduction losses along the supply duct, adding the fraction that the friction losses associated with the increased flow exist for the entire duct system, the 25% conduction losses would probably have the larger impact on fan power as compared to the 30% uniform air-leakage counterpart. The DOE-2 simulation results shown in Table 25 and Table 26 confirm this hypothesis.

Table 26 Impacts of Thermal Losses from Core-System Supply Ducts in Fresno (San Jose)

Parameter	Core System Base Case (VAV) No Leakage Fresno (San Jose)	Percentage Change From Base Case [%]		
		30% Air Leakage from Supply Ducts to Return Plenum	5°F Supply Rise From Plenum to Terminals	
		Coupled Return Uniform Leakage Fresno (San Jose)	Common Return 25% Conduction Loss Fresno (San Jose)	Decoupled Return 25% Conduction Loss Fresno (San Jose)
Core Fan Air Flow [cfm]	57,700 (57,000)	43% (43%)	33% (32%)	33% (32%)
Core Fan Consumption [MWh]	101 (101)	112% (111%)	135% (135%)	135% (135%)
Core Fan Peak Draw [kW]	34 (34)	115% (115%)	138% (135%)	138% (135%)
Annual Core Cooling Load [MBtu]	2,280 (1,225)	13% (-7%)	21% (32%)	18% (27%)
Annual Perimeter Cooling Load [MBtu]	1,890 (805)	-5% (-4%)	-4% (-7%)	-1% (1%)

6. ENERGY SAVINGS OPPORTUNITY ASSESSMENT - CALIFORNIA THERMAL ENERGY DISTRIBUTION SYSTEMS

In Section 3, we disaggregated HVAC-related energy consumption in California commercial buildings according to their means of thermal energy distribution. The results of this project's stock characterization efforts are used to arrive at an energy savings opportunity and comfort improvement assessment. For this discussion, savings opportunities divide into three sub-groups: 1) reducing thermal imperfections, 2) reducing fan energy, and 3) modifying operational parameters. Also discussed is the development of improved designed tools, reduction of uncontrolled airflow in commercial buildings, and indoor air quality implications of thermal distribution systems.

Improvement in duct thermal losses and transport efficiency of thermal distribution systems results in both direct and indirect transport energy (fans and pumps) and heating and cooling energy savings. An energy savings schematic is shown in Figure 7, with solid lines for direct savings and dotted lines for indirect savings.

As an example, reducing duct thermal losses directly produces heating and cooling energy savings. Reducing duct thermal losses can also directly contribute to transport energy savings due to the reduced airflow rates required to deliver the same heating and cooling loads. Another example, improving transport energy efficiency, produces direct transport energy savings, which can also, result in direct heating and cooling energy savings. In addition to direct energy savings, indirect energy savings in heating and cooling energy result from reductions from transport energy and vice-versa. As some energy savings opportunities are assessed in this section, focus will be provided on certain levels of direct and indirect savings.

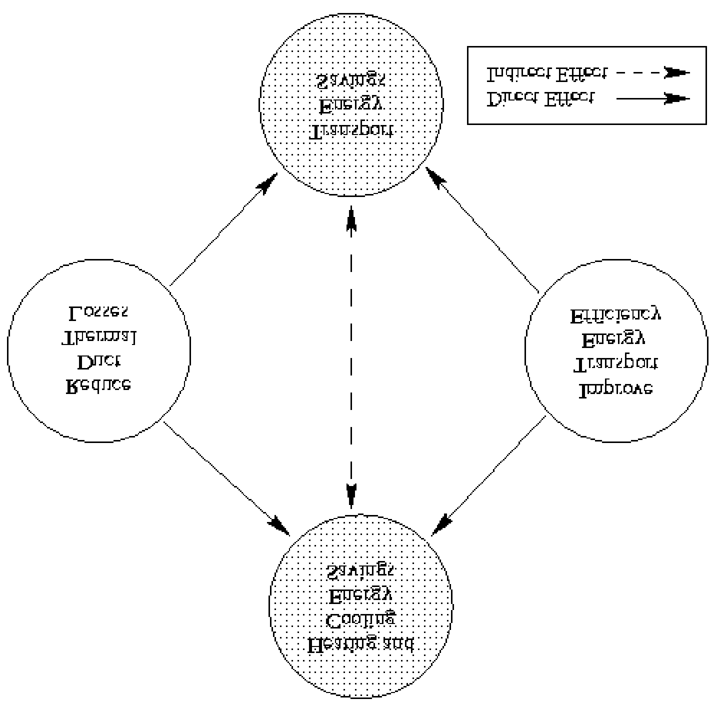

Figure 7 Direct and Indirect Energy Savings

6.1 FAN, PUMP AND COOLING ENERGY CONSUMPTION

As determined by the CEC forecast for 1994, and discussed earlier in this report, the total annual energy consumption for fans and pumps in California commercial buildings is 8,626 GWh in 1994. The corresponding total annual cooling energy consumption is 13,583 GWh. The energy savings discussed in this section of the report are based on their applications to packaged units and central air systems. Following is a summary of these systems' energy consumption, as determined for this project.

6.1.1 Packaged Systems

From results of the packaged-system distribution based upon utility data (Table 16), the electrical consumption for cooling with packaged units in California commercial buildings was approximately 7,000 GWh (Table 18), which equaled approximately 50% of the cooling energy in California commercial buildings in 1993 (CEC 1991a). Packaged systems in large offices had the highest cooling electricity consumption (2,331 GWh), followed by hospitals (822 GWh), retail (685 GWh) and small offices (612 GWh). Natural gas consumption for packaged units in the California commercial buildings was estimated to be approximately 29 trillion BTUs in 1993.

The systems in the case study operated essentially like residential equipment, with the fan cycling on and off with the compressor. This operational pattern resulted in that fans used less than 15% of the total electrical energy consumption of the systems. This means that most of the savings opportunities should be weighted toward the improvement of thermal performance, such as duct insulation and leakage. This also means that the load factor for these systems is relatively low since the entire unit (not just the compressor) cycled on/off during peak-demand periods.

On the other hand, many packaged systems, particularly larger ones, use continuous fan operation, in which case fan energy can be a significant fraction of the total energy consumption. The energy and demand saving potentials would be largely affected by fan operation and air leakage.

Overall the energy and demand saving potentials for packaged units come from reducing air leakage, conduction losses, unnecessary solar gains on the cabinet and exposed ductwork, and taking advantage of the duct systems' thermal storage effects during fan overrun.

6.1.2 Central Air Systems

From the analysis of the system distribution based upon utility data (Table 17), the electrical consumption for cooling with central air systems in California commercial buildings was approximately 2,748 GWh (Table 20). Central air systems in large offices had the highest cooling electricity consumption (1,749 GWh), followed by hospitals (646 GWh), schools (106 GWh), and hotels/motels (101 GWh). Natural gas consumption for central air systems in the California commercial buildings was estimated to be approximately 11 trillion BTUs.

The stock characterization shows that the majority of the California commercial buildings are equipped with CAV systems, and that fan energy is a large fraction of thermal distribution energy in large buildings.

The energy and demand saving potential for large existing buildings comes from reducing excess air flow, reducing the system pressure, removing or reducing thermal imperfections, checking and adjusting operating schedules, and increasing fan and motor efficiency. Another strategy that was identified as a means of addressing this savings potential was the development and use of simplified “tools” for diagnosing the fan energy performance of existing systems.

6.1.3 Cycling Effects

We assume that units that have to provide ventilation (i.e., for offices, schools) do not cycle. In cases where ventilation is provided by other means, or where ventilation is not a task of the HVAC system (i.e., in warehouses), systems might cycle depending on the load.

A comparison between ratios of fan energy consumption to cooling energy consumption for all the packaged units (4,628 GWh vs. 7,000 GWh) and for all central air systems (1,624 GWh vs. 2,748 GWh) shows a relatively higher fan to cooling energy ratio for packaged units. This indicates that fans in packaged units may not cycle too much. As such, the impacts of various thermal retrofit efforts (including improved design tools) may be higher for packaged units while fan efficiency improvements may be more suitable for central air systems.

6.2 IMPACTS OF REDUCING THERMAL IMPERFECTIONS

Reducing thermal imperfections can result in a significant amount of fan and cooling energy savings. Our DOE-2 modeling of the thermally imperfect systems shows that air leakage and conduction losses directly impact the system fans’ peak demand and energy consumption. Our analysis applies to both air-packaged units and air central systems. The approaches to reducing these losses are dependent on our ability to eliminate leakage and provide more efficient insulation to reduce conduction losses, which means different techniques in new construction and retrofit applications. Thermal losses related to leakage can be reduced through repairing and sealing the ducts using remote sealing technologies (i.e., an aerosol-based sealing technology developed for residential buildings). Reducing duct leakage results in lower airflow rates and lower heating and cooling energy required to provide certain space conditioning. Thermal losses related to conduction losses can be reduced through installing additional insulation, and reducing solar gains, and improving the effectiveness of existing insulation (i.e., sealing interior insulation). We have addressed some of these solutions separately and for various types of systems.

As discussed in section 5.3, the thermally-imperfect system analysis conducted as part of this project assumes that thermal energy losses are about 25% of the cooling load, and that the potential energy savings related to completely eliminating these thermal losses are 58% for the fan energy and 8% for the cooling energy. Assuming that eliminating all thermal loss from the ducts is possible for one-third of the package units in the California commercial buildings, approximately 220 GWh (fan) and 50 GWh (cooling) of direct electricity energy savings per year is possible. Similarly, approximately 80 GWh (fan) and 20 GWh (cooling) of direct annual electricity energy savings is possible for one-third of the central systems in the California commercial buildings. Overall, annual electricity energy savings of approximately 300 GWh (fan) and 70 GWh (cooling) are possible if we could eliminate all thermal loss from ducts for one-third of packaged and central systems.

6.2.1 Impacts of Eliminating Air Leakage from Ducts

In order to analyze the air leakage impact on energy loss, we assumes the 25% thermal energy losses is split evenly between air leakage and conduction losses. Assuming that eliminating half of the total thermal loss due to air leakage (12.5%) from ductwork is possible for one-third of the package units in the California commercial buildings, this would result in a 33% reduction in overall fan energy consumption, and about a 5% reduction in total cooling load due to fan heat. Annual electricity energy savings of approximately 130 GWh (fan) and 30 GWh (cooling) are possible for package systems, while 45 GWh (fan) and 10 GWh (cooling) for central systems. Overall, annual electricity energy savings of approximately 175 GWh (fan) and 40 GWh (cooling) are possible if we could preserve the assumed “half of a quarter” air leakage from all ducts for one-third of packaged and central systems.

6.2.2 Impacts of Duct Insulation

Interior duct insulation is installed for both noise and thermal insulation. In pre-fabricated ductwork such as flexible ductwork, the insulation material is a flexible fiberglass assembly covered by an impervious liner. In field-fabricated ductwork, this insulation material can take the form of compressed fiberglass panels that are cut, shaped and glued to the interior surface of the system ductwork. This type of interior insulation often is installed without an impervious interior liner or may not otherwise be sealed to prevent air movement through the insulation. This effect is significant in ducts where air velocities are greater than 1000 fpm: 5% increase in heat transfer at 1000 fpm, 31% at 1500 fpm, 59% at 2000 fpm, and 83% at 2500 fpm (ASHRAE 1993).

While branch ducts to registers are usually designed based on velocities of 500-700 fpm, the average velocity through the main trunk ductwork of air distribution systems may be approximately 1250 fpm for low velocity systems and 2500 fpm for high velocity systems. The average increases in heat transfer across the duct insulation would be approximately 18% for low velocity systems and 83% for high velocity systems, where interior duct insulation exists without an impervious liner or being sealed against through-insulation air flow.

Our savings estimates are based on the assumption that 1) thermal losses are 25% of the total cooling load, 2) conduction losses are approximately half of the total thermal losses, 3) 20% of the ductwork exposed to outside (rooftop and top floor ceiling plenum) is main trunk ductwork having interior insulation with pervious liners and 4) that it is possible to seal 50% of the systems with internal duct insulation. Based on the assumptions, the annual direct savings would be 4 GWh (fan energy) and 6 GWh (cooling energy) for rooftop packaged systems with average air velocity of 1250 fpm in their ducts. Similarly, the annual direct savings would be 3 GWh (fan energy) and 4 GWh (cooling energy) for central systems with average air velocity of 2500 fpm in their ducts. The sealing of interior duct insulation also results in indirect reduction of fan energy due to the reduced airflow resistance and reduced pressure drops over areas with impervious insulation liners.

6.2.3 Impacts of Reducing Solar Gains on Exposed Ductwork and System Equipment

Our rooftop packaged unit case studies found that solar gains on rooftop ductwork and equipment cabinets that were added to the cooling capacity required for each system. Based on our analyses, if steps were taken to reduce the solar gains on rooftop ductwork and equipment cabinets, such as applying white or reflective paint or adding additional insulation, annual fan and cooling savings of 2% could be realized. Assuming that this type of retrofit were applied to 25% of the existing rooftop packaged units, direct annual energy savings of 20 GWh (fans) and 35 GWh (cooling) would be realized.

6.3 MEASURES OF REDUCING FAN ENERGY CONSUMPTION

Fan energy consumption can be reduced by a variety of methods, including reducing excess airflow, reducing system pressures and increasing fan and motor efficiency. Improvements in these areas result in significant savings in central air systems, as they tend to run more continuously in order to provide ventilation for building occupants.

6.3.1 Reducing Excess Air flow

Excess air flow is the amount of air delivered above the air flow rates needed to provide ventilation air and transport heating or cooling energy to a given space. CAV systems, the most prevalent systems in California commercial buildings, deliver a constant volume of air which may at times be higher than that needed to meet the ventilation and heating and cooling needs of the space. This excess airflow rate can be reduced through the use of variable air volume systems (VAV) or hydronic fan coil systems.

6.3.2 Converting Constant Air Volume (CAV) to Variable Air Volume (VAV) Systems

Converting existing constant-air-volume systems to variable air volume can be an involved, yet possible, retrofit activity. The conversion includes modifying or replacing the CAV fan and motor controls for variable speed duty, installing terminal boxes at each sub-zone and running control wiring from the new or modified central controller to the terminal boxes. Due to the extent of the VAV retrofit process, we have assumed that 5% of the existing central air distribution systems might be converted to VAV systems. Based on the DOE-2 analyses of central distribution systems, the savings potential for VAV systems over that of CAV systems could be 50% of both fan and cooling energy. Based on these assumptions, approximately 40 GWh of direct fan energy and 70 GWh of direct cooling energy could be saved by converting 5% of existing central air systems to VAV systems. Indirect cooling savings could also result from the reduced amount of fan and motor heat transferred to the conditioned air stream.

There is a higher potential for variable air volume systems in new construction activities. EPRI's study (Pietsch 1991) indicates a significant trend toward variable air volume systems in buildings with central systems. It also estimates 75% of central systems installed nationally in the last years have been variable air volume systems. Our existing stock characterization shows a variable air volume penetration of 19% for large office buildings. As the existing California commercial buildings includes a large percentage of older buildings with CAV or other types of central systems, it is possible that anywhere between 20% and 50% of new buildings could contain variable air volume systems. Based on this assumption and a projected 3% increase in floor area per year, potential savings could range from 10 GWh (fans) and 14 GWh (cooling energy) to 25 GWh (fans) and 35 GWh (cooling energy) per year of new large-office buildings. Over ten years, the total cumulative electricity savings from cooling and fan operation could be approximately 1320 GWh to 3300 GWh.

6.3.3 Installing Hydronic Fan Coil Systems in New Construction

As determined from the CBECS data, fan-coil systems are usually found in large buildings. The use of hydronic distribution systems is an energy conserving and peak-power reducing alternative to conventional air distribution systems. As discussed above, the fans that are used to transport cool air through the ducts draw a significant amount of the electrical energy used to cool buildings by all-air systems. Part of this fan electricity is heating the conditioned air, and therefore, is part of the internal thermal cooling peak load.

Reducing the fan energy consumption and the fan peak-power requirement by installing hydronic thermal distribution systems would be possible but usually not practical in cases where all-air systems are installed, since the changeover to hydronic fan coil systems from central air systems will be involved for existing buildings. However, the savings potential associated with using hydronic distribution systems could be best achieved in new buildings or buildings undergoing major renovations (i.e., complete removal or replacement of building HVAC systems). Hydronic fan-coil systems include piping networks, boilers, chillers and fan coil units with dedicated ductwork.

Based on the DOE-2 analyses of central distribution systems, the savings potential for hydronic fan-coil systems over that of CAV systems could be 80% of fan energy and 50% of cooling energy consumption. For new buildings with central air-conditioning, assuming 3% additional floor area is constructed per year and a hydronic system market penetration of 20% of new central distribution systems, the potential savings could be 8 GWh (fans) and 8 GWh (cooling energy) per year of new building floor area. Over ten years, the cumulative electricity savings could be approximately 880 GWh.

6.3.4 Reducing System Pressure

Fan energy is often higher than necessary to deliver heating or cooling to buildings, due to overall system pressure drops that are higher than necessary. While pressure ranges are used in designing duct systems, systems are often not designed to have the lowest feasible pressure drops or operating pressures. Factors that affect system pressure include restrictive fittings (sudden contractions and expansions), multiple changes in duct directions, flow restriction from pervious liners on interior duct insulation, etc. Reducing system pressure through careful duct design, fitting selection and installation can result in significant fan energy savings. As was mentioned earlier, providing an impervious liner or seal on interior duct insulation can decrease thermal losses through the duct liner by anywhere from 5% to over 80%, depending on the air velocity through the duct. This effort also reduces the pressure drop over these duct sections, which directly affects the amount of fan energy required.

Assuming that reducing system pressure is implemented in one-third of the additional 3% new construction (package units and central air systems) each year, and that results in a 33% savings in fan energy and a corresponding 20% of cooling energy, annual energy savings would be approximately 20 GWh (fans) and 20 GWh (cooling).

6.3.5 Increasing Fan Efficiency

Our analysis, in conjunction with the CEC estimate of fan/pump energy in commercial buildings suggest that approximately 1,600 GWh (Table 20) are used by fans in commercial buildings with central air handling systems. Assuming that fan and motor efficiency could be improved 15%, on average, for 20% of the fans in service would yield a direct fan energy savings of approximately 50 GWh per year. Applying similar efficiency retrofits to all fans in service would result in fan energy savings of 250 GWh. All savings due to improvements in air system pressure and fan/motor efficiency and some of the savings due to air volume reduction would result in peak demand savings as well as energy savings. Indirect cooling energy savings would also result due to the corresponding reduction in the amount of fan and motor heat transferred into the conditioned air stream.

6.3.6 Improving Operating Schedules

As mentioned earlier, a study (Herzog et al. 1990) found that many buildings' HVAC systems operate from 15% to 100% longer than the hours needed to provide thermal comfort and ventilation for the occupants. Assuming that efforts were made to modify the operating schedules of 20% of the buildings with longer than necessary operating schedules and that an average fan energy savings of 30% and cooling energy savings of 25% were possible (lower cooling savings percentage due to lower internal gains and lower indoor-outdoor temperature difference during unoccupied times), the potential annual direct energy savings would be 520 GWh (fan energy) and 680 GWh (cooling energy).

6.3.7 Adopting Fan-overrun

Our rooftop packaged unit case studies found that two of the systems had fan overrun occurring. This allowed the systems to recover additional cooling energy from the ductwork. For the case study systems, the use of fan overrun to reduce system runtime resulted in an average of a 2% annual fan and cooling energy savings. We found that the length of fan overrun was critical to the amount of energy recovered, with takeback occurring when the overrun period was longer than needed to extract the cooling energy remaining in the ductwork. If correct fan overrun time periods were to be determined and implemented in one-third of the packaged rooftop units in the California commercial buildings, direct annual savings of 30 GWh (fan) and 45 GWh (cooling) would be possible.

6.4 DISCUSSION

6.4.1 Creating Better Design Tools

One issue identified as part of our efforts is the impact of design tools on thermal distribution system energy in commercial buildings. Three key issues were identified: 1) the lack of capabilities to adequately model the integrated energy performance of distribution systems in whole-building energy analysis software (i.e., DOE-2), 2) the lack of incentive within the Title 24 process for reduction of fan power below 0.8 W/cfm (CAV systems) or 1.25 W/cfm (variable air volume systems), and 3) the lack of operating pressure specifications in duct design tools.

The last issue was first brought to our attention by the instructors of an ASHRAE professional development course on Air-System Design and Retrofit, which was attended by several of the research team members. Standard duct-design methods take zonal airflow as inputs, rather than zonal flows plus a target operating pressure. By incorporating the ability to automatically iterate the design to meet a target design pressure in to design tools, designers can work to develop systems having lower pressure drops which lead to lower fan energy consumption.

Taking this one step further, the second issue is the incorporation of distribution-system energy analysis and, perhaps, design, into energy analysis software. In the best of all worlds, one would like to have a program that automatically yields complete energy use predictions from HVAC designs, and provides intelligent suggestions for reducing that predicted energy use, taking into account the expected performance associated with various degrees of commissioning. Short of this utopian vision, one major step would be to incorporate the capability to model distribution-system performance into whole-building simulation tools such as DOE-2. In our energy analysis work, we have found it to be extremely difficult to analyze thermal distribution performance with DOE-2. The energy analysis presented on the fan-power impacts of duct thermal losses to ceiling plenums is an example of what types of effects are typically ignored, as well as what could be incorporated into tools such as DOE-2.

6.4.2 Reducing Uncontrolled Air Flow

An issue that is closely related to thermal distribution system performance is that of uncontrolled airflow in commercial buildings. The connection is the large preponderance of air distribution systems in commercial buildings, and the degree to which these systems typically overwhelm natural infiltration forces. Two related issues/activities have been identified to date in this project.

The first is a project at the Florida Solar Energy Center (FSEC) evaluating uncontrolled airflow patterns due to inadequate design/engineering/commissioning in 70 Florida commercial buildings. One of the issues identified early in that project is the impact of unbalanced flows in hotels and the influence of internal resistance due to closed doorways. This problem, similar to that associated with closed internal doors in residential dwellings, creates unwanted airflow through the building shell, resulting in both excess energy consumption, as well as material degradation problems due to moisture damage in Florida.

Another interesting discovery in our work to date was the surprisingly high envelope leakage in the two strip-mall-store case studies. Their envelope leakage levels were approximately twice those typically found in residences, the result being that small supply/return pressure imbalances will create large infiltration/exfiltration air flows, and that even with well-balanced systems, natural infiltration rates may be excessive. The reduction of this uncontrolled airflow through envelope tightening would result in a reduction of building total heating and cooling loads and fan energy.

6.4.3 IAQ Implications of Thermal Distribution Systems

Another area that seems to merit further investigation is the indoor air quality (IAQ) implications of various thermal energy distribution options. Recent ASHRAE activities have pointed out several issues that can have important implications for energy use by thermal energy distribution systems. Most of these issues revolve around ASHRAE Standard 62, which addresses ventilation for indoor air quality. IAQ implications have been discussed with regard to insulation inside ductwork versus outside ductwork. More specifically, there is increasing concern about the materials inside ducts, and their impact on air quality (principally from the point of view of bioaerosols). The quantity and location of insulation can have a large impact on energy performance, and development of alternative duct materials may come under more careful scrutiny. In addition, recirculation air, while having a big impact on the cost-effectiveness of various distribution systems, also largely affects air quality.

7. SUMMARY AND CONCLUSIONS

Significant amount of energy is used for space conditioning and ventilating in California commercial buildings. This includes approximately 35% of statewide electricity consumption; and 15% of statewide gas consumption in California. Space conditioning in commercial buildings accounts for approximately 18% of their electricity consumption, and 42% of their natural gas consumption. In 1997, fans and pumps used approximately additional 8800 GWh, or 10% of commercial-building electricity consumption.

The characterization of the thermal distribution systems in commercial buildings based on existing literature indicated that about three-quarters of building floor area contain air ducts for space conditioning. Based upon surveys by four utilities in California (1988 through 1993), we found that the most common thermal distribution system is the air-to-air single duct with CAV supply, followed by the multizone system. The electricity use of these systems accounts for over 60% of the statewide fan/pump electricity use by thermal distribution systems in commercial buildings. Commercial office buildings, retail buildings, hotels and health/hospital buildings consume approximately two-third of electricity energy used for space cooling, heating and ventilation. This indicates significant fan-energy saving potentials due to inefficient thermal distribution system including inefficient fan operation. In addition, the direct energy consumption by fans and pumps also results in heating of conditioned air and therefore contributes to excessive internal cooling load. Reducing fan energy has thus dual benefits in energy savings.

Energy conservation measures for HVAC systems (excluding distribution systems) are found in about one-third of the total building floor area, as compared to only one-sixth of building floor area for distribution systems. Although some practicing engineers are aware of available energy-saving measures, first costs of HVAC equipment and thermal distribution systems are the main driving force of system selection in the design stage. There is a need for better design-tools, which take into account of system types and efficiency of thermal distribution energy. The issue of performance compliance for the California Energy Efficiency Standards (Title 24, CEC 1998) for non-residential buildings needs to be addressed to encourage practice and implementation for system energy-efficiency, such as fan energy reduction.

The study shows that saving potentials for HVAC energy use and demand in small commercial buildings with packaged units can be effectively realized by reducing air leakage and conduction losses through duct systems, decreasing unnecessary solar gains on the cabinet and exposed ductwork, and taking advantage of the duct systems' thermal storage effects during fan overrun. The saving potentials for HVAC energy use and demand in large existing commercial buildings comes from reducing excessive system airflow, decreasing the system operating pressure, removing or reducing thermal imperfections, checking and adjusting operating schedules, and increasing fan and motor efficiency. Another strategy that was identified as a means of addressing this savings potential is to develop and to use simplified "tools" for diagnosing the fan energy performance of existing systems.

Through opportunity assessments directed at a subset of technologies, we identified preliminary energy-savings opportunities based upon survey results and energy analyses. In general, three forms of system energy performance are affected by 1) thermal losses induced by air leakage through systems components (i.e., duct, equipment), 2) thermal losses induced by heat conduction, convection, and radiation, and 3) equipment efficiency and control strategies. Thermal losses related to air leakage can be reduced through repairing and sealing the ducts using remote sealing technologies (i.e., an aerosol-based sealing technology developed for residential buildings). Thermal losses related to conduction losses (including convective or radiative losses) can be reduced through installing additional insulation, improving the effectiveness of existing insulation, and reducing solar gains of ducts exposed to sunlight. Reducing duct leakage and conduction loss can result in lower air flow rates, lower fan energy, and lower heating and cooling energy required to provide certain space-conditioning. Although it is not possible to accurately quantify all potential savings induced by the strategies discussed, this study indicates encouraging opportunities for energy saving through the performance improvement of thermal distribution systems in commercial buildings.

For one-third of packaged and central systems in California, annual electricity energy savings of approximately 175 GWh (fan) and 40 GWh (cooling) are possible if we could preserve the assumed “half of a quarter” air leakage from all ducts, and annual electricity energy savings of approximately 300 GWh (fan) and 70 GWh (cooling) are possible if we could eliminate all thermal loss through conduction from ducts.

Assuming that fan and motor efficiency could be improved 15%, on average, for 20% of the fans in service would yield a direct fan energy savings of approximately 50 GWh per year. Applying similar efficiency retrofits to all fans in service would result in annual fan-energy savings of about 250 GWh. In addition, all savings due to improvements in air system pressure and fan/ motor efficiency and some of the savings due to air volume reduction would result in peak demand savings as well as energy savings.

Assuming that efforts were made to modify the operating schedules of 20% of the buildings with longer than necessary operating schedules and that an average fan energy savings of 30% and cooling energy savings of 25% were possible (lower cooling savings percentage due to lower internal gains and lower indoor-outdoor temperature difference during unoccupied times), the potential annual direct energy savings would be 520 GWh (fan energy) and 680 GWh (cooling energy).

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APPENDICES (A-D)