Honeywell

ENERGY CONSERVATION WITH COMFORT

MANUAL AND WORKBOOK

Commercial Comfort & Energy Specialist

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ENERGY CONSERVATION WITH COMFORT

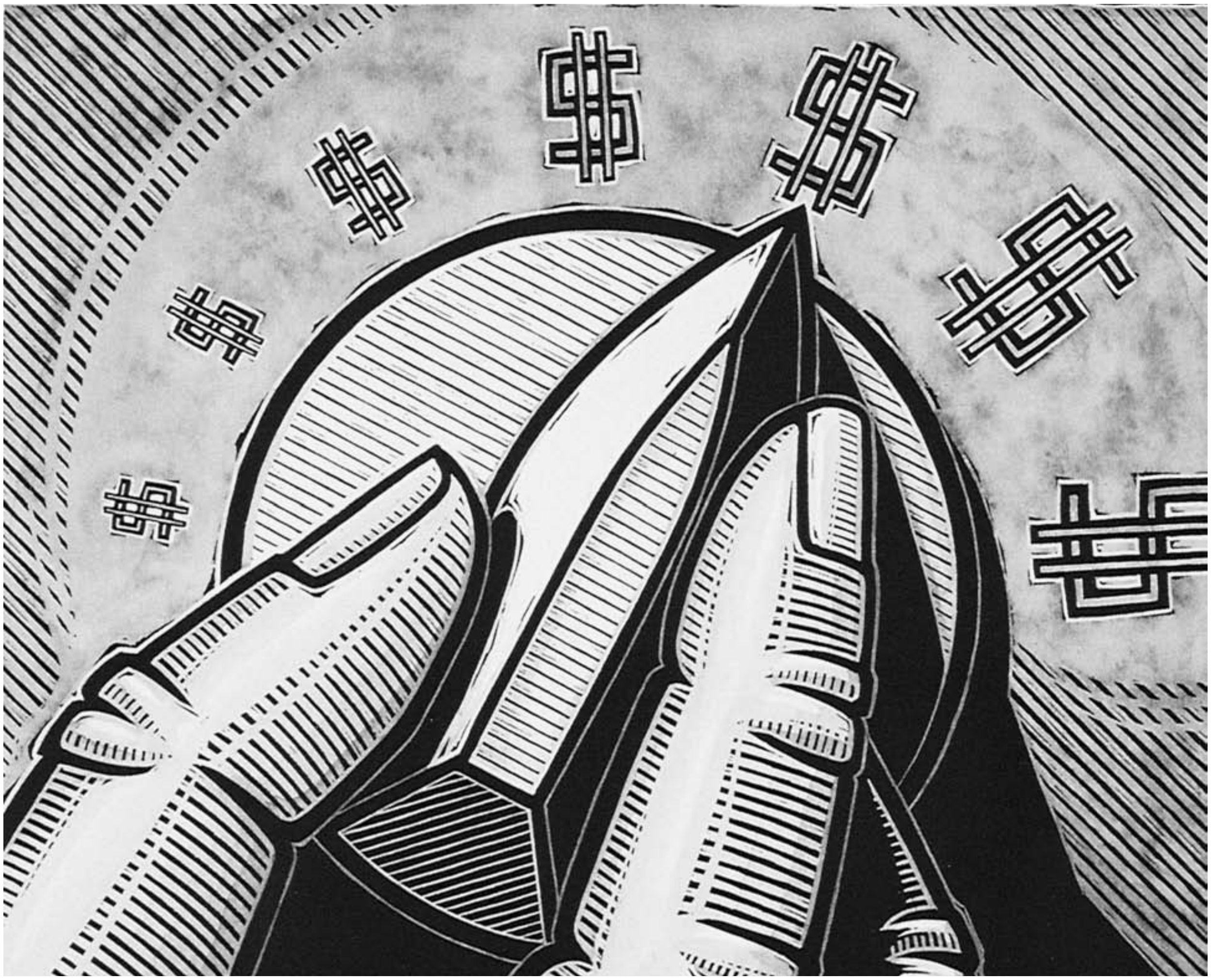
INTRODUCTION

Prior to 1970 energy sources were considered a limitless commodity which allowed design professionals broad latitude in facility and plant design. First cost was a greater concern at that time. In most cases, first cost was depreciated over a span of more than 20 years. Operational cost over the life of the building was a secondary design concern as energy cost was measured in pennies per square foot of building area, rather than the dollars per square foot today. Energy used to be cheap!

Not until the oil embargo of the early ‘70s and US energy crisis did we start to focus on the fact and their physical plants. Only then was serious consideration given to the way we operate buildings and that these measures could possibly pay for conservation expenditures in just a few years.

Over the next 15 years, efficiency was placed at the top of HVAC equipment, controls and facility design criteria; professional organizations studied, reviewed and published new energy efficiency standards; government agencies established tough efficiency codes. The ‘80s identified pollution hazards associated with energy use. By the 1990s, energy conservation measures not only came to affect operational cost. It also became public policy.

The late 1990s continued to be a turbulent period. Some utilities bought and sold contracting firms. Natural gas and electricity became deregulated in certain regions in attempts to follow the telecommunications industry and spur competition. Many service and supply firms have consolidated in order to lower costs and offer better national delivery. The results have been mixed. In some regions heating and electrical costs have doubled in the last one to two years. It’s not clear which strategies will prevail nationally, only that energy costs continue to rise unless individual facility managers take counter measures.



A PLAN FOR THE FUTURE

In 1997, participants in EPA's Energy Star Building and Green Lights Program saved more than 4.7 billion kwhs, enough to meet the yearly electricity needs of nearly half a million American homes. EPA suggests that homeowners can reduce energy by 30 percent when following the EPA Energy Star Program.

Based on Honeywell's experience in schools, healthcare, commercial, industrial and municipal facilities, most buildings could save 20 percent or more on electrical and natural gas costs annually by using some very basic as well as more advanced energy conservation methods.

Heating, ventilation and air conditioning costs have been estimated to consume, on average, up to 40 % of non-residential building energy costs. These costs are expected to remain high and continue to dominate a disproportionate amount of each building's yearly expenditures. Our ability to use energy efficiently while providing for building occupant comfort will be no easy task.

People who pay for energy are well aware of rising costs and uncertain supplies. Most of us have done something in our businesses to conserve energy. Perhaps thermostats have been adjusted or lighting levels have been lowered. Unfortunately, many energy management programs begun years ago have not been maintained. Energy costs were just factored in as a "cost of doing business" and passed onto those who leased the space or as a fixed line item in the facility budget. Efforts to continuously reduce consumption slowed or came to a halt.

The underlying assumption has been that energy costs would remain stable within a reasonable range like they had been throughout most of the 1990s. But, basic energy costs surged in 2000 and 2001. Any buildings not employing a significant number of conservation strategies have seen their energy bills double and triple. And, if the rising cost of energy isn't enough, there's a Catch 22. How do you achieve energy conservation and keep building occupants happy at the same time? How can you manage conservation with comfort?

Twenty years ago Honeywell addressed these issues in its Energy Conservation with Comfort manual. It was considered by many to be a primary resource, an energy manager's handbook on conservation in non-residential buildings. The manual provided Energy Conservation Recommendations (ECRs) applicable to heating, ventilating and air conditioning (HVAC) systems in new and existing buildings. Since the recent dramatic increase in energy costs, media attention has focused on much needed conservation measures. The 2001 edition of Energy Conservation with Comfort is intended to help fill the need for a current resource to assist facility managers in addressing their energy issues.

The information contained in this manual provides building owners and managers with the basic tools to promote energy conservation awareness, evaluate consumption and develop an energy plan to evaluate and implement energy conservation strategies. In short, this manual is intended to provide a process to dramatically reduce energy consumption and costs while maintaining occupant comfort and promote proper ventilation and indoor air quality.

INTRODUCING

How do you achieve conservation and keep occupants happy at the same time? How can you manage conservation with comfort? Energy conservation begins with the basics. Start with thermostats, valves and dampers. Renovate and modernize the parts before you explore what automation can do for the whole. To make the best energy management decisions, you must start with efficient, well-maintained operating controls.

A constant flow of creativity and imagination must be directed to the task of energy management. Use your imagination and that of the team you will organize. There are as many areas to explore as there are rooms in your building and controls in your system. Let's start at the beginning.

BEGIN WITH THE BASICS

Where do you begin? That question has stopped many organizations in their tracks. Energy conservation is not a problem that can be solved overnight. It will take planning, long and short range. But you can begin immediately with some low cost/no cost opportunities that will save energy now while you develop a plan for the future.

IN THE HEATING SEASON:

Heat the building to 70°F when occupied, 60°F or less when unoccupied (be careful that this doesn't start up the air conditioning system).

Lock thermostats at lower (heat) settings.

Turn the heat off during the last hour of occupancy. The building will maintain its heat for a relatively long period of time, and you'll be spared the cost of this additional heating operation. Allow your daily heating investment to "glide."

IN THE COOLING SEASON:

Cool the building to 78° F rather than 74 ° F when occupied. Consider NOT applying mechanical cooling at all when unoccupied, but within a range the mechanical system is capable of recovering prior to the next occupied period.

FOR THE ENTIRE YEAR:

Concentrate off-hours occupancy in a single zone, if possible.

Assure outside air ventilation complies with the American Society of Heating, Refrigerating and Air Conditioning Engineers (ASHRAE) standards of 10-20 cfm per person where safety is not compromised. Incorporate newer "demand control ventilation" (DCV) to avoid costly over-ventilation whenever buildings have lower than design maximum occupancy.

Change air filters on a regular schedule to ensure maximum heating or cooling capacity.

Turn off non-critical exhaust fans so that cooled and heated air isn't exhausted unnecessarily.

Reduce domestic hot water temperatures.

Turn off lighting in unoccupied work areas; reduce lighting when safety is not compromised.

Clean lighting fixtures and replace lamps on a regular schedule — this will help insure maximum illumination for kilowatts expended.

WHAT ARE YOU WILLING TO INVEST TO SAVE ENERGY?

The success of your plan depends on your attitude and commitment.

How do you view the value of your investment of funds in energy conservation versus the alternative possibilities? It takes money to save money, beyond the simple adjustments just offered. You will have to invest money to save energy but it will pay dividends each succeeding year.

To determine the "real" value of your investment in energy conservation, you must consider four things: the cost of energy, the cost of implementing your plan, occupant productivity and your payback.

The key consideration is your energy cost. Without a realistic appraisal of what your system is costing you and where — you can't determine the most worthwhile changes or estimate what your savings might be.

THE MYTHICAL MANAGER & MONEY SAVINGS MAGIC

Your investment in energy conservation can yield great dividends.

Imagine a facility manager in1972 with an 80,000 square-foot building in Chicago which consumed 125,000 BTUs per square foot for heating, cooling, lights, fans and pumps at a cost of $2.50 per million BTUs. His annual energy bill was $25,000.

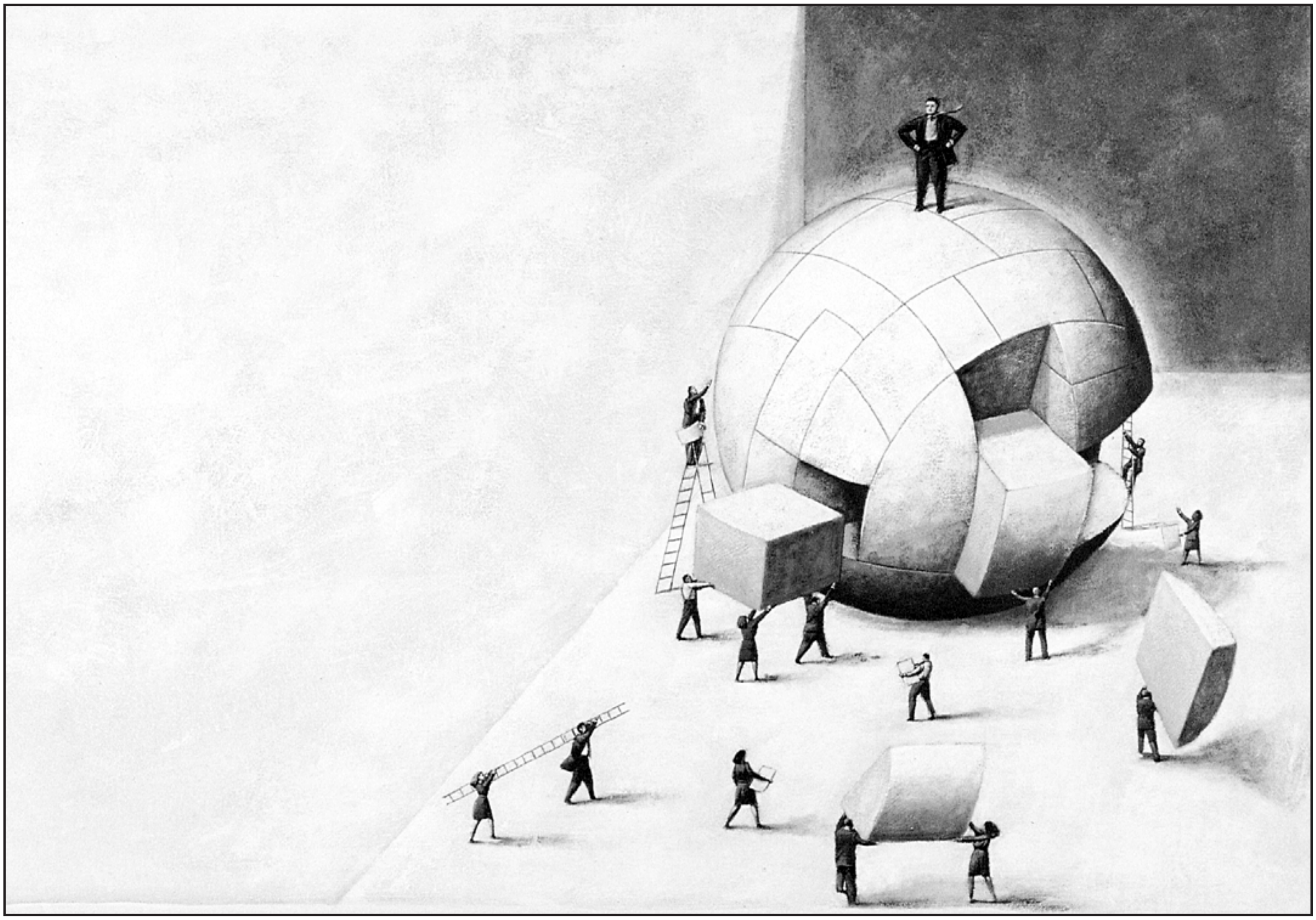
Had they implemented a program to reduce consumption by 10 percent by lowering lighting levels, cutting down on outdoor air, and taking a few low cost or no cost actions mentioned earlier, his accumulated savings could have compounded to more than $20,000 in the first five years.

Or, with no action taken and with the cost of energy having risen close to $10 per million Btus, the energy bill for that same facility would have skyrocketed to more than $100,000. Do you think the facility manager could have found other uses for most of that extra $75,000? Is that facility manager prepared should energy costs rise further to $15 per million BTUs or beyond?

By 2001, electricity has gradually increased from "penny cheap" rates of under $0.05 per kwh to fairly common rates of $0.10 to $0.15 per kwh. Most residents of California experienced much higher rates during 2000 , some paying as much as $0.25 per kwh when demand exceeded regionally produced supply. Even so, rolling brownouts have been common and are expected to be more severe during the higher demand air conditioning season. Many utilities there and around the country impose additional "demand charges" when a buildings established KW levels are exceeded.

During much of the 1990s natural gas evolved to being a "lower cost" fuel for generating electricity and alternate fuel to power large facility chillers. Natural gas heating costs leaped from $0.50 per therm ($6.80 per million BTUs) to over $1.00 per therm ($13.60 per million BTUs) during 2000/2001 over much of the US. This represented over a 100% increase in a single year after being stable for a decade. Rates are not expected to return to their earlier levels any time soon. Now, increased demand for natural gas to generate electricity competes directly with demand as a furnace heating fuel. The result is lower supplies and dramatically increased cost.

So, what is a facility manager or building owner to do? How can these dramatically increased costs be managed effectively? How can the energy cost portion of building expenditures be controlled and reduced? There are a myriad of possible solutions. Many involve how the building and its systems are operated. Many others involve potential upgrades and improvements to the building mechanical and controls systems. But, most important at the beginning, you need a plan and a committed team to tackle the issues and options. And, don't forget that you can call on a Honeywell Energy Specialist for assistance as you form your team and move ahead to develop and implement your energy conservation plan.



ORGANIZING FOR ENERGY CONSERVATION

START BY BUILDING A TEAM

Composition of an energy team will vary as much as organizations vary in size and style of management. You may be the key decision maker or you may manage by committee and report to the Board or Executive Director. In either case, it is imperative that one individual be delegated the responsibility, authority and budget to effectively spearhead conservation efforts.

Ideally, your energy team should be a task force of four to six core participants — experts in their fields. In general, you will need to have management, facility operations engineering and finance represented. You'll need electrical expertise to deal with control wiring, electrical service and electrical distribution issues. Mechanical engineering will address mechanical systems, operation, design and heat transfer problems.

Return on investment and funding are the bailiwicks of the financial representative. It is fundamental that energy programs have the support of top management. Their support of your program can help provide focus, critical funding and add importance to your efforts. Without this support, your program will never get off the ground or be sustainable.

Members of your facility staff might represent most of these areas of expertise. Even if you find all or most areas present within your organization, you may want to seek outside support. Don't forget that local utilities (electrical and gas/oil) have grown to provide much more than basic utility rate information. Call on yours to support your conservation efforts. And, see if there are current rebates or other energy efficiency programs to encourage conservation measures.

And finally, consulting engineering firms offer unbiased technical expertise to tie it all together.

Also, don't neglect one of the sources closest to home — other building owners/managers. There may be other facilities or plants in your own company or organization faced with the same challenges. Be sure to contact them to determine whether they have or are interested in developing an energy conservation program too.

Once you have selected your team, consider these questions: Is the team representative of the internal expertise? Does the team have the support of the upper management and operating departments? What are the team's strengths and weaknesses?

BE SPECIFIC ABOUT YOUR OBJECTIVE

That's not as easy as it sounds. You say you want to save 20 percent over the next 10 years. But 20 percent of what? Cost? Consumption? What's the base? How will you measure it? Setting realistic objectives requires an analysis of your building's energy history and present consumption.

And to determine that, you'll have to do a survey. Chances are your energy plan will build on some historical usage figure. The Energy Survey in the next section provides a tool for surveying your building. Fill in the various checklists and you will have a complete energy history for your building(s). Once the data is gathered and grouped, develop an objective based on your present energy consumption.

Your objective can be stated in a variety of ways: in dollars saved, for example, or as a percentage of energy savings over a particular time period. For the latter, you will have to convert energy to a common unit of measurement. You'll have so many Therms of natural gas, Kwhs of electricity, gallons of oil, etcetera consumed by your existing systems. The common unit of measurement is the Btu, and your energy management objective or goal could then be stated as BTUs saved per square foot.

Your energy organization needs a strong core, so spend time developing a results-oriented team that can work together toward a goal.

BUILD MOMENTUM

After you've formed a team, you'll have to plan your progress toward your conservation goal. The following steps will carry you from initiation to implementation of an energy management program:

1. Set up an Energy Survey Team.
2. Survey your existing buildings to identify energy conservation opportunities.
3. Analyze the potential savings of these opportunities.
4. Prioritize your energy savings opportunities and organize them into a formal long-range plan.
5. Use training and monitoring to orient your organizations to operate in an energy conservation mode.

DRAFT A PLAN

Drafting a plan for energy conservation in new building designs can occur right at the drawing board.

With existing buildings, there is now a clean sheet. In fact, original building plans may no longer be available even though consuming systems are still there. You must evaluate what you have now if you are to institute any changes.

BREAK YOUR PLAN INTO MANAGEABLE UNITS

An overview of any energy plan should describe present energy consumption conditions in applicable units, e.g., individual buildings or project developments. There should be plans of action for each of these units. These should be revised from time to time, based on your evaluation of consumption costs and savings. Periodic reviews and adjustments are a necessity.

Buildings or facilities that offer good potential for savings should be identified and priorities set for budgeting and funding purposes. Identification of individual projects will provide management with a tool to track progress and evaluate results relative to costs.

COMMUNICATE THE PROGRAM

In the beginning, you will need to cultivate support for the program and its objectives. The solution is communication, both up the organizational ladder and down. Some people will have a need to know, because your plans will affect their actions; others will simply want to know. Deal with both in your communications network.

UP THE LADDER AND DOWN

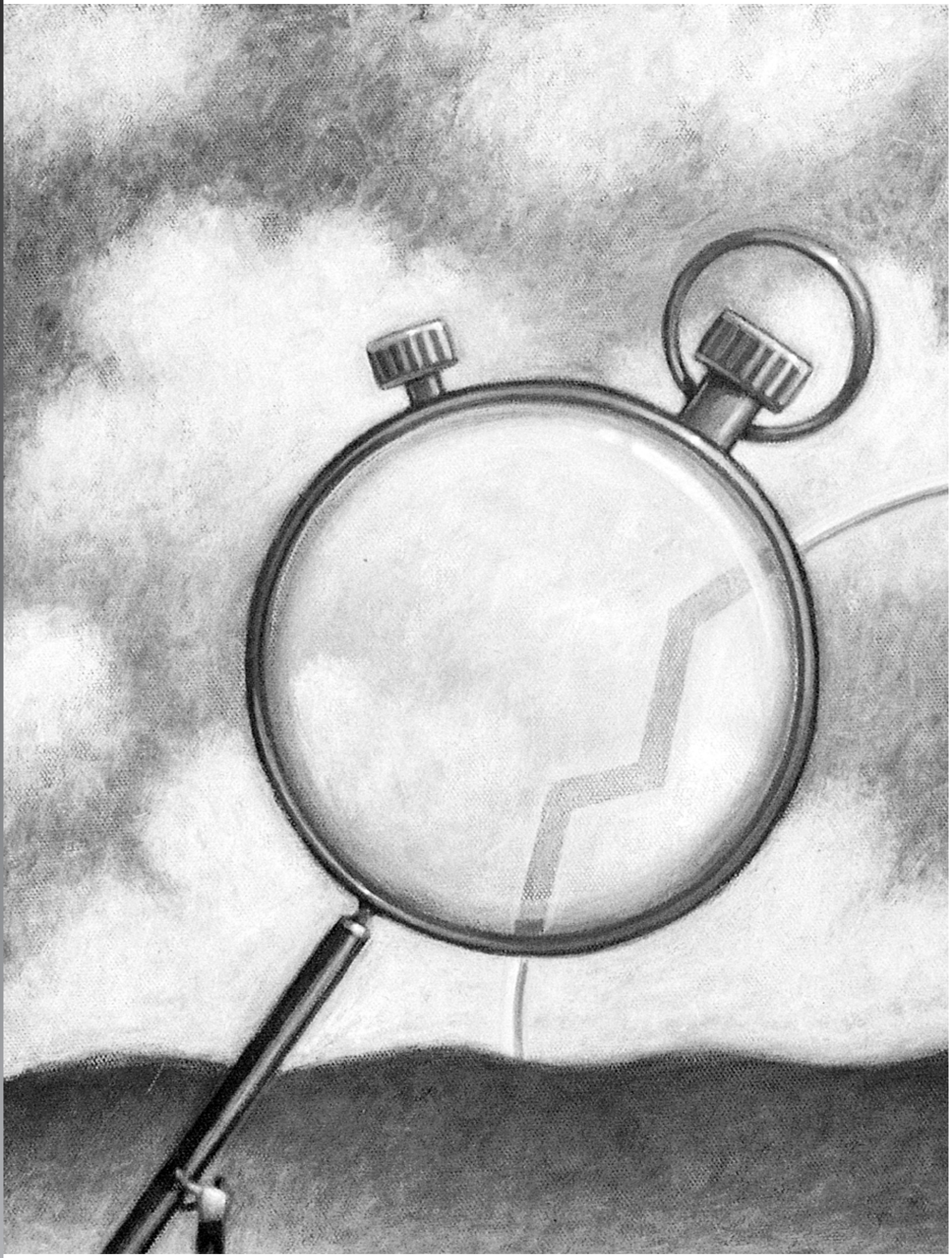
An upwards flow of information is a necessity in any energy management program. It alerts upper management to the progress and goals of the program and encourages their support.

Information can take numerous forms, but the most efficient for communicating with upper management will probably be a periodic report (monthly or quarterly). This report should cover estimated costs and savings of the program, actual costs and savings, explanation of variances, and recommendations for the next period.

Involving your financial personnel assures you of getting support to evaluate and fund the various investment possibilities.

Communicate with the building operators. In some cases, they must furnish needed information. And, their familiarity with your plans will make them less resistant to changes. And don’t forget to involve the general populace of your building. Prominent displays of energy goals, charts, process reports (presented in gallons of oil or kilowatt hours, not BTUs or dollars) will keep goals and progress on their minds. People working together towards common goals get things done.

Involve as many facets of your organization as possible. It will make the job a lot easier to implement because more people will be aware and feel included. Work together toward energy savings.



SURVEYING THE SITUATION

There are three types of surveys your team should expect to conduct. The first will involve an “energy survey” which will provide an overview of past and current costs. The second will involve a one-page summary of “space conditioning equipment and schedules.” The third will involve a detailed look at the building systems/components that consume energy and details of their operation. Don’t short-change or underestimate the need for thorough and accurate survey efforts. Take a professional approach by using people qualified to gather data and answer the questions posed by each survey. You’ll need to establish or contract survey teams in order to plan and implement conservation projects effectively in later steps.

The energy survey is an important first step in implementing an energy conservation plan for your building. Properly done, it can provide a baseline that can be used to determine the success of future conservation efforts.

Delve into your buildings’ energy history, analyze utility bills and study equipment scheduling. This effort is the cornerstone of every effective program. It will pay off in dollar savings and reduce energy consumption.

Energy consumption can be determined by totaling all of the energy types purchased and utilized by your facility over a 12-month period. Working with the calendar year is the most common method, however, any consecutive 12-month period will work. Analyzing two or three years of energy consumption. This will begin to show performance trends over time, not just changes resulting from varying rates.

The information needed from the billing statements includes the amount of energy consumed per fuel type over a length of time, usually in monthly increments. Billing information can be obtained from your own records or requested from the local utility.

The following text describes charges associated with the two common types of energy found in almost all facilities today.

ELECTRICITY

Electricity billing is based on two basic elements, consumption and demand.

Consumption is determined by measuring the amount of watts consumed over a unit of time, usually an hour. In order to keep numbers small and reasonable, electricity is commonly billed in one thousand watts per hour units known as Kilowatt-hours (KWH). KWH is a cumulative measurement of the electrical energy consumed or used by the customer. Consumption is assessed in cents per KWH and accounts for 50 to 60 percent of the average electrical bill each month. Mathematically, the billing equations are as follows: (Total KW) x (Hours On) = (KWH) x (Charge in cents / kwh) = $ billed

The second basic element in electrical billing is the demand charge. The demand charge is intended to compensate the utility company for their capital investment and reserve operating cost needed to serve the peak loads. The Peak Demand determination is the single highest demand in watts recorded at a single moment or during any 15 or 30 minute period of the billing cycle. Not all electrical utilities bill for a demand charge. Check whether yours does and what variation of the above billing strategy they use.

A growing number of power utility companies are already employing RTP (Real Time Pricing). It’s the power industry’s newest means of stretching local generating capacity and reducing harmful stack emission.

Under Real Time Pricing, the local electrical utility establishes its pricing on a day-by-day, hour-by-hour sliding scale based on the variable cost of production or acquisition. The utility provides daily rate schedules to major customers detailing the hourly price of power one day in advance. This schedule may be amended during the operative day, sometimes with no more than one hour notice. Those equipped to receive this information and quickly adapt control strategies to the new pricing profile can take full advantage of transient savings opportunities. Those who cannot will be unable to avoid the shifting rate spikes.

NATURAL GAS

Natural gas is delivered and measured in cubic feet. However, billing statements will refer to the amount of gas consumed based on a unit of measure called a therm. One therm is equal to 100 cubic feet of natural gas, a decatherm is equal to 10 therms or 1000 cubic feet. Consumption is assessed as a cost per unit of natural gas usually expressed in therms or decatherms. Over the past two decades many facilities had shifted their heating source from oil to natural gas because of its increased availability, clean burning characteristics and value versus ever increasing electrical rates. This strategy is no longer so certain with the dramatic increases in natural gas rates since during 2000 and 2001.

BRITISH THERMAL UNITS

British Thermal Units or BTUs aren’t British at all. Technically, a single BTU contains enough heat energy to raise one pound of water one degree Fahrenheit. A less technical definition of a BTU is the heat content of a wooden kitchen match burned end to end. All fuels possess energy that can be measured and described in BTUs. The conservation chart on the next page shows examples of commercial fuels, common units of measure and the BTU content associated with each.

RATING YOUR BUILDING’S ENERGY USE

Rating the efficiency of a building’s energy use follows the same principles as rating your car’s efficiency. The efficiency of a car is derived by dividing the number of miles traveled by the number of gallons used. In a similar manner, a building’s efficiency can be obtained by dividing the total amount of energy consumed in BTUs over a 12-month period by the gross conditioned square footage of the building. Or, stated as the number of BTUs consumed per square foot of conditioned floor space (eated, cooled, ventilated and lit for people) annually (BTUs/ft^2/yr). There is little question that the rating is approximate, but when applied consistently, this value can be very effective for comparing different properties, or one property from year to year. One challenge is to determine the electrical BTUs used for heating, air condiditoning and lighting apart from all other purposes like copy machines, elevators, computer systems.

CONSERVATION CHART A

FUEL TYPE	UNIT OF MEASURE	BTU CONTENT OF FUEL
Natural Gas	1 Cubic Foot	1,032 BTUs
	1 Ccf = 100 Cu Ft	1 Therm = 103,200 BTUs
	1 Mcf = 1,000 Cu Ft	1,032,000 BTUs = 1.032 MMBTU's
	1 Mcf = 10 CCF = 1 Dekatherm	1,032,000 BTUs = 1.032 MMBTU's
Propane	1 Gal	91,600 BTUs
	1 Cu Ft	2,500 BTUs
Fuel Oil	1 Gal #2	139,000 BTUs
	1 Gal #4	145,000 BTUs
	1 Gal #6	150,000 BTUs

WHY NOT USE CUBIC FEET?

Square footage is used rather than cubic footage because buildings of similar construction and amenities are compared, known as building types. Categorically, warehouse and office buildings are examples of two distinct building types. Logically, buildings with increased cubic footage, due to high ceilings, such as warehouses would not be compared to office buildings with low ceilings. Also, building types such as warehouses and office buildings, have different environmental parameters that make them incompatible comparisons.

HOW DO WE USE THESE NUMBERS?

We can use the BTUs/ft^2/yr rating in many ways. The two most common uses are the ability to compare the energy performance of similar building types and the ability to track the energy use in a building or group of buildings, over time.

SAMPLE ENERGY PERFORMANCE SURVEY

On the following page, a sample commercial building Energy Performance Survey has been completed using information from an actual 62,000 square foot athletic/office facility in the Midwest. The form has been constructed for easy entry of the facility's utility information and conversion to BTUs/ft^2/yr. This form serves as an overview of energy usage.

IT'S YOUR TURN

Now that you know how to obtain and enter the energy survey information, it's time to take the leap and complete an energy survey for your building! Several blank Energy Survey forms have been provided in the appendix at the back of this manual for your use. Refer to the survey example and the text in this chapter to refresh yourself as necessary. Now, complete a survey for your build and then you can compare your findings to other building types and sizes.

SAMPLE

Energy Performance Survey

Date: Nov. 2, 1998

Name of Facility: Office/ Athletic Facility

Location: Chicago

	Building Energy Types			
	Electricity		Other Energy Type:	
	kWh Used	Subtotal	Unit Quantity	Subtotal
January	18,171		2,929	
February	17,472		3,203	
March	16,773		3,007	
1st Quarter Subtotal		52,416		9,139
April	11,881		2,929	
May	10,483		3,320	
June	9,086		4,296	
2nd Quarter Subtotal		31,450		10,545
July	6,570		4,687	
August	4,613		3,398	
September	5,591		1,953	
3rd Quarter Subtotal		16,774		10,038
October	9,784		2,343	
November	13,978		3,320	
December	15,375		3,670	
4th Quarter Subtotal		39,137		9,333
Year's Total		139,777		39,055

Unit of purchase notation

BTU Conversion Formulas

Energy Type	**(Units)**	**Year's Total**		**BTUs/unit Bill Coversion**		**BTUs per year**
Electricity	(kWh)	139,777	**X**	3,413	=	4,770,589,901
Natural Gas	(Therm)	39,055	**X**	103,200	=	4,030,476,000
	(Mcf)		**X**	10,320,000	=	
Fuel Oil	(#2 grade)		**X**	139,000	=	
Coal	(Ton)		**X**	245,000,000	=	

Total BTUs per year for all energy used 4,507,534,901

This is the total amount of energy consumed by the building in BTUs per square feet.

BTUs per square foot per year formula

4,507,534,901	/	62,000	=	72,702
Total Yr. BTUs		**Gross facility ft^2**		**Total BTUs/ft^2/year**

62,000 is the number of conditioned square feet in our building example.

SAMPLE SURVEY

The following sample forms have been completed using information from an actual 62,000 square foot Commercial office building in the Midwest. This is an example of how your own forms might turn out. Do not be concerned if you have difficulty locating some of the requested information, there are energy conservation possibilities in all your building systems.

Use the blank forms at the back of the manual for your own building survey. Most of the information can be provided by your physical plant administrator, but don't hesitate to take full advantage of your Energy Survey Team.

SPACE CONDITIONING EQUIPMENT AND SCHEDULES

SIZE, GROSS SQ. FT. 62000

AREA COOLED 62000 AREA HEATED 62000

TYPE(S) OF OCCUPANCY: (% OR SQ. FT.)

Office 100%

Warehouse

Manufacturing

Retail

Lobbies & Mall (Enclosed)

(Other)

(Other)

(Other)

BUILDING USE AND OCCUPANCY

Fully Occupied: (50% or more of normal)

Weekdays (Hours) 7 AM to 6 PM

Weekends (Hours) 7 AM to 12 MIDNIGHT SATURDAY

0 to 0 Sunday

0 to 0 Holidays

Remarks: Describe below if occupancy differs for different floors, areas, buildings:

LIGHTING SURVEY

1. Interior Lighting Type FLUORESCENT Watts/Ft² Offices

Other

Total Install KW 136

On-Off from Breaker Panel? YES

Wall Switches? No Control Switching? None

Operating Schedule ON 7:30 AM OFF 10:00 PM

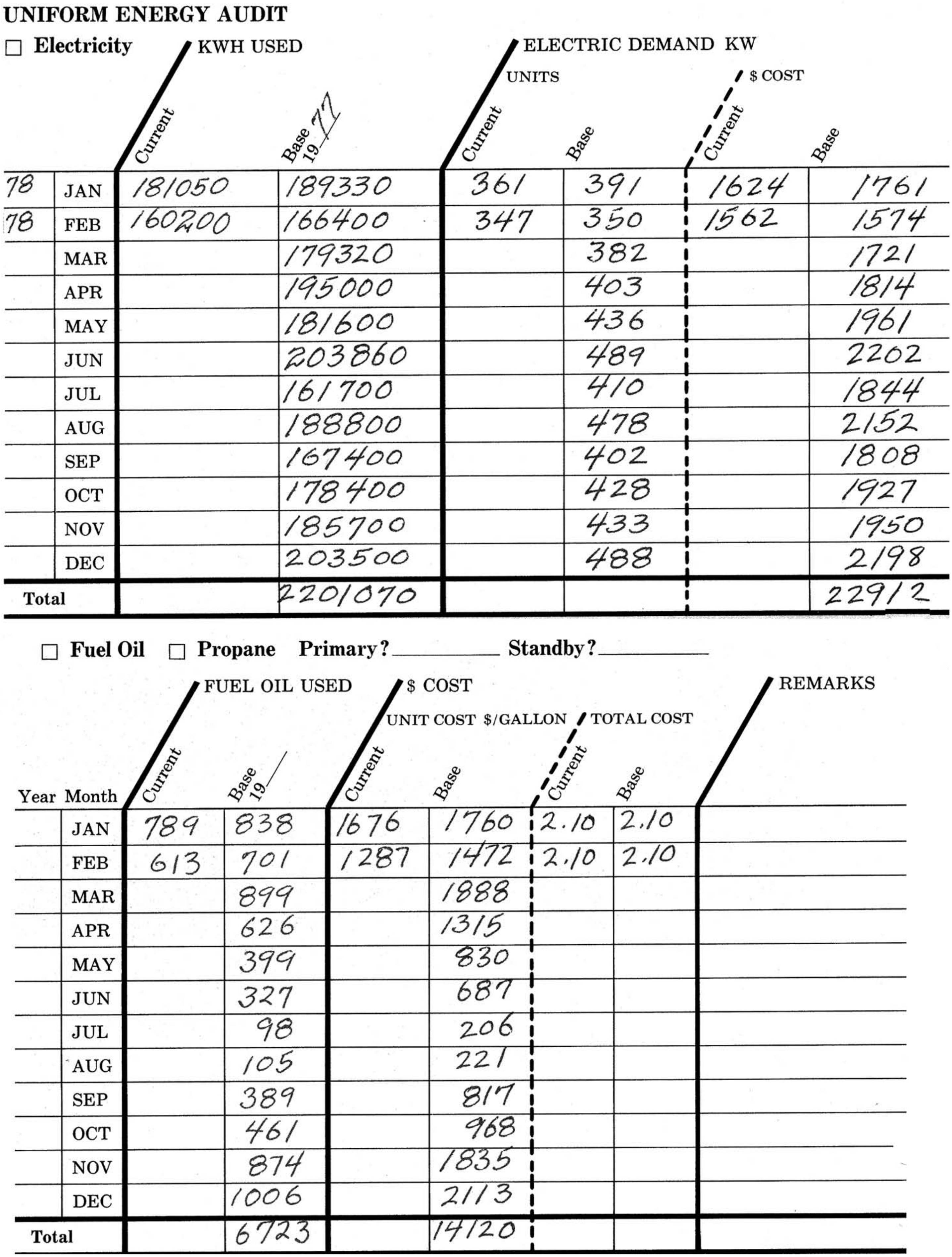
2. Exterior Lighting Type HIGH PRESSURE SODIUM Sq. Ft. Served: 40,000

Total KW 20 Foot-Candles: UNKNOWN

Remarks:

Operating Schedule: ON AND OFF FROM PHOTOCELL

Old Energy Audit Found “Filed Away”



UNIFORM ENERGY AUDIT

☐ **Electricity**

		KWH USED		ELECTRIC DEMAND KW UNITS		$ COST	
		Current	Base 19 77	Current	Base	Current	Base
78	JAN	181050	189330	361	391	1624	1761
78	FEB	160200	166400	347	350	1562	1574
	MAR		179320		382		1721
	APR		195000		403		1814
	MAY		181600		436		1961
	JUN		203860		489		2202
	JUL		161700		410		1844
	AUG		188800		478		2152
	SEP		167400		402		1808
	OCT		178400		428		1927
	NOV		185700		433		1950
	DEC		203500		488		2198
Total			2201070				22912

☐ **Fuel Oil** ☐ **Propane** **Primary?**__________ **Standby?**__________

		FUEL OIL USED		$ COST UNIT COST $/GALLON		TOTAL COST		REMARKS
Year	Month	Current	Base 19__	Current	Base	Current	Base	
	JAN	789	838	1676	1760	2.10	2.10	
	FEB	613	701	1287	1472	2.10	2.10	
	MAR		899		1888			
	APR		626		1315			
	MAY		399		830			
	JUN		327		687			
	JUL		98		206			
	AUG		105		221			
	SEP		389		817			
	OCT		461		968			
	NOV		874		1835			
	DEC		1006		2113			
Total			6723		14120			

$ COST				REMARKS
NIT COST		TOTAL COST		
urrent	Base	Current	Base	
5869	.031	7072	.039	
5158	.031	6408	.040	
5738	.032			
6047	.031			
6538	.036			
7339	.036			
6145	.038			
7174	.036			
6026	.036			
6422	.035			
6500	.036			
7326				
76282	.0347			

THE BUILDING DATA SHEET

These letter symbols (A through J, and TTF) should be determined specifically for your building. Use the information you've listed on the last few pages, as well as the data in the Technical Information section to find the values. These values can then be used in place of the letter symbols in the ECMs that follow:

A (air handling capacity)= ________ $\frac{ft^3}{min}$ or CFM

B (percent ventilation air)= ________ (expressed as a decimal)

C (heating season average outside temperature)= ________ °F

D (cost of heating)= ________ $\frac{\$}{10^6 \text{ Btu}}$

E (heating season length)= ________ $\frac{\text{weeks}}{\text{year}}$

TTF (Thermal Transmission Factor) = ________ $\frac{\text{Btu}}{\text{hr °F ft}^2}$

F (TTF x building perimeter in feet x building height in feet)= ________ $\frac{\text{Btu}}{\text{hr°F}}$

G (cooling season average outside dry bulb temperature)= ________ °F

H (cooling season average outdoor air enthalpy load)= ________ $\frac{10^6 \text{ Btu}}{\text{year 1000 CFM}}$

Assumes a 50 hr week, 10 hrs/day 5 days per week for 1000 CFM of outdoor air brought in and cooled to 55°F.

I (cooling season length)= ________ $\frac{\text{weeks}}{\text{year}}$

J (cost of cooling)= ________ $\frac{\$}{10^6 \text{ Btu}}$

NOTES: a. 1.08 is a constant used in heating calculations
b. 4.5 is a constant used in cooling calculations

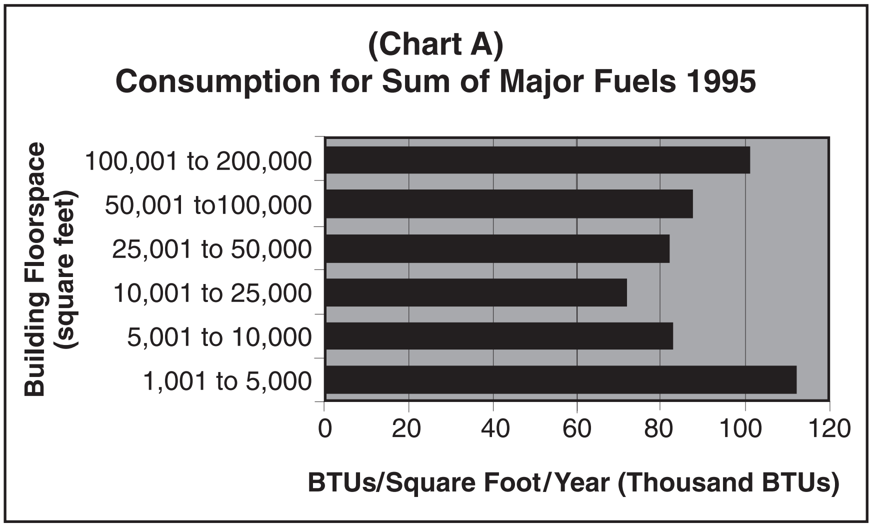
BUILDING SURVEY – EQUIPMENT LIST

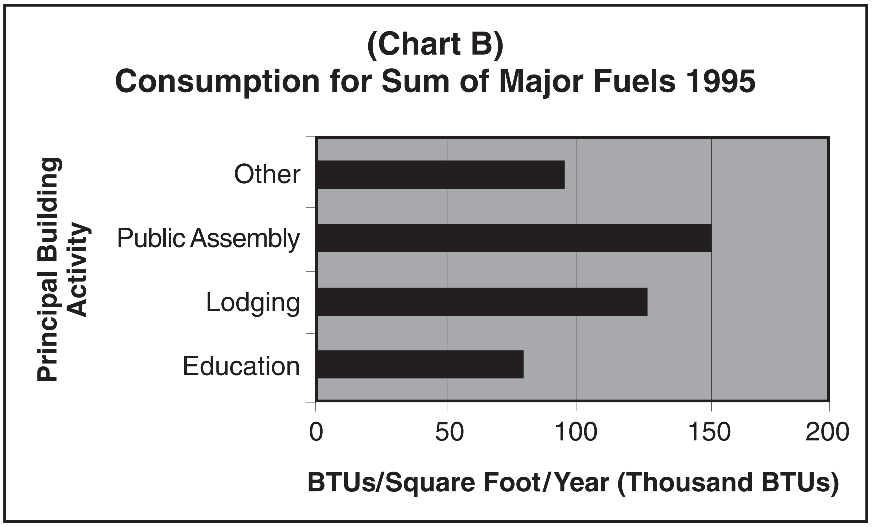
	EQUIPMENT			RATED OUTPUT			RATED INPUT		
	Description	Subsystem	Quantity	☒ HP ☐	CFM	☐ TONS ☐ BTU ☒ KW X	☐ KW ☐	☐ TONS ☐ BTU ☐ X	Area or Served
1	AHU#1								Basem
2		Supply Fan	1	50	24900				
3		Return Fan	1	10	22000				
4									
5	AHU#2								Floors 2
6		Supply Fan	1	75	32000				
7		Return Fan	1	15	29000				
8									
9	AHU#3								Floor 5
10		Supply Fan	1	40	14500				
11									
12	Chiller #1	Recip	1				70		All
13									
14	Chiller #2	Recip	1				70		All
15									
16	Chiller Auxiliaries								
17		Cond Fans	6	30					All
18		CHW Pump	1	15					
19									
20	Boiler#1	Hot Water	1			3600			
21									
22	Pumps,	Misc	7	28					
23									
24	Miscellany		—				13.5		
25									
26	Elevators		2	30					
27									
28									
29									
30									
31									
		Totals							

System: __________ **Date:** __________ **Building/Location:** __________

OPERATING SCHEDULE	OUTSIDE AIR CFM	HOURS PER WK SPACE IS OCCUPIED	NIGHT SET BACK?	REMARKS
Weekdays				
				Double Duct
24 Hours/Day	3000	112	Yes	
Year Around				
		102	Yes	Mulitzone
24 Hrs/Day	3000			
Year Around				
24 Hrs/Day		69	No	Double Duct
Year Around	1500			
24Hrs/Day				
24 Hrs/Day				
24 Hrs/Day				
24 Hrs/Day				Supplies hot deck
				& domestic HW
Varies				15% on time
Varies				
As Required				

HOW DO YOU COMPARE?





AFTER THE SURVEY

The energy audit is an important first step in understanding your facility's energy use. Properly done, it can provide you with a numerical baseline to measure results of future conservation efforts and it will provide you with energy consumption numbers that can be compared to consumption figures from other sources and other similar facilities.

A WORD OF CAUTION

It should be mentioned that low energy usage could be a sign of shut down or inoperable equipment. Investigation of mechanical systems may confirm operation problems. As this point, one might be tempted to leave things as is. However, you have a responsibility to operate your facility equipment as required by governmental and local building codes and provide a safe as well as comfortable environment for building occupants.

ENERGY USE COMPARISONS

The first group of energy data shown in charts A – C was extracted from the Department of Energy's, Energy Information Administration (EIA) June 1995 Energy Consumption Series, Buildings and Energy Report.

Chart A provides energy usage rates (BTUs/sq ft/yr) for different building sizes. Note that the buildings between 10,000 and 25,000 sq ft achieved the lowest rate and the smallest and largest sized facilities comsumed energy at the highest rate, that is, were the least efficient.

Chart B provides a comparison of fuel consumption between four building types that differ by activity. Education buildings consumed at the lowest rate while public/assembly buildings consumed fuel at approximately twice the rate. Building construction and use, HVAC mechanical efficiency, choice of fuels, HVAC control system and operation, occupancy rates and other factors might all contribute to the difference. These and other factors should all be considered during the survey and fact gathering process.

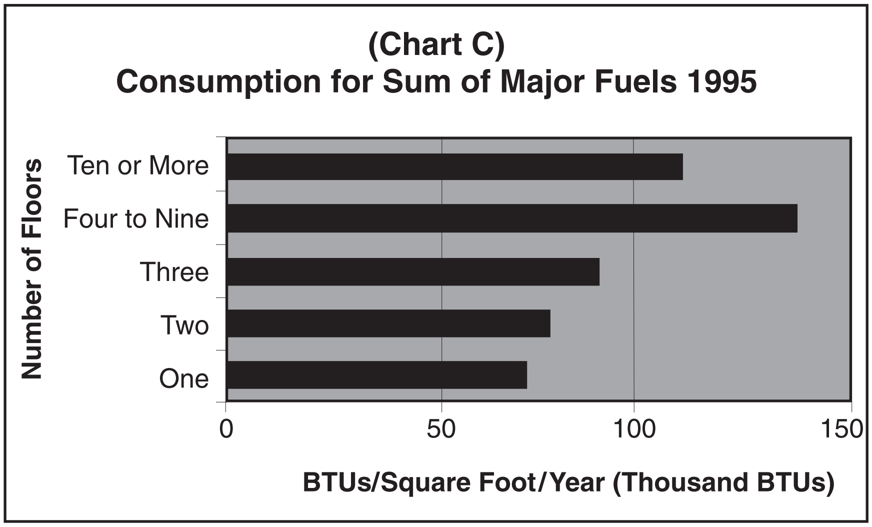


Chart C zeros in on one aspect of building construction, number of floors, that was found to strongly correlate to higher or lower rates of fuel consumption. Note that buildings of one or two stories had the lowest rate of fuel consumption while buildings of four to ten or more stories consumed energy at near twice the rate.

These findings are not exactly intuitive and contrary to many people's expectation. If your building data revealed that your facility uses less or equal amounts of energy, congratulations. If your facility is average or uses more energy than other facilities, don't despair, read on to learn about the implementation of energy plans and what can be done to improve energy efficiency in many types of mechanical and lighting equipment found in virtually every building type and size.

OPERATING IN THE ENERGY CONSERVATION MODE

Your new controls, building automation system, HVAC rehabilitation and building envelope improvements all create new operating parameters for your facility. Operating in the energy conservation mode involves adjustments both to equipment schedules and human habits.

In order to make a smooth transition, you need to consider these key elements in your new operation:
- Operating Procedures
- Training
- Energy Auditing
- Reporting

OPERATING PROCEDURES

When building modifications are complete, new operating procedures must be written and communicated to necessary personnel. The procedure for implementing each operational improvement should clearly define objectives, actions required, personnel responsibility and, if necessary, emergency actions. These procedures should be published in a conveniently indexed reference manual. And the objectives and steps being taken need to be communicated to all facility personnel.

For example: If automatic light control turns off plant lighting at 4:30 p.m., security and janitorial personnel must know how to restore lighting when necessary or a timing system must be in place to accomplish this automatically. And more attention may have to be paid to emergency lighting where main lighting is turned off.

TRAINING: THE KEY TO COOPERATION

Training can play an equally important role in the acceptance of your plans and the efficiency with which they are implemented. Training becomes a form of motivation. It allows you to involve personnel in achieving your goal of energy conservation; you give them a vested interest in working toward the goal.

Training, as one important form of communication, can also break down barriers. What's new can be threatening if it isn't fully understood. Explain the research and objective setting that preceded the implementation of the plan. Let personnel know what you plan to do and why, so they will feel more at ease with being involved in the program. Invite their participation and suggestions for all future improvements. This can and should be a very positive community mission.

Important steps to take in training are:

- Identify any new tasks associated with new energy projects;

- Assign tasks and responsibilities to personnel;

- Train personnel (before tasks are to begin)

- Establish measurable energy goals for trainees;

- Recognize people who have successfully completed training;

- Reinforce and communicate broadly successful performance against objectives.

ENERGY AUDITING

"Energy Auditing" simply means an orderly month-by-month accounting of energy used in a building for comparison against a budget, goal or another standard of performance. You're probably using a form for reviewing various segments of your own business. This tracking is crucial in order to measure, verify and redirect your energy conservation program. It equips you to answer questions posed by upper management, change the program to fit new developments and new construction, and monitor the performance of your plan and the people implementing it.

WHAT CAN ENERGY AUDITING DO FOR YOU?

Here are some things a comprehensive audit can do for you:

- Establish energy norms for your building;

- Determine good or bad performance of HVAC and other systems;

- Discover when and where new conservation measures are needed;

- Measure effectiveness of conservation steps already taken;

- Establish a data base for new conservation steps or capital expenses you wish to implement;

- Measure how well operating practices and schedules are being followed;

- Account for increases or decreases in operating costs;

- Relate energy costs to occupancy and weather;

- Analyze your electric utility rate structure and indicate how to buy your electricity at lower cost.

HOW TO DO IT

There is an ideal way to administer an energy audit and a realistic way. You may have inexperienced personnel or even lack the personnel to administer the audit. If you want (or need) to start simply, following these steps will provide you with a minimum auditing system.

1. Establish a "base year" against which to compare (we suggest going only two or three years);

2. Set a month-by-month energy budget or target, one year in advance;

3. Record all energy purchased or used on a monthly basis, in both energy units and dollars;

4. Report performance against your target or budget on a monthly (or quarterly) basis with the help of your financial department to upper management. Delivering and reporting your progress is one of the only means by which successful energy management programs are sustained. Report your performance both in energy units and dollars.

HOW TO DO IT BETTER

Taking these additional steps will give you a much better insight into where your energy is being used and, incidentally, where it's being wasted. How far you go with these procedures is dictated by how much you have at stake. If your energy bills exceed $20,000 per month, strongly consider implementing every one of these steps.

1. Log all energy used on a daily basis (necessary to track usage against weather and occupancy). Making use of electronic data logging to simplify trending and graphics for presentation;

2. Survey your facility and account for 95 percent or more of your demand for both electricity and fuel.

3. Record electrical demand patterns by making daily or hourly data logging of kilowatts (KW) vs. hours of the day until you have established what is normal for hot, cold and "in between" weather and for occupied and unoccupied times. Identify critical periods of the day and year;

4. Keep an up-to-date file on your utility rates for gas, electricity, purchased steam, etc;

5. Recalculate all electric invoices. Consider duplicate metering. This will give you a better perspective of where your money is going including:
(a) Energy charges
(b) Fuel adjustments
(c) Demand charges and ratchet clauses
(d) Time-of-day metering
(e) Power factor penalties
(f) Taxes
(g) Errors in utility invoices

6. Meet with utility reps at least twice a year to fully understand availability, costs of their product and alternative conservation strategies they promote;

7. Graph your month-by-month energy units showing last year, this year and target for each utility. (This will get employees and other occupants more interested in conservation efforts. Post your graphs in several conspicuous areas, promote your progress at appropriate events and participation in any building efficiency programs;

8. Record heating degree days every day. Record cooling degree days every day. (Often listed in newspaper or available from your local weather service);

9. Consider sub-metering where tenants control energy usage;

10. Seek out reliable computer generated building energy savings simulations to compare their potential usage/savings to your targets;

11. Be sure each major boiler and chiller, whether electric, gas or oil-fired has a separate meter to track energy usage.

REPORTING

The auditing information you collect can serve many purposes. It can be used for management reports, for public relations and for analysis of performance against goals. Exhibits A, B, and C and D provide a few samples of documents using an energy audit for different purposes and audiences. They are typical of those produced by companies that already do energy auditing.

Exhibit A is a monthly report on electricity. Similar logs would be used for fuel oil, propane, natural gas, purchased steam, etc. This is the raw data used to prepare management reports such as Exhibit D.

Exhibit B is a one-day graph showing kilowatt (KW) demand hour by hour. This was taken from a one month computer printout, usually available from the utility after the end of the month. It shows an interesting “lump” commencing at 5 a.m. Why? You should be asking too!

Exhibit C is a graph representative of how a certain building is responding to an energy conservation goal.

Exhibit D shows how data can be presented to management to show performance accomplished against target and cost avoidance.

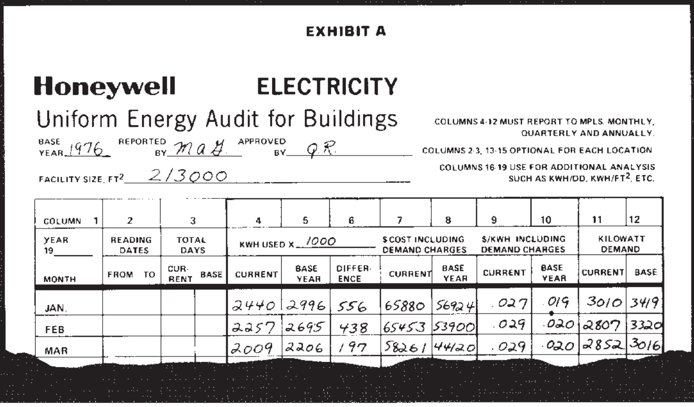


EXHIBIT A

Honeywell ELECTRICITY

Uniform Energy Audit for Buildings

BASE YEAR 1976 REPORTED BY Ma G. APPROVED BY Q.R.

FACILITY SIZE, FT² 213000

COLUMNS 4-12 MUST REPORT TO MPLS. MONTHLY, QUARTERLY AND ANNUALLY.

COLUMNS 2-3, 13-15 OPTIONAL FOR EACH LOCATION

COLUMNS 16-19 USE FOR ADDITIONAL ANALYSIS SUCH AS KWH/DD, KWH/FT², ETC.

COLUMN 1	2		3		4	5	6	7	8	9	10	11	12
YEAR 19	READING DATES		TOTAL DAYS		KWH USED X 1000			$ COST INCLUDING DEMAND CHARGES		$/KWH INCLUDING DEMAND CHARGES		KILOWATT DEMAND	
MONTH	FROM	TO	CUR-RENT	BASE	CURRENT	BASE YEAR	DIFFER-ENCE	CURRENT	BASE YEAR	CURRENT	BASE YEAR	CURRENT	BASE
JAN.					2440	2996	556	65880	56924	.027	.019	3010	3419
FEB					2257	2695	438	65453	53900	.029	.020	2807	3320
MAR					2009	2206	197	58261	44120	.029	.020	2852	3016

EXHIBIT B

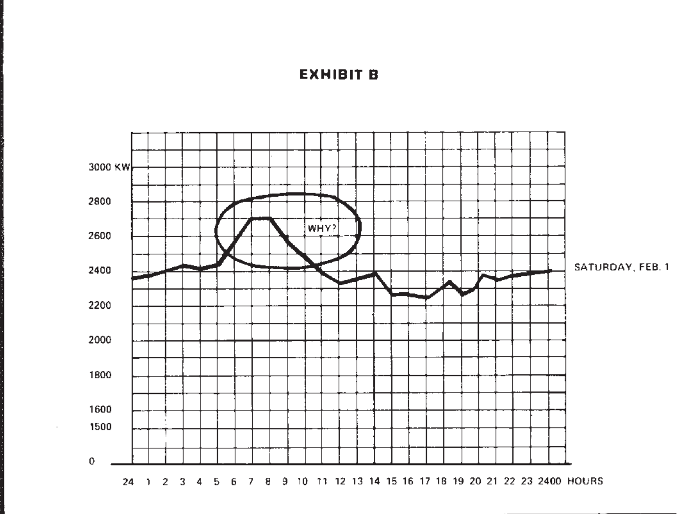
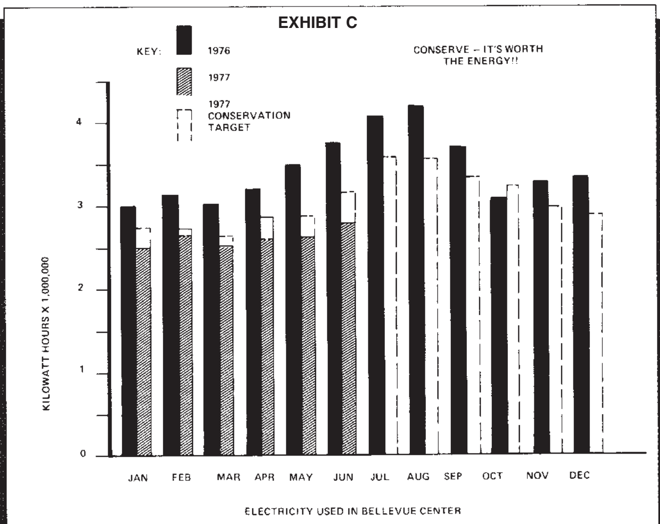


EXHIBIT C



ELECTRICITY USED IN BELLEVUE CENTER

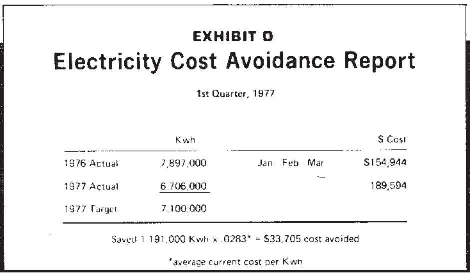


EXHIBIT D

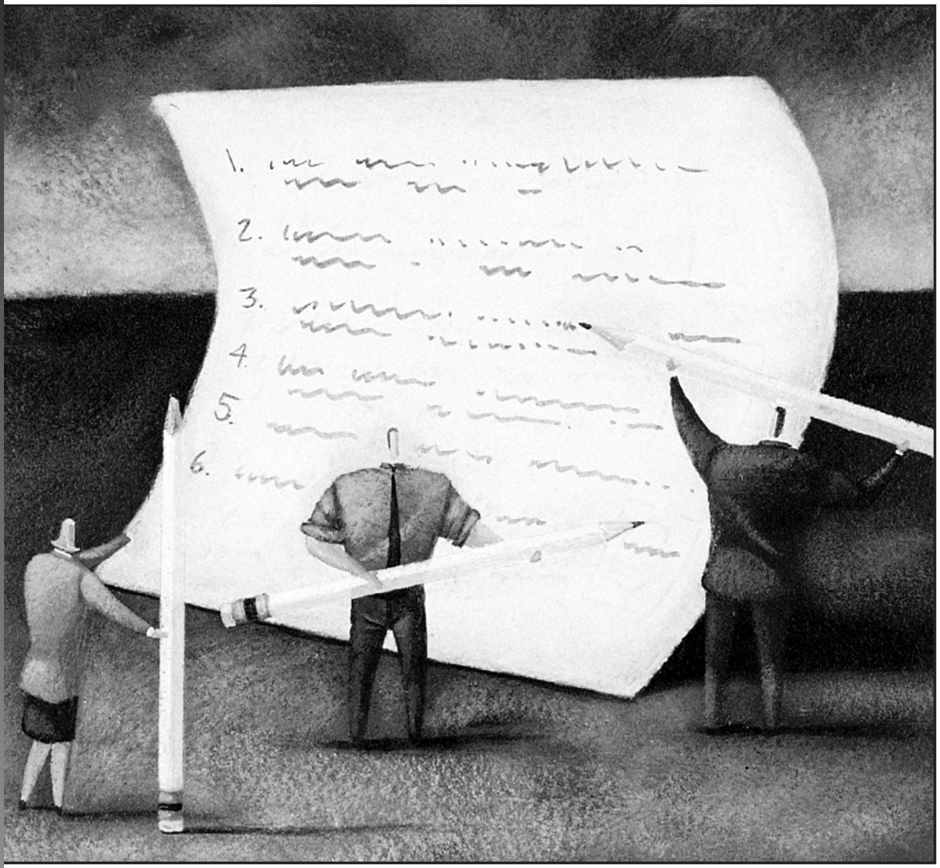
Electricity Cost Avoidance Report

1st Quarter, 1977

	Kwh		$ Cost
1976 Actual	7,897,000	Jan Feb Mar	$154,944
1977 Actual	6,706,000		189,594
1977 Target	7,100,000		

Saved 1 191,000 Kwh x .0283* = $33,705 cost avoided

*average current cost per Kwh



PLANNING AHEAD

A long-range energy plan is essential for energy managers hoping to go beyond the low cost/no cost stages of energy savings. It's essential because to achieve the maximum potential you almost always have to make significant investments (capital expenditures). This elevates energy management to the same level of planning, proposing and budgeting as new construction or major renovation projects. Significant long-term savings call for nothing less.

WHAT IS A LONG-RANGE PLAN?

A completed long-range plan usually meets these criteria:

The plan:

1.Published and updated annually;

2. Contains a statement of long-range goals and objectives;

3. Includes a statement of endorsement by top management;

4. Briefly describes the facility in "energy terms" and includes an energy history, summary of the audit (survey), energy cost and use trends and breakdown of principal energy users (HVAC, lights, office and process equipment);

5. Describes each completed and planned energy project with units and dollars projected or saved, status, time schedule and cost

6. Summarizes the energy projects showing total cost and total savings, all related to the long-range objectives;

7. Seeks and obtains approval of each project, phased so that each may be budgeted at the appropriate time.

The style of your long-range plan is also important. The people who will review and approve the document may not have technical backgrounds. The plan should be written in non-technical languages so it is understandable by the Boards of Directors, financial officers, etc.

Finally, we recommend that your plan be issued by the energy manager (or committee) of the facility. If you are including vendor proposals or an outside consultant's report as a part of the plan, these should be summarized in the body of the text and attached to the appendix.

(SAMPLE)

LONG RANGE ENERGY PLAN FOR XYZ FACILITY
JUNE 15, 1999
UPDATED FEBRUARY 11, 2000

I. Goals

The goals of this energy plan were originally stated on June 1, 1999, in a letter from facilities director, E.L. Howard (See Appendix A), calling for:

- 11% reduction in energy units for fiscal year commencing July 1, 1999
- 20% reduction in energy units by July 1, 2003

These targets are to be met without loss of program quality and with due regard for occupant comfort. Base year is July 1, 1998, to June 30, 1999.

II. Energy Summary

This facility consists of 47,000 square feet of heated and cooled floor space. Base year energy use is tabulated below:

1998 BASE YEAR ENERGY USE

USAGE	ELECTRICITY		NATURAL GAS		COST	
	Kwh	**%**	**CCF**	**%**	**Dollars**	**%**
HVAC	337,328	45	34,067	71	$43,765	55
Lighting	288,480	39	———	0	$18,520	24
Pool	120,392	16	14,045	29	$16,844	21
Totals	746,200	100	48,112	100	$79,129	100

ENERGY COST TRENDS FOR FACILITY

	1998	1999	2000	2001	2002
Electricity (Loaded Cost)	$.0642	$.0699	$.0753	$.0795*	$.080*
Natural Gas ($/CCF)	$.649	$.694	.755	$.768*	$.797*

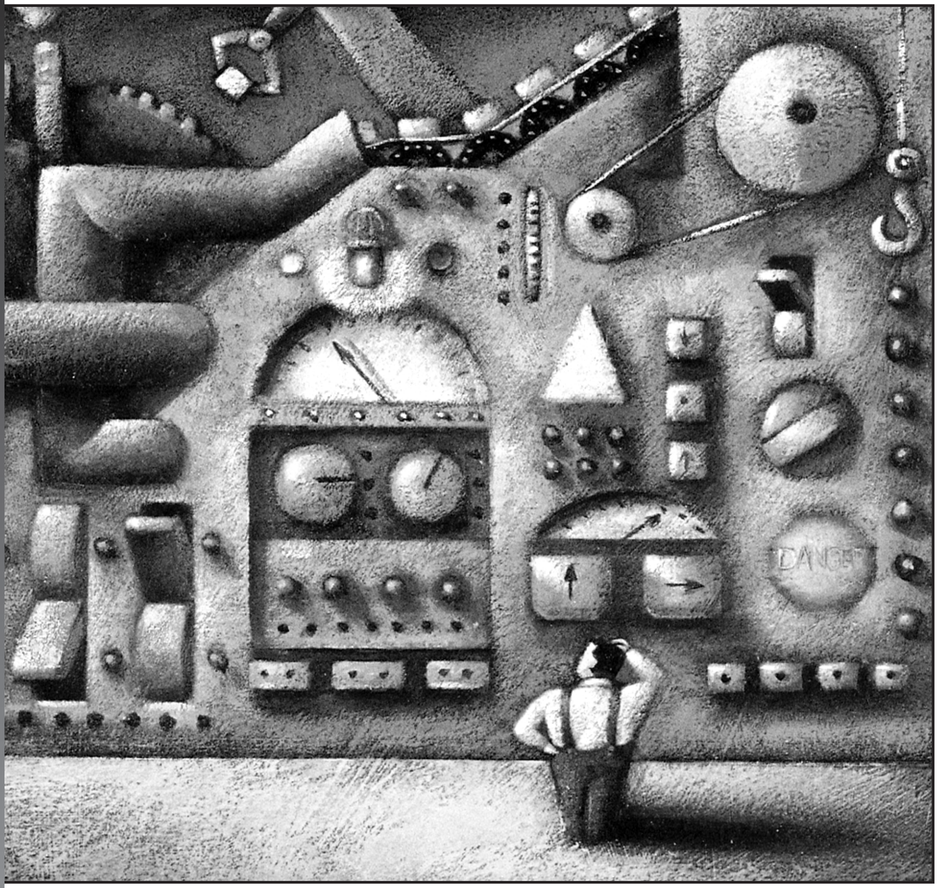
*Projected by suppliers

1996 BASE YEAR ENERGY USE

Item	Unit and Costs Avoided	Status	Start	COMPLETE
131	Light Level Reduction 90,705 Kwh, $6,550	Complete	1-2-98	2-15-99
142	Honeywell Controls 82,080 Kwh, $5,737	Budgeted	5-2-00	3-6-01
158	Add economizers, SF1 & 6 S_________	Study	———	———

III. Actions Requested

1. Include project 142 in 2000 budget.
2. Approve study costs for item 158.



IMPLEMENTING YOUR ENERGY PLAN

WHERE TO START

There are three main areas you can explore in implementing your energy plan:
1.Low cost/no cost items,
2.Mechanical system and controls,
3.Computerized energy management

It makes sense to explore them in the order listed; do everything you can at minimal cost before you invest more substantial sums.

START WITH NO COST/ LOW COST ITEMS

These simple control adjustments and operational tips are extremely cost effective since no direct or new investment is involved. You will find some of these listed in the Introduction. Get the people in your building involved in energy conservation. Assign specific persons to turn lights “off” and “on” at proper times or immediately investigate low cost ways to “automatically” control lights. Schedule janitors and those working beyond normal hours so lights (and HVAC systems) are “on” only one floor or section of the building. Finally, publicize your energy program to building occupants and ask for their support. Posting graphs showing goals vs. actual will emphasize the group effort and accomplishments.

Since these items don’t usually need budgets or high level approvals, they will get your program off to a fast start. And they have the added benefit of involving people in your energy program. Don’t forget to “ask” building occupants for “their” conservation ideas. Then, implement as many as you possibly can.

CONSIDER CONTROLS

People-based solutions alone are not enough. There are many other available to you at modest cost.

Your building's thermostats, control systems, and the boilers, chillers, air conditioning units, and air handlers they serve may be outdated. If they aren't, they may have been selected to minimize initial cost rather than to maximize efficiency. In any case, it makes sense to ensure that your heating and cooling equipment is operating at peak efficiency. This may necessitate control replacement or renovation.

Check each thermostat within the building. Make sure the heating season checks are performed during cold weather and the cooling checks during warm weather. Then, continue to follow this checklist from thermostats through your entire HVAC system. Don't be surprised to find that some system components don't function properly or might actually have been disabled (such as outdoor air damper linkages). Many unexplained practices occur over time. Maybe they "seemed" to make sense to someone back then. But, they may be costing you dearly and/or interfere with the proper operation of basic systems such as proper amounts of ventilation.

CONTROL IS THE KEY WORD

This Energy Conservation Manual and Workbook is made up of ideas you can use to reduce energy consumption, cut heating and air conditioning costs, and still maintain a comfortable working environment. Every one of the recommendations is based on better control of your building operation, through more efficient control devices (thermostats, low-leakage dampers, etc.), improvements in equipment efficiency, sharper scheduling methods or simply avoiding current wasteful practices.

The more Honeywell Energy Conservation Measures (ECMs) you choose to work into your building operation (or into new construction specifications), the more energy efficient your building will be. These ECMs are explained in the next chapter.

But here, control is a key word. The more efficiently you control, the more effective your energy conservation-with-comfort plan is going to be.

ANALYZE AUTOMATION

Once you have increased the mechanical operating efficiency of your temperature control system, evaluate building automation based energy management to regulate your total building system. It represents a substantial investment, but frequently offers significant additional savings to justify the cost. This could be as simply as upgrading to the newest electronic commercial setback thermostats with other advanced features. It might mean one of the lower cost but highly sophisticated light commercial building automation systems based on either constant volume or variable air volume control. Or, for larger facilities or campuses, a comprehensive building automation system with computerized energy management, and potentially integrated access, fire and security with quite sophisticated building HVAC control.

These are the kinds of programs we're talking about:

A PROGRAM TO START AND STOP YOUR EQUIPMENT

A computer based building control system can start and stop your HVAC equipment based on a duty cycling schedule, or simply reduce motor speeds to maintain comfort conditions while trimming significant energy by optimizing electrical energy use. Based on changing seasons and weather extremes, it can also provide optimal ramping up and down of comfort conditions to let your building coast into lower occupancy times of day/night and bring building temperatures up or down just in time for occupancy the next day, avoiding wastefully hours of runtime. Then, during the day when space conditions are satisfied, the building control system might even cycle equipment on and off for short periods or throttle back on airflow in less occupied zones to save even more energy...automatically.

A PROGRAM TO SLASH ELECTRIC DEMAND CHARGES

By constantly monitoring your electric usage rate and comparing this to the previous peak stored in the computer's memory, the system sheds low priority electric loads. This helps conserve on electric demand charges.

A PROGRAM TO HARNESS NATURE'S FREE COOLING AIR SOURCE

Systems are capable of continually checking inside and outside climate conditions, constantly comparing temperature and humidity, searching for when outside air might provide "free cooling" rather than running your mechanical air conditioning system.
Sophisticated sensors constantly seek the air with the lowest total heat content. As the heat content changes, the system automatically adjusts the ventilation damper controls to bring the most economical air source into your building. Automatic savings provided by your building control system without the need for operating personnel, seasonal changes or maintenance procedures beyond operation checks. And the best systems also test that they are operating properly and alert your staff or service contractor if they are not functioning properly. Automatically and without constant checks.

A PROGRAM TO REDUCE CHILLER LOAD

You can take the pressure off your chillers with a program that takes the indoor and outside climate data and issues discharge air temperature reset commands. This permits partial unloading of your chiller plant because somewhat warmer chilled water can be used as outdoor temperatures warm up. The resulting reduction in the temperature can save you significant amounts of energy.

A PROGRAM TO CONTROL LIGHTING

Lighting, as we have discussed earlier, can be one of your biggest energy wasters. Your building control system can be programmed with occupancy times, janitorial schedules and weekend and holiday schedules for centralized lighting control. Lighting is turned "on" only when necessary and "off" as early as possible – automatically. And, a survey will reveal whether a technology "step-up" in lights and ballasts might be long overdue and show how quickly it can pay for itself.

THE LAST RESORT

In spite of your best efforts, control and automation improvements may not save enough to meet your energy goals. Tune-ups for your existing equipment may not be enough. There are situations where major rehabilitation of HVAC mechanical and control systems is practical and necessary to save energy. The following checklist may reveal areas where a consultant or HVAC vendor can help you evaluate and/or design improvements into your basic HVAC system.

Heating Season Checklist for Control System Upgrade

Yes No

DAY OPERATION

☐ ☐ 1. Are thermostats set at 70°F or less in public spaces? (ECM #1 and #2)
☐ ☐ 2. Are they locked?
☐ ☐ 3. If they are non-locking, are there provisions to keep settings at 70°F?
☐ ☐ 4. Does the thermostat work?
☐ ☐ 5. When set below 70°F , does valve, damper or heat source turn on?
☐ ☐ 6. Is thermostat calibrated within 1° to 2°F of setting?
☐ ☐ 7. When set below 70°F, is the cooling source (outside air or refrigeration) locked out? (ECM #3)
☐ ☐ 8. Is ZEB (Zero Energy Band) operation provided for? (ECM #3)
☐ ☐ 9. Have you reduced outdoor air ventilating quantity when the building is occupied, consistent with ASHRAE 62 standards? (20 CFM/person for office areas.) (ECM #5)
☐ ☐ 10. Do you shut off exhaust fans in toilets, kitchens or labs when areas are unoccupied? (ECM #5)

NIGHT/UNOCCUPIED OPERATION

☐ ☐ 1. When unoccupied, is the temperature setting automatically reduced by at least 10°F?
☐ ☐ 2. Are warehouses or storage areas kept at lower temperatures than occupied areas?
☐ ☐ 3. Do you close the outdoor air damper when the building is unoccupied? (ECM #6)
☐ ☐ 4. Do outdoor air dampers close *tightly* during night or unoccupied times? (ECM #7)

STEAM, HOT WATER OR ELECTRICAL SYSTEMS FOR SPACE HEATING

☐ ☐ 1. Have you determined above what outdoor temperature you can shut off heating boilers, heat exchanger pumps and/or electric heat? (ECM #17)
☐ ☐ 2. If #1 differs for different buildings, zones or times of day, have these parameters been determined?
☐ ☐ 3. Do you have procedures in effect to shut off the heat, as determined under 1 and 2?
☐ ☐ 4. For space-heating hot water systems, have you checked to see if water temperatures, or temperature schedules, are at or below original design? (Remember that original design temperatures for secondary hot water were based on a 75°F room temperature requirement.) See chiller/boiler control sections.
☐ ☐ 5. Is domestic hot water set no higher than 110°F? (Certain types of buildings may have higher requirements.)

COOLING SEASON CHECKLIST

☐ ☐ 1. Are thermostats set and locked at 75°F or above?
☐ ☐ 2. If thermostats can't be locked, what provisions have been made to keep them at 75°F?
☐ ☐ 3. During nights, weekends, holidays or when the building is unoccupied, is the mechanical cooling shut off?
☐ ☐ 4. Have provisions been made to prevent full use of cooling at night, when only overtime workers are present?
☐ ☐ 5. Have you raised the "cold deck" temperature (leaving temperature from cooling coil) in terminal reheat systems? (58°F is a suggested trial setting.)
☐ ☐ 6. Have you raised the chilled water temperature, leaving your chillers at least 2°F?
☐ ☐ 7. On all comfort cooling Direct Expansion (DX) compressors and rooftop units, have you readjusted SUCTION pressures to raise suction temperatures by 4°F?
☐ ☐ 8. Have you reduced outside air brought in by rooftops and all other cooling systems to 0 CFM unoccupied?
☐ ☐ 9. 10 to 30 CFM (per person) or moreoccupied?
☐ ☐ 10. Have you adjusted economizer control to use outdoor air for cooling? (ECM #*)
☐ ☐ 11. Have you reduced light levels to reduce the cooling load?
☐ ☐ 12. Have you adjusted cooling tower water to the lowest practical limit, in order to reduce compressor power needed? (Check with compressor manufacturer for trial values.)
☐ ☐ 13. Have you reduced static pressure on high-pressure fan systems to be consistent with air delivery at the farthest unit or space?

HVAC and Electrical Systems Update

Yes No

BOILERS

☐ ☐ 1. Do you regularly run flue gas analysis?

☐ ☐ 2. Do you have a means of evaluating the analysis and taking corrective action?

☐ ☐ 3. Do you have an up-to-date program for both fireside and water-side maintenance of heat exchange surfaces?

CHILLERS

☐ ☐ 1. If chillers are sequenced, are they sequenced so the most efficient unit goes on first?

2. When were heat exchange surfaces last cleaned:

☐ ☐ 3 months ago

☐ ☐ 6 months ago

☐ ☐ 9 months ago

☐ ☐ 3. Are they scheduled for the next cleaning? If so, when? ______________________

____

HVAC SYSTEMS

☐ ☐ 1. Can they utilize outdoor air cooling? (ECM #4)

☐ ☐ 2. Can they use zero outside air when the building (space served) is unoccupied? (ECM #6)

☐ ☐ 3. Can perimeter units take advantage of heat (return air) from interior zones?

☐ ☐ 4. Have you applied "spot cooling" units for small areas that must be used during off hours?

LOADING DOCK

☐ ☐ 1. Have shrouds or air curtain fans been considered?

HAVE HEAT RECOVERY UNITS BEEN CONSIDERED FOR:

Yes No

☐ ☐ 1. Exhaust air?

☐ ☐ 2. Process air?

☐ ☐ 3. Transformer vaults?

☐ ☐ 4. Equipment rooms, computers?

☐ ☐ 5. Air compressor coolers?

☐ ☐ 6. Other wasted heat?

ELECTRICAL SYSTEM – POWER FACTOR CORRECTION

☐ ☐ 1. Do you pay any penalty for low-power factor?

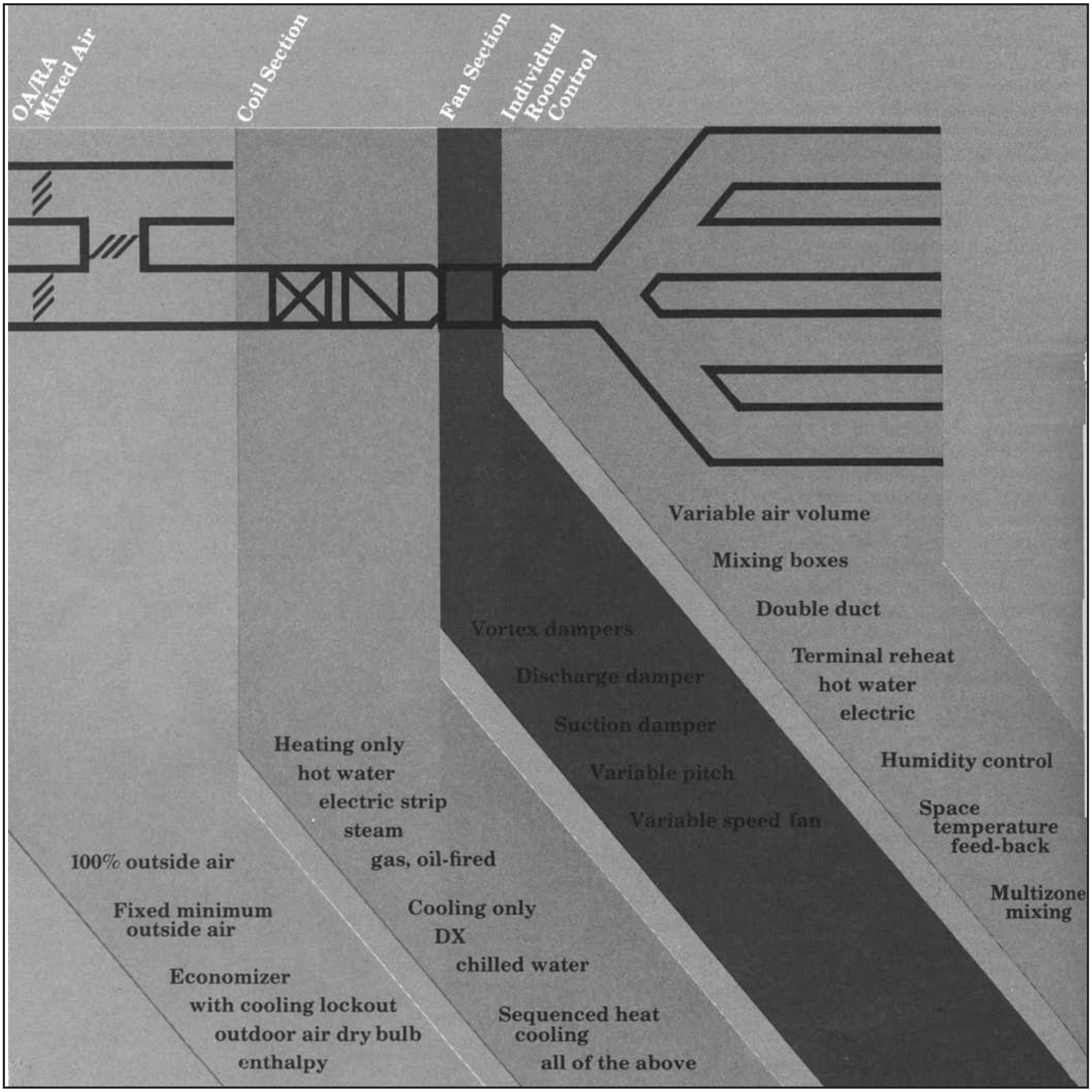
☐ ☐ 2. If yes, has correction via capacitors been considered?

ALTERNATE FUELS AND CONTINGENCY PLANNING

A complete energy plan must provide for contingencies. You may not have all the answers, but you should consider these **"what if"** questions for your facility:

☐ ☐ 1. If I cannot obtain the fossil fuel I rely on to heat the building, is there a written procedure to deal with such a crisis?

☐ ☐ 2. When time-of-day pricing of electricity becomes a reality, do I have a cost analysis for how it affects us and a plan for dealing with it?



ENERGY CONSERVATION MEASURES

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ENERGY CONSERVATION MEASURES

Energy Conservation Measures (ECMs) are proven strategies for both new buildings and modifications of existing systems. They're specifically set up to let you view some of the possible answers to your operation and energy conservation problems in your buildings.

ENERGY IMPROVEMENTS FOR YOUR HVAC SYSTEMS AND BUILDING ENVELOPE

Energy conservation doesn't end with control improvements, though they often represent the fastest payback opportunities. Any complete energy program must go beyond control upgrades to further curtail energy waste. Some of these areas are lighting, insulation, glazing, alternate fuels, heat recovery, thermal storage, spot cooling (or heating), motor speed control, power factor correction and replacement of antiquated or inefficient HVAC equipment. And, don't overlook the likelyhood that portions of your systems may simply need maintenance or repair. An outside air damper may be "blocked open" which causes a huge amount of heating season waste, though it's masked by your systems capacity to heat that frigid air up to room temperature. You'd never know of the waste without a survey or inspection. Or, steam traps may be leaking in a large number of cases forcing your boiler to heat up many times its normal capacity of water rather than simply circulate steam and return condensate. Again, this might double a heating bill, or more, and not be readily recognized.

Honeywell energy specialists and energy specialist partners are trained and equipped to analyze various energy savings improvements from the standpoint of cost effectiveness and return on investment (ROI). In addition to engineering and cost feasibility studies, consulting engineers can also help establish energy conservation priorities, engineer complex system solutions, select vendors, draw up project specifications and bid documents.

BENEFIT/COST ANALYSIS

Almost every energy conserving procedure has a cost associated with it. For a project that entails a low initial cost and/or has immediate savings, the simple payback analysis can be used.

$$\text{Simple Payback Period} = \frac{\text{Project Cost}}{\text{Annual Savings}}$$

Energy saving measures requiring heavy capital investment can be handled in several ways. Two ways are:

1. Payback Period Analysis
2. Life Cycle Costing

Payback Period Analysis
This is simple and straightforward, but *only* takes into account:

c= capital cost
S= annual operating and maintenance savings
r= interest rate
n= number of years to achieve payback

$$n = \frac{\log \frac{S}{(S{-}rc)}}{\log (1 + r)}$$

EXAMPLE: A heat recovery device will cost $11,950 installed. It will save $3,126/year in energy and will cost $884/year to maintain. Interest rate is 9.5%.

c= $11,950
S= $3,126 – $884 = $2,242
r= 9.5% (.095)

$$n = \frac{\log \frac{2242}{2242 - (.095 \times \$11{,}950)}}{\log (1 + .095)}$$

$$n = \frac{\log 2.0259}{\log 1.095} = \frac{.3066}{.0394} = 7.78 \text{ yrs}$$

Simple Payback

$$\text{Payback} = \frac{\$11{,}950}{(\$\text{ savings/yr}) - (\text{yearly O\&M costs})}$$

Example

$$\text{Payback} = \frac{\$30{,}000}{\$15{,}000 - \$3{,}000} = 2.5 \text{ years}$$

Return On Investment - R.O.I.

$$\text{R.O.I.} = \frac{\frac{\$\text{Savings}}{\text{year}} - \frac{\text{Depreciation}}{\text{year}} - \frac{\text{O\&M Cost}}{\text{year}} - \text{Tax Savings}}{\textbf{FIRST COST}}$$

$$\frac{\$15{,}000 - \$2{,}000 - \$3{,}000 - \$1{,}000}{30{,}000} = 30\%$$

LIFE CYCLE COSTING

Life Cycle Costing (LCC) is a method of calculating the total cost of ownership over the life of an asset. LCC is justified when:

1. Investment is large.
2. Life is several years or more.
3. Energy cost is large.
4. Efficiency from maintenance and operation can bring about substantial savings.
5. Two or more alternative systems are being compared.

Taxes, depreciation, cost of maintenance, inflation, and other factors are all a part of LCC.

(LCC analyses are complex and usually require completion by a person trained in this technique.)

SELECTING ECMs AND PRIORITIES

If you have considered all the possible conservation steps, you might come up with a list like this:

- Lighting retrofit
- Lighting controls
- DDC controls upgrade
- Variable speed drives installation

The headings include the areas worth considering as you priortize your actions. Proposals to management should be detailed.

When you propose a project, you should include supporting descriptive data. This should give management necessary background, budget data, implementation techniques, and monitoring systems, etc.

Your presentation to management should follow the same steps that you used to analyze your savings potential. It should identify the energy savings opportunities, prioritize them, justify the expenses associated with them, and motivate management to support your efforts.

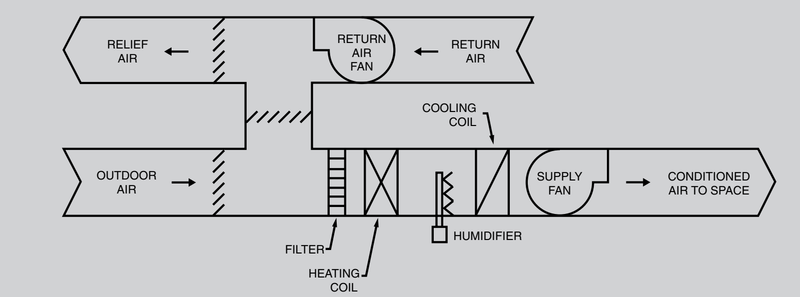
IDENTIFYING SYSTEMS

A variety of heating/cooling system types are mentioned throughout this manual. The following diagrams and description have been included in this workbook to assist in identifying and understanding the systems that serve their buildings.

(We have included definitions of terms in the Glossary.)

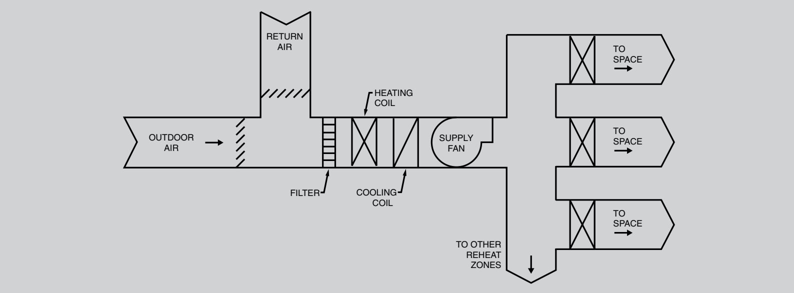
SINGLE ZONE SYSTEM

Single zone systems consist of a mixing, conditioning and fan section. The conditioning section may have heating, cooling, humidifying or a combination of capabilities. Single zone sysems can be factory assembled roof top units, or built up from individual components and may or may not have distributing duct work.



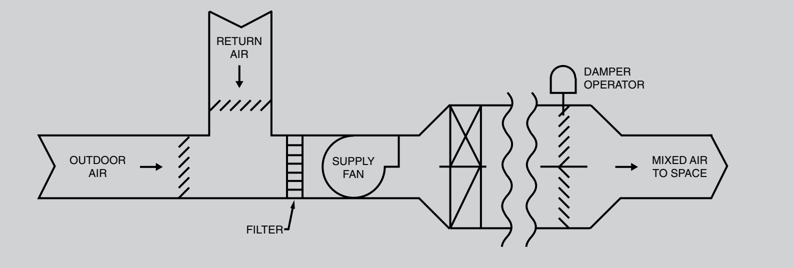
TERMINAL REHEAT SYSTEM

Reheat systems are modifications of single zone systems. Fixed cold temperature air is supplied by the central conditioning system and reheated in the terminal units when the space cooling load is less than maximum. The reheat is controlled by thermostats located in each conditioning space.



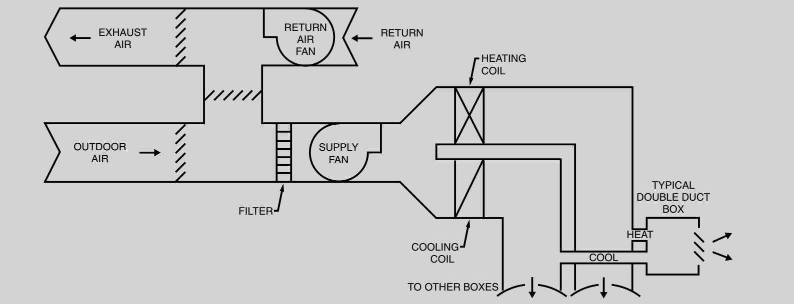
MULTIZONE SYSTEMS

Multizone systems condition all air at the central system and mix heated and cooled air at the unit to satisfy various zone loads as sensed by zone thermostats. These systems may be packaged roof top units or field fabricated systems.



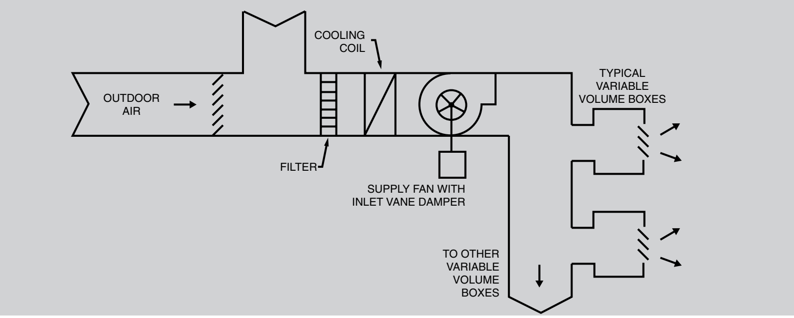
DUAL DUCT SYSTEMS

Dual duct systems are similar to multi-zone systems except heated and cooled air is ducted to the conditioned spaces and mixed as required in terminal mixing boxes.



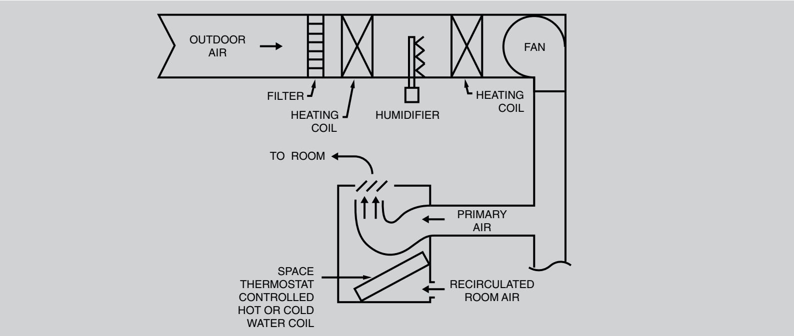
VARIABLE AIR VOLUME SYSTEMS

A variable air volume system delivers a varying amount of air as required by the conditioned spaces. The volume control may be by fan inlet (vortex) damper, discharge damper or fan speed control. Terminal sections may be single duct variable volume units with or without reheat, controlled by space thermostats.



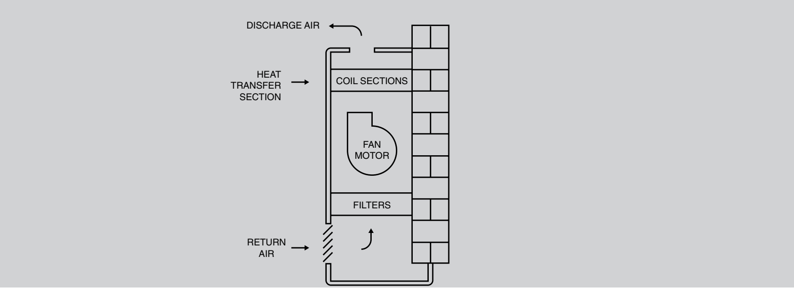
INDUCTION SYSTEMS

Induction systems generally have units at the outside perimeter of conditioned spaces. Conditioned primary air is supplied to the units where it passes through nozzles or jets and by induction draws room air through the induction unit coil. Room temperature control is accomplished by modulating water flow through the unit coil.



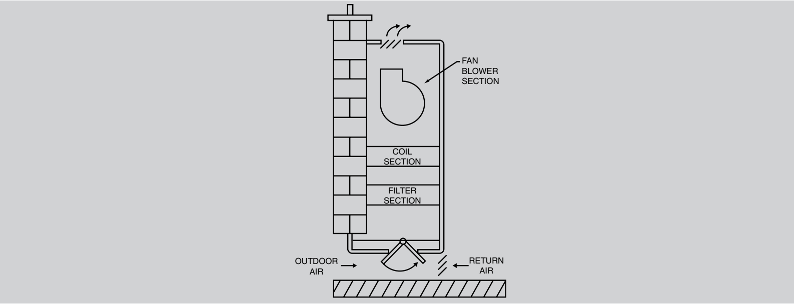
FAN COIL UNITS

A fan coil unit consists of a cabinet with heating and/or cooling coil, motor and fan and a filter. The unit may be floor or ceiling mounted and uses 100% return air to condition a space.



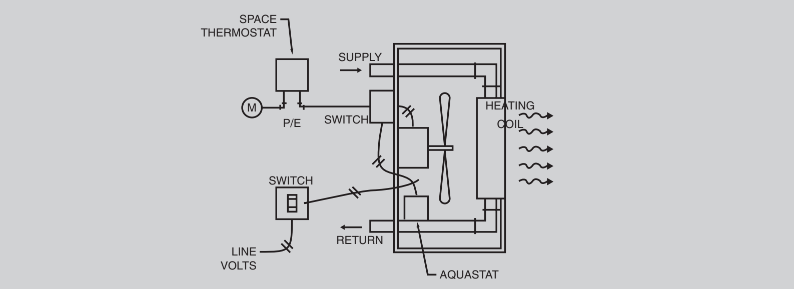
UNIT VENTILATOR

A unit ventilator consists of a cabinet with heating and/or cooling coil, motor and fan, a filter and return air - outside air mixing section. The unit may be floor or ceiling mounted and uses return and outside air as required by the space.



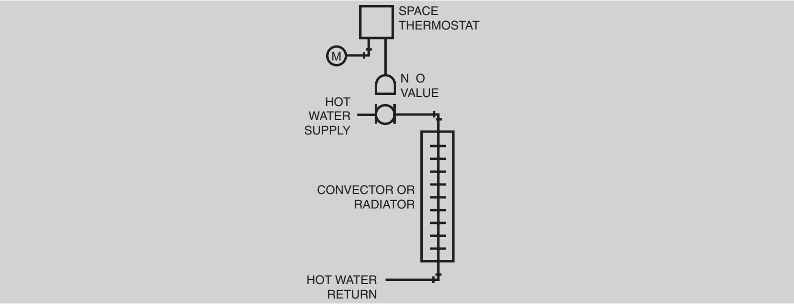
UNIT HEATER

Unit heaters have a fan and heating coil which may be electric, hot water or steam. They do not have distribution duct work but generally use adjustable air distribution vanes. Unit heaters may be mounted overhead for heating open areas or enclosed in cabinets for heating corridors and vestibules.



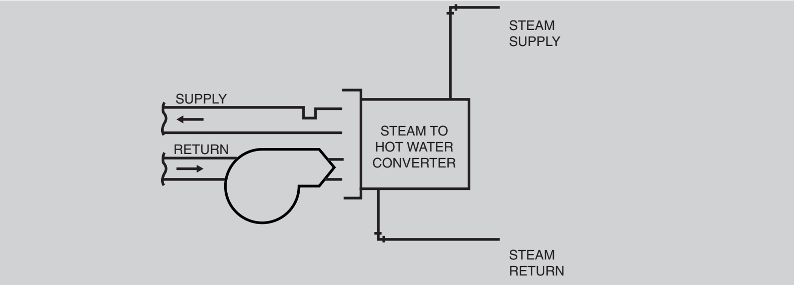
PERIMETER RADIATION

Perimeter radiation consists of electric resistance heaters or hot water radiators usually within an enclosure but without a fan. They are generally used around the conditioned perimeter of a building in conjunction with other interior systems to overcome heat losses through walls and windows.



HOT WATER CONVERTERS

A hot water converter is a heat exchanger that uses steam or hot water to raise the temperature of heating system water. Converters consist of a shell and tubes with the water to be heated circulated through the tubes and the heating steam or hot water circulated in the shell around the tubes.



CONSERVATION WITH THERMOSTATS

(ECM's # 1 – 4)

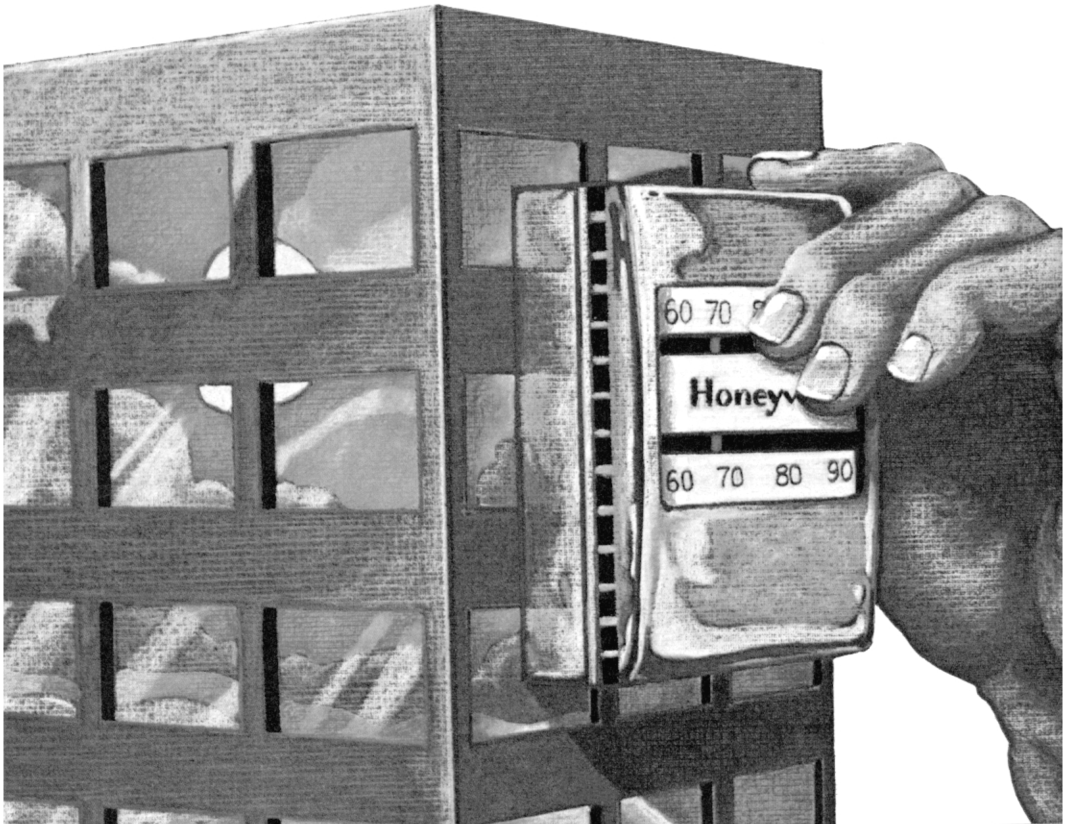
The single most important symbol of your energy conservation program is the room thermostat. This one device represents the entire HVAC system to the occupant who doesn't understand or even care where the hot air or cold air comes from. It is also the easiest place for you to start your conservation with comfort program.

There are four basic ways your thermostat can be used to save energy:

1. Set the thermostat up to 78°F when in the cooling mode. Make sure when you do this that you don't cycle your heating equipment.
2. Set the thermostat down to 68°F (or as low as you can) during the heating mode. Again be sure you do not cycle the cooling equipment in the process.
3. Prevent the simultaneous heating and cooling in systems that provide both on demand such as reheat systems, four pipe fan coil systems and single zone systems.
4. Set the temperature to a lower temperature at night or unoccupied hours during the heating season. Set a higher temperature or shut the system off at night or during unoccupied times in the cooling season.

In each case, the savings result from maintaining space temperature at a level that is compatible with new standards and cannot be easily defeated by occupants. Current ASHRAE Standards suggest a maximum of 72°F in the heating mode and 75°F in the cooling mode. Many buildings use setpoints of 68°F for heating and 75°F to 78°F for cooling to maximize savings with satisfactory occupant comfort. Consider a realistic temperature setting for your particular circumstances.

All of the benefits of implementing these strategies can be undone by allowing free unlimited access to thermostat adjustments. You should select a thermostat that can be locked or enclosed to prevent occupant adjustment.



Larger facilities typically have used pneumatic thermostats. There have been many advances in commercial electronic thermostats during the last fifteen years that make them the “thermostat of choice” for most light commercial buildings and many larger facilitities, especially those using packaged rooftop equipment.

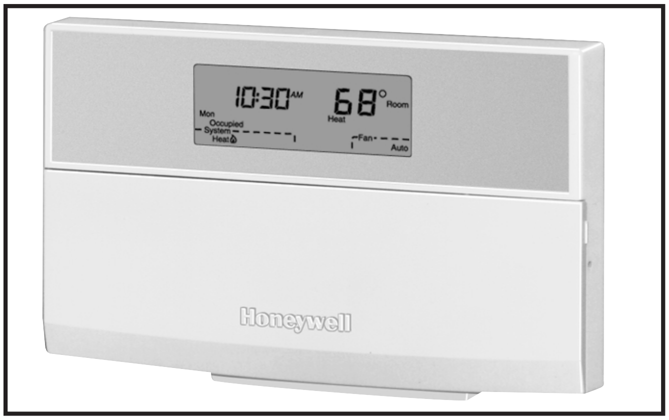
Most electronic thermostats have some specific benefits built into them that help provide control and energy savings. Off-set for unoccupied periods is a “must” feature. Many times people consider “accuracy” alone as the most important feature. They do not realize that many times precise control (if achieved) is done at the expense of equipment life by over cycling the equipment.

Honeywell commercial thermostats provide “intelligent” control well beyond simple precise or accurate sensing. Extensive simulation testing has provided a unique level of optimization that delivers precise control but not at the expense of equipment life. Carefully designed control algorithms that integrate proprietary PID and anticipator functions are unmatched in the industry. The following feature listing is intended to help you be aware of the many features now incorporated in Honeywell commercial electronic thermostats that you should look for in your selection.

- Proportional plus Integral (P+I) control fluctuations
- Intelligent Recovery optimizes equipment start times

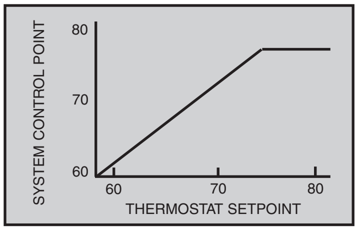
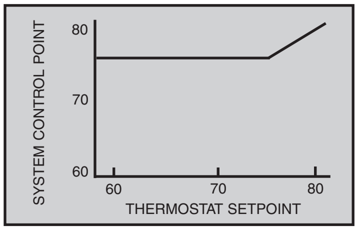
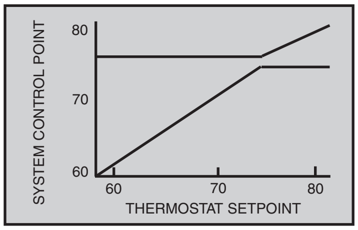
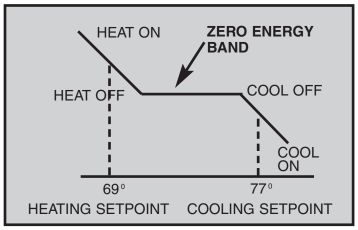
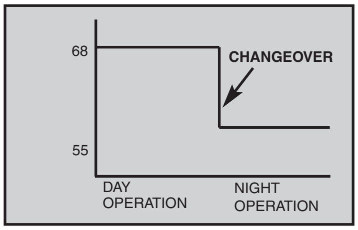
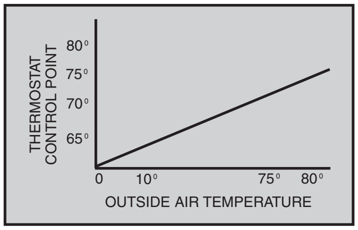
- Keypad lockout
- Remote sensor capability
- Selectable temporary occupied
- Discharge air sensor capability
- Adjustable heating/cooling deadband

- Setpoint range stops
- Networking / Remote
- Intelligent Fan‘ provides occupied period continuously fan operation
- Automatic or manual changeover

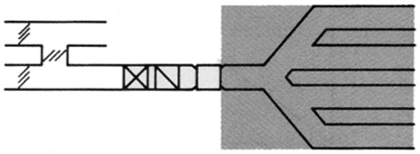


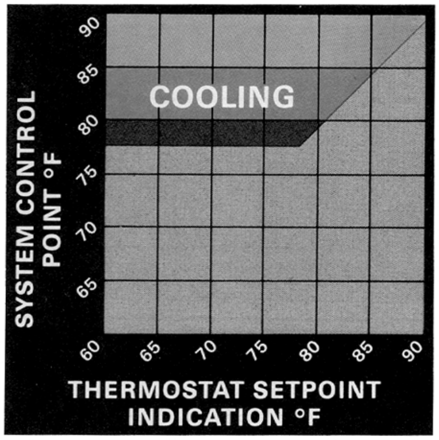
T7300 Electronic Thermostat

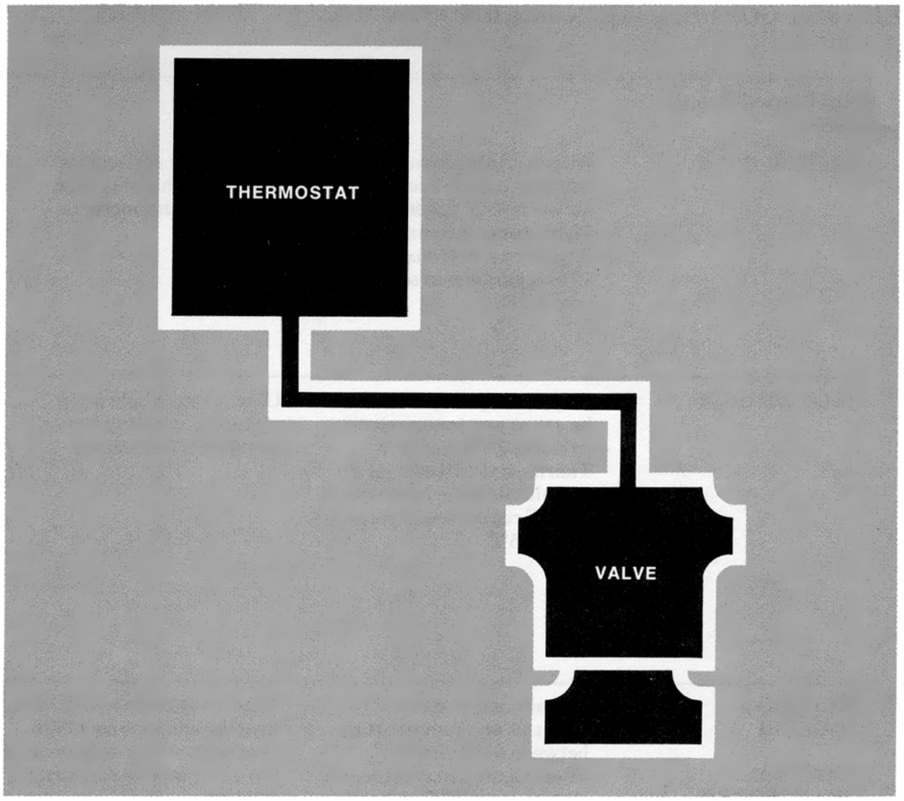
ENERGY CONSERVATION THERMOSTATS SELECTION CHART

Limited Control Range Thermostat HEATING ONLY	Setpoint adjustments can be set at any temperature between 60°F and 90°F. Thermostat will control at setpoint up to a maximum of 75°F regardless of setpoints over 75°F.	Used on any application requiring a heating type pneumatic thermostat.	
COOLING ONLY	Setpoint adjustments can be set at any temperature between 60°F and 90°F. Thermostat will control at setpoint down to a minimum of 75°F regardless of setpoints below 75°F.	Used on any application requiring a cooling type pneumatic thermostat.	
HEATING & COOLING (CENTRAL CHANGEOVER)	Set point adjustments can be set at any temperature between 60°F and 90°F. Thermostat will control at a maximum of 75°F in heating and a minimum of 75°F in cooling.	Used on systems which have heating and cooling where changeover from one mode to the other is done at a central location.	
Zero Energy Band Thermostat HEATING AND COOLING (SIMULTANEOUS OR SEQUENCED)	Separate setpoints for heating and cooling. Cooling and heating systems overlap. Thermostat controls at a maximum of 75°F in heating and a minimum of 75°F in cooling. Zero energy band (no heating or cooling to space) is adjustable.	Used on systems where both heating and cooling are available at the same time. Thermostat prevents simultaneous use of both. Typical use is on system with central cooling and reheat at zones such as VAV with perimeter or terminal reheat, dual duct mixing boxes, etc.	
Day-Night Thermostat HEATING AND UNIT VENTILATORS	Separate setpoints for day and night operation. Changeover is accomplished manually or automatically from time clock or automation system.	Used on heating systems or unit ventilator systems.	
Remote Reset Thermostats HEATING OR COOLING	Thermostat can be set at unit from 60°F to 85°F. Control point is then reset to a higher or lower temperature by outdoor air temperature, manually or from automation system.	Used on heating or cooling systems where it is desirable to remotely change setpoint from another input, i.e. outdoor air temperature or BTU meter.	

ECM #1
SETTING COOLING SET POINT UP DURING SUMMER







$$\boxed{\text{F}}\ \frac{\text{Btu}}{\text{hr °F}} \times \boxed{\text{cooling setup}}\ \text{°F} \times \boxed{\text{operating hrs/wk}}\ \frac{\text{hrs}}{\text{wk}} \times \boxed{\text{I}}\ \frac{\text{wks}}{\text{yr}} = \boxed{\text{COOLING BTU s SAVED PER YEAR}}$$

$$\boxed{\text{COOLING BTU s SAVED PER YEAR}} \times \boxed{\text{J}}\ \frac{\$}{10^6\ \text{Btu}} = \boxed{\text{COOLING \$ SAVED PER YEAR}}$$

EXAMPLE: 80,000 ft² two story building, 200′ x 200′ x 25′, TTF=.81, Chicago weather data, 60,000 cfm. Air handling capacity, 25% ventilation air, gas fuel cost $ 10.0 per 1000 ft³, electric cost .10 per Kwh, air cooled condenser; old thermostat setpoint: 74°F new setpoint: 68°F from 7 A.M. to 5 P.M. 5 days per week and 55° at all other times during the winter; 78°F in cooling mode 10 hours per day 5 days per week.

$$\boxed{16{,}200}\ \frac{\text{Btu}}{\text{hr °F}} \times \boxed{4}\ \text{°F} \times \boxed{50}\ \frac{\text{hrs}}{\text{wk}} \times \boxed{20.9}\ \frac{\text{wks}}{\text{yr}} = \boxed{67.7\ \times 10^6\ \text{COOLING BTU s SAVED PER YEAR}}$$

$$\boxed{67.7\ \times 10^6} \times \boxed{8.33}\ \frac{\$}{10^6\ \text{Btu}} = \boxed{\$564\ \text{COOLING \$ SAVED PER YEAR}}$$

CAUTIONS: Setting the room thermostat to a lower setpoint during the winter on interior zones where cooling is required can increase your energy costs. Likewise setting thermostats up on zones with reheat can cause increased energy usage by cycling the reheat element.

A = air handling capacity $\left(\frac{ft^3}{min}\right)$*; B = present ventilation air (%); C = heating season average outside temp. (°F); D = cost of heating* $\left(\frac{\$}{10^6\,Btu}\right)$*;*

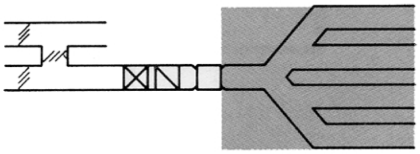
E = heating season length (weeks); F is a calculated value: F = TTF x building perimeter in feet x building height in feet $\left(\frac{Btu}{hr\,°F}\right)$*;*

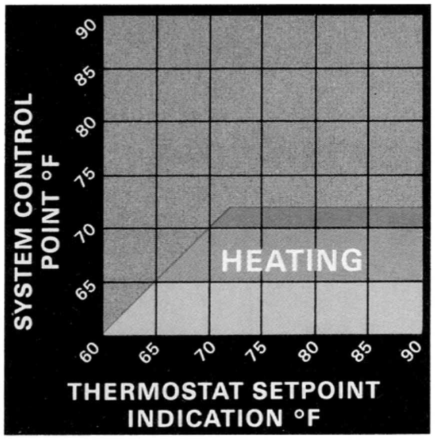
TTF = Thermal Transmission Factor found in the Technical Data Section on page 103; G = cooling season average outside dry bulb temp. (°F);

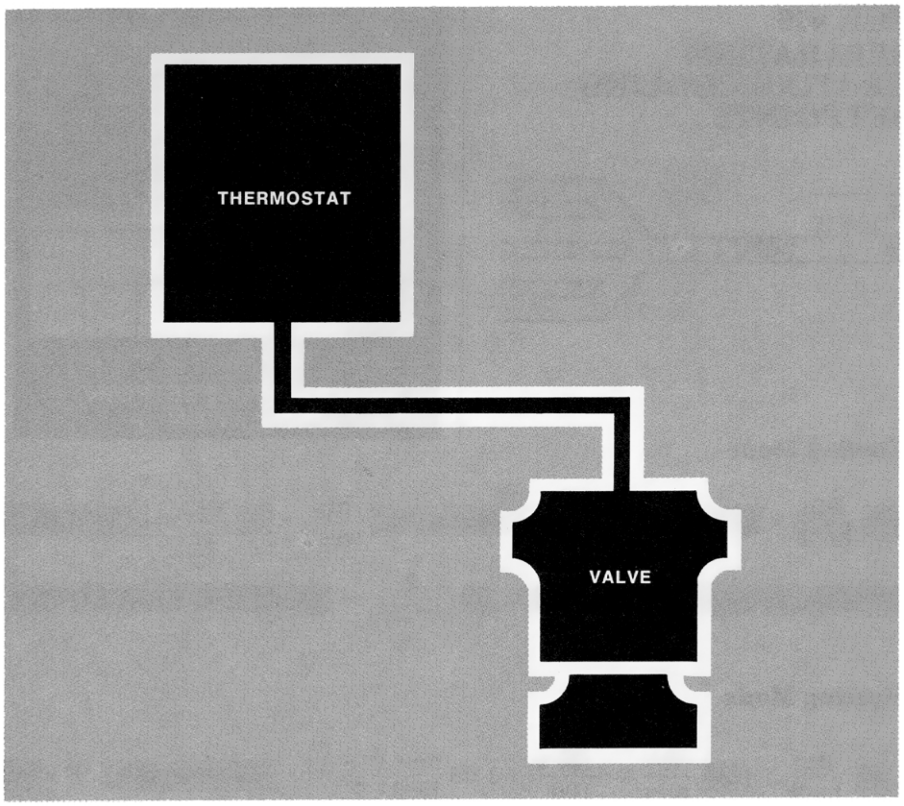
H = seasonal cooling load for outdoor air $\left(\frac{10^6\,Btu}{year\,1000\,cfm}\right)$*; I = cooling season length (weeks); J = cost of cooling* $\left(\frac{\$}{10^6\,Btu}\right)$

K = seasonal cooling savings $\left(\frac{10^6\,Btu}{yr\,1000\,cfm}\right)$

ECM # 2
SETTING HEATING SET POINT DOWN DURING WINTER







$$F\ \frac{Btu}{hr\ °F} + \left(A\ \frac{ft^3}{min} \times \frac{B\ \%}{100} \times 1.08\ \frac{min\ Btu}{hr\ ft^3\ °F} \right)\Bigg] \times \text{heating set down}\ °F \times \text{operating hrs/wk}\ \frac{hrs}{wk} \times E\ \frac{wks}{yr} =$$

HEATING BTU s SAVED PER YEAR

$$\text{HEATING BTU s SAVED PER YEAR} \times D\ \frac{\$}{10^6\ Btu} = \text{HEATING \$ SAVED PER YEAR}$$

EXAMPLE: 80,000 ft² two story building, 200′ x 200′ x 25′, TTF = .81, Chicago weather data, 60,000 cfm. Air handling capacity, B% ventilation air, fuel cost $13.60 per 10⁶ Btu's, electric cost .10 per Kwh, air cooled condenser; old thermostat setpoint: 74°F new setpoint: 68°F from 7 A.M. to 5 P.M. 5 days per week and 60° at all other times during the winter; 78°F in cooling mode 10 hours per day 5 days per week.

$$\Bigg[\ 16{,}200\ \frac{Btu}{hr\ °F} + \left(60{,}000\ \frac{ft^3}{min} \times \frac{18.6\ \overset{*}{\%}}{100} \times 1.08\ \frac{min\ Btu}{hr\ ft^3\ °F} \right)\Bigg] \times 6\ °F \times 50\ \frac{hrs}{wk} \times 30\ \frac{wks}{yr} =$$

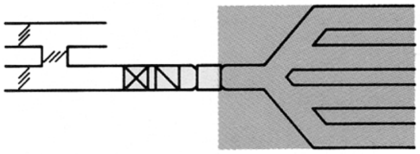
254.3×10^6 HEATING BTU s SAVED PER YEAR

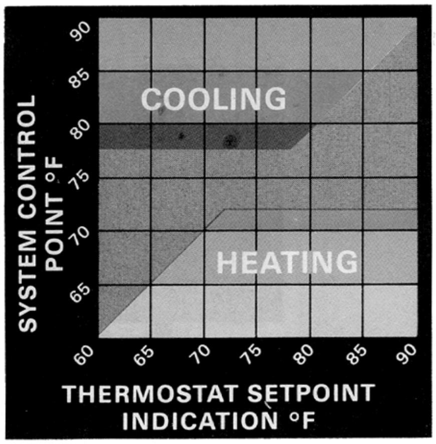
$$254.3 \times 10^6 \times 13.60\ \frac{\$}{10^6\ Btu} = \$3458\ \text{HEATING \$ SAVED PER YEAR}$$

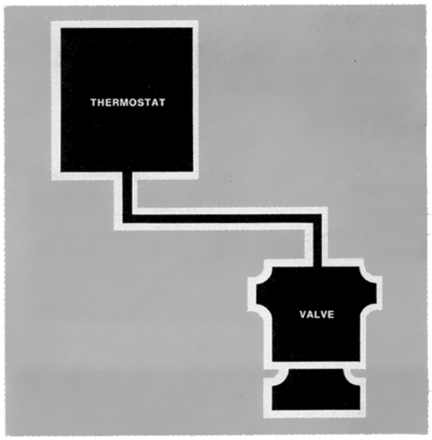
*See ASHRAE tables in appendix. 80K sq ft x 7 people/1000 x 20 CFM/P = 11,200 CFM
min OA 11,200 CFM/60,000 CFM capacity = 18.6% OA minimum.

CAUTIONS: Setting the room thermostat to a lower setpoint during the winter on interior zones where cooling is required can increase your energy costs. Likewise setting thermostats up on zones with reheat can cause increased energy usage by cycling the reheat element.

ECM #3 SEPARATING HEATING/COOLING SETPOINTS







Cooling Mode

$$\text{F}\ \frac{\text{Btu}}{\text{hr}\ ^\circ\text{F}} \times \text{cooling setup}\ ^\circ\text{F} \times \text{operating hrs/wk}\ \frac{\text{hrs}}{\text{week}} \times \text{I}\ \frac{\text{wks}}{\text{yr}} = \text{COOLING BTU s SAVED PER YEAR}$$

$$\text{COOLING BTU s SAVED PER YEAR} \times \text{J}\ \frac{\$}{10^6\ \text{Btu}} = \text{COOLING \$ SAVED PER YEAR}$$

Heating Mode

$$\left[\text{F}\ \frac{\text{Btu}}{\text{hr}\ ^\circ\text{F}} + \left(\text{A}\ \frac{\text{ft}^3}{\text{min}} \times \frac{\text{B}\ \%}{100} \times 1.08\ \frac{\text{min Btu}}{\text{hr ft}^3\ ^\circ\text{F}}\right)\right] \times \text{heating set back}\ ^\circ\text{F} \times \text{operating hrs/wk}\ \frac{\text{hrs}}{\text{wk}} \times \text{E}\ \frac{\text{wks}}{\text{yr}} =$$

$$\text{HEATING BTU s SAVED PER YEAR}$$

$$\text{HEATING BTU s SAVED PER YEAR} \times \text{D}\ \frac{\$}{10^6\ \text{Btu}} = \text{HEATING \$ SAVED PER YEAR}$$

EXAMPLE: 80,000 ft^2 two story building, 200′ x 200′ x 25′, TTF=.81, Chicago weather data, 60,000 cfm. Air handling capacity, B % ventilation air, gas fuel cost \$10.0 per 1000 ft^3, electric cost .10 per Kwh, air cooled condenser; old thermostat setpoint: 74°F new setpoint: 68°F from 7 A.M. to 5 P.M. 5 days per week and 60° at all other times during the winter; 78°F in cooling mode 10 hours per day 5 days per week.

Cooling Mode

$$16{,}200\ \frac{\text{Btu}}{\text{hr}\ ^\circ\text{F}} \times 4\ ^\circ\text{F} \times 50\ \frac{\text{hrs}}{\text{wk}} \times 20.9\ \frac{\text{wks}}{\text{yr}} = 6.7 \times 10^6\ \text{COOLING BTU s SAVED PER YEAR}$$

$$6.7 \times 10^6 \times 10.00\ \frac{\$}{10^6\ \text{Btu}} = \text{\$670 COOLING \$ SAVED PER YEAR}$$

Heating Mode

$$\left[16{,}200\ \frac{\text{Btu}}{\text{hr}\ ^\circ\text{F}} + \left(60{,}000\ \frac{\text{ft}^3}{\text{min}} \times \frac{\overset{*}{18.6\ \%}}{100} \times 1.08\ \frac{\text{min}}{\text{hr ft}^3\ ^\circ\text{F}}\right)\right] \times 6\ ^\circ\text{F} \times 50\ \frac{\text{hrs}}{\text{wk}} \times 30\ \frac{\text{wks}}{\text{yr}} =$$

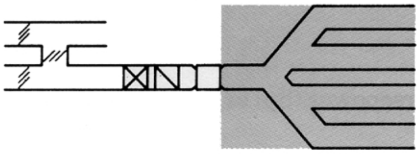
$$254.3 \times 10^6\ \text{HEATING BTU s SAVED PER YEAR}$$

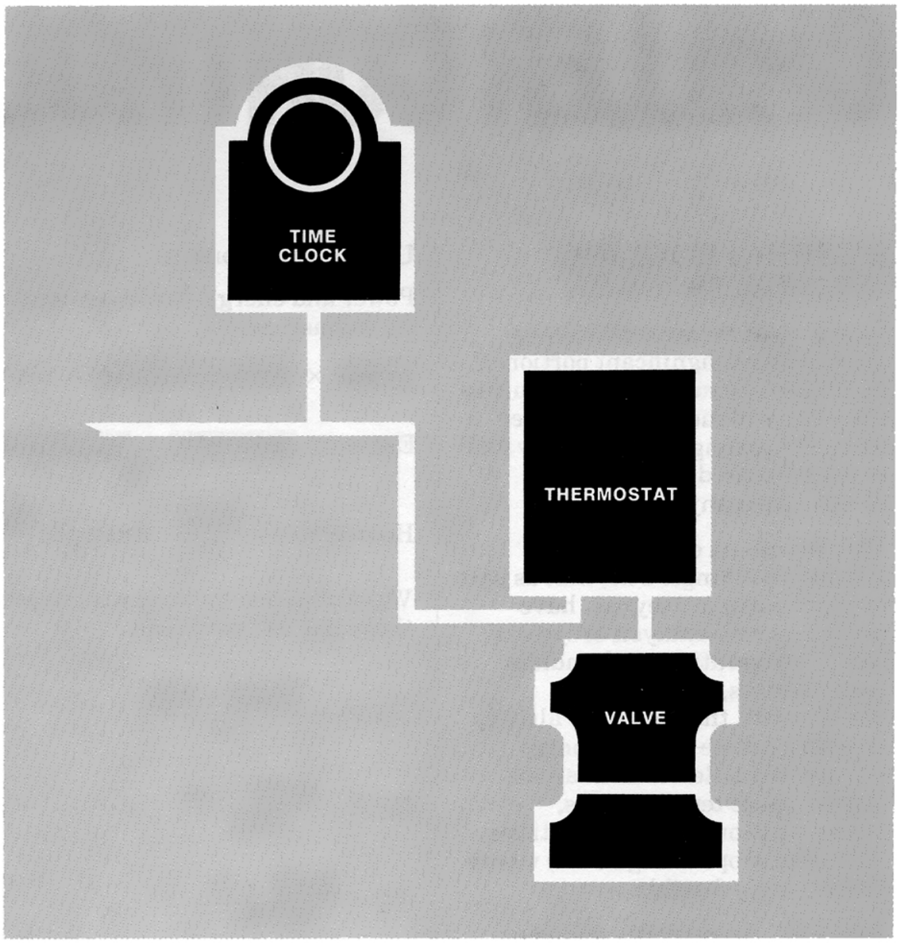
$$254.3 \times 10^6 \times 13.60\ \frac{\$}{10^6\ \text{Btu}} = \text{\$3458 HEATING \$ SAVED PER YEAR}$$

*See ASHRAE tables in appendix. 80K sq ft x 7 people/1000 x 20 CFM/P = 11,200 CFM min OA 11,200 CFM/60,000 CFM capacity = 18.6% OA minimum.

CAUTIONS: Setting the room thermostat to a lower setpoint during the winter on interior zones where cooling is required can increase your energy costs. Likewise setting thermostats up on zones with reheat can cause increased energy usage by cycling the reheat element.

ECM #4
REDUCED HEATING REQUIREMENTS DURING NIGHT OR UNOCCUPIED TIMES





$$\left[\text{F} \ \frac{\text{Btu}}{\text{hr °F}} + \left(\text{A} \ \frac{\text{ft}^3}{\text{min}} \times \frac{\text{B} \ \%}{100} \times 1.08 \frac{\text{min Btu}}{\text{hr ft}^3\text{ °F}} \right) \right] \times \text{night setback °F} \times \text{unoccupied hrs/wk} \ \frac{\text{hrs}}{\text{wk}} \times \text{E} \ \frac{\text{wks}}{\text{yr}} =$$

BTU s SAVED PER YEAR

$$\text{BTU s SAVED PER YEAR} \times \text{D} \ \frac{\$}{10^6 \text{ Btu}} = \text{\$ SAVED PER YEAR}$$

EXAMPLE: 80,000 ft² two story building, 200′ x 200′ x 25′, TTF=.81, Chicago weather data, 60,000 cfm. Air handling capacity, B % ventilation air, gas fuel cost $10.0 per 1000 ft³, electric cost .10 per Kwh, air cooled condenser; old thermostat setpoint: 74°F new setpoint: 68°F from 7 A.M. to 5 P.M. 5 days per week and 60° at all other times during the winter; 78°F in cooling mode 10 hours per day 5 days per week.

$$\left[16{,}200 \ \frac{\text{Btu}}{\text{hr °F}} + \left(60{,}000 \ \frac{\text{ft}^3}{\text{min}} \times \frac{\overset{*}{18.6} \ \%}{100} \times 1.08 \frac{\text{min Btu}}{\text{hr ft}^3\text{ °F}} \right) \right] \times 8 \text{ °F} \times 118 \ \frac{\text{hrs}}{\text{wk}} \times 30 \ \frac{\text{wks}}{\text{yr}} =$$

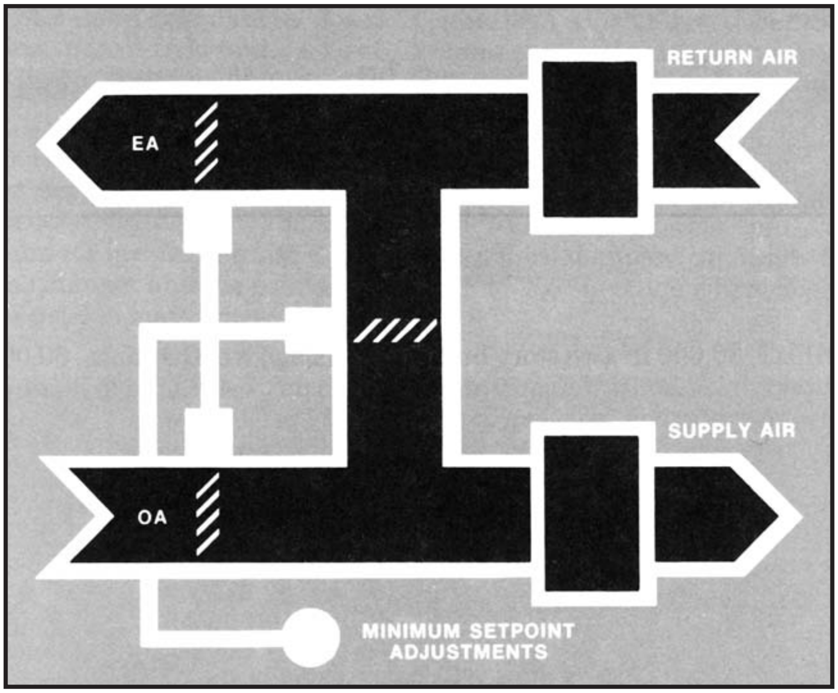
800 × 10⁶ BTU s SAVED PER YEAR

$$800 \times 10^6 \times 13.60 \ \frac{\$}{10^6 \text{ Btu}} = \text{\$10,882 SAVED PER YEAR}$$

*See ASHRAE tables in appendix. 80K sq ft x 7 people/1000 x 20 CFM/P = 11,200 CFM
min OA 11,200 CFM/60,000 CFM capacity = 18.6% OA minimum.

CAUTIONS: Setting the room thermostat to a lower setpoint during the winter on interior zones where cooling is required can increase your energy costs. Likewise setting thermostats up on zones with reheat can cause increased energy usage by cycling the reheat element.

ECM #5
REDUCED MINIMUM OUTDOOR AIR



Reducing the amount of outdoor air for ventilation can reduce the amount of mechanical heating and cooling required for space conditioning. Prior to the energy crisis, the amount of outdoor air (OA) required for ventilation was substantially greater than suggested by current ASHRAE Standards. More recently, ASHRAE 62-1999 Ventilation Standards require ventilation rates of 10 to 60 CFM per person depending on the space application.

Most existing buildings today operate with either too little or too much outside air ventilation. Systems providing inadequate ventilation can directly effect and risk occupant's health and productivity. Many more systems are believed to be over-ventilating or could benefit from reduced ventilation if acceptable measures were taken.

Newly constructed buildings are required to meet state and/or local codes, which most often are similar to those of ASHRAE. But, after construction many of the operational settings, including damper minimum positioning, get changed. One of the most common causes for over-ventilation is improper adjustment of outdoor air dampers. And, it is fairly common to find outdoor air damper minimum positions set for 25% to 50% outdoor airflow which is dramatically greater that occupancy rates require.

Outdoor air can be reduced during occupied periods, meet actually occupants ventilation needs and still comply with ASHRAE standards in several alternative ways. Each offers a significant reduction in energy waste while providing proper ventilation and contribute to improved indoor air quality.

The first method called "demand control ventilation" (DCV) employs carbon dioxide (CO2) sensors placed in the return air or occupied space. Carbon dioxide is a natural byproduct of human respiration. Constant monitoring of its concentration provides an accurate means of measuring changes in building occupancy and controlling the rate of outdoor air ventilation. When CO2 levels exceed a predetermined threshold, outdoor air ventilation is automatically increased to maintain a CFM ventilation rate for that type of building and level of occupancy. The proper amount of outdoor air ventilation can be automatically increased and decreased with changes in building occupancy.

A threshold can be determined by 1) selecting a metabolic rate factor (10600 for office/sedentary) and 2) dividing this factor by the ASHRAE ventilation rate for a given building application (20 cfm per person for an office) and finally 3) adding this to the outdoor CO2 concentration level.

Office example: 10600/20 cfm per person = 530 parts per million (ppm) CO2 above the outdoor (ambient) level. A Chicago ambient level of 400 ppm + 530 ppm = 930 ppm CO2 threshold. CO2 sensors can be added to larger building systems to provide DCV very affordably. They are also available as part of integrated economizer/DCV rooftop control systems to which optimize comfort control strategies and maximize savings.

CO2 sensors should be selected with caution. Choose only from high quality models that avoid very costly yearly or more frequent re-calibration. Lower quality (low bid) sensors drift significantly in a few months and can cost as much to maintain each year as their original installed cost.

Commercial air cleaners offer an alternative method with growing acceptance. Re-circulation with air cleaning systems, often used together with some degree of outdoor air dilution, can be an effective IAQ strategy if contaminants are appreciably reduced. Their use is especially applicable for environments that otherwise might require 100% outdoor air and those with high particulate or biological contaminant concentrations such as the hospitality/gaming industry (smoke) and day care/health centers (spores, bacteria). Stand-alone commercial air cleaners offer the additional advantages of more flexible location nearer occupant concentrations, combinations of high efficiency HEPA filters and large capacity gas phase adsorption and high ventilation rate units that provide increased air changes per hour.

References:

ASHRAE 62-1999 available for purchase from www.ashrae.com for about $50.

"Research Finds Economizing Plus Demand Control Ventilation Delivers Highest Energy Savings," Brandemuehl and Braun 1998, form 63-9058.

ASHRAE 1999 Abstract 4276 "The Impact of Demand-Controlled and Economizer Ventilation Strategies on Energy Use in Buildings," Brandemuehl and Braun, form 63-7063.

"DCV: History, Theory, Myths" by Steven Di Giacomo published in Engineered Systems, form 63-9103.

Detailed information regarding CO2 sensors, free Savings Estimator software (20 MB), Fresh Air Economizer and Demand Control Ventilation systems at: www.Honeywell.com/building/components.

Information regarding Honeywell Commercial Air Products and indoor air quality issues at: www.cleanairfacility.com.

Information regarding indoor air quality measurement and analysis services at: www.myfacility.com.

Winter

$$\Bigg[\left(A\ \frac{ft^3}{min} \times \frac{B\ \%}{100}\right) - \left(\text{required cfm/person} \times \text{\# of people}\right)\frac{ft^3}{min} \times 1.08\ \frac{min\ Btu}{hr\ ft^{3\circ}F} \times \text{operating hrs/wk}\ \frac{hrs}{wk} \times$$

$$\left(\text{average indoor temperature} - C\right)\Bigg] \times E\ \frac{wks}{yr} = \text{BTU s SAVED PER YEAR}$$

$$\text{BTUs SAVED PER YEAR} \times D\ \frac{\$}{10^6\ Btu} = \text{\$ SAVED PER YEAR}$$

Summer

$$\Bigg[\left(A\ \frac{ft^3}{min} \times \frac{B\ \%}{100}\right) - \left(\text{required cfm/person} \times \text{\# of people}\right)\frac{ft^3}{min} \times \frac{H\ \ Btu}{yr\ 1000\ ft^3/min} \times \frac{\text{operating hrs/wk}}{\text{50 operating hrs/wk}} =$$

$$\text{BTU s SAVED PER YEAR}$$

$$\text{BTUs SAVED PER YEAR} \times J\ \frac{\$}{10^6\ Btu} = \text{\$ SAVED PER YEAR}$$

EXAMPLE: 80,000 ft² two story building, 400 people, 60,000 cfm air handling capacity, ASHRAE STD 62-1999 ventilation air, gas fuel cost $13.60 per 10⁶ Btu's, electric cost $0.10 per Kwh, air cooled condenser, Chicago weather data, 55 operating hours per week, 74°F average indoor temperature.

Winter

$$\Bigg[\left(60{,}000\ \frac{ft^3}{min} \times \frac{18.6\ \%}{100}\right) - \left(20 \times 400\right)\frac{ft^3}{min}\Bigg] \times 1.08\ \frac{min\ Btu}{hr\ ft^3\ {}^\circ F} \times 55\ \frac{hrs}{wk} \times \left(74 - 34.2\right){}^\circ F \times 30\ \frac{wks}{yr} =$$

$$224 \times 10^6\ \text{Btu s SAVED PER YEAR}$$

$$224 \times 10^6 \times 13.60\ \frac{\$}{10^6\ Btu} = \text{\$3048 SAVED PER YEAR}$$

Summer

$$\Bigg[\left(60{,}000\ \frac{ft^3}{min} \times \frac{18.6\ \%}{100}\right) - \left(20 \times 400\right)\frac{ft^3}{min}\Bigg] \times \frac{38.791 \times 10^6\ Btu}{yr\ 1000\ ft^3/min} \times \frac{\text{55 operating hrs/wk}}{\text{50 hrs/wk}} =$$

$$135 \times 10^6\ \text{BTU s SAVED PER YEAR}$$

$$135 \times 10^6 \times 10.00\ \frac{\$}{10^6\ Btu} = \text{\$1350 SAVED PER YEAR}$$

*See ASHRAE tables in appendix. 80K sq ft x 7 people/1000 x 20 CFM/P = 11,200 CFM min OA 11,200 CFM/60,000 CFM capacity = 18.6% OA minimum.

SPECIAL DATA NECESSARY. Use data in *Technical Information* Section.

CAUTIONS: When reducing the amount of outdoor air introduced into a building, care should be taken to also reduce the amount of exhaust air to prevent a negative static pressure from developing.

Many state and local codes specify the minimum ventilation air requirements for specific types of buildings and may not agree with ASHRAE 62-73.

A = air handling capacity $\left(\frac{ft^3}{min}\right)$; *B = present ventilation air (%); C = heating season average outside temp. (°F); D = cost of heating* $\left(\frac{\$}{10^6\ Btu}\right)$;

E = heating season length (weeks); F is a calculated value: F = TTF x building perimeter in feet x building height in feet $\left(\frac{Btu}{hr\ {}^\circ F}\right)$;

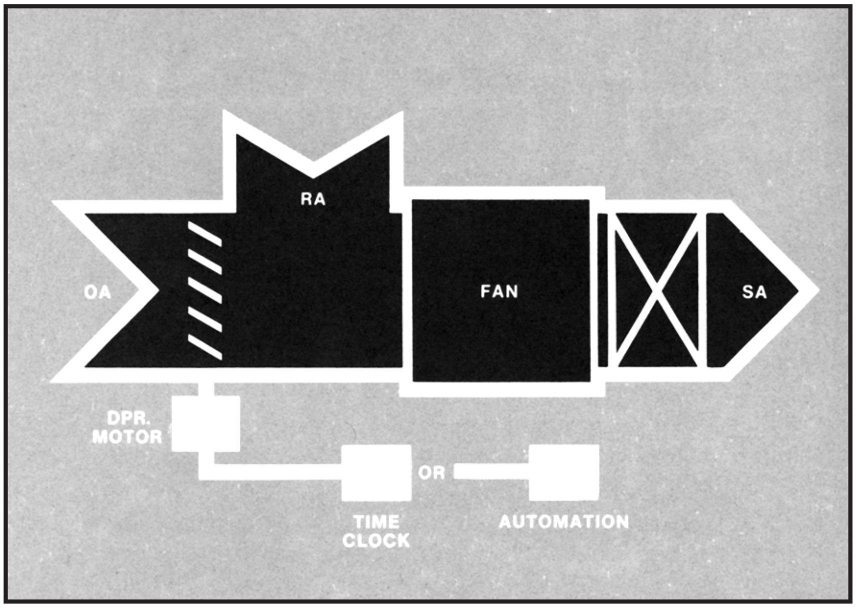
TTF = Thermal Transmission Factor found in the Technical Data Section on page 103; G = cooling season average outside dry bulb temp. (°F);

H = seasonal cooling load for outdoor air $\left(\frac{10^6\ Btu}{year\ 1000\ cfm}\right)$; *I = cooling season length (weeks); J = cost of cooling* $\left(\frac{\$}{10^6\ Btu}\right)$

K = seasonal cooling savings $\left(\frac{10^6\ Btu}{yr\ 1000\ cfm}\right)$

ECM #6

UNOCCUPIED VENTILATION REDUCTION



Considerable energy savings are possible by closing off outside ventilation air during unoccupied periods.

Required ventilation for conditioned spaces need only be supplied while they are occupied. Many fan systems are run intermittently during unoccupied periods to maintain night setpoint temperatures, but very frequently there is no change to reduce the amount of outside air into the space during unoccupied periods. Almost all systems have to be started prior to occupancy to warm up or cool down the space to comfort conditions. By adding a time clock or automation system, the outside air can be shut off during these unoccupied periods. The result is a lower demand on HVAC mechanical system equipment runtime to heat or cool that unnecessary outside air.

Note: Certain new construction state and local codes allow total close-off of outdoor ventilation only if provision is made for space air change prior to the next occupied period. Otherwise, a reduced rate of ventilation may be required.

$$\text{A}\ \frac{\text{ft}^3}{\text{min}} \times \left(\frac{\text{B\%} - \text{damper leakage\%}}{100}\right) \times 1.08\ \frac{\text{min Btu}}{\text{hr ft}^3\ {}^\circ\text{F}} \times \left(\text{average indoor temperature} - \text{C}\right) {}^\circ\text{F} \times \text{unoccupied hrs/wk}\ \frac{\text{hrs}}{\text{wk}} \times$$

$$\text{E}\ \frac{\text{wks}}{\text{yr}} = \text{BTU s SAVED PER YEAR}$$

$$\text{BTUs SAVED PER YEAR} \times \text{D}\ \frac{\$}{10^6\ \text{Btu}} = \text{\$ SAVED PER YEAR}$$

EXAMPLE: 80,000 ft² office building, Chicago weather data, 60,000 cfm air handling capacity, ASHRAE STD 62-1999 ventilation air, gas fuel cost $13.60 per 10^6 Btu's, inside temperature 60°F, 118 unoccupied hours per week, damper leakage .5%.

$$60{,}000\ \frac{\text{ft}^3}{\text{min}} \times \left(\frac{18.6 - \overset{*}{.5}\ \%}{100}\right) \times 1.08\ \frac{\text{min Btu}}{\text{hr ft}^3\ {}^\circ\text{F}} \times \left(60 - 34.2\right) {}^\circ\text{F} \times 118\ \frac{\text{hrs}}{\text{wk}} \times 30\ \frac{\text{wks}}{\text{yr}} =$$

$$1071 \times 10^6 \times 13.60\ \frac{\$}{10^6\ \text{Btu}} = \text{\$14,568 SAVED PER YEAR}$$

CAUTIONS: This example assumes that the space is maintained at 60°F and that the fan operates throughout the unoccupied hours. If this is not the case, the appropriate space temperature and the unoccupied hours when the fan operates should be used.

Cooling savings are not estimated because outside temperatures are typically low during unoccupied times.

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*Most commercial dampers leak 5% to 30% even when closed. Adjust your calculations accordingly.
See "Technical Information" page for a procedure to determine your damper leakage.

A = air handling capacity $\left(\frac{ft^3}{min}\right)$; *B = present ventilation air (%); C = heating season average outside temp. (°F); D = cost of heating* $\left(\frac{\$}{10^6\ Btu}\right)$;

E = heating season length (weeks); F is a calculated value: F = TTF x building perimeter in feet x building height in feet $\left(\frac{Btu}{hr\ {}^\circ F}\right)$;

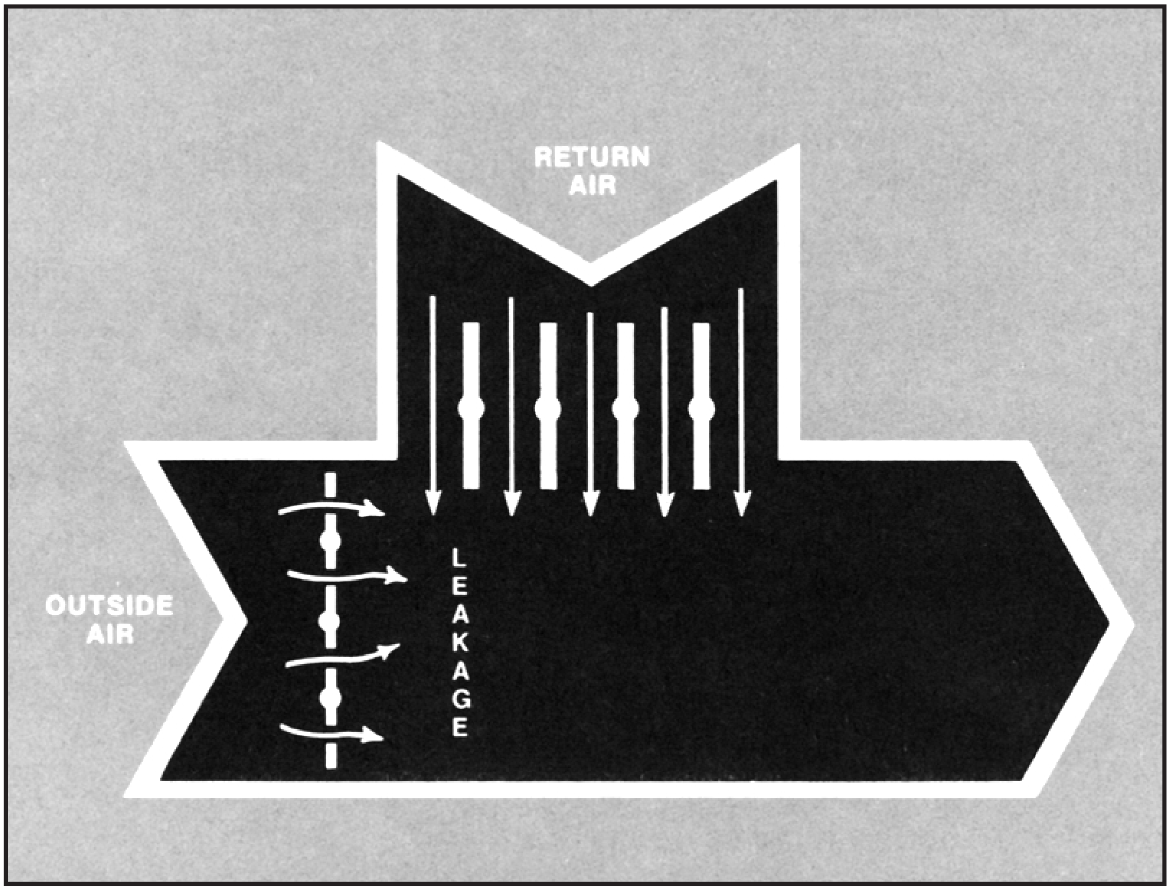
TTF = Thermal Transmission Factor found in the Technical Data Section on page 103; G = cooling season average outside dry bulb temp. (°F);

H = seasonal cooling load for outdoor air $\left(\frac{10^6\ Btu}{year\ 1000\ cfm}\right)$; *I = cooling season length (weeks); J = cost of cooling* $\left(\frac{\$}{10^6\ Btu}\right)$

K = seasonal cooling savings $\left(\frac{10^6\ Btu}{yr\ 1000\ cfm}\right)$

ECM #7

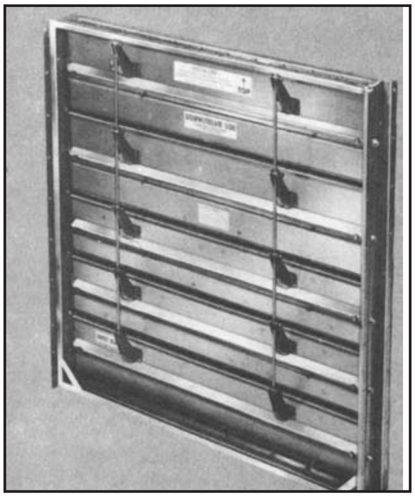
LOW LEAKAGE DAMPERS



Allowing unwanted outdoor air into a building during periods when the outdoor air damper should be closed will cause boilers and chillers to run and unnecessarily add to the mechanical system load.

This situation can occur during those periods when the outdoor air damper is supposed to be closed, such as unoccupied periods. Another common situation is when a specialized damper has been installed to control minimum air and the primary damper has been driven to a closed position, but leaks. Excessive ventilation will also occur during occupied hours when the outdoor air dampers are designed to close for freeze protection at a predetermined temperature, but the dampers do not provide tight shutoff and risk consequential damage as well as waste energy.

The use of low leakage dampers will reduce this wasteful load if they restrict leakage to less than 1 percent. Standard dampers can allow from 5 percent to 30 percent leakage when closed.



Low Leakage Damper

$$A \frac{ft^3}{min} \times \left(\frac{\text{estimated damper leakage } \% - 1\ \%}{100} \right) \times 1.08 \frac{min\ Btu}{hr\ ft^3\ {}^\circ F} \times \left(\text{average indoor temperature} - C \right) {}^\circ F \times$$

$$\text{unoccupied hrs/wk when fan is operating} \frac{hrs}{wk} \times E \frac{wks}{yr} = \text{BTU s SAVED PER YEAR}$$

$$\text{BTUs SAVED PER YEAR} \times D \frac{\$}{10^6\ Btu} = \text{\$ SAVED PER YEAR}$$

EXAMPLE: 80,000 ft² two story building, Chicago weather data, 60,000 cfm air handling capacity, 118 hours per week unoccupied hours, fan operates 24 hours/day, estimated leakage of the existing damper is 10%, average indoor temperature is 74°F, gas fuel costs $13.60 per 10^6 Btu's.

$$60{,}000 \frac{ft^3}{min} \times \left(\frac{10\ \% - 1\%}{100} \right) \times 1.08 \frac{min\ Btu}{hr\ ft^3\ {}^\circ F} \times \left(74 - 34.2 \right) {}^\circ F \times 118 \frac{hrs}{wk} \times 30 \frac{wks}{yr} =$$

$$822 \times 10^6 \text{ BTU s SAVED PER YEAR}$$

$$822 \times 10^6 \times 13.60 \frac{\$}{10^6\ Btu} = \text{\$11,179 SAVED PER YEAR}$$

SPECIAL DATA. Use data in *Technical Information* Section.

CAUTIONS: Savings shown will be reduced if the building is maintained at a lower temperature during unoccupied hours. This calculation is good only when fan is operating. When fan is off the amount of air leakage through the dampers is reduced.

A = air handling capacity $\left(\frac{ft^3}{min}\right)$; *B = present ventilation air (%); C = heating season average outside temp. (°F); D = cost of heating* $\left(\frac{\$}{10^6\ Btu}\right)$;

E = heating season length (weeks); F is a calculated value: F = TTF x building perimeter in feet x building height in feet $\left(\frac{Btu}{hr\ {}^\circ F}\right)$;

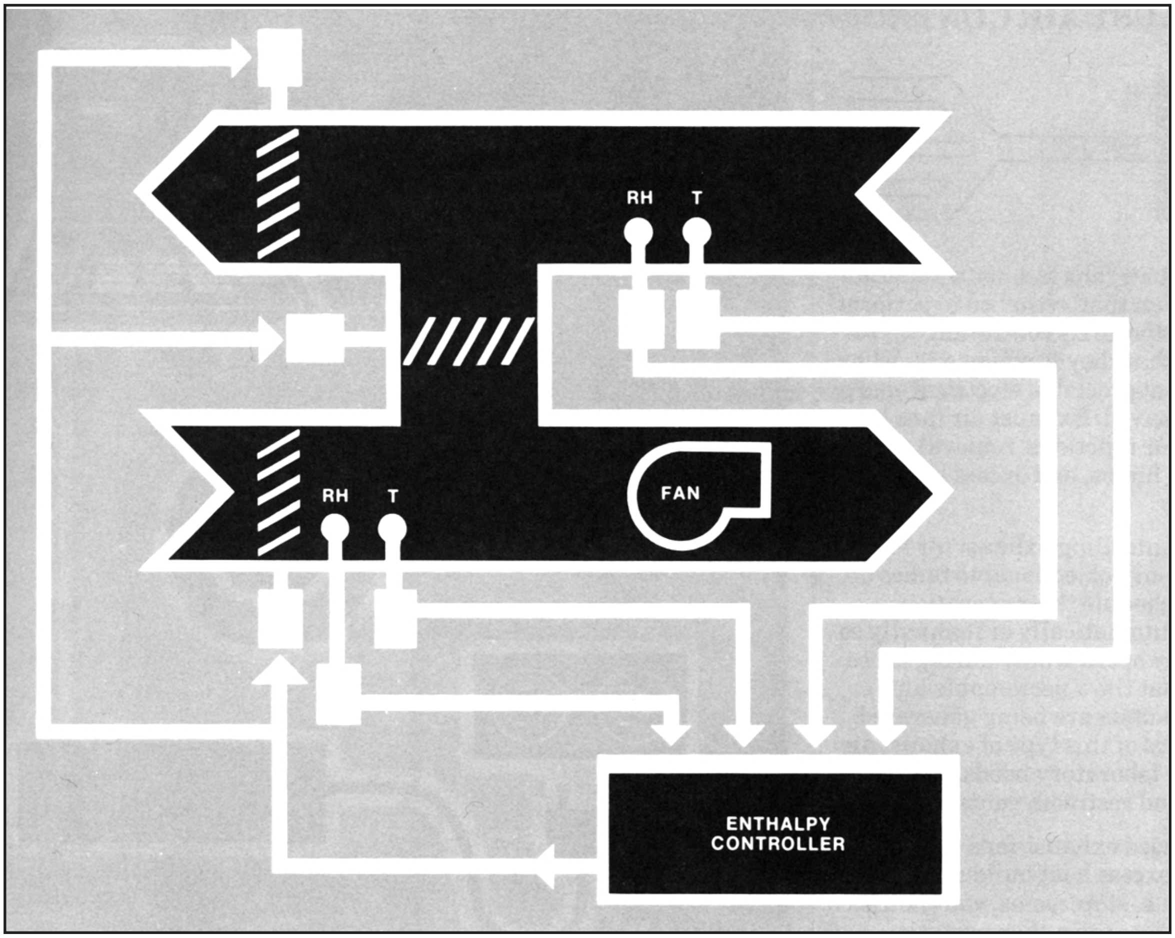
TTF = Thermal Transmission Factor found in the Technical Data Section on page 103; G = cooling season average outside dry bulb temp. (°F);

H = seasonal cooling load for outdoor air $\left(\frac{10^6\ Btu}{year\ 1000\ cfm}\right)$; *I = cooling season length (weeks); J = cost of cooling* $\left(\frac{\$}{10^6\ Btu}\right)$

K = seasonal cooling savings $\left(\frac{10^6\ Btu}{yr\ 1000\ cfm}\right)$

ECM #8

ENTHALPY CONTROL/DRY BULB ECONOMIZER



The load on a cooling coil for an air handling system is a function of the total heat content of air entering the coil. Total heat is a function of two measurements, dry bulb (DB) and relative humidity (RH) or dew point (DP). Using suitable outdoor air to cool or provide an initial stage of cooling for a building results in lower mechanical refrigeration load whenever outdoor air has a lower total heat content (enthalpy) than the return air.

Many older economizer systems, referred to as “wild economizers,” provided outdoor air whether or not there was a call for cooling. Often, the outdoor air dampers were left wide open during the fall and winter. An “integrated economizer” control strategy should be employed to access this pre-mechanical “free cooling” stage, but only operate when the space calls for cooling. Otherwise, the result will be dramatically higher heating costs. The economizer control sequence should be integrated further with any “demand control ventilation” strategy. Operating them independently will sub-optimize the energy savings potential each offers.

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Dry Bulb Economizer Caution. Temperature based changeover in economizers has been a lowest cost and well accepted strategy for decades, and many facilities staff and contractors highly value the "reliability" of their operation. However, recent research determined that energy savings, based on temperature changeover, is drastically lower than previously believed. Rather than delivering 15% to 25% savings, temperature changeover delivers a more modest 2% to 8% in most climates.

Even climates such as California's central valley and Los Angeles basin have sufficiently humid or rainy seasons so that savings generated in the dry season are mostly offset in the wet season. Typically in winter conditions when there is a call for cooling, the outdoor dampers are opened if the outdoor air is below the maximum temperature setting, commonly 72° F. But, whenever there is fog or rain, this places an additional burden on the mechanical cooling system. Rather than providing the intended "free cooling," the air conditioning system must increase its runtime to dehumidify unlimited outdoor air (very little re-circulated).

Temperature changeover is sometimes used to meet the "spec" in new construction, but not the intent of most codes. Using temperature changeover allows a firm to bid rooftop equipment or built-up control systems at lower initial cost but with substantially greater yearly operating cost.

Enthalpy Control

Systems using enthalpy changeover control measure temperature and humidity conditions. Many systems are installed with only outdoor air sensors. The installer must decide whether to select a very aggressive setting that risks higher temperature/humidity conditions in the space or a more conservative setting that delivers much lower savings.

Differential enthalpy control systems measure temperature and humidity both in the outdoor and return air on a continuous basis. On a call for cooling a controller or logic module automatically computes which air source would impose the lowest load on the cooling system. If outside air is the lowest load, the controller modulates outdoor air dampers beyond their minimum position to maintain a steady mixed or discharge air temperature, typically 55 ° F. Or, if outdoor air is not suitable, dampers are driven to their minimum position and return air is re-circulated.

Integration With Demand Control Ventilation

For new construction and larger retrofit projects, installation of any economizer system typically must provide minimum outdoor air for "design maximum occupancy." If code allows, additional controls can be installed to provide a minimum outdoor air volume based on "actual" occupancy. See ECM #5 and #6 for discussions of incorporating "demand control ventilation"

For convenient estimated savings calculations of building setback/setup, economizing and demand control ventilation control strategies, request a "Savings Estimator" cd-rom by e-mail from info@honeywell.com.
Be sure to include your name, position, organization and complete mailing address.

Enthalpy Control Savings Calculations

Savings are based on the assumption that the system previously had either a) no 100% O.A. damper or b) a fixed minimum outdoor air setting (whenever the fan operates) sufficient for ventilation purposes, it is also assumed that c) minimum outdoor air has already been reduced (see ECM# 5, Ventilation Reduction), and the minimum damper opening will be at the *new value* determined in ECM#6.

Step 1. Determine minimum cfm of outdoor air to be used during occupied hours.
Step 2. Calculate annual savings.

$$\text{A}\ \frac{\text{ft}^3}{\text{min}}\left(1-\frac{\text{B}\ \%}{100}\right)\times \text{K}\ \frac{10^6\ \text{Btu}}{\text{yr 1000 cfm}}\times\frac{\text{operating hrs/wk}}{50}\times \text{J}\ \frac{\$}{10^6\ \text{Btu}} = \text{\$ SAVED PER YEAR}$$

This equation is used to calculate the savings resulting from enthalpy control of outdoor air. This savings will generally be greater than the savings resulting from a dry bulb economizer. To estimate dry bulb economizer, multiply the enthalpy savings by .93

EXAMPLE—80,000 ft² two-story office building, 60,000 cfm air handling capacity, electrical cost $.10/kWh, air-cooled condensing unit. Chicago location, 55 operating hours per week. 100% dampers in-place, now set at fixed minimum at 3.33%.

$$60{,}000\left(1-\frac{3.3}{100}\right)\times 12.434\ \frac{10^6}{\text{yr 1000 cfm}}\times\frac{55}{50}\times 10.00\ \frac{\$}{10^6\ \text{Btu}} = \text{\$7936 SAVED PER YEAR}$$

SPECIAL DATA NECESSARY None

CAUTIONS—Be sure that provisions are made for:
a) Freeze protection of 100% O.A. duct and dampers are added.
b) Mixed air controller to prevent temperatures less than 55°F during cold weather. (This may be increased as weather gets colder.)

A = air handling capacity $\left(\frac{ft^3}{min}\right)$; *B = present ventilation air (%); C = heating season average outside temp. (°F); D = cost of heating* $\left(\frac{\$}{10^6\ Btu}\right)$;

E = heating season length (weeks); F is a calculated value: F = TTF x building perimeter in feet x building height in feet $\left(\frac{Btu}{hr\ °F}\right)$;

TTF = Thermal Transmission Factor found in the Technical Data Section on page 103; G = cooling season average outside dry bulb temp. (°F);

H = seasonal cooling load for outdoor air $\left(\frac{10^6\ Btu}{year\ 1000\ cfm}\right)$; *I = cooling season length (weeks); J = cost of cooling* $\left(\frac{\$}{10^6\ Btu}\right)$

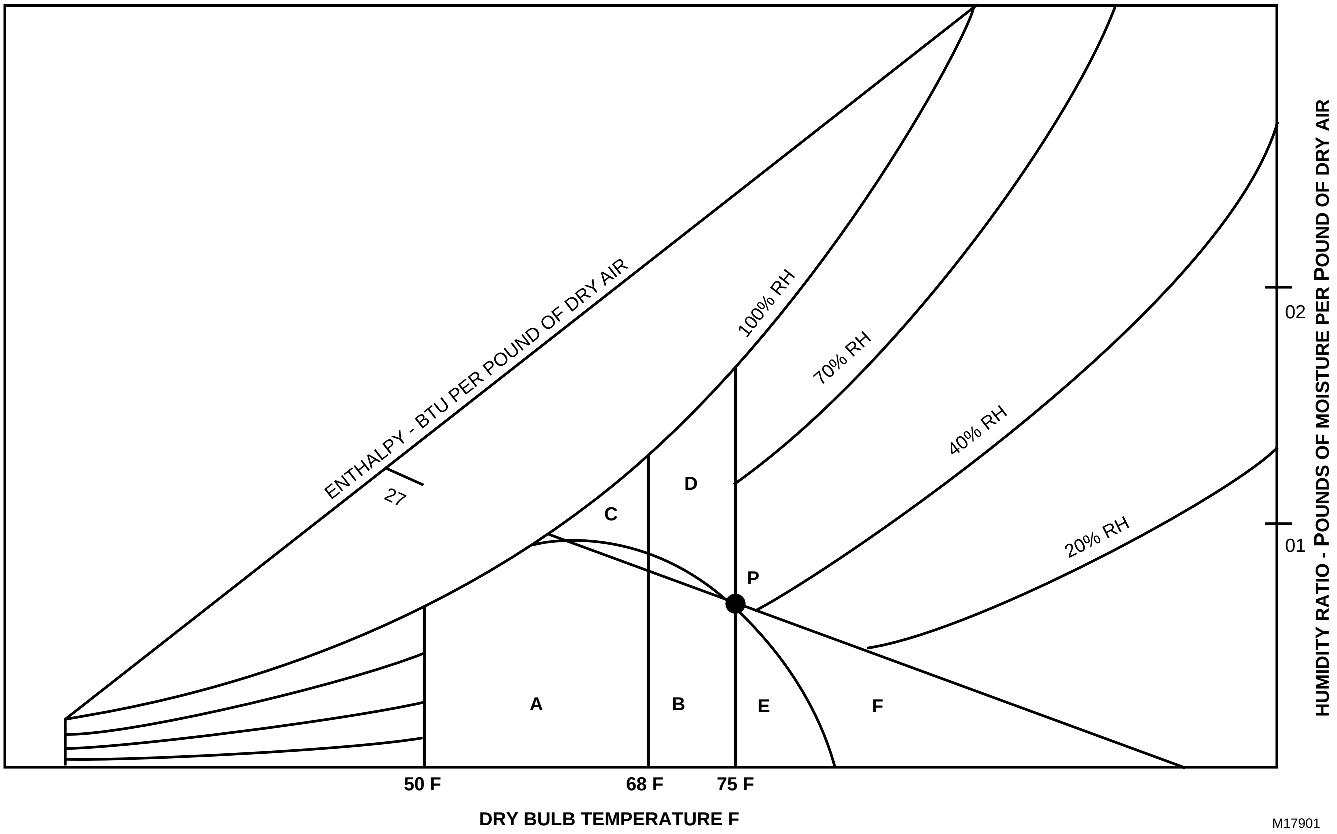
K = seasonal cooling savings $\left(\frac{10^6\ Btu}{yr\ 1000\ cfm}\right)$

Comfort and savings issues can also be explained with the help of a psychrometric chart, as shown below. For example, the return air of 75°F (24°C) and 40% RH was chosen and shown as point P. Intersecting point P is a line of constant enthalpy (straight diagonal) and a line of constant temperature (vertical). The curved line intersecting P is the enthalpy decision curve used in this study. The final line is a constant dry-bulb temperature line at 68°F (20°C) line represents the setting that differential dry-bulb economizer would choose for this case. Area B represents outdoor air conditions that save energy with economizer operation and would be chosen by the enthalpy and differential dry bulb strategies, but not the single dry-bulb strategy. Areas C and D are outdoor conditions that would be chosen by C. Outdoor air from the C and D regions of the chart is not desirable to use for two reasons. First, since the thermostat is only a dry-bulb controller, the humidity can rise to an uncomfortable level when dry-bulb cooling requirements can be satisfied with only the humid outdoor air. Second, if the heat gain in the space is high enough that the compressor needs to run, the higher enthalpy outdoor air will cause the compressor to run longer than if the return air was used. The extra run time is for the dehumidification of the humid outdoor air. Usually what is expected is a combination of discomfort and excess energy use since loading conditions vary throughout the day. For a humid area of the country, an economizer using a dry-bulb strategy will in fact lose the energy gained when outdoor conditions fall into area B when the conditions of areas C and D exist. The converse of this, however, is also true. For dry areas of the country, the amount of time the outdoor air spends in area B will be less than the amount of time spent in areas C and D.

Areas E and F represent outdoor air that a true enthalpy sensing economizer would use. This air is desirable at all times, except when the HVAC equipment does not provide any latent cooling. The curve that separates areas E and F was selected such that the undesirable lower enthalpy outdoor air lies in area F for most applications, including the building modeled in this study. This graphic examination explains the results of the simulation that enthalpy sensing provides the most savings in all types of clomates without the potential for loss of space comfort.

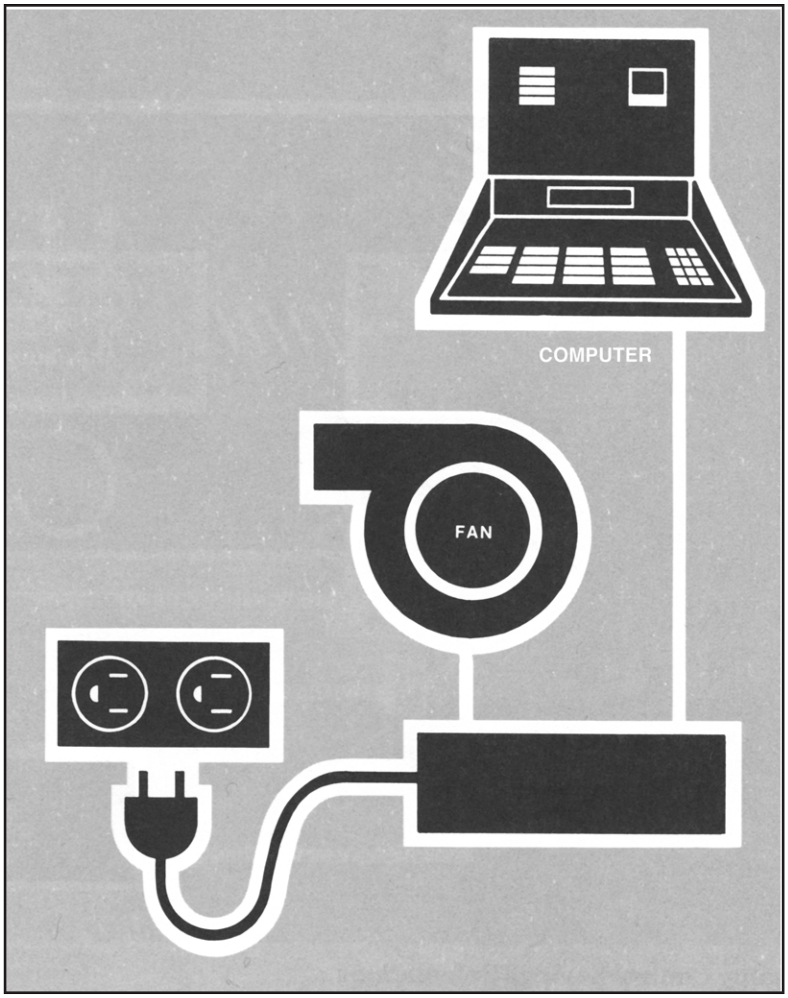
See pages 111 and 112 for detailed psychrometric charts.

PSYCHROMETRIC CHART



ECM#9

EXHAUST AIR CONTROL



Exhaust air fans frequently operate more often than required to perform their purpose. By controlling these fans so that they function only when needed, appreciable electrical energy may be saved. Exhaust air fans perform three major functions: exhaust odors and fumes; maintain building pressure; remove excess heat build-up.

When controlling exhaust air fans that remove objectionable fumes or odors, schedule their operation, either automatically or manually, so that they operate only during occupied periods and those times that the objectionable air contaminants are being generated. Manual spring timers, interlocks with space lighting and space occupancy sensors have all been successfully used to limit unnecessarily prolonged venting. Laboratory hoods, kitchen vents and rest room vents are examples of common application candidates.

Place exhaust fans used to control excess heat build up in spaces (i.e. storerooms, warehouses and garages) on a thermostatic control. In some applications, the exhaust air fan can be nterlocked with the supply air fan serving the same area to maintain building pressure.

Installation of backdraft dampers in the exhaust air fan discharge will prevent uncontrolled outside air from entering the conditioned space or prevent the loss of conditioned air from the space. This depends on the relative static pressure within the building when the exhaust fan is off. See ECM #7 regarding low leakage, tight close-off dampers.

$$\frac{\textbf{exhaust air fan wattage}}{1000} \times \textbf{hours off per day}\ \frac{\text{hrs}}{\text{day}} \times \textbf{operating days/wk}\ \frac{\text{days}}{\text{wk}} \times \textbf{operating wks/yr}\ \frac{\text{wks}}{\text{yr}} =$$

KWH s SAVED PER YEAR

$$\textbf{KWHs SAVED PER YEAR} \times \textbf{electrical cost}\ \frac{\$}{\text{Kwh}} = \textbf{\$ SAVED PER YEAR}$$

EXAMPLE A ½ horsepower Copy Room exhaust fan now operates 24 hours a day every day of the year. It is determined that it needs to operate only from 8:00 a.m. to 12:00 a.m. and 1:00 p.m. to 5:00 p.m. for five days per week. The nameplate on the motor gives 4.9 amps at 230 volts. The electricity costs 10¢/Kwh.

Watts = .85 x 230 x 4.9 = 958

$$\frac{\textbf{958}}{1000} \times \textbf{16}\ \frac{\text{hrs}}{\text{day}} \times \textbf{5}\ \frac{\text{days}}{\text{wk}} \times \textbf{52}\ \frac{\text{wks}}{\text{yr}} = \textbf{3985 KWHs SAVED PER YEAR}$$

$$\textbf{3985} \times \textbf{.10}\ \frac{\$}{\text{Kwh}} = \textbf{\$ 398.50 SAVED PER YEAR}$$

DEFINITION OF TERMS

Watt usage of exhaust fan.
For single phase motor: watts = 0.85 × volts × amps.
For three phase motor: watts = 1.732 × .85 × volts × amps; volts and amps secured from motor nameplate.
(0.85 is an assumed power factor.)
1.732 = $\sqrt{3}$

CAUTION: Specific areas where dangerous or combustible fumes may accumulate will require continuous ventilation and/or other monitoring steps.

A = air handling capacity $\left(\frac{ft^3}{min}\right)$; *B = present ventilation air (%); C = heating season average outside temp. (°F); D = cost of heating* $\left(\frac{\$}{10^6\,Btu}\right)$;

E = heating season length (weeks); F is a calculated value: F = TTF x building perimeter in feet x building height in feet $\left(\frac{Btu}{hr\,°F}\right)$;

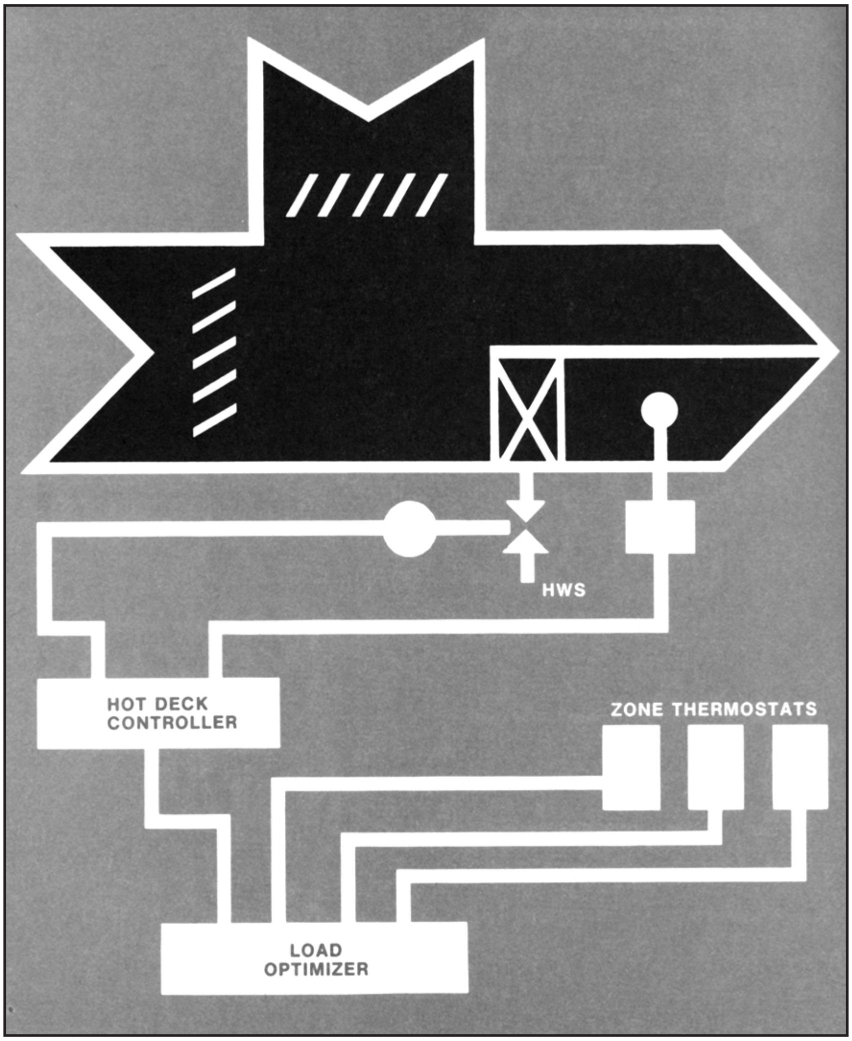
TTF = Thermal Transmission Factor found in the Technical Data Section on page 103; G = cooling season average outside dry bulb temp. (°F);

H = seasonal cooling load for outdoor air $\left(\frac{10^6\,Btu}{year\ 1000\ cfm}\right)$; *I = cooling season length (weeks); J = cost of cooling* $\left(\frac{\$}{10^6\,Btu}\right)$

K = seasonal cooling savings $\left(\frac{10^6\,Btu}{yr\ 1000\ cfm}\right)$

ECM #10

RESETTING HOT DECK TEMPERATURE



Many older or original dual duct systems using a hot deck and a cold deck rely on fixed hot deck/cold deck temperatures or reset the hot deck from outdoor temperature, leaving the cold deck at a fixed temperature. None of the above control strategies takes actual zone loads into account, and therefore wastes energy by mixing air that's too hot with air that's too cold. If a variable air volume retrofit isn't being considered, the deck temperatures of the original system can be optimized as an interim measure to reduce energy waste. (See ECMs #13, #14 and #15 to explore more aggressive energy conservation measure options.)

Money and energy can be saved by having hot deck temperature controls respond to the zone having the greatest heating demand. As the demand is reduced, the hot deck temperature is lowered accordingly, reducing energy required by the hot deck energy supply. A load analyzer compares the output of each zone thermostat and produces an output equal to the zone of greatest heating demand. The hot deck temperature is reset to satisfy that zone.

This concept should be used with reset of cold deck temperature in multi-zone or dual duct systems as well. Deck temperature becomes responsive to load rather than a schedule based on peak conditions and the lessened need for mixing saves money. (See ECM #11)

$$\left(\text{A} \frac{ft^3}{min} \times \frac{\text{\% hot deck air}\ \%}{100} \times 1.08 \frac{min\ Btu}{hr\ ft^3\ {}^\circ F} \right) \times \left(\text{hot deck reset in summer}\ {}^\circ F \times \text{I} \frac{wks}{yr} + \text{hot deck reset in winter}\ {}^\circ F \times \right.$$

$$\left. \text{E} \frac{wks}{yr} \right) \times \text{operating hrs/wk} \frac{hrs}{wk} = \text{BTU s SAVED PER YEAR}$$

$$\text{BTUs SAVED PER YEAR} \times \text{D} \frac{\$}{10^6\ Btu} = \text{\$ SAVED PER YEAR}$$

EXAMPLE: 80,000 ft² two story building, Chicago weather data, 60,000 cfm air handling capacity, fuel cost \$3.87 per 10⁶ Btu's, 60 occupied hours per week, 10 °F reset in summer, 3 °F reset in winter, and 25% of airflow in hot deck.

$$\left(60{,}000 \frac{ft^3}{min} \times \frac{25\ \%}{100} \times 1.08 \frac{min\ Btu}{hr\ ft^3\ {}^\circ F} \right) \times \left(10\ {}^\circ F \times 20.9 \frac{wks}{yr} + 3\ {}^\circ F \times 30 \frac{wks}{yr} \right) \times 60 \frac{hrs}{wk} =$$

$$290.6 \times 10^6 \text{ BTU s SAVED PER YEAR}$$

$$290.6 \times 10^6 \times 13.60 \frac{\$}{10^6\ Btu} = \$\ 3952 \text{ SAVED PER YEAR}$$

SPECIAL DATA NECESSARY: Use data in *Technical Information* Section.

CAUTIONS: Since each thermostat is capable of controlling the hot deck temperature, caution is advised. Misadjustment of thermostats and the upsetting influence of highly variable loads such as conference rooms may occur. These problems may be avoided through use of locking thermostat covers and elimination of variable load zones to reset deck temperatures.

DEFINITION OF TERMS

"Hot deck reset in summer" is an estimate of reset possible during cooling season. If heating is turned off use zero. Recommended value is 1°F if estimate unavailable.

"Hot deck reset in winter" is an estimate of reset possible during heating season. Recommended value is 2°F if estimate unavailable.

"% hot deck air" is the proportion of total air passing through hot deck. Recommended value is 50% if estimate unavailable.

A = air handling capacity $\left(\frac{ft^3}{min}\right)$*; B = present ventilation air (%); C = heating season average outside temp. (°F); D = cost of heating* $\left(\frac{\$}{10^6\ Btu}\right)$*;*

E = heating season length (weeks); F is a calculated value: F = TTF x building perimeter in feet x building height in feet $\left(\frac{Btu}{hr\ {}^\circ F}\right)$*;*

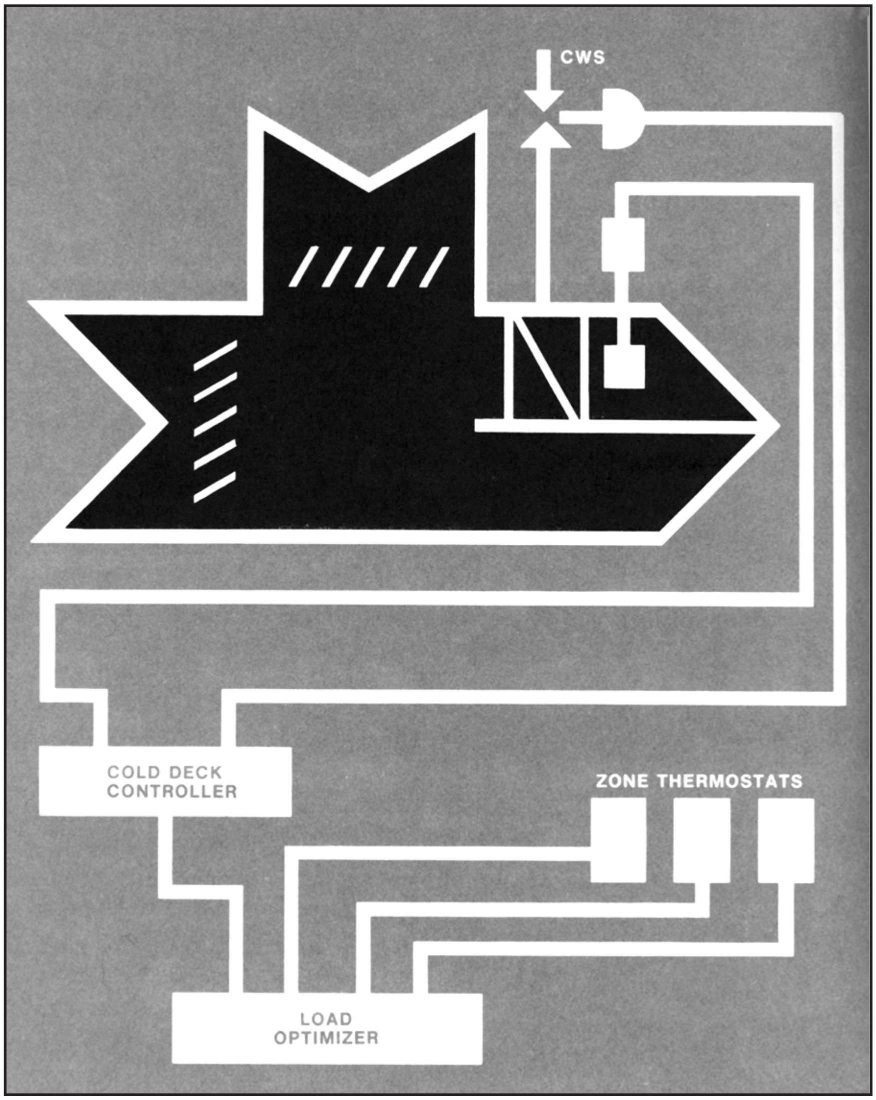
TTF = Thermal Transmission Factor found in the Technical Data Section on page 103; G = cooling season average outside dry bulb temp. (°F);

H = seasonal cooling load for outdoor air $\left(\frac{10^6\ Btu}{year\ 1000\ cfm}\right)$*; I = cooling season length (weeks); J = cost of cooling* $\left(\frac{\$}{10^6\ Btu}\right)$

K = seasonal cooling savings $\left(\frac{10^6\ Btu}{yr\ 1000\ cfm}\right)$

ECM #11

RESETTING COLD DECK TEMPERATURE



Fixed cold deck temperatures usually result in a waste of cooling energy since the air will be reheated or mixed with warm air at all times except during maximum (design) load. If a variable air volume retrofit isn't being considered, the deck temperatures of the original system can be optimized as an interim measure to reduce energy waste. (See ECMs #13, #14 and #15 to explore more aggressive energy conservation measure options.)

Cooling energy can be saved if cold deck temperature controls respond to the area control having the greatest cooling demand. As the demand is reduced, the cold deck temperature is raised accordingly. A load analyzer compares the output of each zone thermostat and produces an output proportional to the greatest cooling demand. The cold deck temperature is reset upward to minimally satisfy that zone.

This concept should be applied with the hot deck reset in multizone or dual duct systems as well. By reducing hot deck temperature less mixing of hot and cold air results, which saves energy. Deck temperature is responsive to load rather than a schedule based on peak conditions. (See ECM # 6)

The calculation is based on an assumed percentage of total air in cold deck and the expected savings in cold deck reset during the cooling season.

$$A\ \frac{ft^3}{min} \times \frac{\text{\% cold deck air}\ \%}{100} \times 4.5\ \frac{min\ lb}{hr\ ft^3} \times \text{enthalpy reset } (\Delta H)\ \frac{Btu}{lb} \times I\ \frac{wks}{yr} \times \text{operating hrs/wk}\ \frac{hrs}{wk} =$$

BTU s SAVED PER YEAR

$$\text{BTU s SAVED PER YEAR} \times J\ \frac{\$}{10^6\ Btu} = \text{\$ SAVED PER YEAR}$$

EXAMPLE: 80,000 ft² two story building, Chicago weather data, 60,000 cfm air handling capacity, electric cooling cost $.10 per Kwh, air-cooled condenser, 60 occupied hours per week, 1.5 Btu/lb reset, and 70% of air flow in cold deck.

$$60{,}000\ \frac{ft^3}{min} \times \frac{70\ \%}{100} \times 4.5\ \frac{min\ lb}{hr\ ft^3} \times 1.5\ \frac{Btu}{lb} \times 20.9\ \frac{wks}{yr} \times 60\ \frac{hrs}{wk} = 356 \times 10^6 \text{ BTU s SAVED PER YEAR}$$

$$356 \times 10^6 \times 10.00\ \frac{\$}{10^6\ Btu} = \text{\$3560 SAVED PER YEAR}$$

SPECIAL DATA NECESSARY: Use data in *Technical Information* Section.

CAUTIONS: Since each thermostat is capable of controlling the cold deck temperature, caution is advised. Misadjustment of thermostats and the upsetting influence of highly variable loads such as conference rooms may occur. These problems may be avoided through use of locking thermostat covers and elimination of variable load zones to reset deck temperatures.

DEFINITION OF TERMS

"Enthalpy reset" is an estimate of reset possible during cooling season. Measurement of actual cold deck conditions before and after trial reset settings is recommended. Then the "before and after enthalpy" can be estimated with the help of a psychrometric chart. Recommended value is 1.5 Btu/lb if estimate unavailable.

"% cold deck air" is an estimate of the portion of total air passing through the cold deck during summer cooling. Recommended value is 50% if estimate is unavailable.

A = air handling capacity $\left(\frac{ft^3}{min}\right)$*; B = present ventilation air (%); C = heating season average outside temp. (°F); D = cost of heating* $\left(\frac{\$}{10^6\ Btu}\right)$*;*

E = heating season length (weeks); F is a calculated value: F = TTF x building perimeter in feet x building height in feet $\left(\frac{Btu}{hr\ {}^\circ F}\right)$*;*

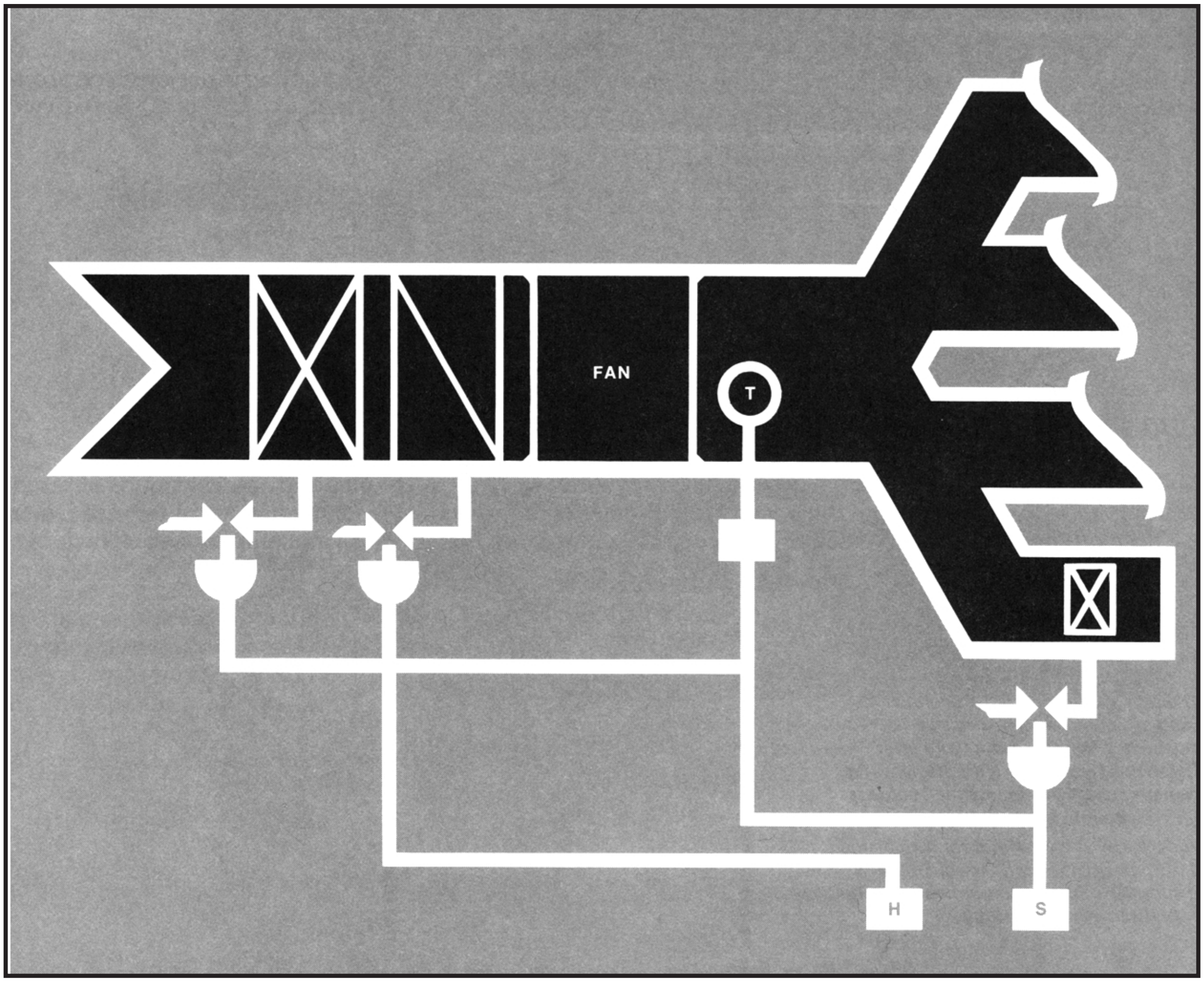
TTF = Thermal Transmission Factor found in the Technical Data Section on page 103; G = cooling season average outside dry bulb temp. (°F);

H = seasonal cooling load for outdoor air $\left(\frac{10^6\ Btu}{year\ 1000\ cfm}\right)$*; I = cooling season length (weeks); J = cost of cooling* $\left(\frac{\$}{10^6\ Btu}\right)$

K = seasonal cooling savings $\left(\frac{10^6\ Btu}{yr\ 1000\ cfm}\right)$

ECM #12

ZONE OPTIMIZATION IN REHEAT SYSTEMS



Most older terminal reset (reheat) systems ordinarily use a fixed cold deck temperature and apply thermostat controlled reheat in each zone to maintain comfort. If every zone is reheated, the cold deck is too cold and cooling energy, as well as reheat energy, is being wasted. Money and energy can be saved by installing zone-optimizing load analyzers on your temperature control system.

The demand of each zone is measured and cold supply air temperature is set upward to minimize the amount of reheat required. This makes the HVAC system flexible enough to satisfy the zone of greatest demand and reduces the total amount of both cooling and heating energy needed.

Summer Cooling Savings

$$\text{A}\ \frac{\text{ft}^3}{\text{min}} \times \text{operating hrs/wk}\ \frac{\text{hrs}}{\text{wk}} \times \text{I}\ \frac{\text{wks}}{\text{yr}} \times 4.5\ \frac{\text{min lb}}{\text{hr ft}^3} \times \text{enthalpy reset } (\Delta \text{H})\ \frac{\text{Btu}}{\text{lb}} = \text{BTU s SAVED PER YEAR}$$

$$\text{BTU s SAVED PER YEAR} \times \text{J}\ \frac{\$}{10^6\ \text{Btu}} = \text{\$ SAVED PER YEAR}$$

Summer Reheat Savings

$$\text{A}\ \frac{\text{ft}^3}{\text{min}} \times \text{operating hrs/wk}\ \frac{\text{hrs}}{\text{wk}} \times \text{I}\ \frac{\text{wks}}{\text{yr}} \times 1.08\ \frac{\text{min Btu}}{\text{hr ft}^3\ {}^\circ\text{F}} \times \text{dry bulb temperature reset during summer } (\Delta T_1)\ {}^\circ\text{F} =$$

$$\text{BTU s SAVED PER YEAR}$$

$$\text{BTU s SAVED PER YEAR} \times \text{D}\ \frac{\$}{10^6\ \text{Btu}} = \text{\$ SAVED PER YEAR}$$

Winter Reheat Savings

$$\text{A}\ \frac{\text{ft}^3}{\text{min}} \times \text{operating hrs/wk}\ \frac{\text{hrs}}{\text{wk}} \times 1.08\ \frac{\text{min Btu}}{\text{hr ft}^3\ {}^\circ\text{F}} \times \text{E}\ \frac{\text{wks}}{\text{yr}} \times \text{dry bulb temperature reset during winter } (\Delta T_2)\ {}^\circ\text{F} =$$

$$\text{BTU s SAVED PER YEAR}$$

$$\text{BTU s SAVED PER YEAR} \times \text{D}\ \frac{\$}{10^6\ \text{Btu}} = \text{\$ SAVED PER YEAR}$$

EXAMPLE: 10,000 cfm system, operating continuously, $10.00/$10^6$ Btu cooling costs, heating cost $13.60 per 10^6 Btu's Chicago weather data. ΔH enthalpy reset realized during summer months, 3.0 Btu/lb. ΔT_1 is the dry bulb temperature reset during summer operation, 4°F. ΔT_2 is the dry bulb temperature reset during winter operation, 7°F, ranges from 5° to 10°F.

Summer Cooling Savings

$$10{,}000\ \frac{\text{ft}^3}{\text{min}} \times 168\ \frac{\text{hrs}}{\text{wk}} \times 20.9\ \frac{\text{wks}}{\text{yr}} \times 4.5\ \frac{\text{min lb}}{\text{hr ft}^3} \times 3\ \frac{\text{Btu}}{\text{lb}} = 474 \times 10^6\ \text{BTU s SAVED PER YEAR}$$

$$474 \times 10^6 \times 10.\ \frac{\$}{10^6\ \text{Btu}} = \text{\$4740 SAVED PER YEAR}$$

Summer Reheat Savings

$$10{,}000\ \frac{\text{ft}^3}{\text{min}} \times 168\ \frac{\text{hrs}}{\text{wk}} \times 20.9\ \frac{\text{wks}}{\text{yr}} \times 1.08\ \frac{\text{min Btu}}{\text{hr ft}^3\ {}^\circ\text{F}} \times 4\ {}^\circ\text{F} = 151.7 \times 10^6\ \text{BTU s SAVED PER YEAR}$$

$$151.7 \times 10^6 \times 13.60\ \frac{\$}{10^6\ \text{Btu}} = \text{\$2063 SAVED PER YEAR}$$

Winter Reheat Savings

$$10{,}000\ \frac{\text{ft}^3}{\text{min}} \times 168\ \frac{\text{hrs}}{\text{wk}} \times 1.08\ \frac{\text{min Btu}}{\text{hr ft}^3\ {}^\circ\text{F}} \times 30\ \frac{\text{wks}}{\text{yr}} \times 7\ {}^\circ\text{F} = 381.0 \times 10^6\ \text{BTU S SAVED PER YEAR}$$

$$381.0 \times 10^6 \times 13.60\ \frac{\$}{10^6\ \text{Btu}} = \text{\$ 5182 SAVED PER YEAR}$$

A = air handling capacity $\left(\frac{ft^3}{min}\right)$; *B = present ventilation air (%); C = heating season average outside temp. (°F); D = cost of heating* $\left(\frac{\$}{10^6\ Btu}\right)$;

E = heating season length (weeks); F is a calculated value: F = TTF x building perimeter in feet x building height in feet $\left(\frac{Btu}{hr\ {}^\circ F}\right)$;

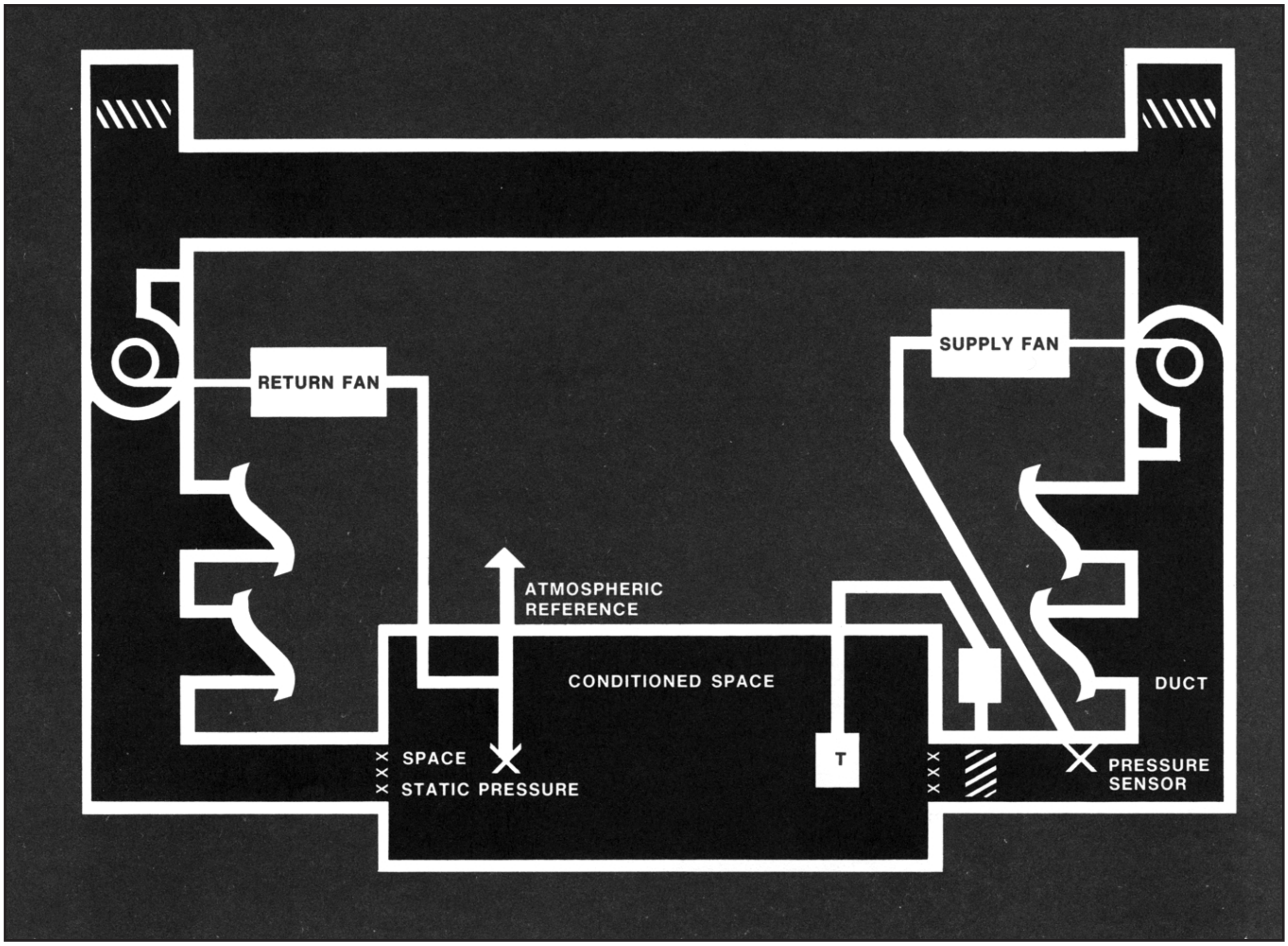
TTF = Thermal Transmission Factor found in the Technical Data Section on page 103; G = cooling season average outside dry bulb temp. (°F);

H = seasonal cooling load for outdoor air $\left(\frac{10^6\ Btu}{year\ 1000\ cfm}\right)$; *I = cooling season length (weeks); J = cost of cooling* $\left(\frac{\$}{10^6\ Btu}\right)$

K = seasonal cooling savings $\left(\frac{10^6\ Btu}{yr\ 1000\ cfm}\right)$

ECM #13

RETROFIT OF CENTRAL FANS FOR VARIABLE VOLUME USAGE



Conversion of constant volume air distribution systems to variable air volume (VAV) has been a cornerstone of most building energy retrofits since the 1970s. It often requires that action be taken to prevent the central fans from operating in critical areas and producing excessive negative static in the building. A central premise of VAV retrofits is the change from constant volume systems that typically ran heating and cooling systems simultaneously year around to a VAV system based on delivery of heating/cooling to the building perimeter but delivering only varying volumes of a constant temperature air to interior spaces. VAV systems dramatically reduce the excessive waste common to larger building constant volume systems .

More importantly, the retrofit of the central fan system to a VAV system results in considerable energy savings in fan horsepower. This ECM will only address those fans which have forward curved impellers or are equipped with some means of flow adjusters, such as vortex vanes.

Fans with air foil or backward curved impellers without vortex vanes require the addition of vanes or some other means of volume control, such as an eddy current clutch or a variable speed drive, before the advantage of the VAV operation can be realized. This requires an onsite evaluation to select the least costly approach.

FORWARD CURVED FAN SYSTEMS

Most typical of rooftop and packaged fan systems, the supply fan needs no retrofit since the throttling action of the VAV terminal units automatically results in reduction of fan horsepower and energy consumed.

The return fan may require the addition of a damper in the return duct system if building static pressures become objectionable. A space static sensor controls this damper with a reference to outdoor pressure.

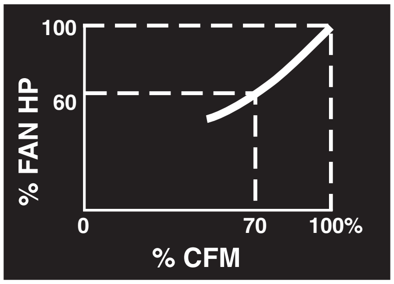
FANS WITH VORTEX VANES

Fans with vortex vanes require a different approach. The throttling effect of the terminal units, although producing some energy savings, may push the fan operation into the area not recommended by the fan manufacturer. Therefore, static pressure control should be installed on both the supply and return fans.

The supply fan static control maintains the minimum supply duct static required to deliver the minimum required cfm.

The return fan static control senses changes in space pressure relative to outdoors and maintains the space at a balanced or slightly positive value to minimize infiltration.

A VAV system delivers air in quantities based on the actual instantaneous load requirements of multiple individual zones. The more variable the building load, the greater the savings by converting to VAV. In a typical office building, the difference in fan cfm between a constant volume system and the average flow in a VAV system is typically 30-40 percent. Reduction in required fan horsepower and the dramatic resulting electrical savings are compelling reasons to retrofit wherever possible. The graph above shows the typical power savings possible by operating a vortex vane equipped fan in a VAV mode. By operating an average cfm of 70 percent of the original, we can realize approximately a 40 percent reduction in fan power requirements. (A more exact estimate may be obtained by using your system fan flow/horsepower curves. Refer to the affinity law equations for theoretical maximum savings. A reduction of average fan speed by 15% reduces energy consumption by nearly 40%.)



fan(s) operating horsepower hp × % fan power reduction % × .8 $\frac{\text{Kw}}{\text{hp}}$ × operating hrs/wk $\frac{\text{hrs}}{\text{wk}}$ × operating wks/yr $\frac{\text{wks}}{\text{yr}}$ =

KWH s SAVED PER YEAR

KWH s SAVED PER YEAR × electrical cost $\frac{\$}{\text{Kwh}}$ = **$ SAVED PER YEAR**

EXAMPLE: A constant volume system with supply fan delivery of 70,000 cfm at 4″ static (approximately 60 brake horsepower) and return fan handling 60,000 cfm at 1.5″ static (20 brake horsepower) with a 30% reduction (40% fan brake horsepower reduction) in flow by converting to V.A.V., 55 hours a week, 52 week year, $.10/Kwh electrical cost.

80 hp × 40 % × .8 $\frac{\text{Kw}}{\text{hp}}$ × 55 $\frac{\text{hrs}}{\text{wk}}$ × 52 $\frac{\text{wks}}{\text{yr}}$ = **73216 KWHs SAVED PER YEAR**

73216 × .10 $\frac{\$}{\text{Kwh}}$ = **$7322 SAVED PER YEAR**

CAUTIONS

1. Throttled fan motors change power factor. If your installation is cost sensitive to this change, consider adding capacitor banks to minimize effects.
2. Space static reference should be in a large representative area or in the common return plenum that is as close to the space as possible.

DEFINITION OF TERMS

.8 is an average value for converting horsepower to kilowatt hours.

A = air handling capacity $\left(\frac{ft^3}{min}\right)$*; B = present ventilation air (%); C = heating season average outside temp. (°F); D = cost of heating* $\left(\frac{\$}{10^6\,Btu}\right)$*;*

E = heating season length (weeks); F is a calculated value: F = TTF x building perimeter in feet x building height in feet $\left(\frac{Btu}{hr\,°F}\right)$*;*

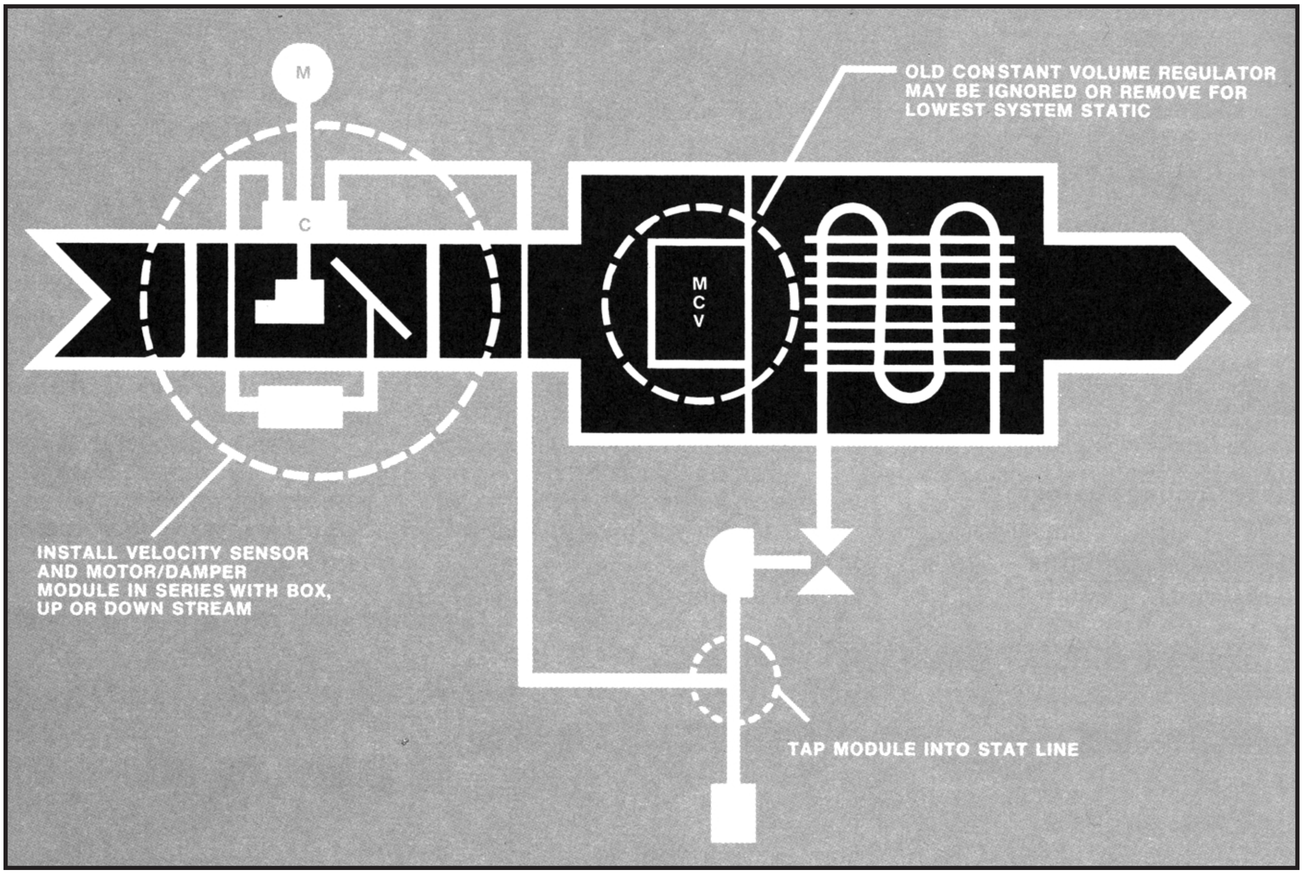
TTF = Thermal Transmission Factor found in the Technical Data Section on page 103; G = cooling season average outside dry bulb temp. (°F);

H = seasonal cooling load for outdoor air $\left(\frac{10^6\,Btu}{year\,1000\,cfm}\right)$*; I = cooling season length (weeks); J = cost of cooling* $\left(\frac{\$}{10^6\,Btu}\right)$

K = seasonal cooling savings $\left(\frac{10^6\,Btu}{yr\,1000\,cfm}\right)$

ECM #14

CONVERSION OF SINGLE DUCT CONSTANT VOLUME REHEAT TO VAV



A single duct reheat system provides cold air continuously in quantities to satisfy peak load conditions. Under all other operating conditions it utilizes additional energy to reheat this air to meet the reduced load. (Whether reheat is based on steam, hot water or electric resistance, it is costly.)

Converting to a variable air volume (VAV) system will totally eliminate the reheat action in most interior zones and significantly reduce it in exterior zones. Simply adding a variable air volume sensor/controller to the terminal unit is all that’s needed in most cases. If heat is occasionally required in some zones, the reheat may be retained and sequenced with the variable air volume control.

The calculations assume an average office space where the architectural design had to allow for flexible usage and, therefore, provided a 25% margin in air supply to handle peak loads. Actual average loading, including early start-ups, lunch vacancies, etc., average 60% peak load. If other energy conservation measures were taken, such as lighting reductions or setpoint changes, the calculated savings would be increased accordingly. Also, if the space selected is a highly variable load, such as a meeting room, the savings could be several times the average calculated.

VAV conversion supplies the actual BTUs required. Therefore, conversion to a VAV system results in savings from "cancelled" cold air energy as well as from the elimination of reheat energy required.

The calculations for savings must cover two conditions:
1.The reduction in cold air provided (during cooling season);
2. The reheat energy saved.

Cooling Savings

$$\frac{30\%}{100} \times \text{air volume delivered by terminal unit}\ \frac{ft^3}{min} \times \left(\text{return air temperature} - \text{cold duct temperature} \right) °F \times 1.08\ \frac{min\ Btu}{hr\ ft^3\ °F} \times$$

$$\text{operating hrs/wk}\ \frac{hrs}{wk} \times \ \frac{wks}{yr} = \text{COOLING BTU s SAVED PER YEAR PER 1000 ft}^3\text{/min OF TERMINAL UNIT AIR FLOW}$$

$$\text{COOLING BTU s SAVED PER YEAR} \times \text{J}\ \frac{\$}{10^6\ Btu} = \text{COOLING \$ SAVED PER YEAR (PER TERMINAL UNIT AIR FLOW)}$$

Heating Savings

$$\frac{20\%}{100} \times \text{air volume delivered by terminal unit}\ \frac{ft^3}{min} \times \left(\text{return air temperature} - \text{cold duct temperature} \right) °F \times 1.08\ \frac{min\ Btu}{hr\ ft^3\ °F} \times$$

$$\text{operating hrs/wk}\ \frac{hrs}{wk} \times 52\ \frac{wks}{yr} = \text{HEATING BTU s SAVED PER YEAR (PER TERMINAL UNIT AIR FLOW)}$$

$$\text{HEATING BTU s SAVED PER YEAR} \times \text{D}\ \frac{\$}{10^6\ Btu} = \text{HEATING \$ SAVED PER YEAR (PER TERMINAL UNIT AIR FLOW)}$$

EXAMPLE: A Chicago area general office with single duct electric reheat at approximately 1333 square feet of floor space, the terminal unit providing 1000 cubic feet per minute air supply. Return air 78°F, cold duct 60°F, 12 hour operating days, 6 day week and electric .10 ¢/Kwh. Water cooled condensers.

Cooling Savings

$$\frac{30\%}{100} \times 1000\ \frac{ft^3}{min} \times \left(78 - 60 \right) °F \times 1.08\ \frac{min\ Btu}{hr\ ft^3\ °F} \times 72\ \frac{hrs}{wk} \times 20.9\ \frac{wks}{yr} =$$

$$8.8 \times 10^6 \text{ COOLING BTU s SAVED PER YEAR}$$

$$8.8 \times 10^6 \times 8.35\ \frac{\$}{10^6\ Btu} = \text{\$73 COOLING \$ SAVED PER YEAR}$$

Heating Savings

$$\frac{20\%}{100} \times 1000\ \frac{ft^3}{min} \times \left(78 - 60 \right) °F \times 1.08\ \frac{min\ Btu}{hr\ ft^3\ °F} \times 72\ \frac{hrs}{wk} \times 52\ \frac{wks}{yr} =$$

$$14.5 \times 10^6 \text{ HEATING BTU s SAVED PER YEAR}$$

$$14.5 \times 10^6 \times 29.31\ \frac{\$}{10^6\ Btu} = \text{\$426 HEATING \$ SAVED PER YEAR}$$

SPECIAL DATA NECESSARY: Use data in *Technical Information* Section.

CAUTIONS: Conversions of a large percentage of the building's terminal units may push the fan operating point into its unstable area resulting in surges in flow and excessive noise. Fans equipped with volume adjustments should be readjusted. For additional energy savings, the fan should be equipped with automatic variable air volume controls which would allow complete conversion of the building to variable air volume if desired. This will provide additional savings in fan energy requirements. (See ECM #13)

Without volume adjustments, the fan operating point may be improved by a speed reduction. Forward curved fans need no special consideration and additional energy savings will be realized in fan power consumption automatically.

A = air handling capacity $\left(\frac{ft^3}{min}\right)$*; B = present ventilation air (%); C = heating season average outside temp. (°F); D = cost of heating* $\left(\frac{\$}{10^6\ Btu}\right)$*;*

E = heating season length (weeks); F is a calculated value: F = TTF x building perimeter in feet x building height in feet $\left(\frac{Btu}{hr\ °F}\right)$*;*

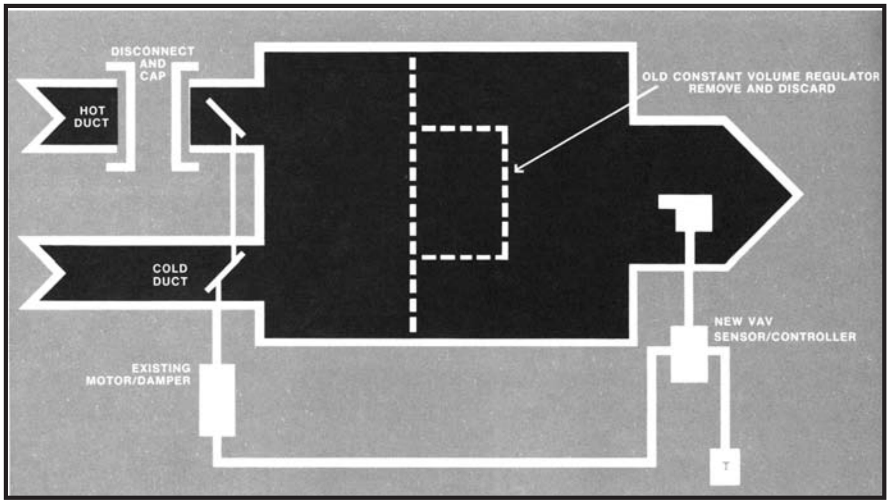
TTF = Thermal Transmission Factor found in the Technical Data Section on page 103; G = cooling season average outside dry bulb temp. (°F);

H = seasonal cooling load for outdoor air $\left(\frac{10^6\ Btu}{year\ 1000\ cfm}\right)$*; I = cooling season length (weeks); J = cost of cooling* $\left(\frac{\$}{10^6\ Btu}\right)$

K = seasonal cooling savings $\left(\frac{10^6\ Btu}{yr\ 1000\ cfm}\right)$

ECM #15

CONVERSION OF DUAL DUCT CONSTANT VOLUME TO VAV



Dual duct constant volume systems continually mix air from the hot and cold ducts to achieve a precise, desired temperature that satisfies the load and space occupants. This is true except on the hottest or coldest days when the design conditions are occurring and only hot or cold air is provided.

This mixing is wasteful. It takes air you expended energy to cool down and mixes it with air you expended energy to heat up. And, under all but extreme design conditions, every Btu generated in the heat duct must be offset by additional cooling. Space temperature could have been better maintained by reducing the air volume instead of its temperature.

The interior zones of practically all dual duct constant volume systems have hot air supplied solely to temper the cold air being delivered during off peak operation. By converting these to single duct variable volume systems, the need for hot air supply in these areas is totally eliminated, along with the wasteful mixing.

The calculations assume an average office space where the architectural design had to allow for flexible usage and, therefore, provided a 25% margin in air supply to handle peak load usage. Additionally, actual average loading, including early start-ups, lunch vacancies, etc., average 60% of peak load. If other energy conservation measures were taken, such as lighting reductions or setpoint changes, the calculated savings would be increased accordingly.

Also, if the space selected has a highly variable load, such as a meeting room, the savings in these areas could be several times the average calculated.

Since the actual Btus required are provided with the VAV conversion, the savings are in the elimination of hot air energy as well as the cold air energy cancelled. This occurs in the 40% of the air volume (average reduction) not delivered and is divided between the hot and cold air in inverse proportion to their Btu content. The Btu content can be computed frrom the hot and cold duct temperatures compared to the space setpoint.

The calculations for savings must cover two conditions:

1.The heating season when free cooling is available and the savings are in the hot air requirements.
2.The cooling season when the hot deck uses return air and the savings are ing the elimination of cold air “cancelled.”

Heating Savings

$$\frac{\left(\text{winter thermostat setpoint} - \text{winter cold deck temperature}\right)^\circ\text{F}}{\left[\left(\text{winter hot deck temperature} - \text{winter thermostat setpoint}\right)^\circ\text{F} + \left(\text{winter thermostat setpoint} - \text{winter cold deck temperature}\right)^\circ\text{F}\right]}$$

$$\times \frac{20\%}{100} \times \text{cfm volume of terminal unit} \ \frac{\text{ft}^3}{\text{min}} \times \left(\text{winter hot deck temperature} - \text{winter mixed air temperature}\right)^\circ\text{F} \times 1.08 \ \frac{\text{min Btu}}{\text{hr ft}^3\ ^\circ\text{F}} \times$$

$$\text{operating hrs/wk} \ \frac{\text{hrs}}{\text{wk}} \times \text{E} \ \frac{\text{wks}}{\text{yr}} = \textbf{HEATING BTU s SAVED PER YEAR}$$

$$\text{HEATING BTU s SAVED PER YEAR} \times \text{D} \ \frac{\$}{10^6\ \text{Btu}} = \textbf{HEATING \$ SAVED PER YEAR} \ \text{(PER TERMINAL UNIT AIR FLOW)}$$

Cooling

$$\frac{\left(\text{summer hot deck temperature} - \text{summer thermostat setpoint}\right)^\circ\text{F}}{\left[\left(\text{summer hot deck temperature} - \text{summer thermostat setpoint}\right)^\circ\text{F} + \left(\text{summer thermostat setpoint} - \text{summer cold deck temperature}\right)^\circ\text{F}\right.}$$

$$\times \frac{30\%}{100} \times \text{cfm volume of terminal unit} \ \frac{\text{ft}^3}{\text{min}} \times \left(\text{summer mixed air temperature} - \text{summer cold deck temperature}\right)^\circ\text{F} \times 1.08 \ \frac{\text{min Btu}}{\text{hr ft}^3\ ^\circ\text{F}} \times$$

$$\text{operating hrs/wk} \ \frac{\text{hrs}}{\text{wk}} \times \text{I} \ \frac{\text{wks}}{\text{yr}} = \textbf{SUMMER BTU s SAVED PER YEAR} \ \text{(PER TERMINAL UNIT AIR FLOW)}$$

$$\text{SUMMER BTU s SAVED PER YEAR} \times \text{J} \ \frac{\$}{10^6\ \text{Btu}} = \textbf{SUMMER \$ SAVED PER YEAR} \ \text{(PER TERMINAL UNIT AIR FLOW)}$$

EXAMPLE: A dual duct constant volume terminal unit in a building in the Chicago area, approximately 1333 ft^2 of floor having an air flow of 1,000 cfm. Electrical costs \$.10/Kwh. Fuel oil costs \$.1.40 gal. Heating season: hot duct 110°F, mixed air 60°F, space temperature 72°F. Cooling season: cold duct 60°F., mixed air 80°F., space temperature 72°F. Water-cooled condensing unit used. Assume a 12 hour day with a 6 day week.

Heating Savings

$$\frac{\left(72 - 60\right)\,°\mathrm{F}}{\left[\left(110 - 72\right)°\mathrm{F} + \left(72 - 60\right)°\mathrm{F}\right]} \times .2 \times 1000\,\frac{\mathrm{ft}^3}{\mathrm{min}} \times \left(110 - 60\right)°\mathrm{F} \times 1.08\,\frac{\mathrm{min\ Btu}}{\mathrm{hr\ ft^3\ °F}} \times 72\,\frac{\mathrm{hrs}}{\mathrm{wk}} \times$$

$$30\,\frac{\mathrm{wks}}{\mathrm{yr}} = 5.6 \times 10^6 \text{ HEATING BTU s SAVED PER YEAR}$$

$$5.6 \times 10^6 \times 14.32\,\frac{\$}{10^6\,\mathrm{Btu}} = \$80.19 \text{HEATING \$ SAVED PER YEAR}$$

Cooling

$$\frac{\left(80 - 72\right)\,°\mathrm{F}}{\left[\left(80 - 72\right)°\mathrm{F} + \left(72 - 60\right)°\mathrm{F}\right]} \times .3 \times 1000\,\frac{\mathrm{ft}^3}{\mathrm{min}} \times \left(80 - 60\right)°\mathrm{F} \times 1.08\,\frac{\mathrm{min\ Btu}}{\mathrm{hr\ ft^3\ °F}} \times 72\,\frac{\mathrm{hrs}}{\mathrm{wk}} \times$$

$$20.9\,\frac{\mathrm{wks}}{\mathrm{yr}} = 3.9 \times 10^6 \text{ SUMMER BTU s SAVED PER YEAR}$$

$$3.9 \times 10^6 \times 8.33\,\frac{\$}{10^6\,\mathrm{Btu}} = \$32 \text{ SUMMER \$ SAVED PER YEAR}$$

SPECIAL DATA NECESSARY: Use data in *Technical Information* Section.

CAUTIONS: Conversions of a large percentage of the building's terminal units may push fan operation into undesirable surge areas. If fans are equipped with volume adjustments, they should be readjusted. For additional energy savings, the fan should be equipped with automatic variable air volume controls which will allow complete conversion of the building to variable air volume if desired and provide additional savings in fan energy requirements. (See ECM #13)

Without volume adjustments, the fan operating point may be improved by a speed reduction. Forward curved fans need no special consideration and additional energy savings will be realized in fan power consumption.

A = air handling capacity $\left(\frac{ft^3}{min}\right)$*; B = present ventilation air (%); C = heating season average outside temp. (°F); D = cost of heating* $\left(\frac{\$}{10^6\,Btu}\right)$*;*

E = heating season length (weeks); F is a calculated value: F = TTF x building perimeter in feet x building height in feet $\left(\frac{Btu}{hr\ °F}\right)$*;*

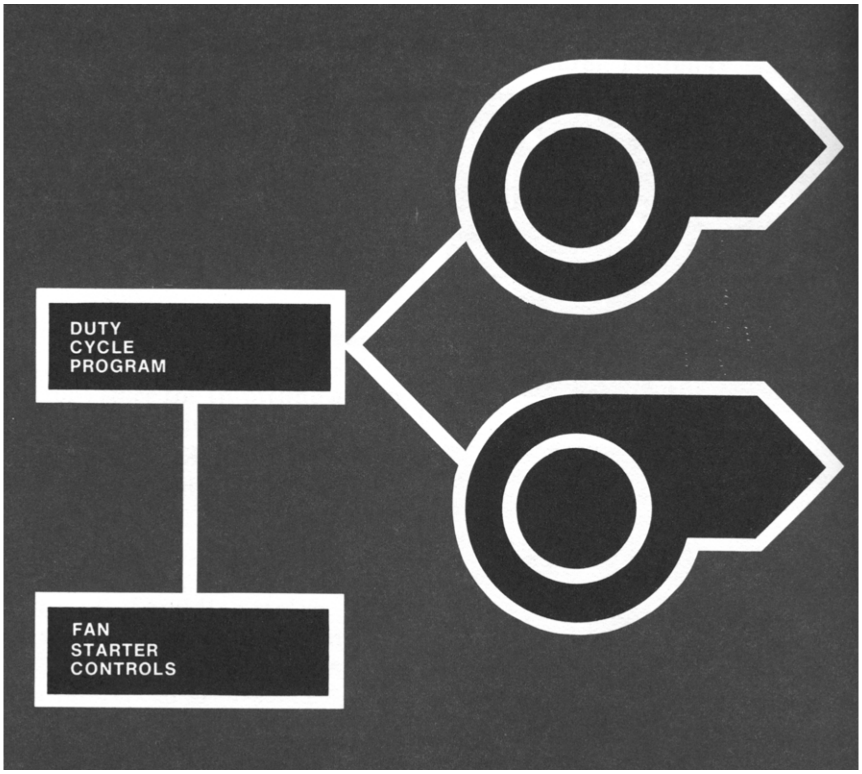
TTF = Thermal Transmission Factor found in the Technical Data Section on page 103; G = cooling season average outside dry bulb temp. (°F);

H = seasonal cooling load for outdoor air $\left(\frac{10^6\,Btu}{year\ 1000\ cfm}\right)$*; I = cooling season length (weeks); J = cost of cooling* $\left(\frac{\$}{10^6\,Btu}\right)$

K = seasonal cooling savings $\left(\frac{10^6\,Btu}{yr\ 1000\ cfm}\right)$

ECM #16

FAN CONTROL WITH DUTY CYCLING



A duty cycle program conserves energy by shutting down a fan system for a portion of its normal operating period. Several programs are available from a simple fixed “off” interval per time cycle period to a modulated “off” time based on space temperature and outdoor conditions. Duty cycle programs are normally applied with building automation systems. Specialized duty cycling controllers are also available.

Duty cycle programs are often used where several air handlers or packaged rooftops provide space conditioning to the same area and an “off” cycle for a single unit will not significantly effect occupant comfort.

Any constant volume single or dual duct central fan system can be considered for a duty cycle program. Fan systems serving areas where ventilation is critical, such as rooms with high or consistent occupancy, are not good candidates for consideration. However warehouses, gymnasiums, large warehouse style retail stores and large athletic stadiums are good candidates.

Duty cycling was more common in the 1970s and 1980s prior to the advent of VAV systems. It is also a predecessor strategy to the application of variable frequency drive (VFD) fan control. Duty cycling is now less favored because occupant comfort suffers if misapplied and frequent on/off motor switching increases motor wear, causes surges in building Kw demand and provides less precise space comfort control.

Duty cycling is normally not recommended for VAV systems which are designed to automatically reduce system fan horsepower.

Fan Electrical Savings

$$\text{fan horsepower} \ \text{hp} \times .8 \frac{\text{kw}}{\text{hp}} \times \text{hours fan is cycled off} \ \frac{\text{hrs}}{\text{wk}} \times 52 \frac{\text{wk}}{\text{yr}} = \text{KWH s SAVED PER YEAR}$$

$$\text{KWH s SAVED PER YEAR} \times \text{electrical cost} \ \frac{\$}{\text{Kwh}} = \text{\$ SAVED PER YEAR}$$

Yearly Heating Ventilation Savings

$$\text{A} \ \frac{\text{ft}^3}{\text{min}} \times \frac{\text{B} \ \%}{100} \times 1.08 \frac{\text{min Btu}}{\text{hr ft}^3 \ ^\circ\text{F}} \times \left(\text{average space temperature} - \text{C} \right) ^\circ\text{F} \times \text{E} \ \frac{\text{wks}}{\text{yr}} \times \text{hours fan is cycled off} \ \frac{\text{hrs}}{\text{wk}} =$$

$$\text{HEATING BTU s SAVED PER YEAR}$$

$$\text{HEATING BTU s SAVED PER YEAR} \times \text{D} \ \frac{\$}{10^6 \ \text{Btu}} = \text{HEATING \$ SAVED PER YEAR}$$

Yearly Cooling Ventilation Savings

$$\text{A} \ \frac{\text{ft}^3}{\text{min}} \times \frac{\text{B} \ \%}{100} \times \frac{\text{H} \quad \text{Btu}}{\text{yr } 1000 \ \text{ft}^3/\text{min}} \times \frac{\text{hrs fan cycled off/wk}}{50 \ \text{hrs/wk}} = \text{COOLING BTU s SAVED PER YEAR}$$

$$\text{COOLING BTU s SAVED PER YEAR} \times \text{J} \ \frac{\$}{10^6 \ \text{Btu}} = \text{COOLING \$ SAVED PER YEAR}$$

EXAMPLE: 80,000 ft² two story building, Chicago weather data, 60,000 cfm air handling capacity, electric cost $.10 per Kwh, air cooled condenser, fuel cost $13.60 per 10^6 Btu's, 50 horsepower fan system, 15% ventilation air, 7 hours cycled off per week.

Fan Electrical Savings

$$50 \ \text{hp} \times .8 \frac{\text{kw}}{\text{hp}} \ 7 \ \frac{\text{hrs}}{\text{wk}} \times 52 \frac{\text{wk}}{\text{yr}} = \text{14560 KWH s SAVED PER YEAR}$$

$$14560 \times .10 \ \frac{\$}{10^6 \ \text{Btu}} = \text{\$456 SAVED PER YEAR}$$

Yearly Heating Ventilation Savings

$$60{,}000 \ \frac{\text{ft}^3}{\text{min}} \times \frac{15 \ \%}{100} \times 1.08 \frac{\text{min Btu}}{\text{hr ft}^3 \ ^\circ\text{F}} \times \left(68 - 34.2 \right) ^\circ\text{F} \times 30 \ \frac{\text{wks}}{\text{yr}} \times 7 \ \frac{\text{hr}}{\text{wk}} =$$

$$69 \times 10^6 \ \text{HEATING BTU's SAVED PER YEAR}$$

$$69 \times 10^6 \times 13.60 \ \frac{\$}{10^6 \ \text{Btu}} = \text{\$938 HEATING \$ SAVED PER YEAR}$$

Yearly Cooling Ventilation Savings

$$60{,}000 \ \frac{\text{ft}^3}{\text{min}} \times \frac{15 \ \%}{100} \times \frac{38.791 \times 10^6 \ \text{Btu}}{\text{yr } 1000 \ \text{ft}^3/\text{min}} \times \frac{7 \ \text{hrs/wk}}{50 \ \text{hrs/wk}} = 48.9 \times 10^6 \ \text{BTU s SAVED PER YEAR}$$

$$48.9 \times 10^6 \times 10.00 \ \frac{\$}{10^6 \ \text{Btu}} = \text{\$489 COOLING \$ SAVED PER YEAR}$$

SPECIAL DATA NECESSARY: Use data in *Technical Information* Section.

CAUTIONS: Consult your local Honeywell representative for recommendations on how often a particular fan motor can be safely cycled and how to determine if it serves a non-critical ventilation area.

Centrifugal fans have high inertia loads and require careful consideration of belt drives to prevent damage to belts and/or motor windings and motor characteristics.

DEFINITION OF TERMS For estimating purposes, savings can be calculated using: Kilowatts=0.8 x Nameplate horsepower

A = air handling capacity $\left(\frac{ft^3}{min}\right)$*; B = present ventilation air (%); C = heating season average outside temp. (°F); D = cost of heating* $\left(\frac{\$}{10^6 \ Btu}\right)$*;*

E = heating season length (weeks); F is a calculated value: F = TTF x building perimeter in feet x building height in feet $\left(\frac{Btu}{hr \ ^\circ F}\right)$*;*

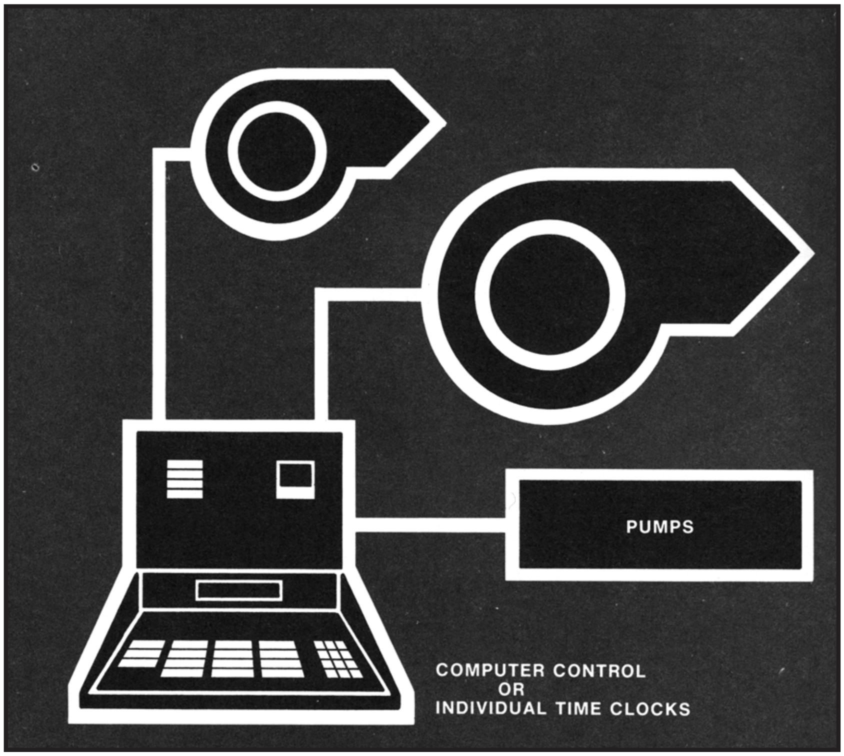
TTF = Thermal Transmission Factor found in the Technical Data Section on page 103; G = cooling season average outside dry bulb temp. (°F);

H = seasonal cooling load for outdoor air $\left(\frac{10^6 \ Btu}{year \ 1000 \ cfm}\right)$*; I = cooling season length (weeks); J = cost of cooling* $\left(\frac{\$}{10^6 \ Btu}\right)$

K = seasonal cooling savings $\left(\frac{10^6 \ Btu}{yr \ 1000 \ cfm}\right)$

ECM #17

EQUIPMENT SCHEDULING



Automatic scheduling of HVAC equipment saves energy in many areas. The four main areas include:

- Electric energy for fans and/or pumps
- Cooling energy for ventilation (outdoor) air
- Heating energy for ventilation air
- Lowered night temperature during heating season

Establishing a weekly schedule where equipment is turned on and off to conform to the hours the building is normally occupied will result in significant energy savings.

OPTIMUM EQUIPMENT

Scheduling produces additional savings by varying equipment start and stop times so morning warm-up times (or cool-down times) are no longer than necessary, as measured by space and outdoor temperatures. Flexible scheduling is a "must" for any building having frequent or occasional changes in occupancy patterns. Within this feature, automatic schedules tend to reflect "worst case" occupancy hours.

Equipment that is controlled by a building automation systems can be quickly and easily changed without use of tools and can be accomplished from a central or remote location with a computer control system.

Yearly Fan or Pump Electrical Savings

$$\text{hours saved per week*}\ \frac{\text{hrs}}{\text{wk}} \times \text{weeks per year operation}\ \frac{\text{wks}}{\text{yr}} \times .8\,\frac{\text{kw}}{\text{hp}} \times \text{fan or pump horsepower hp} = \text{KWH s SAVED PER YEAR}$$

$$\text{KWHs SAVED PER YEAR} \times \text{cost of electricity}\ \frac{\$}{\text{Kwh}} = \text{\$ SAVED PER YEAR}$$

Yearly Heating Savings

$$\text{hours saved per week*}\ \frac{\text{hrs}}{\text{wk}} \times \text{E}\ \frac{\text{wks}}{\text{yr}} \times \text{A}\ \frac{\text{ft}^3}{\text{min}} \times \frac{\text{B}}{100}\,\% \times 1.08\,\frac{\text{min Btu}}{\text{hr ft}^3\,{}^\circ\text{F}} \times \left(\text{average indoor temperature} - \text{C}\right){}^\circ\text{F} =$$

$$\text{HEATING BTU s SAVED PER YEAR}$$

$$\text{HEATING BTUs SAVED PER YEAR} \times \text{D}\ \frac{\$}{10^6\,\text{Btu}} = \text{HEATING \$ SAVED PER YEAR}$$

Yearly Cooling Savings

$$\frac{\text{hours saved/wk}}{50\ \text{hrs/wk}} \times \frac{\text{A ft}^3}{\text{min}} \times \frac{\text{B}}{100}\,\% \times \frac{\text{H}\quad\text{Btu}}{\text{yr 1000 ft}^3/\text{min}} = \text{COOLING BTU s SAVED PER YEAR}$$

$$\text{COOLING BTUs SAVED PER YEAR} \times \text{J}\ \frac{\$}{10^6\,\text{Btu}} = \text{COOLING \$ SAVED PER YEAR}$$

EXAMPLE An 80,000 square foot two-story building, Chicago weather data, 60,000 cfm air handling capacity. Electrical costs $.10 per Kwh, air cooled condensers, fuel at $13.60 per 10^6 Btu's, 50 hp fan system, 18.6 % ventilation air, occupied 8 a.m. to 6 p.m. 5 days a week, currently running 6 a.m. to 11 p.m. 5 days a week.

Be sure we allow 1 hour/day warm up or cool down.

17 running hours/day
—10 occupied hrs/day
— 1 hr startup/day
6 hours saved/day or 5 x 6 = 30 hrs/wk

Yearly Fan or Pump Electrical Savings

$$30\ \frac{\text{hrs}}{\text{wk}} \times 52\ \frac{\text{wks}}{\text{yr}} \times .8\,\frac{\text{kw}}{\text{hp}} \times 50\ \text{hp} = \text{62,400 KWH s SAVED PER YEAR}$$

$$62{,}400 \times .10\ \frac{\$}{\text{Kwh}} = \text{\$6240 SAVED PER YEAR}$$

Yearly Heating Savings

$$30\ \frac{\text{hrs}}{\text{wk}} \times 30\ \frac{\text{wks}}{\text{yr}} \times 60{,}000\ \frac{\text{ft}^3}{\text{min}} \times \frac{18.6}{100}\,\% \times 1.08\,\frac{\text{min Btu}}{\text{hr ft}^3\,{}^\circ\text{F}} \times \left(60\ {}^\circ\text{F} - 34.2\ {}^\circ\text{F}\right) =$$

$$280 \times 10^6\ \text{HEATING BTU s SAVED PER YEAR}$$

$$280 \times 10^6 \times 13.60\ \frac{\$}{10^6\,\text{Btu}} = \text{\$3808 HEATING \$ SAVED PER YEAR}$$

Yearly Cooling Savings

$$\frac{30\ \text{hrs/wk}}{50\ \text{hrs/wk}} \times \frac{60{,}000\ \text{ft}^3}{\text{min}} \times \frac{18.6}{100}\,\% \times \frac{38.791 \times 10^6\ \text{Btu}}{\text{yr 1000 ft}^3\ \text{min}} =$$

$$259.7 \times 10^6\ \text{COOLING BTU s SAVED PER YEAR}$$

$$259.7 \times 10^6 \times 10.0\ \frac{\$}{10^6\,\text{Btu}} = \text{\$ 2597 COOLING \$ SAVED PER YEAR}$$

SPECIAL DATA NECESSARY

1. *Warm Air Temperature.* Except for interior zones having a year-round cooling load, it may be assumed that each cfm of outside air brought in by an air handling system will be warmed up at least to space temperature before leaving via the exhaust system or by building leakage. Therefore, the outdoor air adds to the heat loss of the building.
2. *Occupied* hours must be carefully estimated. A good rule is: If the air conditioning and/or heating must be maintained at comfort levels for any valid reason, then the space must be considered occupied. If no ventilation air is needed, and if temperatures can be allowed to vary above and below the 68° to 78°F range, it may be considered unoccupied.
3. *Warm-up/cool-down* hours per week. If an *optimum* start time apparatus is to be applied, use these guidelines.
   (a) Winter: Subtract earliest start time required in cold weather from latest practical start time in mild weather (50°F).
   (b) Summer: Subtract earliest start time required in hot weather from occupied time, minus one-half hour.
   (c) It is preferred that actual on-site testing be used to determine latest possible start time.
4. Weeks/year might be reduced due to holidays and plant shut-downs.
5. Use data in *Technical Information* Section.

A = air handling capacity $\left(\frac{ft^3}{min}\right)$; *B = present ventilation air (%); C = heating season average outside temp. (°F); D = cost of heating* $\left(\frac{\$}{10^6\,Btu}\right)$;
E = heating season length (weeks); F is a calculated value: F = TTF x building perimeter in feet x building height in feet $\left(\frac{Btu}{hr\,{}^\circ F}\right)$;
TTF = Thermal Transmission Factor found in the Technical Data Section on page 103; G = cooling season average outside dry bulb temp. (°F);
H = seasonal cooling load for outdoor air $\left(\frac{10^6\,Btu}{year\ 1000\ cfm}\right)$; *I = cooling season length (weeks); J = cost of cooling* $\left(\frac{\$}{10^6\,Btu}\right)$
K = seasonal cooling savings $\left(\frac{10^6\,Btu}{yr\ 1000\ cfm}\right)$

BOILER CONTROL

FUEL CAN BE CRUEL WHEN THE HEAT'S ON

The cost of fossil fuel for heating far outweighs cooling costs for commercial buildings in the United States. How this valuable resource is managed should be a high priority for building owners.

Once again, the assumption will be made that you have already studied and adopted suitable conservation steps to save steam or hot water consumed for space heating, air handlers, domestic water and processes. This section will deal with ways to obtain more heat out of each gallon, pound or cubic foot of fuel.

Boilers offer three fuel conservation opportunities:

1. Improvement of Combustion Efficiency
2. Reduction of Losses External to the Combustion Chamber
3. Better Control of the Furnace or Boiler

IMPROVING COMBUSTION EFFICIENCY

Combustion efficiency is a function of many factors, but once a fuel burning apparatus is installed, two variables can still be improved:
•Fuel Atomizing & Mixing with Combustion air
•Air-Fuel Ratio Adjustment

Measurement of combustion efficiency is a well-developed science, and its simplest form consists of measuring:
1) Stack Temperature
2) Flue Gas Composition (O2 or CO2)

All furnaces and boilers for commercial buildings, regardless of size, should have routine measurements of stack temperatures and flue gas analysis at frequent intervals and at low, medium and high fire situations where applicable.

FLUE GAS TEMPERATURE

High flue gas temperature indicates low efficiency. This may result from excess combustion air, dirty heat exchange surface or both. An "ideal" flue temperature should be established at full firing rate, after all fireside surfaces are cleaned and after the combustion air has been adjusted to its optimum point (see Flue Gas Analysis).

FLUE GAS COMPOSITION

"Batch" type portable analyzers to measure either oxygen (O2) or carbon dioxide (CO2) are readily available. They show how much excess air is going up the stack. Generally provided with the analyzer are charts showing the combustion efficiency for various stack temperatures. For large boilers, continuous reading analyzers relieve the operator from the chore of taking readings with the portable devices. It will also insure that consistent readings are documented whether or not staff are available.

It is not unusual to find burners using 60 to 100 percent excess air, especially in buildings where flue gas is not analyzed regularly, or where excess air is adjusted to entirely eliminate smoke.

The O2 analysis is preferred over CO2 analysis. It is more accurate because CO2 (ORSATT) measurements are insensitive to excess air measurements under 10 percent. O2 measurements on the other hand, are equally accurate at all quantities below 60 percent excess air. (See Figure 1 for a comparison.)

BURNERS WITH FIXED FIRING RATE

This adjustment should be made by a qualified person experienced in commercial burner operation. Proper maintenance offers another potential wealth of savings opportunities.

Improvements on the order of 3 to 5 percent in efficiency can usually be made in furnaces or boilers that have not been cleaned, checked and adjusted regularly.

BURNERS WITH HIGH-LOW OR VARIABLE FIRING RATE

The linkages in these burners simultaneously open the fuel valve and the combustion air shutters or valves. Flue gas analysis should be made and recorded at various firing rates and the linkages adjusted (by a qualified person) to obtain the lowest attainable stack temperature and excess O2. Microprocessor based simultaneous adjustment of the fuel valve and combustion air shutters or valves have only recently been introduced to the market. These should be investigated in order to determine whether a retrofit might be beneficial.

USING FLUE GAS O_2 AND TEMPERATURE READINGS TO IMPROVE EFFICIENCY

The ideal O2 concentration in flue gas is in the 3 to 5 percent range. While you might think 0 percent would give you perfect combustion, lower than 3 to 5 percent is impractical and unsafe.

Higher percentages can also be inefficient. First, cold air must be heated up to 400 to 500° F. Second, excess air flow through the furnace means it doesn't stay in contact with the tubes long enough to give up its heat to the water being heated. To put it another way, more gas is being pushed through the heat exchange surface than it was designed to handle and heat is being generated without exchanging it to the water to be distributed to the building.

For this discussion we'll deal in O2 measurements only. If your present instruments read CO2, use Figure 2 to convert from CO2 to O2 readings.

FUEL OIL

For oil, the O2 concentration should be reduced to 3 to 5 percent if this is possible without causing objectionable smoke.

NATURAL GAS

For natural gas, the flue O2 reading can usually be held down to the same level, providing the burner is in excellent condition and the carbon monoxide (CO) emission can be maintained below 440 ppm.

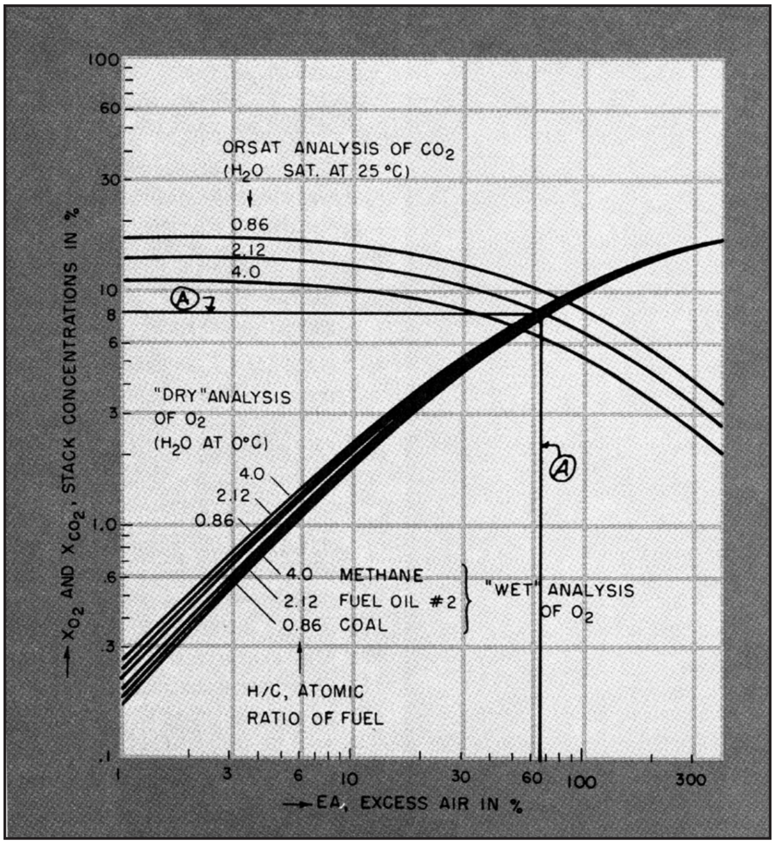


Figure 1

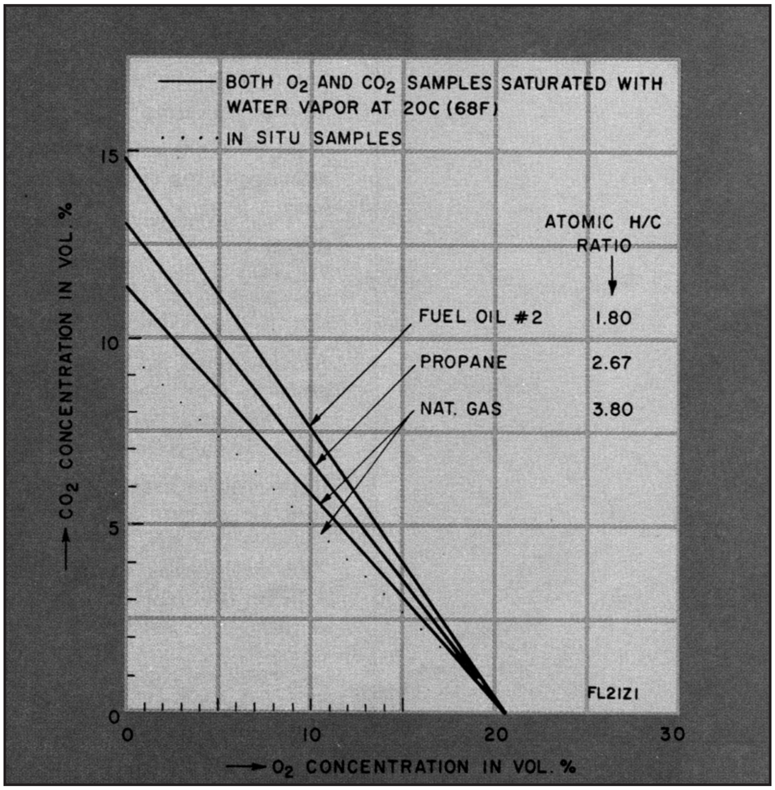


Figure 2

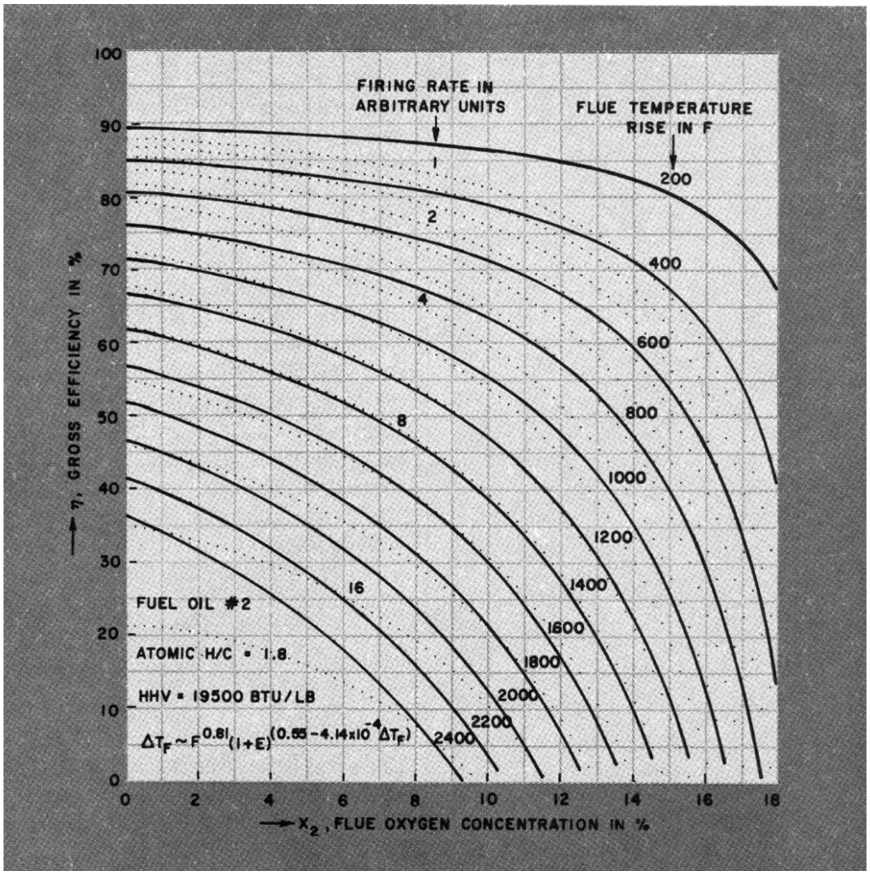


Figure 3

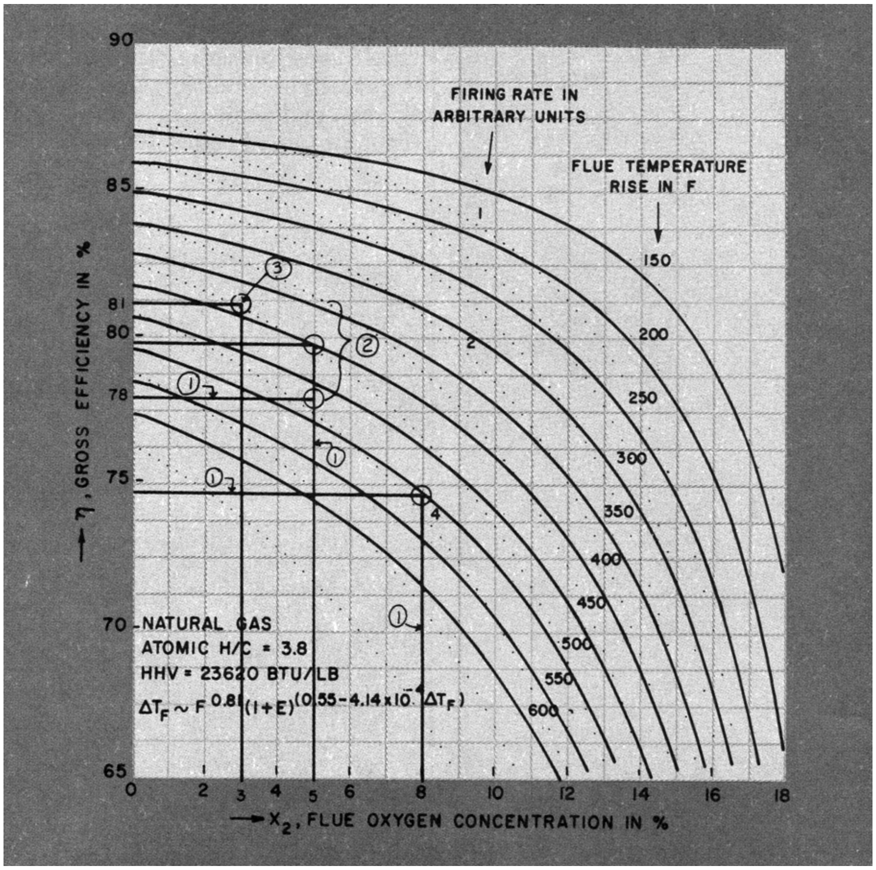


Figure 4

GROSS EFFICIENCY VS. O2, FLUE TEMPERATURE AND FIRING RATE

Figure 3 shows the relationship between these variables for furnaces and boilers using #2 fuel oil. See figure 4 for natural gas.

There are many clues on Figure 4 for analyzing combustion efficiency and finding ways to improve it.

Step 1. Lower O2 readings produce higher efficiency at fixed firing rate (dotted line).

EXAMPLE 1: An O2 reading of 8 percent was obtained at a flue gas temperature of 580°F, on boiler "X" using 1,000,000 BTU/hr of gas. Ambient temperature in boiler room is 80° F.

ANALYSIS: Figure 1 shows that 8 percent O2 means 65 percent excess air is being used. Figure 4 shows efficiency is 75 percent at intersection of 500° F line and 8 percent O2. If we reduce O2 reading to 5 percent, with the firing rate remaining the same, efficiency will be increased to 78 percent.

Step 2. Lower firing rates for a given furnace or boiler result in increased efficiency.

EXAMPLE 2: It is observed boiler "X" fires only 60 percent of the time during the coldest weather. If you decide to reduce gas fed to the burner from 1,000,000 BTU to 700,000 BTU/hr, what will be the new percent "on" time?

.6 x 1,000,000 = 600,000- this is average firing rate required at full load.

600,000 / 700,000 = .857, or 86% "on" time with 7,000,000 Btu input

Assuming a 5 percent O2 reading is possible after readjusting combustion air, what efficiency and flue temperature will result?

Referring to Figure 4 again, our "old" firing rate was 10 and the new is 7, or 70 percent of the old. The "old" firing rate corresponds to 3.77 on the arbitrary firing rate scale. The new firing rate is .7 x 3.77 or 2.64. Going up the 5 percent O2 line to 2.64 firing rate (arbitrary) would correspond to 350º F flue temperature (430º F actual). However, minimum flue temperature is limited by the formula on Figure 4 (DT ºF – F.81 etc. etc.). Since E (excess air) is constant, F.81 is the significant term. This works out to a flue temperature of 397º F.

$$315 - 3.77^{.81}$$

and $\Delta\, T^{\circ}F - 2.64^{.81}$

$$\frac{2.64^{.81}}{3.77^{.81}} = .7003^{.81} = 7.49$$

$$.749 \text{ x } 315 = 236\ T$$
(above 240˚F)

$$240 + 236 = 476^{\circ}F$$
$$476 - 80 = 396^{\circ}F$$

③ on Figure 4

Following the 396ºF line, the new gross efficiency is found to be 80 percent – a gain of 2 percent over the old firing rate.

Step 3: Optimum percent of O2 is in the range of 3 to 5 percent. Burners may need replacement or improvement if this O2 reading can't be obtained without excess CO (400ppm).

EXAMPLE 3: If the burner was improved to allow an O2 reading as low as 3 percent without changing firing rate, an 81 percent efficiency might be possible.

Note: Similar steps can be analyzed using Figure 3 for oil-fired situations.

COMBUSTION CONTROLS FOR LARGE BOILERS

Most boilers having firing rates in excess of 1,000,000 BTU/hr are equipped with a combustion control systems which regulate draft, fire box pressure and fuel-air ratio.

Systems are now available that regulate combustion air to maintain a fixed O2 reading. The O2 content and temperature of the flue gas are measured continuously, along with smoke level (such as Ringelmann numbers). These measurements control combustion efficiency at a high level, on the order of 80 percent or higher, regardless of firing rate. Boilers using fuel in excess of $100,000/year can often justify upgrading the combustion control system by upgrading a closed loop oxygen control system. These systems use zirconium oxide sensors and usually have a smoke sensor as a low limit control.

BURNER RETROFIT

With the advent of more costly fuel, many burners are being retrofitted to achieve better atomization and mixing of air and fuel. This allows more complete combustion, less soot and smoke, less frequent cleaning, and better combustion efficiency.

Oil burners that cannot be adjusted to achieve O2 readings down to 5 percent excess without causing smoke (or high CO in the case of gas burners) are candidates for replacement burners. These burners are capable of more efficient atomizing of oil or mixing of gas with combustion air.

OIL-WATER EMULSION FUEL SYSTEMS

Patented systems using water to emulsify the fuel oil (usually heavier grades) are available. Claimed advantages are better atomizing of fuel, less smoke, fewer pollutants (Nox), and improved combustion efficiency. However, vaporization of the water into steam reduces the energy available to the boiler. The "break-even" point is a function of percent water content and the amount that excess air can be reduced because of better atomizing.

For example, an emulsion with 20 percent water with a stack temperature of 500º F would require that excess air would have to be reduced by more than 25 percent in order to achieve higher efficiency. (Burners using less than 20 percent water are available).

It should also be noted that the firing rate of the burner may be reduced by adding water. Reducing firing rate does increase efficiency (Figure 3). However, there are methods of reducing firing rate without adding water.

For further information search ASHRAE technical papers and abstracts for recent studies regarding efficiency of external combustion systems, limitations of firing fuel-oil-water emulsion.

REDUCING LOSSES EXTERNAL TO THE COMBUSTION CHAMBER

Chief sources of loss for a boiler plant consist of:

- Stack loss during firing (a function of combustion control)
- Stack loss during off cycle
- Loss from furnace walls, casing, breaching, etc.
- Piping losses

Each of the above losses can be reduced, depending on the economics in each situation. Some of these energy conservation techniques are:

1. Preheat combustion air with flue gas.
2. Preheat feed-water from flue gas.
3. Install dampers to prevent off-cycle stack loss
4. Draw combustion air from hottest point in boiler room.
5. Insulate piping, especially in tunnels and other areas not needing heat.
6. Insulate casings and other furnace parts that prevent loss of heat to the boiler room.

SPOT HEATING

Large boilers are often operated 12 months a year because they supply domestic hot water, or other needs that use only a small fraction of boiler capacity. For these cases, consider implementing a smaller boiler or furnace to avoid the fixed losses from piping, boiler casings, etc., associated with a large plant. In some cases, electric heaters can be justified for small “cold spots” in lieu of operating a large boiler.

To evaluate spot heating, carefully measure fuel consumed and heat delivered to the summer load. Then calculate the summer seasonal cost per BTU. Starting with this number, say $10 per million BTU, the economics of “spot heating” will be a straightforward calculation and a smaller heating unit may be justifiable.

BOILER CONTROL

For hot water boilers supplying water directly to space heating equipment, such as radiation, reheat, or preheat coils, or fan coil units, outdoor reset of supply water temperature is useful for two reasons:

A: Closer control in the spaces without wasteful "overshooting" of the desired control point.

B: Reduced piping losses.

CHILLER CONTROL

KEEPING COOLING KILOWATTS "COOL"

Energy used for air conditioning accounts for a significant portion of the kilowatt hours consumed during the year. And fuel bills skyrocket when the cooling load sets daily and seasonal peak demands on the electric company.

This discussion of cooling will assume two things. One, that as part of your energy survey, you have made a careful analysis of systems using refrigeration, air handlers, roof-top units, fan coil units, etc. And two, that you have already applied suitable control Energy Conservation Measures (higher space temperatures, reduction of outside air, reduction of operating hours, utilizing "free cooling" and demand control ventilation) where economically feasible.

Your goal, as an energy manager, is cooling efficiency; getting more cooling (tons) for less money.

CENTRAL CHILLER PLANTS

Central chiller plants usually supply chilled water to a variety of air handlers. These systems are quite large—usually over 500 tons. All parts of the chilled water system, including compressor, pumps, and cooling towers, represent opportunities for energy conservation. We will review each of these system elements individually.

Water chillers use about 1horsepower for each ton of capacity at full load — .746 kilowatt per horsepower. Older chillers consume energy at rates over 1 Kw/ton. Some newer, more efficient chillers consume at this rate or about .4 Kw/ton. During the 1990s when natural gas was a reliable, more affordable fuel in much of the country, many chillers were designed to be operated by gas driven motors. Another popular and sound strategy would be to consider replacing a large capacity chiller with several smaller, higher efficiency units. A smaller modular unit can operate far more efficiently when it is near fully loaded than a much larger capacity chiller only partially loaded. Plan to investigate whether a chiller retrofit would benefit your facility.

Useful Equations

Power and energy: to convert amps to horsepower (hp) or kilowatts (Kw) for 3 phase motors.

volts x amps per phase x $\sqrt{3}$ x power factor ÷ 1000 = KWs

Example: 460 volts x 115 amps x 1.732 x .9 ÷ 1000 = 82.46 KWs

$$\text{Horsepower} = \frac{\text{Kw}}{.746} \quad \text{Example: } \frac{82.46}{.746} = 110.5 \text{ HP}$$

Water flow and tonnage are related to temperature difference.
We call the following three the "24" equations.

GPM = gallons per minute
TONS = 12.000 Btu/hr
TD = temperature difference, in and out

$$\text{TONs} = \frac{\text{GPM x TD}}{24} \qquad \text{GPM} = \frac{\text{TONs x 24}}{\text{TD}} \qquad \text{TD} = \frac{\text{TONs x 24}}{\text{GPM}}$$

Example: Chiller is rated at 600 tons cooling with water in at 600F, out at 45 °F. How many gallons per minute (GPM) will it use?

$$\text{GPM} = \frac{600 \text{ x } 24}{15} = 960 \text{ GPM}$$

In the above example, if the temperature difference (TD) across cooling coils is increased from 15 to 17 °F (62 in, 45 out), how much would the flow through the pumps be reduced? What effect on energy costs would this represent?

$$\text{GPM} = \frac{600 \text{ x } 24}{17} = 864 \text{ GPM}$$

This represents a reduction of 96 GPM or 10% flow. Increasing temperature difference only 2 °F would mean options of either downsizing the pump or operating the pump at a lower speed with a variable frequency drive (VFD) to provide the 10% lower, optimized flow.

Both fans and pumps follow the "affinity laws" for fluid dynamics where a reduction in flow brings about a greater than 1-for-1 reduction in required power. Running a pump or fan at "full speed" is absolutely the most expensive practice. Any meaningful reduction in speed can result in a 2X to 3X reduction in required power (Kw) and a very significant reduction in the electrical charges (Kwh). Stated simply, new power requirement = new speed cubed x original speed cubed.

$$\text{Power} = \frac{\text{New speed } X^3}{\text{Original speed } Y^3}$$

For example a 10% speed reduction:

$\frac{9}{10} = \frac{729}{1000} =$ **73% power required or, 27% reduction !**

This is a theoretical maximum savings and it's probably wise to only expect about 90% to 95% of this due to mechanical loss in efficiency. But, reduce pump speed 10% delivers about a 25% reduction in Kwh electrical energy consumption (less any energy cost for the 2 °F additional temperature drop).

Chiller systems offer BIG opportunities to reduce energy consumption. In most instances, retrofitting a VFD is only about the cost of full pump replacement. And, why replace a reasonably efficient pump without gaining the long-term benefits of speed control? Do the math to see just what is best in your facility. Any control strategies that allow you to take advantage of "partial load" conditions and translate that into reduced fan/pump speed will pay immediate benefits.

You will need to measure and verify your current chiller and pump capacities and current operations in order to establish a baseline necessary to project any meaningful energy savings. Data loggers and current transformers (CT) on motors will help you determine the facts and not depend on anecdotal or subjective "data." Local utilities are often very helpful during the measurement phase of your project.

Start with the chiller catalog ratings.

Table 1 is an example of a chiller capacity rating.

Water chillers use about 1horsepower for each ton of capacity at full load — .746 kilowatt per horsepower. For more accuracy; refer to catalog data or specifications furnished with your chiller(s), or supplied by the chiller company representative. Obtain current model brochures and specification data to get a better idea of how efficiently current chillers are.

Suppose your model 90 chiller is operating at 42° F leaving chilled water temperature. Refer to the preceding table. With 85°F leaving condenser water, the chiller can produce 425 tons using 313 kilowatts (KW) of power. Assume the entering chilled water temperature is observed to be 57° F. This is a temperature difference of 15° F. When these numbers are plugged into our equation, we find:

$$\text{GPM} = \frac{425 \text{ x } 24}{15} = 680 \text{ GPM}$$

This is within the mm-max range of 274 to 768 GPM listed in the rating table. If you have a flow meter on your chiller, you can check actual tonnage delivered by using this equation:

$$\text{TONS} = \frac{\text{GPM x TD}}{24}$$

Chilled Water Pumps

The chilled water pump is likely to be the second biggest energy consumer in your plant. But just how big a culprit is it? Let's look at an example.

Assume the nameplate on a chilled water pump lists the horsepower at 50. Pressure gauges on either side of the pump indicate it is pumping against 150 feet of head at a full load of 530 gallons per minute (GPM). The best way to check power consumed is to use a pump curve from the pump manufacturer or a portable watt meter. (An ammeter is not as accurate because you must guess at the power factor.) Using the pump curves you would find the brake horsepower needed at 150 feet of head is 39.5. The following calculation converts the incoming power to operating power, taking into consideration an efficiency factor.

39.5 hp x .746 Kw/hp x 1/ .9 efficiency = 32.7 KW

Compare these numbers to the original mechanical plans and schedules if you have them.

Condensing Water Pumps

A useful equation for determining gallons per minute needed in condensers is:

$$\text{GPM} = \frac{\text{tons (chiller capacity) x 30}}{\text{TD}}$$

The factor of 30, rather than 24, is used because condensers must usually remove 125% of the heat rejected by the evaporator. The extra 25% is due to the compressor inefficiency and other factors.

Assume you observed a 12 degree temperature rise through the condenser at the full load condition. For our example, the gallons per minute would be calculated as:

$$\frac{425 \text{ tons x } 30}{12} = 1062 \text{ GPM}$$

If you can measure the flow of the condenser water, you can verify the amount of heat rejected (in tons) using the "24" equations. (Remember, however, the condensing water pump has only 30 pounds of head to pump against because the tower is nearby.)

Again, pump curves give the best picture of the brake horsepower needed to move this water. Assume you observed a 30 hp nameplate on the condensing water pump and found pump curves indicating 21.6 brake horsepower (bhp) needed. Plug the new numbers into our conversion equation.

21.6 HP x .746 Kw/hp x 1/.9 efficiency = 17.9 KW

Use this figure for full load condensing water power.

Cooling Tower Fans

For this job, we will employ a different example. Assume that the fan nameplate says 12.5 hp. and all other data is missing. Since this motor is relatively small, assume .8Kw/hp (approximate).

8x12.5 =10 KW

(The .8 factor allows for partial loading (90%) in respect to nameplate rating and 85% motor efficiency at that load.)

We can summarize our preliminary findings on our chiller as follows:

Model 90 centrifugal chiller
Rated tons: 425 (313 Kw) at 4° F leaving,
57° F entering with 680 GPM (chw) flow at full load
Motor: 256 Kw, 460 volts, 3 phase 60 Hertz
Full load amps: 360 (from nameplate)

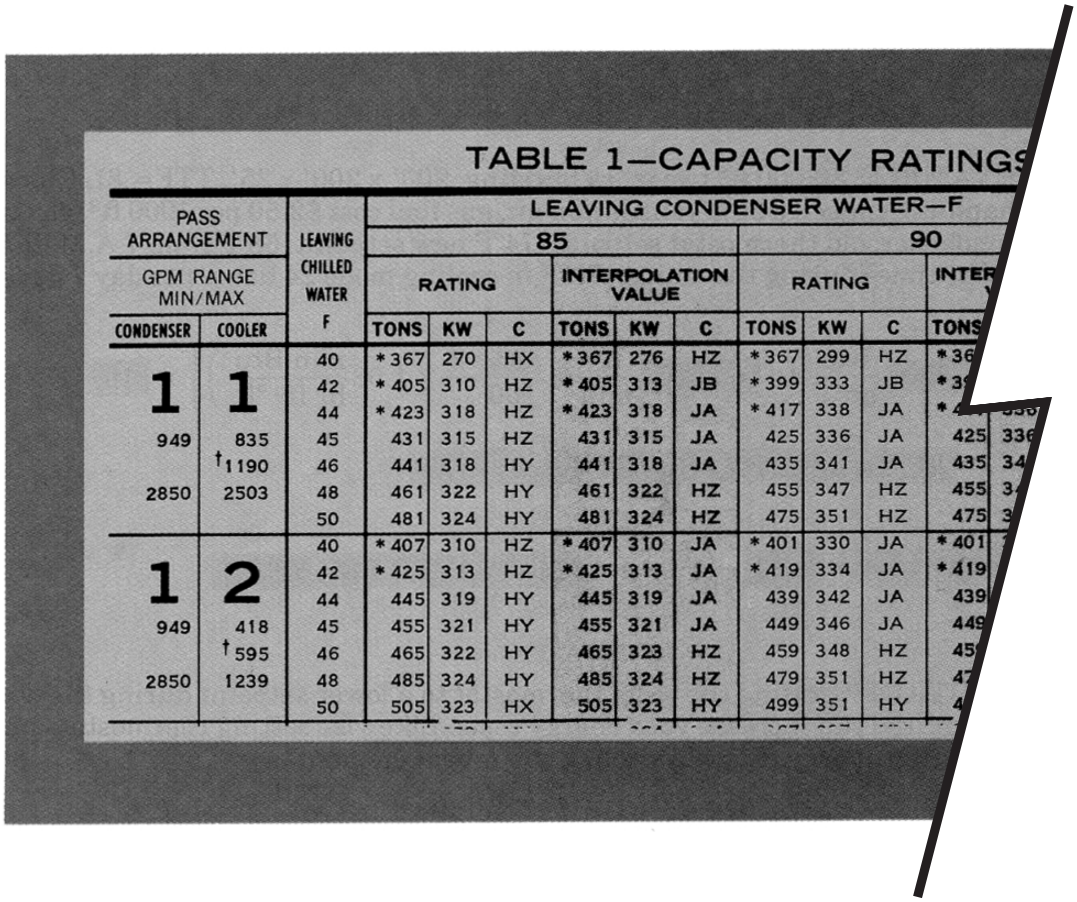


TABLE 1—CAPACITY RATINGS

PASS ARRANGEMENT		LEAVING CHILLED WATER F	LEAVING CONDENSER WATER—F										
GPM RANGE MIN/MAX			85						90				
			RATING			INTERPOLATION VALUE			RATING			INTER	
CONDENSER	COOLER		TONS	KW	C	TONS	KW	C	TONS	KW	C	TONS	
1	**1**	40	*367	270	HX	*367	276	HZ	*367	299	HZ	*36	
		42	*405	310	HZ	*405	313	JB	*399	333	JB	*3	
		44	*423	318	HZ	*423	318	JA	*417	338	JA	*4	
949	835	45	431	315	HZ	431	315	JA	425	336	JA	425	336
	†1190	46	441	318	HY	441	318	JA	435	341	JA	435	34
2850	2503	48	461	322	HY	461	322	HZ	455	347	HZ	455	34
		50	481	324	HY	481	324	HZ	475	351	HZ	475	3
1	**2**	40	*407	310	HZ	*407	310	JA	*401	330	JA	*401	
		42	*425	313	HZ	*425	313	JA	*419	334	JA	*419	
		44	445	319	HY	445	319	JA	439	342	JA	439	
949	418	45	455	321	HY	455	321	JA	449	346	JA	449	
	†595	46	465	322	HY	465	323	HZ	459	348	HZ	45	
2850	1239	48	485	324	HY	485	324	HZ	479	351	HZ	47	
		50	505	323	HX	505	323	HY	499	351	HY	4	

1—chilled water pump:
rated 680 GPM at 150 ft. head, 32.7 Kw

1—condenser water pump:
1062 GPM, 30 psig, 17.9 Kw

1—cooling tower fan: 12.3 hp, 10 Kw

Total power at full load 313 + 32.7
+ 17.9 4 + 10 = 373.6 Kw.

Using a Chiller Plant Audit to Estimate Seasonal Costs

The next step in a chiller plant analysis is to estimate the seasonal costs involved. This will reveal the total dollars you spend for chilled water, as well as the dollars per unit of cooling. Both of these figures are needed if you are to evaluate any capital improvements to your plant.

Assume a chiller plant having two - 300 ton centrifugal, electric drive chillers. What is the annual cost of running the plant if the cooling season is 25 weeks out of the year? What is the cost of a million BTU's of cooling produced? Use the audit form that follows for obtaining these answers.

The audit, (at the left) for this particular plant, shows that:

- cooling energy costs total $27,441 (icluding demand, tax, fuel adjmt)
- $8.33 per million BTUs (Btu x 106)
- chillers used 58 % of the annual energy ($15,920)
- pumps and cooling towers used 42% of the annual energy ($11,280)

The Kwh/ton hour "rule of thumb" would put the cost of cooling at $8.33/Btu x 10^6

1 Kwh/1 ton hour x $0.10/Kwh x 1 ton hour/12,000 Btu x 10° Btu/10° Btu =
$8.33 per million BTUs

This shows that it is important to consider run time and kilowatt hours used for all components of the central system in order to come up with "real" energy costs.

Obviously, a Kwh meter serving the central plant, plus Btu meters measuring cooling produced, would give you more accurate results, but this method is useful for chillers not equipped with such instrumentation.

Controls May Be The Answer

There are several basic ways to conserve chiller plant energy
1.Reduce compressor head,
2.Utilize chilled and condenser water reset,
3.Control electric demand imposed by the chiller plant,
Improve sequencing and local control of chillers, pumps, and cooling towers provide for variance in load.

Many of the following control strategies utilize control sequencing and speed control of chillers and fan and pump motors. Please refer to Honeywell publication 63-7062, "Variable Frequency Drive Application Guide" for a more thorough and detailed discussion on some recommended strategies and cautions.

COMPRESSOR HEAD REDUCTION

Compressor head reduction reduces the pumping effort required by the compressor. You can accomplish this either by raising chilled water temperature (leaving or entering), or by lowering condenser water temperature (entering or leaving).

Rule of Thumb:
Reducing condenser water in, or chilled water out, will save 1% to 2% compressor motor energy per 10 F change. More accurate results can be obtained from chiller catalog data.

CONTROL LOGIC FOR CHILLED WATER RESET

To determine optimum chilled water temperatures, follow these "fine-tuning" calibration steps:

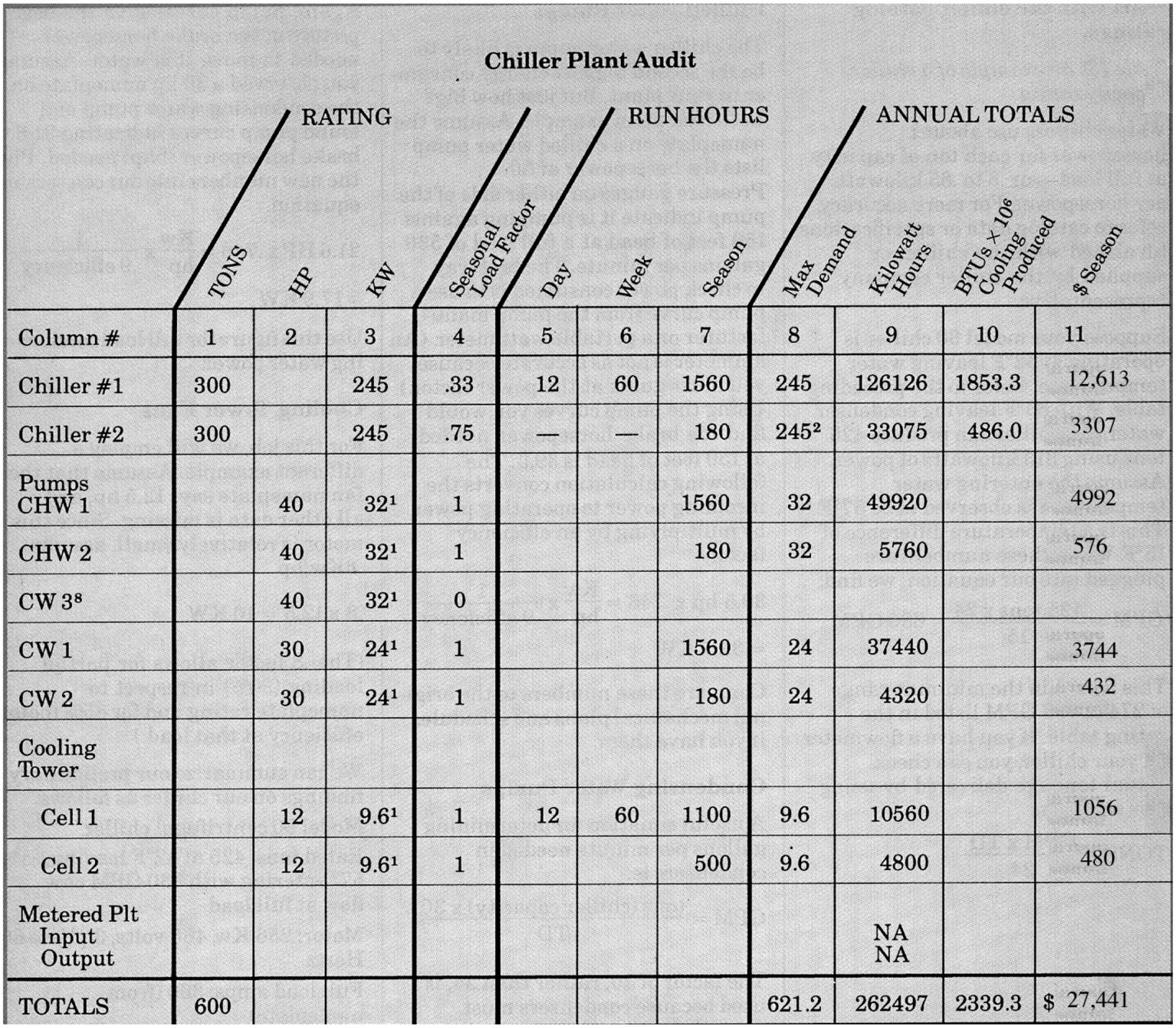
1)Raise "leaving" chilled water temperature one degree at a time and wait for system to settle out.

2)Observe positions of all valves supplying major (or sample) loads in each zone.

3)Continue raising chilled water temperature a degree at a time until one valve (heaviest load) is wide open.

4)If a second valve opens wide, start reducing chilled water temperature.

CAUTION: For multi-building plants having long chilled water mains and large pumps, there is an optimum point where the sum of chilled water pumping and compressor motor costs reach a minimum. Raising chilled water temperatures above this point may actually increase energy use.



Chiller Plant Audit

	RATING				RUN HOURS			ANNUAL TOTALS			
	TONS	HP	KW	Seasonal Load Factor	Day	Week	Season	Max Demand	Kilowatt Hours	BTUs × 10^6 Cooling Produced	$ Season
Column #	1	2	3	4	5	6	7	8	9	10	11
Chiller #1	300		245	.33	12	60	1560	245	126126	1853.3	12,613
Chiller #2	300		245	.75	—	—	180	245^2	33075	486.0	3307
Pumps CHW 1		40	32^1	1			1560	32	49920		4992
CHW 2		40	32^1	1			180	32	5760		576
CW 3^8		40	32^1	0							
CW 1		30	24^1	1			1560	24	37440		3744
CW 2		30	24^1	1			180	24	4320		432
Cooling Tower											
Cell 1		12	9.6^1	1	12	60	1100	9.6	10560		1056
Cell 2		12	9.6^1	1			500	9.6	4800		480
Metered Plt Input Output									NA NA		
TOTALS	600							621.2	262497	2339.3	$ 27,441

Average electric cost $.10/Kwh includes Kwh ☒ demand ☒ fuel adjustment & tax ☒

weeks in cooling season 26 cooling D.Ds/season 1302

NOTES: 1. Calculated at 8 Kw/hp (nameplate)

2. #2 chiller only used when outside temp is above 90°F
3. Column 3 x Column 4 x Column 7
4. Average electric cost is annual dollars divided by annual Kwh
5. Average load on chiller when running through entire cooling season
6. Column 9 x average electric cost of $. 10 /Kwh
7. Column 1 x Column 4 x Column 7 equals ton hours per season.
Ton hours x 112,000 equals Btu's per season
8. Spare pump

CONTROL LOGIC FOR CONDENSER WATER RESET

Set the local control operating tower by-pass valves and tower fans at lowest point permitted (by chiller manufacturer's recommendation) at full load, say 700 F. If data is available for part load, you may reduce chilled water temperature further.

CAUTION: if condensing water causes head pressure to drop too low, refrigerant won't flow into the evaporator and the compressor may surge.

CHILLER ELECTRIC DEMAND CONTROL

Since chillers represent a large increment of electrical usage for many buildings, it is important to control kilowatt "demand." Utilities often impose "ratchet" clauses that impose a penalty in additional demand charges as much as a year after a new Kw limit is reached. Kw monitoring and load shed priorities is a MUST if you have demand charges in your area. Most new chillers feature "soft start" capability. If yours doesn't, explore VFD speed control. VFD speed control for pumps/fans also provides this "soft start" capability to avoid extreme spikes in Kw demand. These large motors need to be "sequenced" on/off to avoid creating a new peak demand for the facility. Also, chillers and for that matter system pumps MUST be sequenced to avoid entire systems spikes, say all coming on one hour prior to building occupancy. There are two additional ways to consider limiting demand:

1)Shutdown selected, non-critical air handlers using chilled water when load shedding must occur.

2)Limit chiller kilowatts by limiting opening of suction inlet vanes (space demands on the chiller system).

The first method is preferred since it won't allow truly critical loads to be shut down.

The second method is recommended where there are no critical loads, and/or where it is desired to limit chiller loads when the plant is first turned on after shutdown.

CHILLER SEQUENCING AND LOCAL CONTROL

In a plant having multiple chillers, whether they are the same sizes or types or not, there is an optimum combination of chillers, pumps, and towers for each increment of plant.

Dealing with the Multiple Chiller Plant

Use this procedure to establish control and sequencing logic for a multiple chiller plant after you have established the optimum operating points for the chilled water leaving and condensed water entering.

STEP 1 Matrix Method

Assume three chillers, designated A, B, and C:
Chiller A 750 tons
Chiller B 750 tons
Chiller C 300 tons

Part load curves for each chiller indicate poor efficiency below 25% load and best efficiency between
50 and 90% load. The matrix lists possible combinations that will keep each chiller above 25% capacity.

STEP 2

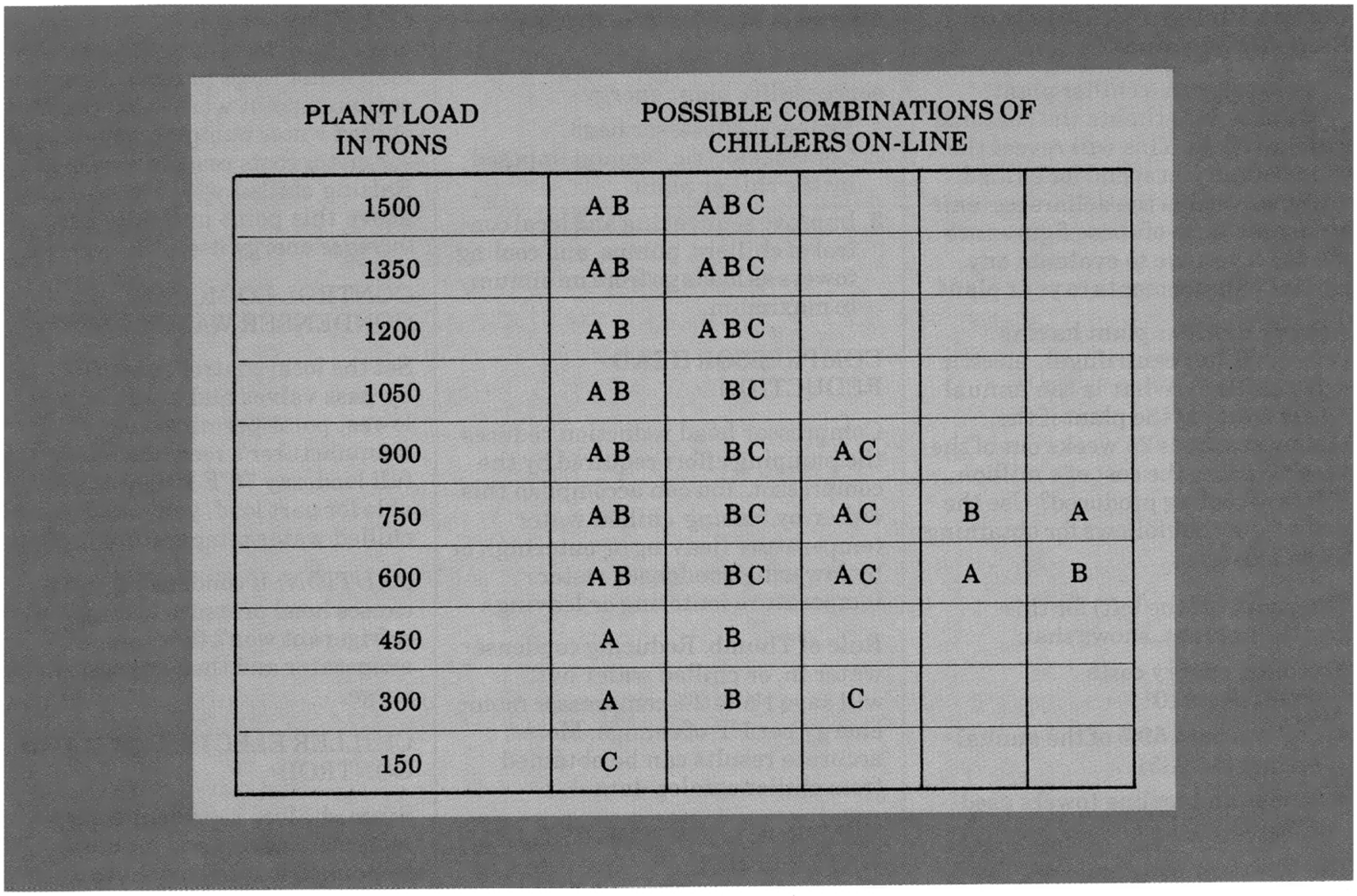
Have your chiller and pump catalog and specification data, including part load curves, within easy reach. Use this information to record the estimated kilowatt load including chillers, pumps, and towers in each square in the matrix. Bear in mind the constraints of the water circuits (for example, each chiller may have one pump which must run with it).

STEP 3

Find the square on each line of the matrix with lowest kilowatt total, then build a control and sequencing scheme based on these combinations. You can further refine this by assuming cooler condensing water and warmer chilled water temperatures as plant load varies directly as a function of outdoor conditions.

CAUTION: Many variables will affect total plant kilowatt load for each combination, and catalog data is not always valid or available at part loads. For this reason, you might wish to verify the combinations selected by taking kilowatt, or kilowatt-hour, readings on each major component while manually controlling the sequence. Again, measurement and verification is key to determining the "as is" current state of operations and the "future" improved state once measures are implemented. It is the only way to know if your goals have been achieved, and hopefully surpassed.

Once you have corrected the matrix and verified your entries by field measurements, automatic controls can be evaluated and installed for the sequence selected.



PLANT LOAD IN TONS	POSSIBLE COMBINATIONS OF CHILLERS ON-LINE				
1500	A B	A B C			
1350	A B	A B C			
1200	A B	A B C			
1050	A B	B C			
900	A B	B C	A C		
750	A B	B C	A C	B	A
600	A B	B C	A C	A	B
450	A	B			
300	A	B	C		
150	C				

Reducing Chiller Energy by Rehabilitation

This section is intended only to identify areas that can be explored to determine whether further energy savings are possible. A qualified consultant will be of great assistance in a more thorough evaluation before major capital expense is incurred.

Free Cooling Cycles

In cool weather, even in winter most buildings still require cooling (chilled water) in interior zones, or where cooling loads are unrelated to outside weather. If condenser water in the range of 35 to 500F is available, sprays and pumps can be added to chillers that allow "free cooling." Without running the compressor, under these conditions, pressure in the evaporator will be greater than the pressure in the condenser. Up to one-third rated capacity can be obtained in this manner with only a fraction of the normal cost. This is much like "free cooling" for a constant volume packaged rooftop unit when there is a "call" for cooling in an interior zone and the outdoor air is cool/dry enough to provide sufficient "cooling" without even running the compressor. These refinements can be the gems of any energy conservation program.

Cooling With Condenser Water

Under the same conditions as above, it is possible for you to use condenser water as the cooling medium, in lieu of the chilled water. Since condenser water is not clean, you will need special sophisticated filtering and water treatment, in addition to changeover valves and piping, to avoid contamination of the chilled water circuits. In some cases, a heat exchanger is an excellent strategy to avoid the filtering and treatment problem and expense.

Chilled Water Main Revisions

Chiller plants often employ a primary loop to maintain constant flow through the chillers that are on-line. A secondary loop draws water from the primary loop and circulates to the remote zones, floors, or other buildings. In some systems, the secondary loop also has constant flow, and each building entrance draws water from these mains to use within the buildings. Such systems need careful analysis, based on these general guidelines, to see if piping or pumping changes are needed.

1. Is primary loop water colder than secondary loop (supply side) water? If so, chillers may be using more power than needed.
2. Is primary loop water flow greater than the flow through the active chillers? If so, primary pump costs may be higher than necessary.
3. Is water in the mains supplying remote buildings, or mechanical equipment rooms, colder than water circulated within the building? If so, piping losses and chiller costs may be higher than necessary. Is flow in these mains substantially greater than the sum of the flows to each building served? If so, pumping costs may be too high.

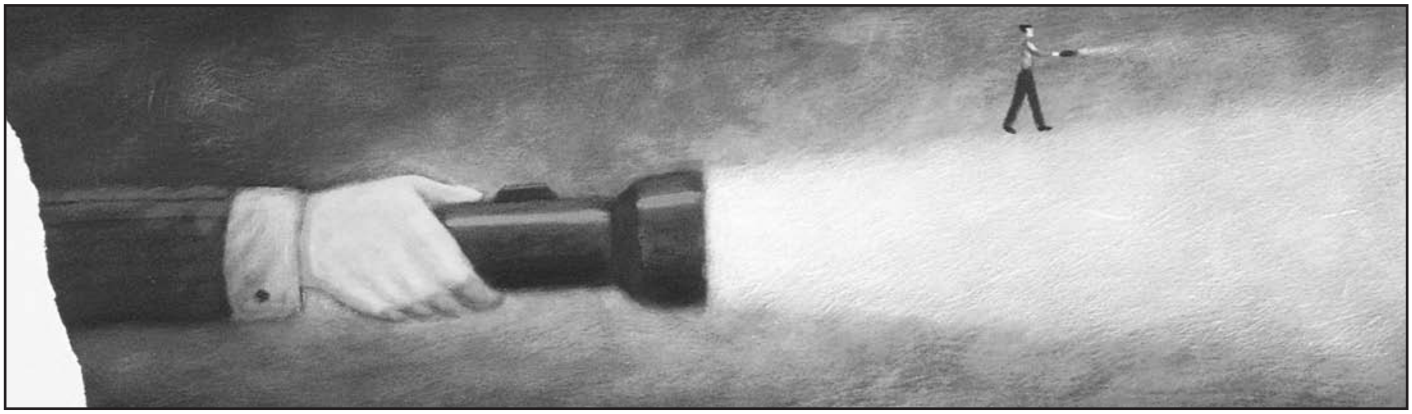
Spot Cooling

In large chiller installations, you may find small zones, floors, or buildings requiring "unusual hours" of cooling. This may necessitate running a 500 ton chiller to supply 10 tons of cooling! Do not continue this practice! Run the numbers and you will find it often is unjustifiable. Re-doing the chiller plant audit, only considering the 10 ton load situation, will reveal the dollar cost per million Btu's of chilled water, as well as the annualized cost for supplying the 10 ton load. It may be costing you a staff member of two! Don't forget labor casts if the chiller plant must be manned when operating.

Isolate the cost of cooling these areas. Now, compare this cost to installing a small "packaged" air conditioner, or small chiller to supply the after-hours load. (The air conditioner or chiller would usually be located at or near the after-hours load area.)

Don't Overlook "Up On The Rooftop"

Most of this section has presumed a large chiller plant for a facility like a large office complex, hospital or medical center, college or university campus. There are a growing number of "chillers" hidden up on building rooftops. Don't overlook evaluating these mechanical systems in addition to those at ground level. They offer most of the same opportunities for significant energy savings. And, in your equipment survey process if you find that many don't operate with chiller water systems, you may be able to implement many of the no or lower cost conservation measures even more quickly. No plumbing/pumping concerns may mean that you have only the ventilation fan equipment to address. Much of the rest may be optimizing, or "fine-tuning" the rooftops operation.



SHEDDING SOME LIGHT ON LIGHTING

Lighting is a major source of energy consumption in your building. Unfortunately, some of that consumption is wasted. If you don't include lighting as part of your plan, you will be missing some excellent conservation opportunities.

OFTEN MISUSED OR UNUSED

Even though many modern office buildings utilize windows for architectural aesthetics, we seldom see natural light used in task lighting. Lights have to be on during all occupied hours. Unfortunately, they often remain on after working hours even though only a few people occupy the building. And, there are often rooms or whole portions of buildings being lit without occupants.

In the past thirty years, lighting design standards have changed and our buildings reflect the trend toward increased lighting levels. About 20% of the total electrical consumption in the United States is used for lighting. In commercial and institutional buildings, lighting can account for 15% up to 60% of electrical energy consumption. However, lighting technology has changed so dramatically that the overall lighting electrical consumption can be reduced even with increased lighting levels.

FOUR PROBLEM AREAS

You can cut that percentage, but first you must identify the problems. Lighting can contribute to your energy waste in the following ways:

1. Lighting levels may be too high for their intended use.
2. Lights may be left on in a larger area when the space is not fully occupied.
3. Unnecessary heat from lights may add to the cooling load. Lights may contribute heat during the heating season, but much escapes without effecting the occupied space. This is particularly true in single story buildings. Extra heat from fossil fuels is a more cost efficient alternative.
4. Some light sources produce fewer luminaires (lumens per watt) than others. Incandescent lighting is the least efficient. Its inefficiency may be compounded by indirect lighting designs that further reduce usable light. Fluorescent lights are more efficient, though metal halide and high pressure sodium offer the highest luminaire levels.

For instance, compare 15 lumens per watt for incandescent lighting with 130 lumens per watt for high pressure sodium lighting. That's a factor of over 8 to 1.

LIGHTING LEVELS

Saving lighting energy is easy, just attack the four areas outlined above. Step one is to reduce lighting levels to a level consistent with the use of the area. Detail work and extensive reading require higher lighting levels than reception areas, for instance. There are three sets of guidelines to consider before establishing lighting levels in your building.

The General Services Administration (GSA) has mandated the following levels for government buildings.

- 50 foot-candles at occupied work stations,
- 30 foot-candles in work areas,
- less than 10 foot-candles in areas such as hallways and corridors

These recommended levels apply to office buildings, administration spaces, retail stores, schools and warehouses. Visually difficult tasks are allowed higher levels (75-100 foot-candles).

A complete discussion of these levels can be found in Lighting and Thermal Operations, Document 0-562-404, published by the Federal Energy Agency. Industrial tasks are covered in the American National Standards Institute publication ANSI-All.11973.

The Illuminating Engineers Society publishes the IES Lighting Handbook.1 This handbook has been used extensively by lighting designers and recommends levels of lighting based on “worst case” situations. For example, setting office lighting, at a minimum level of 100 foot-candles to do visually difficult tasks with minimum errors.

A third source for you to consider is ASHRAE standards for space lighting. They deal with lighting as a part of your “energy budget” for new buildings. ASHRAE refers to the IES Lighting Handbook to determine maximum budgets for lighting, but encourages task lighting with considerably lower levels where no visual tasks occur.

Lastly, consider information available from The Association of Energy Engineers or AEE. They offer a substantial number of useful publications on many aspects of energy use, HVAC retrofits and quite a number on lighting. Visit their website at: www.aeecenter.org.

If you are designing or modifying lighting systems in your buildings, we advise using all three sources to aid you in your plans. These sources will give you the most comprehensive view of lighting as it impacts energy conservation. (From a more practical standpoint, general area lighting in the range of 50-70 foot candles at work surface is widely accepted and practiced by energy conserving building owners.

1 Available from Illuminating Engineering Society, 345 E. 47th St., New York, NY 10017

2 Available from American Society of Heating, Refrigerating, and Air Conditioning Engineers, Inc., 345 E 47th St., New York, NY 10017

LIGHTING CONTROL

Once you have your lighting levels established, you can move onto lighting control. Though lighting control can take various forms, the goal remains the same – lights on only when and where necessary. The simplest solution is manual control, but this option requires the assignment of definite responsibilities and some form of monitoring. Otherwise, it fails.

A second control scheme involves automatic switching with local overrides for people working late or in areas of a facility with intermittent use.

A third variation provides different levels of lighting for different tasks carried on in the same space. For example, during the day you might provide 70 foot-candles for office work. In the same space, after working hours, 30 foot-candles would be adequate for janitorial tasks.

A fourth method involves modulating the amount of light provided to an area. This can be accomplished either manually or automatically.

Lighting control is similar to that of a thermostat. A thermostat is set to control a space at a fixed temperature by modulating the HVAC equipment output to maintain that temperature. Similarly, light sensors provide a signal to a power controller, which, in turn, modulates the amount of power delivered to the ballast. This special ballast controls the light at some fixed level. As light output is reduced, power to the ballast and fixture is also reduced proportionately.

When modulating control of lighting is combined with a computerized building automation system, light levels can be adjusted downward by the system to reduce peak power demand in the building. In comparison to conventional on/off control of lighting, these systems provide the following benefits:

- Easy adjustment of light level to task requirements
- Automatic compensation for availability of perimeter area sunlight
- More efficient ballasts
- Extended useful lamp light
- Ability to incorporate lighting control in peak demand load shedding programs

REDUCING THE COOLING LOAD

Your third step toward lighting efficiency involves reducing the unnecessary heat from lights that adds to your cooling load. First you must determine if it causes an unnecessary or unmanageable cooling load. Ask yourself these questions:

- Is the excess heat useful and economical during the heating season?
- Is there a way to get reduce it during the cooling season?

If the answer to the first question is "no", then you should begin looking for ways to reduce that excess heat. If the answer is "yes", you should evaluate the relative worth of heat from lighting.

Let's look at an example. First, how much heat is involved? A 1000 ft2 office area in a one-story building uses 2.8 watts of fluorescent lighting (ballast included) per square foot. This produces 60-80 foot-candles at desktop.

$$\frac{1000 \text{ x } 2.8}{1000} = 2.8 \text{ Kw}$$

These lights will produce

$$\frac{2.8 \text{ kW x } 3415 \text{ BTUs}}{\text{kWh}} =$$

9554 BTUs/hour of heating energy.

This translates to 9554 Btu/hr of heating energy. Generally, producing the same amount of heat (Btu) from electricity will cost more than producing it from another source such as natural gas. In addition, a large portion of the lighting heat will be lost through the ceiling.

Is that heat really worth the cost? You can't answer that until you consider the cost of the same heat during the cooling season. Suddenly, that heat is not contributing to heating your building, but rather compounding your cooling load.

During the cooling season, your lights are still producing 9554 BTU/hour of energy. With the loss through false ceilings taken into consideration, it is realistic to assume that only half of this heat is being added to your cooling load. In a multi-story building less heat would escape (depending on whether the ceiling is used as a return plenum). The design load for this 1000 ft2 example would be about 30,000 BTU/hour of cooling load. And it stays constant, regardless of the weather and occupancy, as long as the lights are on.

The solution? That same false ceiling can work to your advantage. Some HVAC configurations allow heat from lights to be retained during the heating season and exhausted during the cooling season by means of false ceiling return plenums. Another alternative is a special luminaire having a return air duct connection.

SELECTING EFFICIENT LIGHT SOURCES

The fourth problem area requires additional consideration. To determine what type of light source to use in a particular space, you must weigh the space function and number of hours of use. The goal is to provide the reduced levels of lighting mentioned in step one by using the most efficient fixture. A more efficient fixture gives you more lumens per watt.

For example, school gymnasiums requiring many hours of use each week may be excellent candidates for replacing incandescent fixtures with high pressure sodium lighting. On the other hand, office areas or classrooms in the same building may be more efficiently lit with fluorescent tubes.

Fluorescent light has changed dramatically in the last decade. During the 1980s most existing fluorescent lights were a larger "T12" type tube. Much more efficient "T8" tubes have been a standard element of lighting retrofit programs during the 1990s. At about _ the diameter of the older lights, T8s provide greater illumination at about 20% lower operating consumption.

Make sure you and your electrical consultant check to see what rebates might be available for lighting and other electrical retrofit projects from the electrical provider. Also, check out the many sources of lighting products on the web, such as "gelighting.com," for more detailed information on choices and savings estimation tools.

Use the following lighting ECM calculations to determine how you might conserve energy by reducing hours lights are on and reducing power supplied for lighting.

ECM CALCULATIONS

$$\textbf{yearly electrical savings} = \frac{\textbf{hrs.}}{\textbf{wk}} \ \textbf{x} \ \frac{\textbf{wks.}}{\textbf{yr}} \ \textbf{x} = \textbf{\$ SAVED PER YEAR}$$

EXAMPLE An 80,000 square-foot two-story building is equipped with 280 kW of fluorescent lighting. The lighting levels in all areas range from 100 to 130 footcandles. Lights are turned on and off at lighting panels by the security force: on at 7 a.m., off at 11 p.m. when janitors leave, five days a week. How much can be saved by reducing lighting to a tota of 224 kW and operating all lights from 8 a.m. to 5 p.m., with one-third of lights on from 5 p.m. to 11 p.m. for custodial work?

A. Before reducing light levels, the annual cost would be:

$$80 \frac{\text{hours}}{\text{week}} \text{ x } 51 \frac{\text{weeks}}{\text{year}} \text{ x } 280 \text{ kW x } \frac{\$.10}{\text{kWh}} = \frac{\$114{,}240}{\text{Year}}$$

B. After reducing levels and "on" hours, annual lighting cost would be:

Full lighting, 9 hours a day, 224 kW:

$$45 \frac{\text{hours}}{\text{week}} \text{ x } 51 \frac{\text{weeks}}{\text{year}} \text{ x } 224 \text{ kW x } \frac{\$.10}{\text{kWh}} = \$51{,}408$$

One-third lighting, 5 pm to 11 pm:

$$30 \frac{\text{hours}}{\text{week}} \text{ x } 51 \frac{\text{weeks}}{\text{year}} \text{ x } {}^{1}/_{3} \text{ x } 224 \text{ kW x } \frac{\$.10}{\text{kWh}} = \$11{,}413$$

Total Cost: $51,408 + $11,413 = $62,821 PER YEAR

Savings: $114,240 – $62,821 = $51,419 (45%) Saved PER YEAR

C. The cooling load saved by reducing lighting levels and hours of operation may be calculated, but requires analysis of what fraction of the heat actually enters the space. For this reason, it is usually omitted from lighting savings calculations.

D. Heat supplied by lights can be credited as heating energy saved only to the extent that is is produced during fully occupied periods. No savings should be credited for lighting levels that are in excess of that need for the task performed in the space.

SPECIAL DATA NECESSARY: For fluorescent lighting, add the watt rating of the tubes, then add 20% for the ballast.

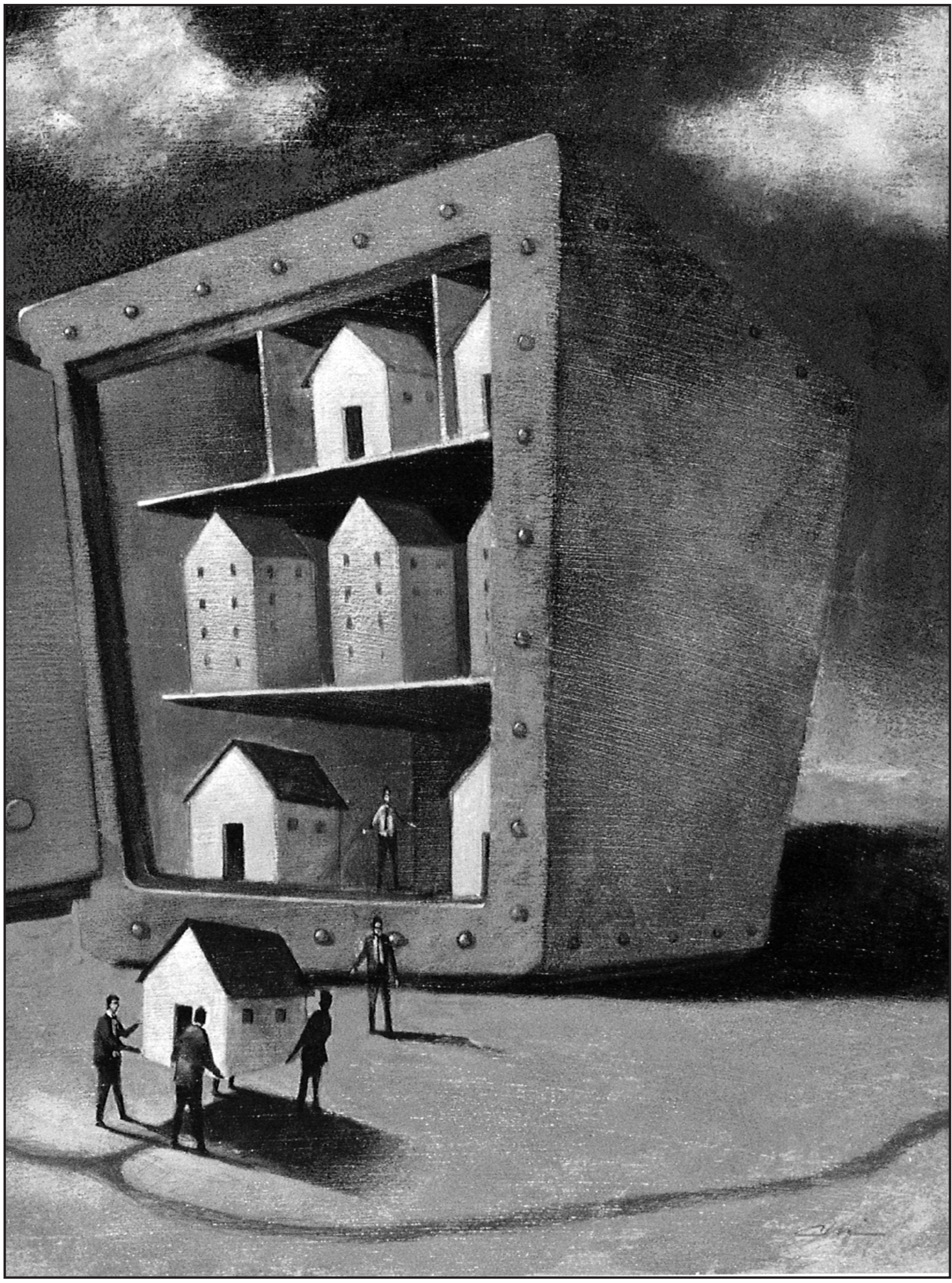
Example: A fixture has three 40-watt tubes. Tube wattages is 120 watts.

Load: 120 x 1.20 = 144 watts

Heat added by lighting is 3,412 BTUs per watt.

$$\text{Example: 144 watts x } 3{,}412 \frac{\text{BTU}}{\text{watt hour}} = 491{,}328 \frac{\text{BTU}}{\text{hour}}$$

CAUTION: When automatic lighting control is applied, make sure that enough night lights remain on for safety after the main lighting is turned off.



INSULATION: KEEPING ENERGY INSIDE THE ENVELOPE

In this section, we will offer some guidelines on how to approach energy savings through an analysis of the building structure. This represents one of the more difficult challenges you face as an energy manager. Not only is it difficult to measure energy gains and losses, it is also hard to determine whether the calculated return on investment is sufficient to justify proceeding with a project to lower heat losses or gains. A more in-depth treatment of the subject may be obtained from your insulation supplier or your consulting engineer.

The roof and walls, windows, and doors are the most obvious places to look for energy losses in a commercial building structure. But they represent only three of six critical areas. You should be aware of all six in order to minimize structural losses. They are:

INFILTRATION:* Air leaking in through doors, window frames, cracks, loading dock doors, etc.

POOR INSULATION: Heat leaking in or out through walls and roofs.

SINGLE GLAZING: Allowing high transmission loss in winter/solar gain plus transmission in summer.

LACK OF SHADING: Increasing solar loads in summer.

HVAC EQUIPMENT: Losses through piping, duct work stacks, roof-top units, etc.

PROCESS EQUIPMENT**: Allowing losses through poor insulation or wasting of exhausted fluids.

*Air leaking in through poorly aligned dampers or excessive ventilation is not treated here.

** In this context, process is intended to mean anything not directly related to providing an environment for people. A hotel laundry and data processing computers in an office building would be examples of process equipment.

PUTTING STRUCTURAL LOSSES INTO PERSPECTIVE

Insulation often comes to mind as the place to begin energy conservation efforts. In a residential building this might be true. Heat leaks in and out through walls, windows and ceilings-period. Internal gains and losses are of minor proportion and importance.

In a commercial structure, on the other hand, insulation is only one culprit. Table 1 is a heat/gain analysis of an actual building in Roanoke, Virginia. It shows a 48% heat loss and 22% heat gain from structural sources. Structure played a part, but lights, ventilation and people accounted for the majority of heat loss and gain.

CAUTION: Table 1 is not representative of all buildings. It is an example used to show that the building structure may not always be a major cause of high energy costs. Each building is unique; include structural considerations in the energy survey of your building.

TABLE 1 – BASIC COMPONENTS CONTRIBUTING TO THE HEAT GAIN AND HEAT LOSS OF A STRUCTURE.

COMPONENT	PERCENT OF TOTAL HEAT GAIN	PERCENT OF TOTAL HEAT LOSS
Conduction and solar effect:		
• Walls	0.2	48
• Windows	19.0	
• Floor and roof	3.1	
Lights		
Ventilation air:		
• Sensible heat	13.7	52
• Latent heat	24.0	
People:		
• Sensible heat	9.0	
• Latent heat	6.0	

*Reprinted by permission from HEATING/PIPING/AIR CONDITIONING, SEPTEMBER, 1995

Design considerations: Heat loss 70°F indoor/outdoor temperature difference. Heat gain: 80°F, 50% relative humidity indoor; and 95°F, 48% relative humidity outdoor. Lights and Miscellaneous: 214.3 Kw, or 2.22 watts/ft2. Floor area: 87,552 ft2; six stories. Volume 1,150,000 ft3. Dimensions: 152 x 96 ft. Construction: 12 in. brick wall; no thermal wall insulation; no exterior shading devices; windows are double glazed, and cover 40% of exterior; built in Roanoke, VA in 1948.

ECONOMICS OF STRUCTURAL ENERGY IMPROVEMENT

Adding insulation and glazing to existing buildings is not often economical in terms of payback period analysis. Ideal payback occurs in 2-3 years; it often takes much longer to realize savings from structural improvement. Therefore, changes are usually made during remodeling, repair, or renovation.

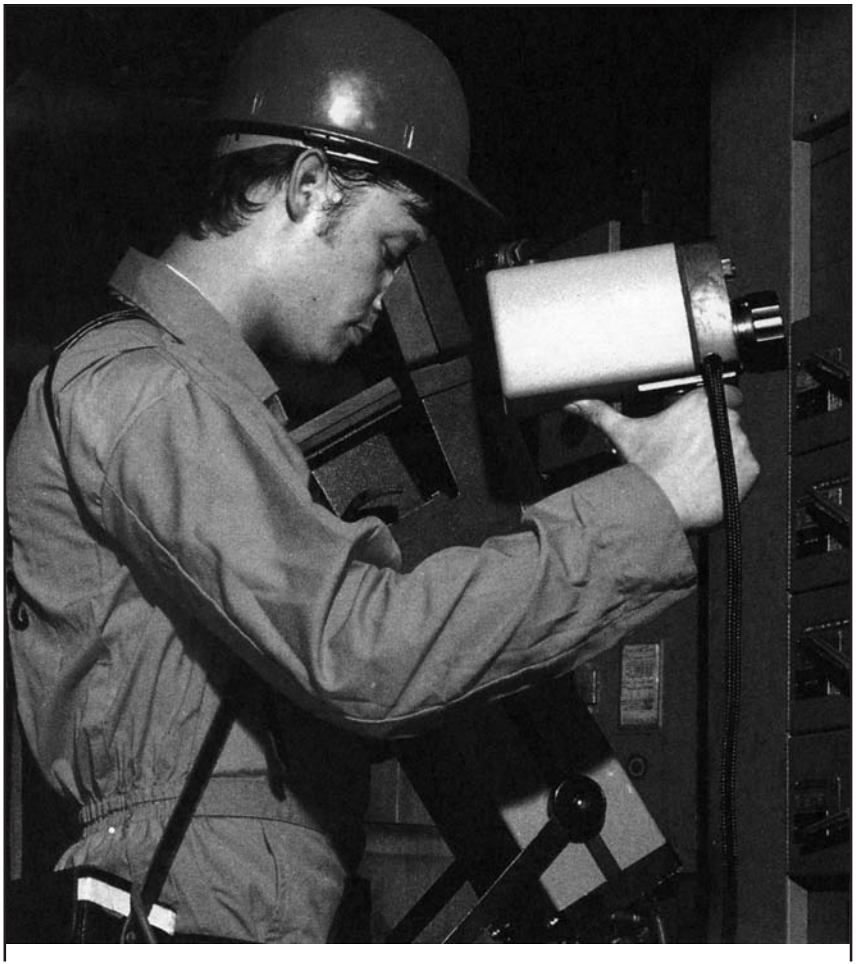
For example, roofing insulation is added when a new roof is needed. There is one exception. Weather stripping, shading, and closing up leaks can often be done at modest cost and with good payback and immediate results which can limit other potential damage.

Detailed economics for any of these procedures is complex and beyond the scope of this section. However, the guidelines that follow may provide a "first look" at your building structure's six critical areas.

GUIDELINES FOR REDUCING STRUCTURE LOSSES

INFILTRATION

This is the easiest loss to detect - if you are willing to wait for a cold day. Feel around doors and windows and other building openings for drafts and leaks. For a more sophisticated approach, an infrared thermograph can be used to detect leaks around building openings. More sophisticated methods are also available for detecting infiltration in terms of air changes per hour. One to consider would be outdoor air-intake volume analysis and CO2 measurement to determine the rate of ventilation which would help estimate the total leakage. Finding the locations of greatest leakage would still need to be done.



Scanning to detect heat loss from HVAC equipment. In actual thermograms the lighter or white areas show areas of energy loss. (Photos courtesy of Energy Conservation Consultants, Inc., Bloomington, Minn.)

POOR INSULATION

Poor insulation can be detected by means of infrared scanning. Some scanning is done with aerial photography, but had held scanners can also detect hot spots. Scans (or thermographs) are particularly useful inside buildings, where they pick up cold spots on walls, windows or ceilings and pinpoint wet or missing insulation. (As a fringe benefit, infrared can spot heat from poor connections or overloaded electrical conductors, too.)

SINGLE GLAZING

Single glazing, as a rule of thumb, has a "U" factor of about 1.0** (1 BTU/ft2/ºF). Double glazing can reduce this to about 0.6**. As a rule, double glazing is installed during major remodeling, repair, or renovation for maximum economy.

** See ASHRAE Handbook of Fundamentals for Accurate Data

SHADING

Awnings, window coatings, reflective glass, plantings, or structural shading devices can all reduce solar gain. Outside expertise should be solicited for an analysis of the benefits and economics of shading.

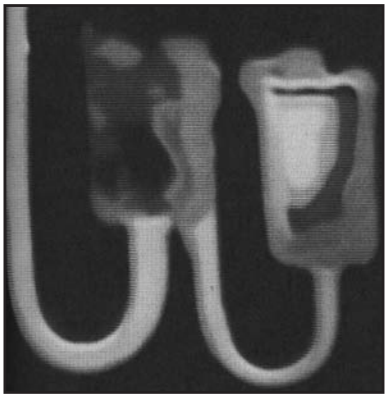
Buildings having a large (over 25%) per cent of glass area should be analyzed with two questions in mind:

1. How much heat loss and gain occurs through windows each year?
2. What are the "ball park" costs involved in shading or glazing improvements and what reduction in heat loss/gain would result?

The answers will help you determine whether fenestration improvements can be considered.

HVAC EQUIPMENT LOSSES

HVAC equipment losses can usually be detected by inspection. You can insulate pipes carrying hot or cold fluids or improve existing insulation. TIMA (Thermal Insulation Manufacturers Association, 7 Kirby Place, Mt. Kisco, NY 10549) in the past had a program to analyze the cost-effectiveness of insulating piping. Table 2 is an example of the information the TIMA program can process. Contact them directly to see what they currently offer.



Additonal Insulation (Inches)	Heat Loss (BTU/HR)	Heat Cost ($/YR)	Insulation Cost ($/YR)	Total Cost ($/YR)
0.0	365,305	$7,271	0	$7,271
1.0	280,582	$5,584	918	$6,503
2.0	233,792	$4,653	1,684	$6,337
3.0	203,918	$4,059	2,450	$6,508

TABLE 2
1,000 ft. 6" pipe with 2" of existing calcium silicate.

Capital investment: $20/lb steam/hr. Remaining depreciation period: 10 yr. Heat cost: $1,76 /M lb steam. Inflation rate: 7%/yr. Base insulation cost: $4.85/lin ft for 2 1/2 x 1 1/2. Cost of money: 10%/yr. Surface temperature: 75ºF.

Economic thickness of additional insulation: 2 inches.

Energy savings over the depreciation period - 11,520.5 million BTU 115,205 therms, natural gas; 76,804 gallons, No 6 fuel oil.

Economic thickness of additional insulation: 2 inches

Energy savings over the depreciation period - 11,520.5 million BTU; 115,205 therms, natural gas; 76,804 gallons, No. 6 fuel oil

Table 2* shows how adding economic thickness of insulation reduces total owning and operating to the lowest annual amount. In this example, the addition of 2 in. of calcium silicate to a line already insulated with 2 in. of the same insulation would bring total annual cost down to $6337 for the indicated operating and cost factors. Both lesser and higher levels of insulation would increase overall cost.

*Reprinted by permission from ASHRAE Journal October 1975

Note: Above example is based on a 1975 heat cost @ $1.76/1000 lbs. steam. Costs have risen both for heat ($5 - $10/1000 lb. steam and insulation. Consult your utility and insulation supplier to determine current savings/payback.

PROCESS LOSSES

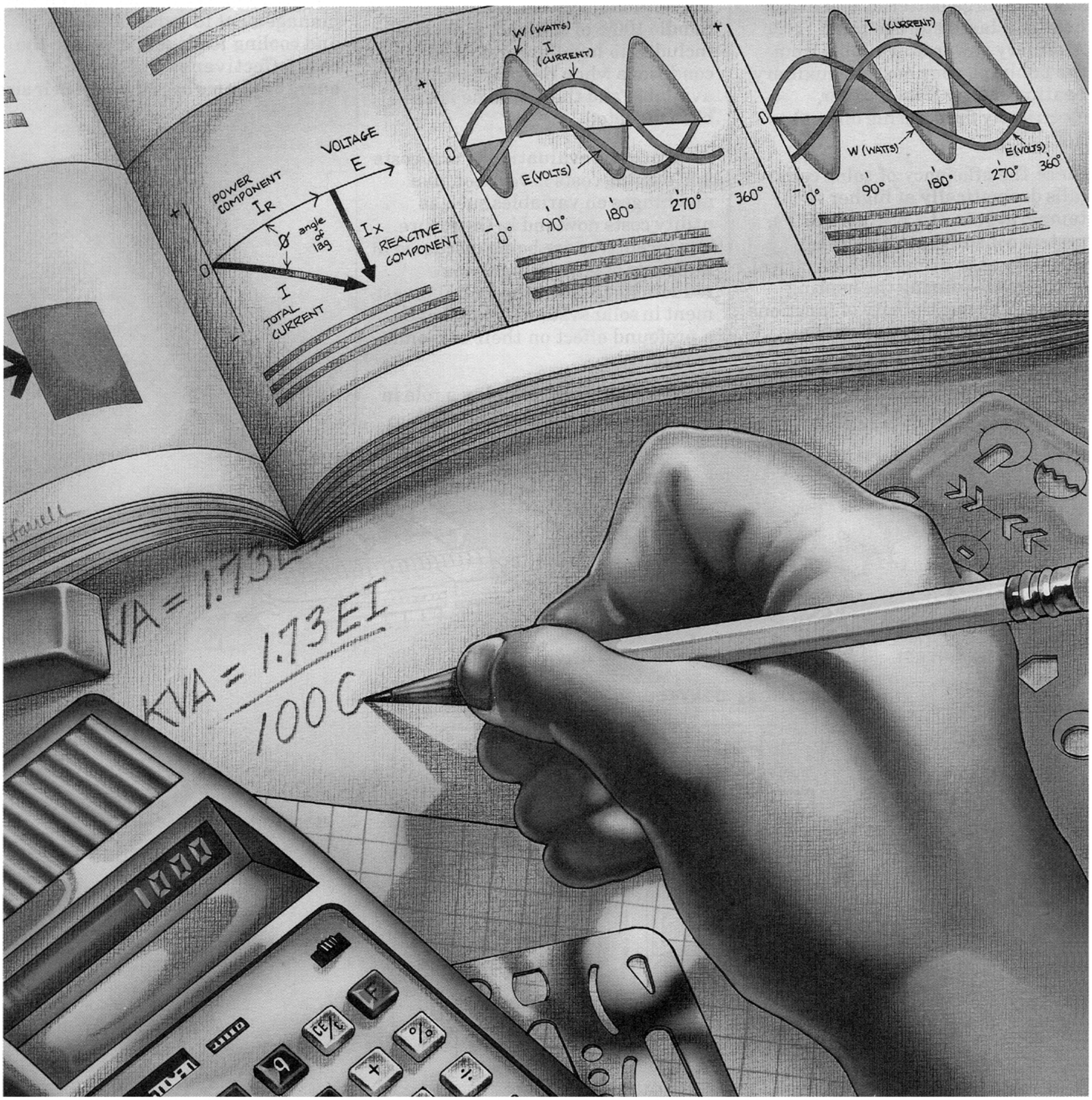
Processes that add to the heating and/or cooling load of structure should be given careful attention (e.g. exhaust hoods or paint spray booths).

Process equipment should be considered in an energy survey of your building. If the equipment is not separately metered, consumption and costs should be estimated.

You can cut down process losses by insulation and/or heat recovery. Identify fluids (air, water, other) leaving the premises carrying higher or lower temperatures than fluids entering. Note spaces that are hot or cold because of process equipment operation.

Consider all six of the critical areas when you evaluate the conservation opportunities offered in the building structure. Weigh the potential savings and prioritize the possibilities in terms of first costs and payback. Though you may find that there are other areas that will give you a faster return on investment, insulation should not be overlooked. If you are anticipating renovation or remodeling, insulation may give you an excellent opportunity to save energy at reasonable cost.

POWER FACTOR—
ITS SIGNIFICANCE FOR BUILDING OWNERS



WHAT IS "POWER FACTOR"?

Power Factor is the relationship (phase) of current and voltage in AC electrical distribution systems. Under ideal conditions, current and voltage are "in phase" and the power factor is "1.0". If inductive loads (motors) are present, power factors *less than 1.0* (typically .80 to .90) can occur. A technical explanation of power factor is included later in this section.

EFFECTS OF LOW POWER FACTOR ON BUILDING EFFICIENCY

Low power factor, electrically speaking, causes *heavier current* to flow in power distribution lines in order to deliver a given number of kilowatts to an electrical load.

Example: A building using 2000 Kw at a given instant, at unity power factor (1.0), would require 278 amps in a 3 phase 4160 volt, 60 hertz feeder to that building. If the power factor were poor, say .85, 327 amps would be needed to supply the same 2000 Kw load.

The effects?

- Power distribution system in the building, or between buildings, can be overloaded by excess (useless) current.
- Electrical costs can be increased if electric utility charges a penalty for low power factor.

EFFECTS OF LOW POWER FACTOR ON ELECTRIC UTILITY

Generating and power distribution systems owned by an electric utility have their capacity measured in KVA (kilovolt amps).

$$\text{KVA} = \frac{\text{volts x amps x }\sqrt{3}}{1000} \quad \text{(three phase system)}$$

In the foregoing example, with unity power factor (1.0), it would take 2000 KVA of generating and distribution network capacity to deliver 2000 Kw.

$$\frac{\text{4160 volts x 277.58 amps x }\sqrt{3}}{1000}$$

=2000KVA

If the power factor dropped to .85, however, 2353 KVA of capacity would be needed.

$$\frac{\text{4160 volts x 326.56 amps x }\sqrt{3}}{1000}$$

=2353 KVA

Thus we see that low power factor has an adverse effect on generating and distribution capacity.

Furthermore, ordinary electric meters as found in residential and small commercial buildings, *do not register power factor* or KVA. The net result on the electric utility?

- Low power factor *overloads* generating and distribution networks with excess KVA.
- Electric utilities, in all but larger users, *cannot measure or be compensated* for low power factor.
- Electric utilities almost always require large users to either keep their power factor above .9 (or .95) or pay a penalty, usually in the form of extra demand charges.

WHY SHOULD YOU CONSIDER POWER FACTOR CORRECTION?

If you own a large building, you should consider correcting poor power factor for either or both of these reasons:

- To reduce power factor "penalty" charges from the electric utility.
- To restore the (KVA) capacity of overloaded feeders within the building or building complex.

HOW DO YOU CORRECT POWER FACTOR?

There are several methods of correcting low power factor. Commonly used are:

1. *Capacitor banks*—You may install these on the primary or secondary side of the substation (transformers) serving a building. (Primary side is preferred, since the higher the voltage, the smaller the capacitor bank needed.)

   Inductive loads tend to cause a lagging power factor and capacitors tend to cause a leading power factor. Therefore, a properly sized capacitor bank can restore a lagging power factor to near unity (.95 to 1.0 is considered satisfactory).
2. *Switched Capacitors*—Plants equipped with very large, intermittent inductive loads, such as large motors, compressors, etc., may require switched capacitors; that is, capacitors are connected to individual motors or groups of motors. Therefore, they are only in action when the motor load is turned on. Or, capacity may be switched on and off at the substation, depending on measured power factor. The switching feature is only required if the capacitors needed are so large that they cause an undesirable *leading* power factor during times when large motors are turned off.
3. *Synchronous Motors*—Synchronous motors are sometimes used to correct power factor. Large motors of this type have the same electrical effect on the distribution network as capacitors. They are seldom applied outside of heavy industrial applications.
4. *Correction by Electric Utility*—Utilities often install capacitors on their own lines where it is not practical to measure power factor, or to assess charges to owners. It is common to see capacitor banks installed on utility poles in residential neighborhoods.

POWER FACTOR—HOW IT IS MEASURED

There is no instrument readily available to measure power factor directly. The usual way is to measure two sides of the vector diagram shown in Figure 1. In a right triangle, two sides will allow computing the third side, as well as the angle of lag, the cosine of which equals power factor.

If a KVARH meter is hooked into the power line, it is capable of measuring the reactive component only (KVAR) of power fed to the building. (See figure 2.)

From the two meter readings, cosine ϕ can be computed and the power factor obtained.

Except for large plants, where the electric utility installs KVARH meters, it is not economical for you to meter or monitor power factor. Rather, you should consult the electric utility annually, and ask what extra costs, if any, are being incurred because of low power factor.

POWER FACTOR—A TECHNICAL EXPLANATION

A KILOWATT is the usual measure of power and equals 1000 watts. It equals 1.34 horsepower, thus 746 watts equals 1 horsepower. A kilowatt-hour (KWH) is the usual measure of energy consumption and is equivalent to using 1000 watts average energy for a period of one hour.

APPARENT POWER *(VA) (KVA)*— If the current and voltage are not in phase, that is, do not reach corresponding values at the same instant, the resultant product of current and voltage is *apparent power* instead of actual power. Apparent power is measured in volt-amperes (VA) or kilovolt amperes (KVA = VA/1000).

Apparent power is measured as follows:

On single phase service VA = EI

$$KVA = \frac{EI}{1000} \text{ (Formula 9)}$$

On three phase service VA = 1.73 EI

$$KVA = \frac{1.73EI}{1000} \text{ (Formula 10)}$$

Actual power (KW) is the product of KVA and the power factor (expressed as a decimal).

Thus, KW = KVA · PF (Formula 11)

REACTIVE KVA *(RKVA) (KVAR)*— In any alternating current circuit it is frequently desirable to consider the current as being composed of *two components,* one in phase with the voltage and one lagging (or leading) by 90 degrees. This would be represented in polar coordinates as per Figure 3. In this case E is assumed to be at zero degrees. Current I lags by angle ϕ (30 degrees). The in-phase, or power, component is I_R which is equal to $I \cdot \text{cosine}\, \phi$.

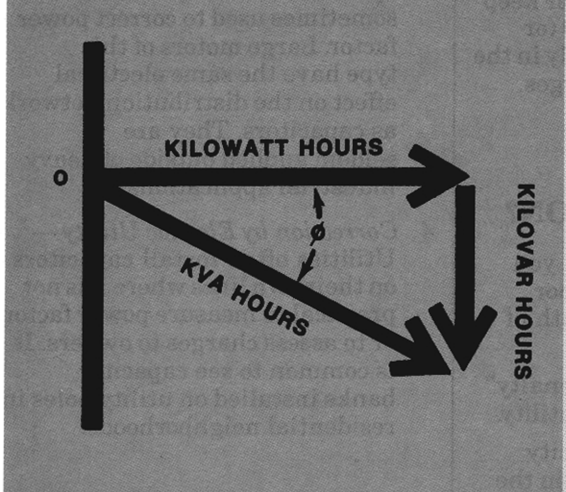
The out-of-phase, or magnetizing, component I_x, is called the *reactive* component. In the case of induction motors, transformers and other magnetic devices the magetizing component serves the important function of *magnetizing* the equipment during one quarter cycle. The energy thus transmitted is returned during the next quarter cycle, as per Figure 3, when the device demagnetizes. The Power Factor is always the ratio of the in-phase component to total current and equals $I_R \div I$ or cosine ϕ. The out-of-phase component $I_x = I$ sine ϕ.

The ratio of out-of-phase to in-phase component of current on three phase service is:

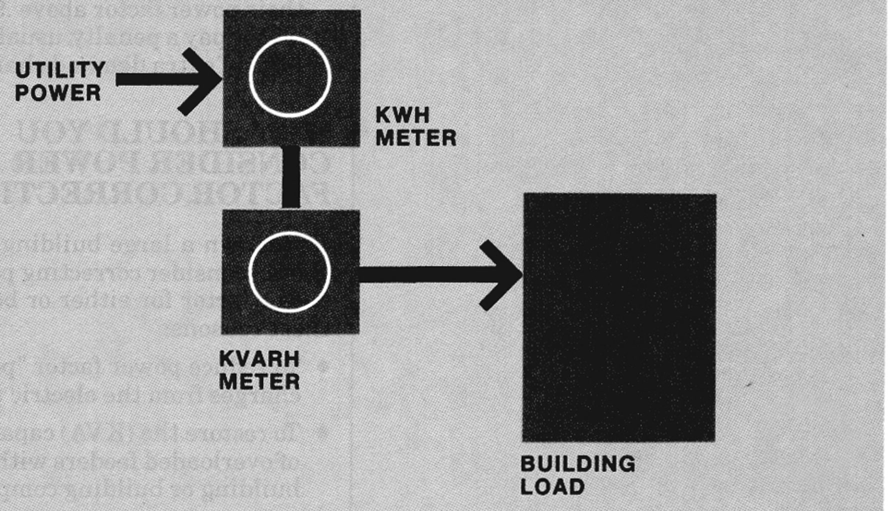
$$\frac{1.73\,E \cdot I_x}{1000} = RKVA \text{ (Formula 12)}$$

$$\frac{RKVA}{KW} = \text{tangent}\ \phi \text{ (Formula 13)}$$

POWER FACTOR *(pf)* is the factor by which apparent power is multiplied to obtain actual power (watts).



(Figure 1)



(Figure 2)

In most magnetic circuits as mentioned above, the current will lag behind the voltage. A typical case represented in Figure 4 where the current changes lag 60 degrees (1/6 cycle) behind corresponding voltage changes. In that part of the current wave and voltage wave when both are positive, or both negative, the resulting power is *positive* (E · I = W, or − E · − I = W). This is represented by the white area above the zero line.

When either the current wave or the voltage wave is negative and the other is positive, the resulting power is *negative* (− E · I =− W, or E · − I = − W). This is represented by the white area below the zero line. The *net power* is the positive area minus the negative area. The *apparent power* (volt amperes) is the sum of the two areas. The *power factor* is the net power divided by the apparent power. *Thus power factor is the factor by which we multiply apparent power to determine actual power.*

Mathematically, power factor is equal to the cosine of the angle by which the current lags (or in rare cases leads) the voltage. In Figure 4 one angle of lag is 60 degrees. The cosine of this angle and thus the corresponding power factor, for that amount of lag is 0.50 corresponding to 0.5 (or 50%) lagging power factor.

LOW POWER FACTOR IS UNDESIRABLE. The capacity of generators, transformers, transmission lines and distribution lines is usually fixed by heating limits or by permissible voltage drop. In any case the current is usually the limiting factor. Since power factor equals KW/KVA, any decrease in power factor means an increase in KVA (and therefore amperes) for a given KW load.

MAGNETIZING CURRENT— Transformers, motors and other electro-magnetic devices containing iron in the magnetic circuit must be magnetized in order to operate. It is customary to speak of the lagging inductive component, referred to above, as a *magnetizing current.*

The load component registers on the watthour meter and does the actual work. The magnetizing component puts energy into the magnetic circuits of the apparatus during one-fourth cycle and returns it to the system the next quarter cycle. This is illustrated in Figure 5 where positive and negative areas representing power are equal and cancel out. Except for losses, the net power interchange due to the magnetizing component is zero.

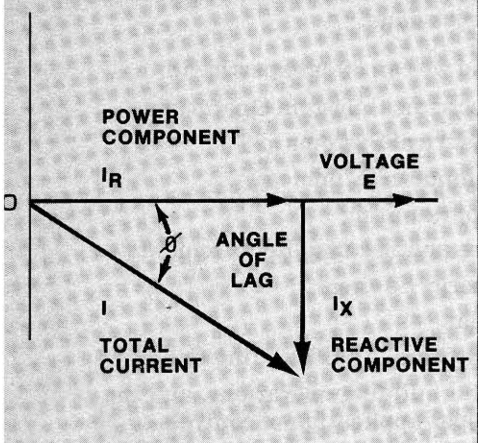
HOW YOU PAY FOR POOR POWER FACTOR

The most common way utilities charge for low power factor is to compute electric demand in KVA, (kilovolt amps) rather than Kw (kilowatts).

Again, using the previous example where peak demand is 2000 Kw, and where demand is charged for at $4.50/KVA, the effect of power factor would be: (Listed below)

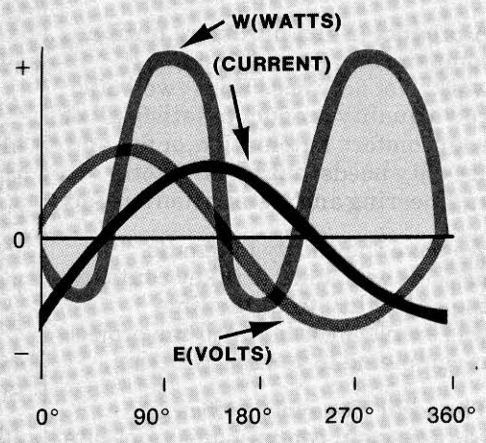
There are many other rate schedules based on power factor. Your electric utility representative will explain the effects of low power factor upon request, and the reduction in cost, if any, that you qualify for by correcting the power factor in your building. It pays to ask!

POWER FACTOR	KW DEMAND	KVA DEMAND	MONTHLY COST
1.0	2000	2000	$ 9000
.90	2000	2222	$ 9999
.85	2000	2353	$10589
.80	2000	2500	$11250



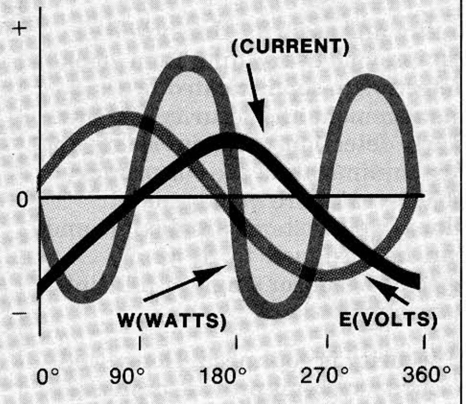
(Figure 3)

Relationship of power components in a-c circuit when current lags behind the voltage.



(Figure 4)

Watts at any instant is volts times amperes. Net power is positive minus negative area.



(Figure 5)

When current lags (or leads) voltage by 90 degrees the net power in watts is zero.

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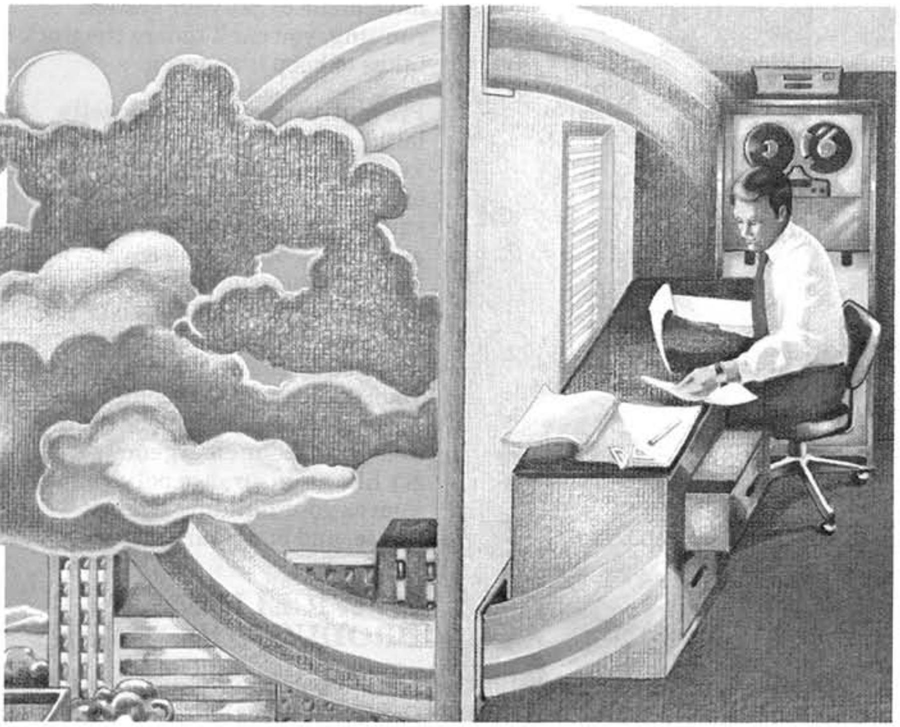
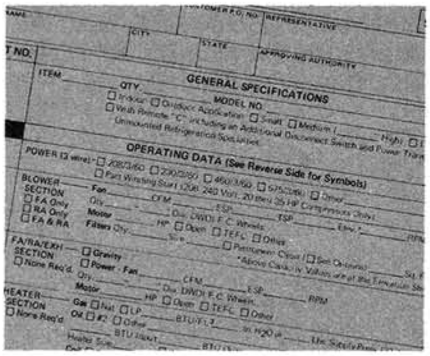
TECHNICAL INFORMATION



You'll find the data needed to fill out THE BUILDING DATA SHEET in this section. When you have additional questions, please check with your local Honeywell representative. (The one nearest you is listed on the back cover of this workbook.)

FIGURING OUT AIR HANDLING CAPACITY



AME
CUSTOMER P.O. NO.
REPRESENTATIVE
CITY
STATE
T NO.
APPROVING AUTHORITY
ITEM
GENERAL SPECIFICATIONS
QTY.
MODEL NO.
☐ Indoor ☐ Outdoor Application
OPERATING DATA (See Reverse Side for Symbols)
POWER (3 wire) ☐ 208/3/60 ☐ 230/3/60 ☐ 460/3/60 ☐ 575/3/60 ☐ Other
BLOWER SECTION
☐ FA Only
☐ RA Only
☐ FA & RA
Fan
Qty.
CFM
ESP
Motor
TSP
HP ☐ Open ☐ TEFC ☐ Other
Filters Qty.
FA/RA/EXH SECTION
☐ None Req'd.
☐ Gravity
☐ Power - Fan
Qty.
CFM
ESP
Motor
HP ☐ Open ☐ TEFC ☐ Other
RPM
HEATER SECTION
☐ None Req'd
Gas ☐ Nat ☐ LP
Oil ☐ #2 ☐ Other
BTU Input

Because each building is different, the air handling capacity of your HVAC system(s) may have to be determined from original engineering drawings or building plans. If your building is using factory-built air handling systems, and such information cannot be found on the drawings, use 400 cfm per ton of air conditioning. Examine engineering data sheets to determine the CFM rating (the air handling capacity) of the HVAC equipment.

DETERMINING VENTILATION "PERCENT AIR"

Your present ventilation air is often "locked in" to your HVAC system, either by local building codes or by adherence to other standards (such as ASHRAE). So you could use the *specified* value for percent ventilation air.

However, even in relatively new buildings, it's been found that some kinds of dampers tend to leak far more than their manufacturers specify. Even when fully closed, they may *leak* as much as the specified ventilation air percentage, if not more (which is a good case by itself for installing Honeywell's MODUFLOW low-leakage dampers).

To make the work you're doing here accurate, you could measure and calculate the percent outside air, instead of assuming it. Here's how it's done.

1. Close your outside air (OA) damper to its fixed minimum position.
2. Measure the following three (3) temperatures, and write the numbers in the space below.
   a. Measure outside air temperature:_______°F (= the value T_o).
   b. Measure return air (RA) temperature:_______°F (= the value T_r).
   c. Measure the mixed air (MA) temperature:_______°F (= the value T_m). Make sure you measure the true MA temperature—on cold days, temperature layers can occur in the mixed air plenum, which is where you'll be taking the measurement.
3. Plug the temperatures for the values T_0, T_r, and T_m into the following formula:

$$B = \frac{T_r - T_m}{T_r - T_o}$$

Work out the arithmetic, and you've got the value for B, the percent ventilation air, that you need to work out ECMs like #5. (To determine leakage, work the same formula with your outside dampers fully closed. This will give you the figure needed to calculate ECM#6.)

HEATING SEASONS AND AVERAGE TEMPERATURES

Use the weather data table (at right) to find the values for C, E, G, H, and I for your area. This data's been generated from United State Air Force weather information for 150 locations in the United States and 13 locations in Canada. Simply pick the location nearest you, that fits your own weather, and use those values in the ECM formulas.

THE TTF, BUILDING SIZE, AND THE VALUE OF F

The Thermal Transmission Factor (TTF) is a pre-determined value. Use the value that matches your building description most closely, together with building perimeter and height measurements for calculating the value of F. For example, for a 200′ x 200′ x 25′ building with a TTF = .69 F = .69 x 800 x 25 = 13,800.

BUILDING DESCRIPTION	TTF VALUE	EXTERIOR WALL CONSTRUCTION	FENESTRATION	ROOF CONSTRUCTION
Low-rise Apartment Building	.48	½″ lapped wood siding; ½″ plywood sheathing; 2″ x 4″ stud framing (16″ c.c.); 2¼″ fiberglass insulation; ½″ Gypsum wallboard.	Single-strength sheet; 30% sidewalls; 0% end walls.	Asphalt shingles; ½″ plywood sheathing, 3½″ fiberglass insulation; ½″ Gypsum wallboard; ventilated attic; roof slope 3 in. 12.
Low-rise Apartment Building	.77	4″ common brick; ½″ plywood sheathing; light framing; no insulation ½″ Gypsum wallboard.	Single-strength sheet; 30% sidewalls; 0% end walls.	Asphalt shingles; ½″ plywood sheathing; 3″ fiberglass insulation; ½″ Gypsum wallboard; ventilated attic; roof slope 3 in. 12.
Office Building	.69	6″ precast concrete panels.	¼″ plate; 30% all walls.	4-ply built-up roofing with gravel; 2″ rigid insulation; steel decking; open web joists; ½″ softboard.
Office Building	.81	1″ insulated sandwich panel with aluminum mullions; structural steel framing.	¼″ plate; 50% all walls.	Metal deck; 4″ poured concrete roofing; structural steel framing; ½″ softwood hung ceiling.
Retail Store	2.0	12″ concrete block, painted both sides.	¼″ plate; 60% South wall; 0% all other walls.	4-ply built-up roofing with gravel; 2″ rigid insulation; steel decking; open web joists; ½″ softboard.
School	.71	4″ common brick, 1″ fiberglass insulation, 4″ concrete block.	Single-strength sheet; 20% all walls.	4-ply built-up roofing with gravel; 2″ rigid insulation; steel decking; open web joists; ½″ softboard.
School	1.1	4″ common brick, no insulation, 4″ concrete block.	Single-strength sheet; 20% all walls,	4-ply built-up roofing with gravel; 1″ rigid insulation; 4″ concrete plank; structural steel framing; ½″ softboard.

HOW MUCH YOUR HEATING AND COOLING COST

For the values D and J, we have put together some common figures, based on the costs of the power involved. Determine your heating and cooling methods, and then find the energy cost that most closely matches yours. Reading to the right, you'll find the approximate cost figure to use in your ECR formula workups.

D—The Cost of Heating

For example, assume that you're using natural gas for heating. Your cost for natural gas is $10. per 1000 cubic feet (MCF). Reading to the right, your D value is then $13.60 per 10^6 Btu.

D = dollar cost per 1,000,000 Btu ($/$10^6$ Btu) for HEATING

IF $/MCF NATURAL GAS IS:	$/THERM	D Value THEN $/$10^6$ BTU x 10^6 (if burned at 70% seasonal efficiency and with 1050 Btu/ft^3) IS:
$ 5.00	.50	$ 6.80
6.00	.60	8.16
7.50	.75	10.20
9.00	.90	12.24
10.00	1.00	13.60
12.00	1.20	16.32
14.00	1.40	19.04
16.00	1.60	21.76
18.00	1.80	24.48

IF $/KWH FOR ELECTRICITY IS:	THEN RESISTANCE HEATING (at 3412 Btu/Kwh) COSTS:	THEN HEAT PUMP (at 7200 Btu/Kwh) COSTS:
$ .01	$ 2.93	$ 1.39
.03	8.79	4.17
.05	14.65	6.94
.07	20.51	9.72
.08	23.45	11.11
.09	26.38	12.50
.10	29.31	13.89
.11	32.24	15.28
.12	35.17	16.67
.15	43.95	20.82
.20	58.60	27.76
.25	73.25	34.70

IF $/GALLON IS:	THEN $/$10^6$ BTU #2 FUEL OIL (at 139,600 Btu/Gallon burned at 70% efficiency) IS:	THEN $/ BTU x 10^6 PROPANE (at 91,500 Btu/Gallon burned at 70% efficiency) IS:
$ .70	$ 7.16	$ 10.93
.80	8.19	12.49
.90	9.21	14.05
1.00	10.23	15.61
1.10	11.25	17.17
1.20	12.28	18.73
1.30	13.30	20.29
1.40	14.33	21.85
1.50	15.35	23.42
1.60	16.37	24.98
1.70	17.39	26.54
1.80	18.41	28.10

IF $/MLB STEAM (Thousand Pounds) IS:	THEN $/BTU x 10^6 DISTRICT STEAM (at 1000 Btu/Lb) IS:
5.00	5.00
6.00	6.00
7.00	7.00
8.00	8.00
9.00	9.00
10.00	10.00
11.00	11.00
12.00	12.00
15.00	15.00
20.00	20.00
25.00	25.00
30.00	30.00

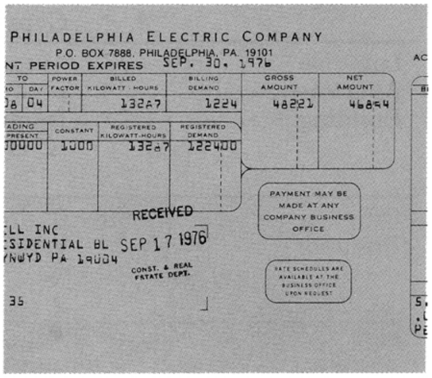
J—The Cost of Cooling

Determine cooling costs the same way (J = dollar cost of cooling per 1,000,000 Btu). For example, if your electricity cost per Kwh is 5¢, then cooling using water-cooled condensers costs $4.17 per 10^6 Btu.

IF $/KWH FOR ELECTRICITY IS:	THEN $/BTU x 10^6 COOLING USING WATER-COOLED CONDENSERS IS:	COOLING USING AIR-COOLED CONDENSERS COSTS:
$ .01	$ .83	$ 1.00
.03	2.50	3.00
.05	4.17	5.00
.07	5.83	7.00
.08	6.67	8.00
.09	7.50	9.00
.10	8.33	10.00
.11	9.17	11.00
.12	10.00	12.00
.15	12.51	15.00
20	16.68	20.00
.25	20.85	25.00

WHERE TO FIND YOUR ENERGY COSTS

For keeping track of your energy costs, with the forms provided on page 14 through 17, use your monthly utility bills. They give you the counts and the amounts of the various types of energy you're using. If necessary, you can call on your Honeywell representative to help figure out the specifics.



PHILADELPHIA ELECTRIC COMPANY
P.O. BOX 7888, PHILADELPHIA, PA. 19101
NT PERIOD EXPIRES SEP. 30. 1976

TO O	TO DAY	POWER FACTOR	BILLED KILOWATT-HOURS	BILLING DEMAND	GROSS AMOUNT	NET AMOUNT
8	04		13287	1224	48221	46854

ADING PRESENT	CONSTANT	REGISTERED KILOWATT-HOURS	REGISTERED DEMAND
0000	1000	13287	122400

RECEIVED
SEP 17 1976
CONST. & REAL ESTATE DEPT.

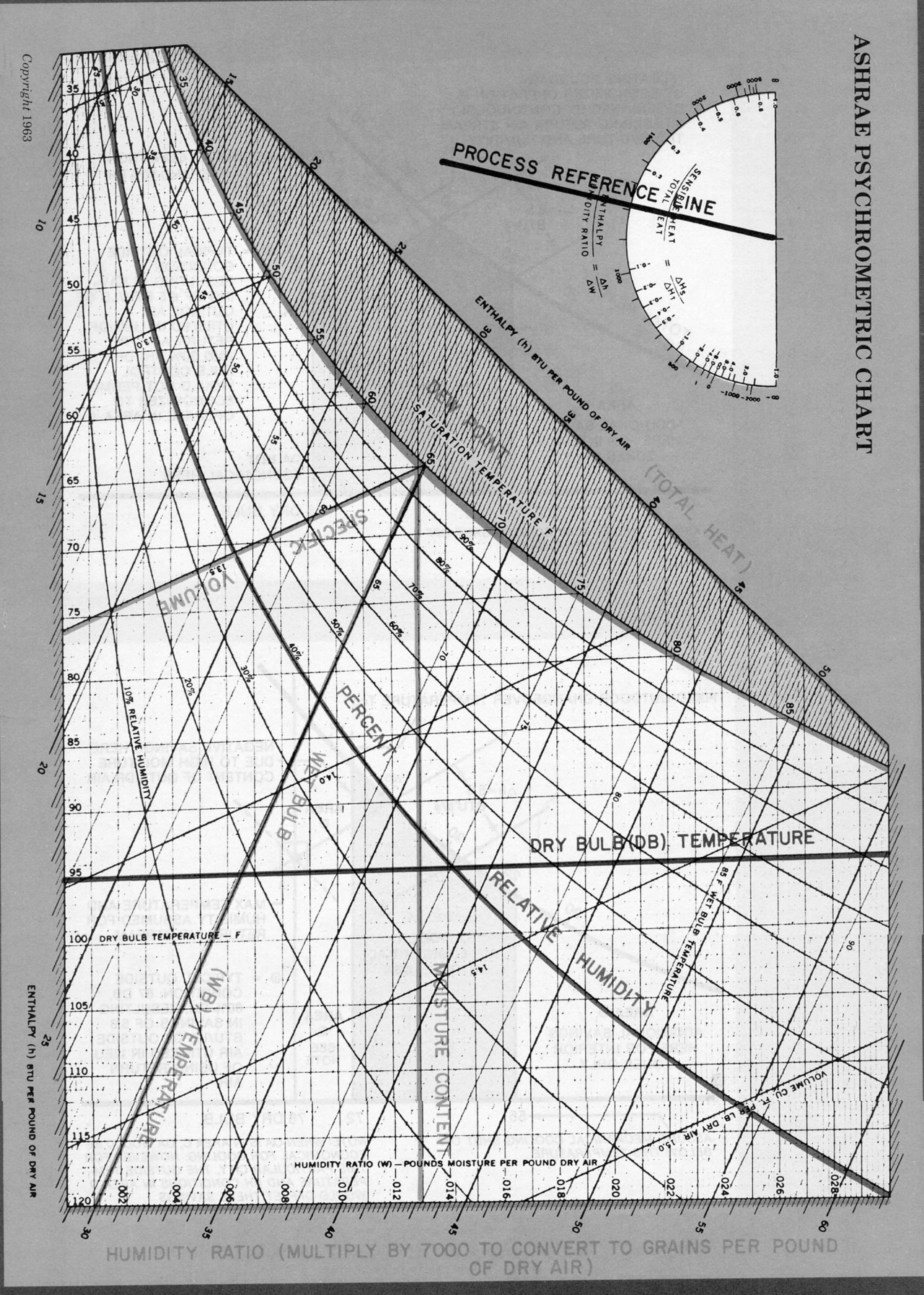
LL INC
SIDENTIAL BL
NWYD PA 19004

35

PAYMENT MAY BE MADE AT ANY COMPANY BUSINESS OFFICE

RATE SCHEDULES ARE AVAILABLE AT THE BUSINESS OFFICE UPON REQUEST

ASHRAE PSYCHROMETRIC CHART



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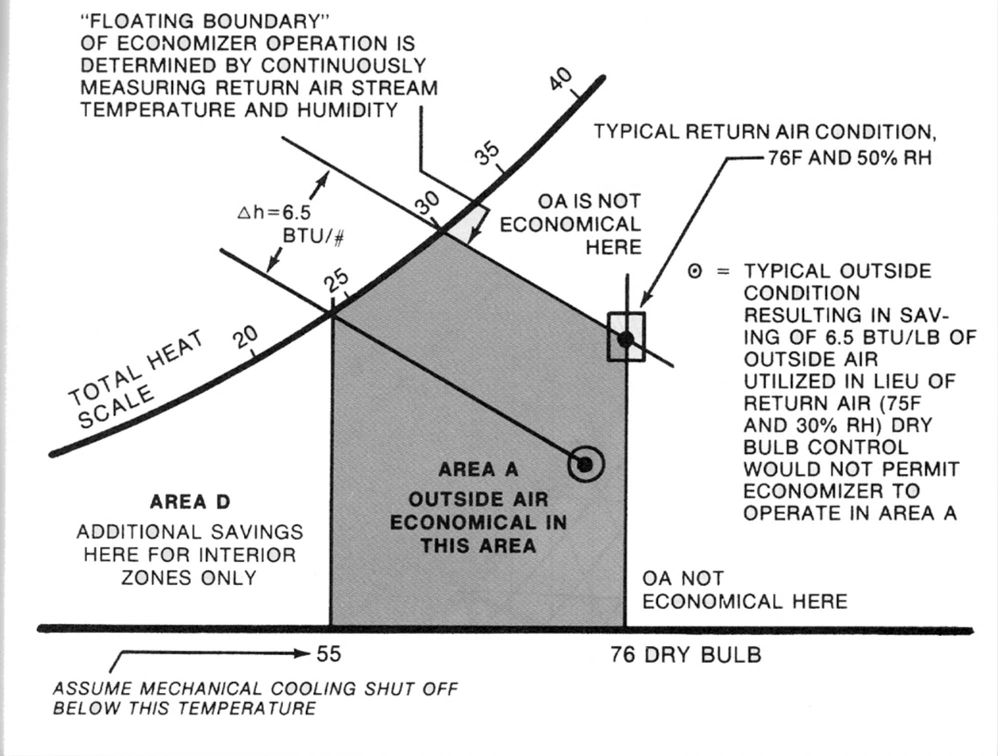
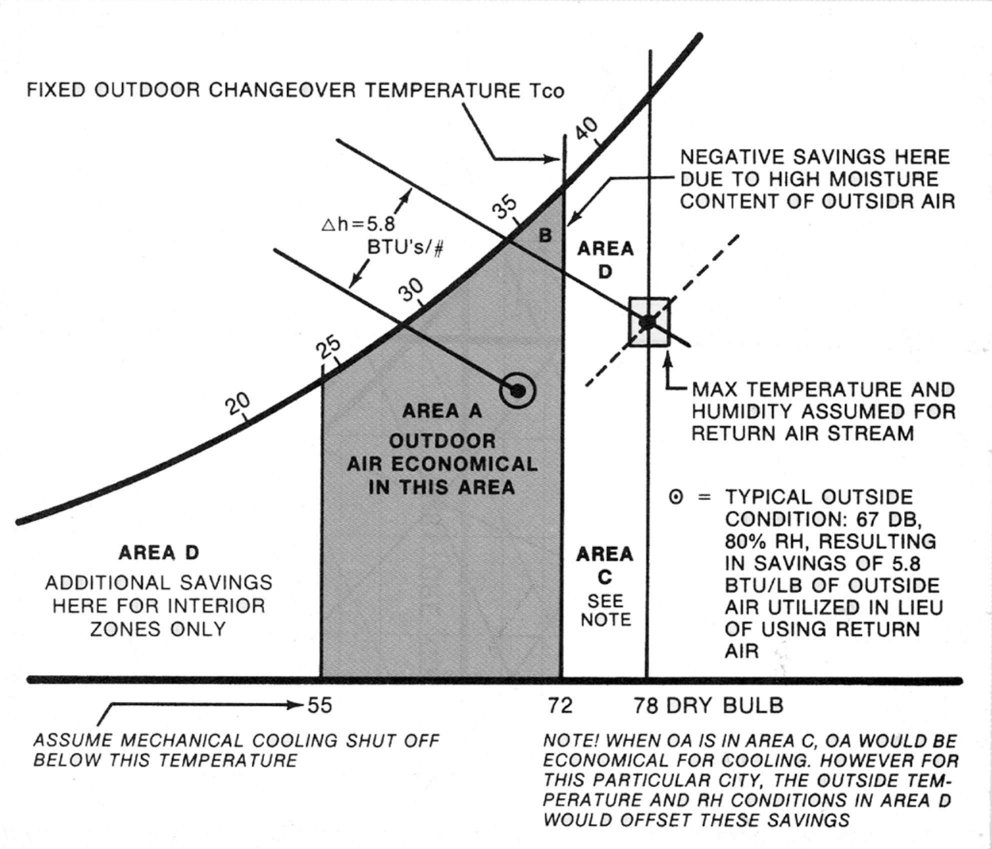
ENTHALPY CONTROL OF ECONOMIZER ANALYSIS FOR "SAMPLE" CITY



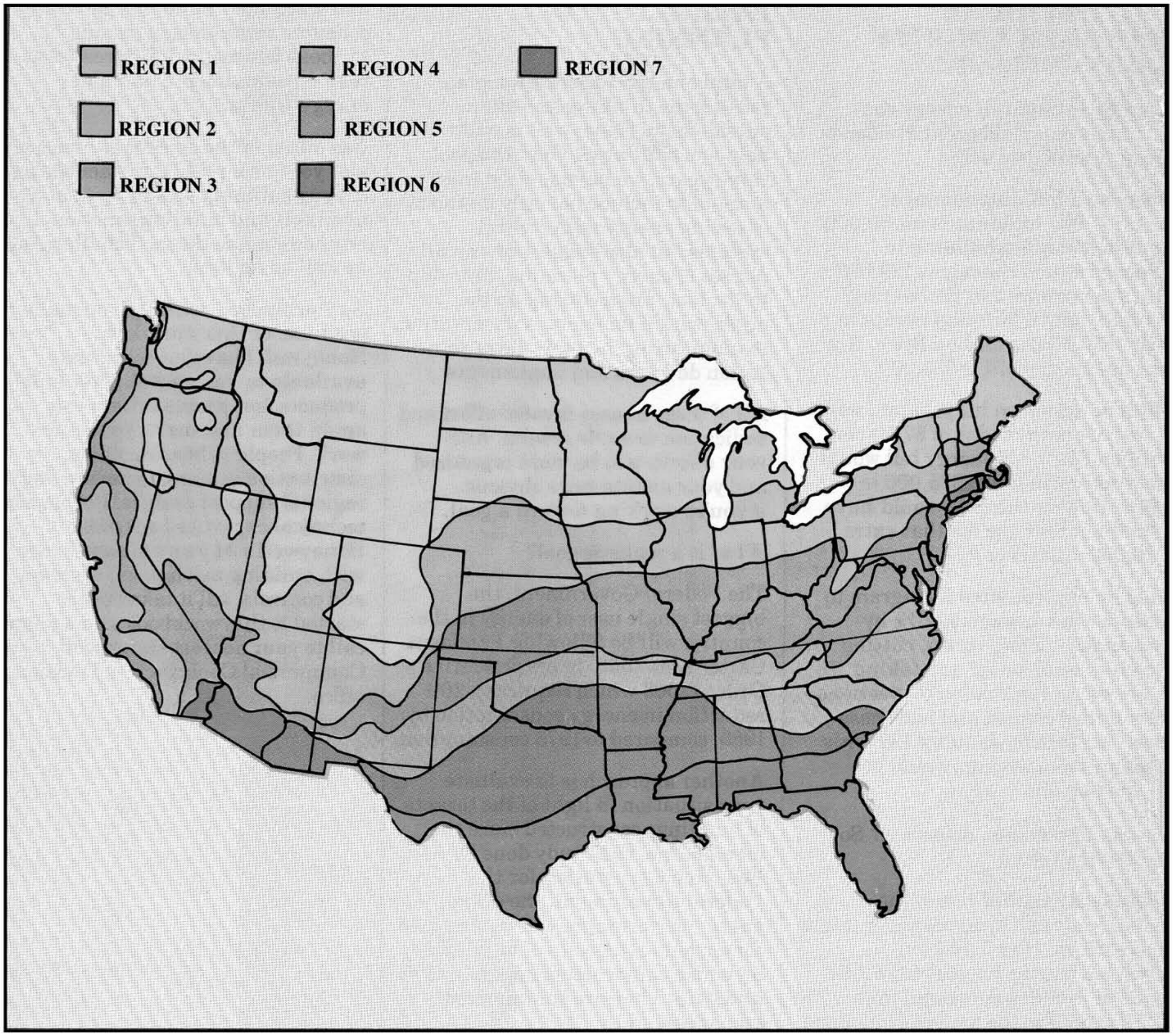
ASSUME MECHANICAL COOLING SHUT OFF BELOW THIS TEMPERATURE

DRY BULB ECONOMIZER CONTROL ANALYSIS FOR "SAMPLE" CITY



ASSUME MECHANICAL COOLING SHUT OFF BELOW THIS TEMPERATURE

NOTE! WHEN OA IS IN AREA C, OA WOULD BE ECONOMICAL FOR COOLING. HOWEVER FOR THIS PARTICULAR CITY, THE OUTSIDE TEMPERATURE AND RH CONDITIONS IN AREA D WOULD OFFSET THESE SAVINGS


REGION 1
REGION 4
REGION 7
REGION 2
REGION 5
REGION 3
REGION 6

ANNUAL ENERGY PERFORMANCE IN 000s BTU's/SQ. FT.

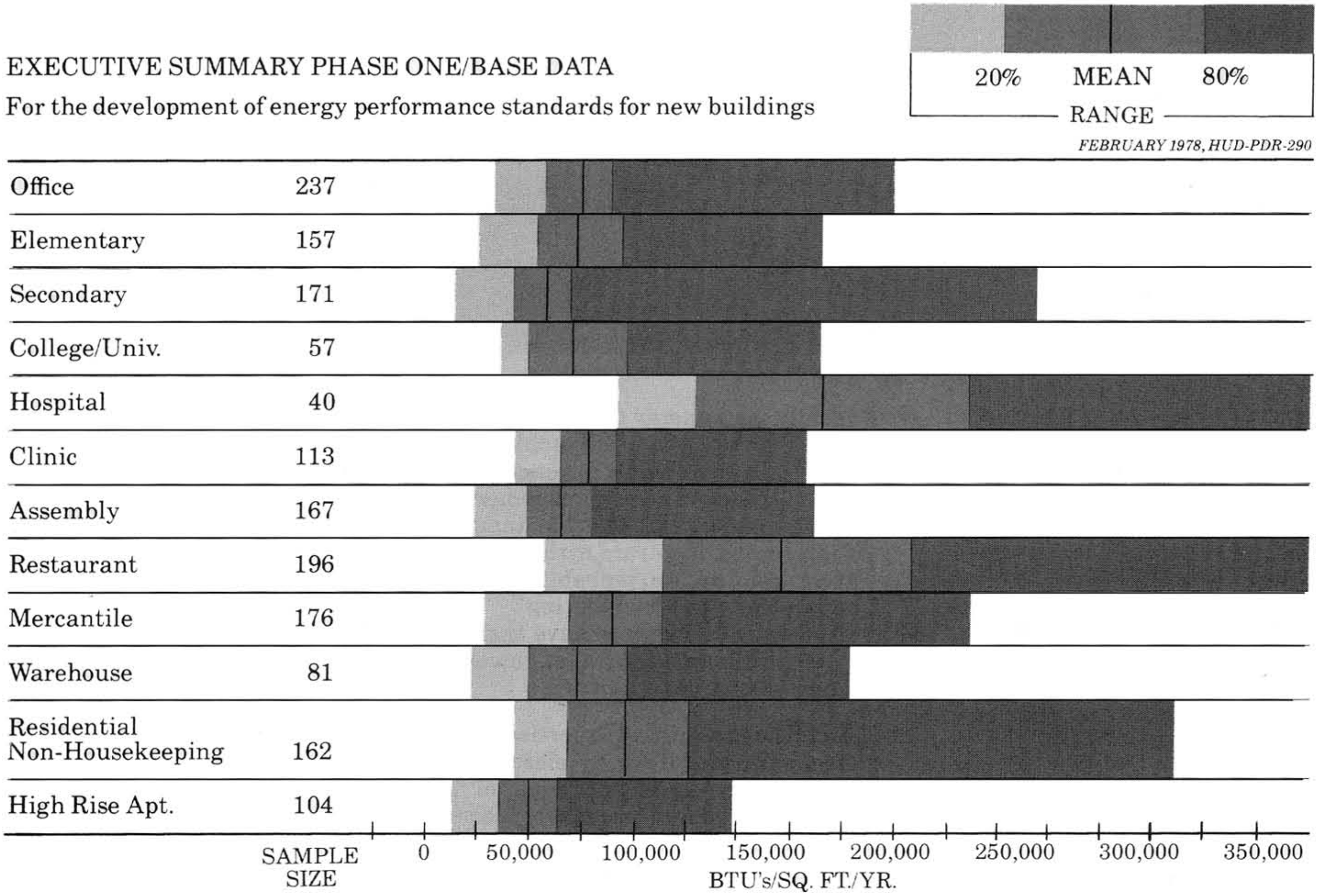
BUILDING TYPE	HEATING & COOLING DEGREE DAY REGION							
	NATIONAL	REGION 1	REGION 2	REGION 3	REGION 4	REGION 5	REGION 6	REGION 7
Office	Mean 84	Mean 85	Mean 76	Mean 65	Mean 61	Mean 51	Mean 50	Mean 64
Elementary	Mean 85	Mean 114	Mean 70	Mean 68	Mean 70	Mean 53	Mean 48	Mean 57
Secondary	Mean 52	Mean 77	Mean 65	Mean 55	Mean 51	Mean 37	Mean 41	Mean 34
College/Univ.	Mean 65	Mean 67	Mean 70	Mean 46	Mean 59	Mean *	Mean *	Mean 83
Hospital	Mean 190	Mean *	Mean 209	Mean 171	Mean 227	Mean 207	Mean *	Mean 197
Clinic	Mean 69	Mean 84	Mean 72	Mean 71	Mean 65	Mean 61	Mean 59	Mean 59
Assembly	Mean 61	Mean 58	Mean 76	Mean 68	Mean 51	Mean 44	Mean 68	Mean 57
Restaurant	Mean 159	Mean 162	Mean 178	Mean 186	Mean 144	Mean 123	Mean 137	Mean 137
Mercantile	Mean 84	Mean 99	Mean 98	Mean 86	Mean 81	Mean 67	Mean 83	Mean 80
Warehouse	Mean 65	Mean 75	Mean 82	Mean 65	Mean 50	Mean 38	Mean 37	Mean 39
Residential Non-Housekeeping	Mean 95	Mean 99	Mean 84	Mean 94	Mean 125	Mean 90	Mean 93	Mean 106
High Rise Apt.	Mean 49	Mean 53	Mean 53	Mean 52	Mean 53	Mean 34	Mean 20	Mean *

*Sample size insufficient to calculate 20%, 80% and/or mean.

FEBRUARY 1978, HUD-PDR-290

EXECUTIVE SUMMARY PHASE ONE/BASE DATA

For the development of energy performance standards for new buildings



FEBRUARY 1978, HUD-PDR-290

WEATHER DATA	**WINTER**		**SUMMER**		**10^6 BTU***		**10^6 BTU***
STATE	AVG. DB WINTER TEMP	LENGTH IN WEEKS	AVG. DB SUMMER TEMP	LENGTH IN WEEKS	TO COOL AIR	OPTIMUM CHANGEOVER TEMP FOR DRY BULB ECONOMIZER**	COOLING SAVINGS
CITY	**C**	**E**	**G**	**I**	**H**		**K**
Alabama							
Birmingham	41.9	16.6	80.6	32.9	72.502	72	16.007
Montgomery	43.5	14.1	81.1	35.3	84.679	72	14.359
Huntsville	40.3	18.8	80.5	30.9	67.713	72	14.648
Mobile	44.7	10.4	79.4	38.4	89.391	67	13.173
Arizona							
Tucson	46.2	12.4	83.5	40.1	61.276	77	20.137
Flagstaff	35.6	33.4	73.5	18.6	20.262	82	22.345
Phoenix	46.4	11.4	86.0	41.3	69.363	82	19.797
Arkansas							
Blytheville	39.5	20.4	80.5	29.7	70.961	72	13.128
Little Rock	41.7	18.1	81.6	31.3	71.929	72	14.288
Ft. Smith	40.5	18.0	81.0	30.5	72.314	72	13.662
California							
Los Angeles	50.2	8.9	72.0	32.6	42.111	72	37.315
San Diego	50.5	7.0	70.9	29.8	40.858	72	38.346
Santa Barbara	49.6	23.9	69.7	12.2	16.386	82	51.450
Bishop	40.2	21.3	82.2	30.4	45.019	82	19.932
Barstow	42.6	20.6	83.7	32.3	49.157	82	21.236
San Francisco	48.2	18.4	71.1	22.2	23.164	79	39.345
Sacramento	46.1	19.4	79.9	28.4	41.610	82	25.558
Colorado							
Denver	35.2	29.4	77.9	22.6	27.052	82	19.493
Colorado Springs	35.4	30.4	76.9	21.6	26.491	82	20.100
Trinidad	36.2	27.7	78.5	25.4	34.007	82	20.410
Grand Junction	36.3	27.5	80.3	23.7	32.076	82	16.074
Delaware							
Dover	38.4	25.2	77.5	23.6	50.337	67	13.533
Wilmington	38.2	26.0	77.5	23.7	46.409	72	14.161
Florida							
Pensacola	44.7	10.4	79.4	38.4	106.944	67	13.511
Miami	49.3	1.6	80.4	50.1	143.912	67	4.011
Jacksonville	45.6	8.6	80.4	41.6	102.703	67	12.370
Orlando	48.5	3.0	78.5	46.2	113.377	67	9.440
Tampa	47.0	4.0	78.5	46.0	111.000	67	9.479
Georgia							
Atlanta	41.1	19.8	78.7	30.0	62.733	72	16.727
Augusta	42.6	16.0	80.7	35.1	80.520	72	16.159
Macon	43.3	14.5	80.3	34.8	77.085	72	15.065
Valdosta	45.0	10.7	80.0	38.9	94.043	72	15.355
Savannah	44.0	12.0	80.0	38.0	92.996	72	15.870
Idaho							
Boise	38.1	31.4	78.8	19.7	27.709	82	16.613
Pocatello	35.1	33.3	78.6	18.8	23.551	82	16.107
Lewiston	40.2	29.7	78.8	18.9	26.574	82	17.155
Illinois							
Chicago	34.2	30.0	77.0	20.9	38.791	72	12.434
Champaign	33.3	27.3	77.9	23.6	38.770	72	12.849
Peoria	34.0	26.0	78.0	24.0	38.770	72	12.080
Rockford	32.0	29.0	77.0	21.0	36.867	72	12.672

**1000 cfm to 55°F for cooling season, based on 10 hr/day, 5 days/wk.*

** CAUTION: See ECM #8

Weather Data, con't.

STATE CITY	**WINTER**		**SUMMER**		**10^6 BTU***		**10^6 BTU***
	AVG. DB WINTER TEMP C	LENGTH IN WEEKS E	AVG. DB SUMMER TEMP G	LENGTH IN WEEKS I	TO COOL AIR H	OPTIMUM CHANGEOVER TEMP FOR DRY BULB ECONOMIZER **	COOLING SAVINGS K
Indiana							
Fort Wayne	34.8	28.5	77.7	22.5	42.551	72	12.500
South Bend	34.2	29.1	77.1	21.4	38.559	72	12.396
Indianapolis	35.8	26.7	78.0	23.9	45.515	72	12.548
Terre Haute	36.8	26.2	78.7	24.8	50.516	72	12.849
Iowa							
Des Moines	32.1	28.0	78.4	21.9	41.213	72	12.645
Mason City	29.8	31.1	76.7	19.7	43.252	72	12.750
Sioux City	31.2	28.9	79.0	22.2	40.447	72	12.283
Council Bluffs	32.1	27.2	78.5	23.0	42.081	72	13.229
Kansas							
Dodge City	35.9	25.4	81.4	25.6	40.237	77	15.436
Goodland	34.3	29.1	81.0	23.6	34.644	82	16.585
Kansas City	36.5	23.6	80.5	25.7	51.300	72	13.197
Wichita	37.0	22.6	81.2	27.0	51.367	72	13.956
Kentucky							
Louisville	38.4	23.5	79.9	26.6	54.019	72	13.700
Covington	36.8	25.1	78.2	24.4	43.478	72	13.678
Hopkinsville	38.2	22.0	79.7	28.4	60.109	72	13.752
Louisiana							
New Orleans	46.4	9.4	79.8	39.6	111.480	67	11.137
Alexandria	43.7	13.3	81.0	37.2	95.036	67	12.600
Shreveport	42.6	15.2	81.8	35.2	84.524	72	14.034
Lake Charles	45.5	10.4	80.4	39.2	107.632	67	10.886
Maine							
Portland	34.5	33.7	74.4	15.5	20.847	77	16.547
Massachusetts							
Boston	35.1	31.1	76.0	19.8	32.537	72	14.644
Springfield	34.6	30.5	76.3	20.1	31.320	72	14.268
Michigan							
Lansing	34.0	30.4	76.0	19.5	30.728	72	13.921
Grand Rapids	34.4	30.5	75.0	19.0	28.563	72	13.700
Traverse City	33.0	32.8	75.3	17.0	25.498	72	14.573
Sault Ste Marie	30.2	37.0	73.4	12.8	16.702	72	15.238
Detroit	33.8	30.5	75.8	19.2	32.997	72	12.892
Minnesota							
Duluth	28.0	37.0	73.5	12.7	16.700	77	15.573
International Falls	25.5	36.8	73.8	14.1	18.041	77	15.290
Minneapolis	29.3	31.0	76.8	18.8	30.661	72	12.968
Mississippi							
Biloxi	45.2	10.1	79.8	37.6	111.068	67	13.063
Jackson	43.0	14.8	81.1	35.3	86.108	72	14.250
Columbus	41.6	16.9	81.2	33.8	80.746	72	13.697

**1000 cfm to 55°F for cooling season, based on 10 hr/day, 5 days/wk.*

** CAUTION: See ECM #8

Weather Data, con't.

STATE	WINTER		SUMMER		10^6 BTU* TO COOL AIR	OPTIMUM CHANGEOVER TEMP FOR DRY BULB ECONOMIZER**	10^6 BTU* COOLING SAVINGS
	AVG. DB WINTER TEMP	LENGTH IN WEEKS	AVG. DB SUMMER TEMP	LENGTH IN WEEKS			
CITY	C	E	G	I	H		K
Missouri							
Kansas City	36.5	23.6	80.5	25.7	49.828	72	13.197
Columbia	36.1	24.4	80.2	25.7	51.319	72	13.148
Springfield	36.7	23.4	79.6	26.9	55.280	72	13.441
St. Louis	36.1	24.2	79.6	26.3	52.665	72	12.934
Montana							
Billings	34.6	32.1	78.1	18.4	22.470	82	17.211
Glasgow	27.9	33.5	77.8	17.5	18.250	82	15.687
Helena	32.9	36.0	76.1	15.5	17.479	82	17.186
Great Falls	33.8	33.8	76.6	16.9	19.523	82	17.374
Nebraska							
Omaha	32.1	27.2	78.5	23.0	42.081	72	13.229
Grand Island	32.6	28.6	79.4	22.7	37.486	77	13.892
North Platt	32.4	29.7	79.1	22.0	32.114	77	15.035
Nevada							
Las Vegas	43.7	15.6	86.8	35.4	60.567	82	20.457
Ely	33.4	35.0	77.7	20.2	24.441	82	17.751
Winnemucca	36.2	31.9	80.4	22.3	31.783	82	17.697
Reno	35.0	33.0	79.0	21.0	25.515	82	19.097
New Hampshire							
Manchester	32.0	32.0	75.0	19.0	28.344	72	14.644
New Jersey							
Trenton	37.5	26.9	77.1	22.9	40.689	72	13.772
New Mexico							
Albuquerque	39.7	23.9	80.4	27.3	37.576	82	18.378
Alamogordo	41.0	19.2	81.8	32.5	45.039	82	20.855
Clovis	38.7	23.1	79.9	29.4	39.173	82	20.644
New York							
Albany	33.8	30.5	76.4	19.5	31.521	72	13.846
Buffalo	34.5	31.1	75.0	18.8	32.916	72	13.376
Syracuse	34.0	30.2	76.1	19.4	30.910	72	14.218
New York City	38.0	27.5	76.0	20.0	40.689	72	14.825
North Carolina							
Greensboro	40.1	21.6	79.0	28.1	55.027	72	15.967
Raleigh	41.0	20.0	79.0	30.0	66.635	72	15.879
Wilmington	43.6	15.2	78.5	33.6	88.000	72	14.161
North Dakota							
Bismarck	27.4	33.5	77.8	18.3	23.351	77	14.795
Grand Forks	24.6	34.4	76.1	16.9	22.163	77	14.133
Minot	27.2	34.7	76.4	16.2	19.282	82	15.380
Fargo	27.2	35.0	77.0	17.0	24.251	77	14.100
Ohio							
Cleveland	34.0	29.4	76.5	21.0	34.066	72	12.640
Dayton	36.2	25.4	78.4	24.3	41.818	72	13.163
Columbus	37.8	25.5	77.6	23.8	41.818	72	13.170
Toledo	33.8	29.5	76.8	21.3	32.997	72	12.647
Cincinnati	36.8	25.1	78.2	24.4	48.502	72	13.678

**1000 cfm to 55°F for cooling season, based on 10 hr/day, 5 days/wk.*

** CAUTION: See ECM #8

Weather Data, con't.

	WINTER		SUMMER		10^6 BTU*		10^6 BTU*
STATE CITY	AVG. DB WINTER TEMP C	LENGTH IN WEEKS E	AVG. DB SUMMER TEMP G	LENGTH IN WEEKS I	TO COOL AIR H	OPTIMUM CHANGEOVER TEMP FOR DRY BULB ECONOMIZER**	COOLING SAVINGS K
Oklahoma							
Altus	39.5	19.5	83.2	31.2	59.470	72	15.151
Oklahoma City	38.9	20.0	81.2	29.5	57.904	72	14.921
Tulsa	39.0	20.2	81.7	29.7	64.398	72	14.702
Enid	37.9	21.6	81.9	28.4	59.745	72	14.662
Oregon							
Burns	35.7	36.3	76.5	17.3			
Medford	41.9	30.9	78.7	21.2	27.150	82	20.122
Pendleton	40.1	29.9	78.1	20.0	25.596	82	20.122
Portland	44.0	30.8	73.5	15.8	18.097	82	25.334
Eugene*	44.0	30.8	74.0	15.0	21.073	82	25.003
Pennsylvania							
Pittsburgh	35.1	28.2	76.0	21.9	35.361	72	14.056
Scranton	35.3	29.7	76.2	20.1	32.081	72	14.072
Williamsport	36.4	28.9	77.2	21.0	34.234	72	13.788
Philadelphia	38.2	26.0	77.5	23.7	40.689	72	14.161
Rhode Island							
Providence	37.6	28.8	74.7	18.7	34.426	72	15.203
South Carolina							
Charleston	43.3	14.2	78.7	36.0	90.941	72	15.437
Columbia	43.2	16.0	79.7	33.4	73.867	72	16.15
Myrtle Beach	43.0	15.9	77.9	32.3	83.593	67	15.704
South Dakota							
Rapid City	32.6	30.7	78.8	19.6	24.746	82	17.353
Huron	28.5	31.4	78.9	20.4		72	12.404
Sioux Falls	29.2	30.4	78.0	20.5	32.166	72	13.279
Tennessee							
Memphis	40.5	18.9	81.1	30.4	71.798	72	13.553
Nashville	39.3	23.3	79.7	28.4	62.170	72	13.100
Knoxville	39.5	21.5	80.0	29.0	56.695	72	14.727
Texas							
Amarillo	38.1	23.0	80.4	28.4	40.959	82	17.748
Lubbock	39.1	20.8	80.3	30.8	41.000	82	16.900
Dallas	42.5	15.1	82.8	34.6	74.244	72	15.239
San Antonio	46.0	8.9	82.7	41.3	88.888	72	14.457
Corpus Christi	48.1	4.8	80.3	43.0	128.001	67	9.795
Houston	47.0	6.0	80.3	42.0	111.808	67	10.700
Utah							
Salt Lake City	36.5	30.2	79.0	19.9	25.170	82	15.364
Wendover	36.7	27.7	79.7	21.6	30.035	82	16.085
Vermont							
Burlington	31.3	33.1	74.8	16.7	22.432	72	15.368
Virginia							
Richmond	40.9	20.9	77.8	26.8	60.639	72	15.122
Roanoke	39.8	23.4	78.9	26.3	45.823	72	15.287

**1000 cfm to 55°F for cooling season, based on 10 hr/day, 5 days/wk.*

** CAUTION: See ECM #8

Weather Data, con't.

STATE	WINTER		SUMMER		10^6 BTU*		10^6 BTU*
CITY	AVG. DB WINTER TEMP C	LENGTH IN WEEKS E	AVG. DB SUMMER TEMP G	LENGTH IN WEEKS I	TO COOL AIR H	OPTIMUM CHANGEOVER TEMP FOR DRY BULB ECONOMIZER **	COOLING SAVINGS K
Washington							
Seattle	43.7	37.3	70.9	9.4	15.768	77	25.142
Spokane	36.6	34.8	76.1	15.6	19.325	82	16.884
West Virginia							
Charleston	38.4	23.7	78.4	26.1	46.953	72	14.291
Clarksburg	36.5	29.1	75.2	22.5	35.264	72	15.430
Wisconsin							
Madison	31.5	30.7	76.9	20.2	34.867	72	12.672
Green Bay	31.1	33.0	75.2	17.5	29.028	72	13.512
Milwaukee	33.0	30.0	77.0	20.9	38.700	72	12.400
Wyoming							
Casper	33.6	33.5	78.3	19.2	23.792	82	15.695
Cheyenne	34.4	33.9	76.0	18.3	21.923	82	18.224
Rock Springs	31.7	35.3	75.3	16.7	17.716	82	16.718

Weather Data, con't.

PROVINCE	WINTER		SUMMER		10^6 BTU*		10^6 BTU*
CITY	AVG. DB WINTER TEMP C	LENGTH IN WEEKS E	AVG. DB SUMMER TEMP G	LENGTH IN WEEKS I	TO COOL AIR H	OPTIMUM CHANGEOVER TEMP FOR DRY BULB ECONOMIZER	COOLING SAVINGS K
Alberta							
Calgary	30.6	39.1	72.6	10.7	12.697	82	24.000
Edmonton	27.6	38.0	72.8	10.7	12.768	82	23.000
British Columbia							
Vancouver	43.2	34.1	69.7	9.9	11.717	82	26.100
Manitoba							
Winnipeg	34.2	35.5	81.6	14.9	20.193	77	13.500
New Brunswick							
Moncton	32.0	36.8	72.4	11.6	16.476	72	14.900
Newfoundland							
St. John's	35.5	41.7	69.8	5.3	7.425	72	14.500
Nova Scotia							
Halifax	34.0	36.5	71.5	9.6	13.375	72	15.700
Ontario							
London	32.6	32.8	74.4	17.2	27.066	72	12.700
Ottawa	31.0	32.1	74.5	15.7	12.562	72	12.900
Toronto	33.2	32.7	82.4	17.0	31.663	67	10.600
Quebec							
Montreal	30.3	32.9	74.1	15.9	24.600	72	13.400
Quebec	29.3	36.5	72.5	12.7	18.753	72	14.700
Saskatchewan							
Regina	24.4	36.5	75.3	15.500	19.578	82	20.900

**1000 cfm to 55°F for cooling season, based on 10 hr/day, 5 days/wk.*

** CAUTION: See ECM #8

GLOSSARY

Various terms that are standard in the HVAC industry are used in this manual. You will find definitions for many of these terms in this glossary. For additional help, consult the ASHRAE Handbook.

ABBREVIATIONS

AHU Air Handling Unit

BTU British Thermal Unit

CCF 100 Cubic Feet

Cfm Cubic Feet per Minute

ΔT Temperature Differential or Change

CU Coefficient of Utilization

DDC Direct Digital Control

DPR Motor Damper Motor

GSA General Services Administration

HHV Highest Heating Value

IES Illuminating Engineers Society

KVA 100 Volt Amps

KVAR Kilovar

MCF 100 Cubic Feet

PPM Parts per Million

Psi Pounds per square inch

TES Thermal Energy Storage

TIMA Thermal Insulation Manufacturers Association

VAV Variable Air Volume

TERMS

Air Conditioning
The process of treating air so as to control simultaneously its temperature, humidity, cleanliness and distribution to meet requirements of the conditioned space.

ASHRAE 90-75
ASHRAE standard for energy conservation in new building design.

ASHRAE 95-61
ASHRAE standard for building ventilation.

Boiler Capacity
The rate of heat output in BTU/h (W) measured at the boiler outlet at the design inlet, outlet, and rated input.

Building Envelope
The elements of a building which enclose conditioned spaces through which thermal energy may be transferred to or from the exterior.

Cold Deck
The portion of the duct containing the chilled water coil or DX coil. Generally parallel with a bypass deck or hot deck.

Degree Day, Heating
A unit, based upon temperature and time, used in estimating fuel consumption and specifying nominal heating load of a building in winter. For any one day, when the mean temperature is less than 65°F (18°C), there exist as many degree days as there are Fahrenheit (Celsius) degrees difference in temperature between the mean temperature for the day and 65°F (18°C).

Domestic Hot Water
Hot water for domestic commercial purposes other than comfort heating and industrial processes (e.g. hot tap water, showers, etc.).

Dry Bulb Temperature
The temperature as measured by a conventional (dry bulb) thermometer.

Duty Cycling
Period cycling off of electrical loads to reduce overall Kw demand level and Kwh consumption.

Economizer Cycling
Used in cooling systems capable of bringing in 100% outdoor air. A control sequence which allows the selection of outside or return air for cooling based on lowest dry bulb temperature or enthalpy.

Energy
The capacity for doing work; taking a number of forms which may be transformed from one into another, such as thermal (heat),

mechanical (work), electrical, and chemical. In customary units, measured in kilowatt-hours (Kwh) or British Thermal Units (BTU).

Energy Efficiency Ratio (EER)
The ratio of net cooling capacity in BTU/h to total rate of electric input in watts under designated operating conditions.

Enthalpy
Total heat of air measured in BTU per pound of dry air.

Fan Inlet (Vortex) Damper
An air valve placed on the inlet to a fan used to modulate the CFM delivered by the fan.

Heat
The form of energy that is transferred by virtue of a temperature difference.

Heat Pump
A refrigeration machine which is arranged to either heat or cool a building by using heat from the condenser section or by using cooling from the evaporator section.

Humidistat
An instrument which measures humidity and controls a device(s) for maintaining a desired humidity.

Infiltration
The uncontrolled inward air leakage through cracks and joints in any building element and around windows and doors of a building, caused by the pressure effects of wind and/or the effect of differences in the indoor and outdoor air density.

Latent Heat
The amount of heat necessary to change a given quantity of water at 212ºF (100ºC) from liquid to vapor at constant barometric pressure.

Load Analyzer
A device which selects and transmits the highest and lowest signals from space thermostats. Used to reset discharge temperature controllers from zones with highest heating and cooling demands.

Load Shedding
The turning off of electrical loads to limit peak electrical demand.

Mixing Box
A box containing dampers in the hot and cold air stream, mixing the two and delivering the air to a space at a specified temperature.

Peak Load
The maximum electrical or thermal load reached during an arbitrary period of time.

Photovoltaic Cells
A solar cell in which electrons flow from one layer of material to the other when exposed to light, thereby converting sunlight directly to electricity.

Power
In connection with machines, power is the time rate of doing work. In connection with the transmission of energy of all types, power refers to the rate at which energy is transmitted. In customary units it is measured in watts (W) or British thermal units per hour (BTU/h).

Recovered Energy
Energy utilized which would otherwise be wasted from an energy utilization system.

Reheat
The application of sensible heat to supply air that has been previously cooled below the temperature of the conditioned space by either mechanical refrigeration or the introduction of outdoor air to provide cooling.

Reset
Adjustment of the set point of a control instrument to a higher or lower value automatically or manually to conserve energy.

Sensible Heat
The heat which changes the temperature of the air without a change in moisture content. Changes in dry bulb thermometer readings are indicative of changes in sensible heat.

Space Temperature Feedback
The process of sensing multiple space temperatures, selecting those spaces with the greatest cooling or heating requirements and using these signals to reset a controller.

Thermostat
An instrument which measures temperature and controls devices for maintaining a desired temperature.

Thermograph
An infrared scan that can be used to detect heat loss due to poor insulation.

U Factor
The overall heat transmission coefficient, or quantity of heat in BTUs transmitted per hour through one square foot of a building section (wall, roof, window, floor, etc.) for each degree F of temperature difference between the air on the warm side and the air on the cold side of the building section.

Variable Air Volume (VAV)
A method used to cool or heat a space or zone by varying the amount of air delivered to that space as conditions change (versus holding the amount of air constant and changing the air temperature).

Ventilation Air
That portion of supply air which comes from outside plus any recirculated air that has been treated to maintain the desired quality of air within a designated space.

Wet Bulb Temperature
The temperature reading obtained from a standard thermometer with its bulb encased in a wick saturated with water at air temperature and exposed to air moving at sufficient velocity to bring fresh samples of air successively to the wick. The thermometer reading drops to a minimum which is dependent on the dry bulb temperature and moisture content of the air.

Zone
A space or group of spaces within a building with heating and/or cooling requirements sufficiently similar so that comfort conditions can be maintained throughout by a single controlled device.

Excerpt From ASHRAE Ventilation Standard 62-1999

TABLE 2
OUTDOOR AIR REQUIREMENTS FOR VENTILATION*
2.1 COMMERCIAL FACILITIES (offices, stores, shops, hotels, sports facilities)

Application	Estimated Maximum** Occupancy P/1000 ft^2 or 100 m^2	Outdoor Air Requirements				Comments
		cfm/ person	L/s· person	cfm/ft^2	L/s·m^2	
Dry Cleaners, Laundries						Dry-cleaning processes may require more air.
Commercial laundry	10	25	13			
Commercial dry cleaner	30	30	15			
Storage, pick up	30	35	18			
Coin-operated laundries	20	15	8			
Coin-operated dry cleaner	20	15	8			
Food and Beverage Service						
Dining rooms	70	20	10			
Cafeteria, fast food	100	20	10			
Bars, cocktail lounges	100	30	15			Supplementary smoke-removal equipment may be required.
Kitchens (cooking)	20	15	8			Makeup air for hood exhaust may require more ventilating air. The sum of the outdoor air and transfer air of acceptable quality from adjacent spaces shall be sufficient to provide an exhaust rate of not less than 1.5 cfm/ft^2 (7.5 L/s·m^2).
Garages, Repair, Service Stations						
Enclosed parking garage				1.50	7.5	Distribution among people must consider worker location and concentration of running engines; stands where engines are run must incorporate systems for positive engines exhaust withdrawal. Contaminant sensors may be used to control ventilation.
Auto repair rooms				1.50	7.5	
Hotels, Motels, Resorts, Dormitories				cfm/room	L/s·room	Independent of room size.
Bedrooms				30	15	
Living rooms				30	15	
Baths				35	18	Installed capacity for intermittent use.
Lobbies	30	15	8			
Conference rooms	50	20	10			
Assembly rooms	120	15	8			
Dormitory sleeping areas	20	15	8			See also food and beverage services, merchandising, barber and beauty shops, garages.
Gambling casinos	120	30	15			Supplementary smoke-removal equipment may be required.
Offices						
Office space	7	20	10			Some office equipment may require local exhaust.
Reception areas	60	15	8			
Telecommunication centers and data entry areas	60	20	10			
Conference rooms	50	20	10			
Public Spaces				cfm/ft^2	L/s·m^2	
Corridors and utilities				0.05	0.25	
Public restrooms, cfm/wc or cfm/urinal		50	25			Normally supplied by transfer air.
Locker and dressing rooms				0.5	2.5	Local mechanical exhaust with no recirculation recommended.
Smoking lounge	70	60	30			
Elevators				1.00	5.0	Normally supplied by transfer air.

* Table 2 prescribes supply rates of acceptable outdoor air required for acceptable indoor air quality. These values have been chosen 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.

** Net occupiable space.

Table 2 reprinted from ASHRAE STANDARD 62-1999 pages 7 to 9.

Excerpt From ASHRAE Ventilation Standard 62-1999

TABLE 2
OUTDOOR AIR REQUIREMENTS FOR VENTILATION* (*Continued*)
2.1 COMMERCIAL FACILITIES (offices, stores, shops, hotels, sports facilities)

Application	Estimated Maximum** Occupancy P/1000 ft² or 100 m²	Outdoor Air Requirements				Comments
		cfm/ person	L/s· person	cfm/ft²	L/s·m²	
Retail Stores, Sales Floors, and Show Room Floors						
Basement and street	30			0.30	1.50	
Upper floors	20			0.20	1.00	
Storage rooms	15			0.15	0.75	
Dressing rooms				0.20	1.00	
Malls and arcades	20			0.20	1.00	
Shipping and receiving	10			0.15	0.75	
Warehouses	5			0.05	0.25	
Smoking lounge	70	60	30			Normally supplied by transfer air, local mechanical exhaust; exhaust with no recirculation recommended.
Specialty Shops						
Barber	25	15	8			
Beauty	25	25	13			
Reducing salons	20	15	8			
Florists	8	15	8			Ventilation to optimize plant growth may dictate requirements.
Clothiers, furniture				0.30	1.50	
Hardware, drugs, fabric	8	15	8			
Supermarkets	8	15	8			
Pet shops				1.00	5.00	
Sports and Amusement						
Spectator areas	150	15	8			When internal combustion engines are operated for maintenance of playing surfaces, increased ventilation rates may be required.
Game rooms	70	25	13			
Ice arenas (playing areas)				0.50	2.50	
Swimming pools (pool and deck area)				0.50	2.50	Higher values may be required for humidity control.
Playing floors (gymnasium)	30	20	10			
Ballrooms and discos	100	25	13			
Bowling alleys (seating areas)	70	25	13			
Theaters						Special ventilation will be needed to eliminate special stage effects (e.g., dry ice vapors, mists, etc.)
Ticket booths	60	20	10			
Lobbies	150	20	10			
Auditorium	150	15	8			
Stages, studios	70	15	8			
Transportation						Ventilation within vehicles may require special considerations.
Waiting rooms	100	15	8			
Platforms	100	15	8			
Vehicles	150	15	8			
Workrooms						
Meat processing	10	15	8			Spaces maintained at low temperatures (−10°F to + 50°F, or −23°C to + 10°C) are not covered by these requirements unless the occupancy is continuous. Ventilation from adjoining spaces is permissible. When the occupancy is intermittent, infiltration will normally exceed the ventilation requirement. (See Reference 18).

---

* Table 2 prescribes supply rates of acceptable outdoor air required for acceptable indoor air quality. These values have been chosen 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.

** Net occupiable space.

Table 2 reprinted from ASHRAE STANDARD 62-1999 pages 7 to 9.

TABLE 2
OUTDOOR AIR REQUIREMENTS FOR VENTILATION* (*Continued*)
2.1 COMMERCIAL FACILITIES (offices, stores, shops, hotels, sports facilities)

Application	Estimated Maximum** Occupancy P/1000 ft² or 100 m²	Outdoor Air Requirements				Comments
		cfm/ person	L/s· person	cfm/ft²	L/s·m²	
Photo studios	10	15	8			
Darkrooms	10			0.50	2.50	
Pharmacy	20	15	8			
Bank vaults	5	15	8			
Duplicating, printing				0.50	2.50	Installed equipment must incorporate positive exhaust and control (as required) of undesirable contaminants (toxic or otherwise).
2.2 INSTITUTIONAL FACILITIES						
Education						
Classroom	50	15	8			
Laboratories	30	20	10			Special contaminant control systems may be required for processes or functions including laboratory animal occupancy.
Training shop	30	20	10			
Music rooms	50	15	8			
Libraries	20	15	8			
Locker rooms				0.50	2.50	
Corridors				0.10	0.50	
Auditoriums	150	15	8			
Smoking lounges	70	60	30			Normally supplied by transfer air. Local mechanical exhaust with no recirculation recommended.
Hospitals, Nursing and Convalescent Homes						
Patient rooms	10	25	13			Special requirements or codes and pressure relationships may determine minimum ventilation rates and filter efficiency. Procedures generating contaminants may require higher rates.
Medical procedure	20	15	8			
Operating rooms	20	30	15			
Recovery and ICU	20	15	8			
Autopsy rooms				0.50	2.50	Air shall not be recirculated into other spaces.
Physical therapy	20	15	8			
Correctional Facilities						
Cells	20	20	10			
Dining halls	100	15	8			
Guard stations	40	15	8			

* Table 2 prescribes supply rates of acceptable outdoor air required for acceptable indoor air quality. These values have been chosen 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.

** Net occupiable space.

Table 2 reprinted from ASHRAE STANDARD 62-1999 pages 7 to 9.

Energy Performance Survey

Date:

Name of Facility:

Location:

	Building Energy Types			
	Electricity		Other Energy Type:	
	kWh Used	Subtotal	Unit Quantity	Subtotal
January		■		▒
February		■		▒
March		■		▒
1st Quarter Subtotal	■		▒	
April		■		▒
May		■		▒
June		■		▒
2nd Quarter Subtotal	■		▒	
July		■		▒
August		■		▒
September		■		▒
3rd Quarter Subtotal	■		▒	
October		■		▒
November		■		▒
December		■		▒
4th Quarter Subtotal	■		▒	
Year's Total	■		▒	

BTU Conversion Formulas

Energy Type	**(Units)**	**Year's Total**		**BTU/Units Bill Coversion**		**BTUs per year**
Electricity	(kWh)	____________	**X**	3,413	=	____________
Natural Gas	(Therm)	____________	**X**	103,200	=	____________
	(Mcf)	____________	**X**	10,320,000	=	____________
Fuel Oil	(#2 grade)	____________	**X**	139,000	=	____________
Coal	(Ton)	____________	**X**	245,000,000	=	____________

Total BTUs per year for all energy used ____________

BTUs per square foot per year formula

____________	/	____________	=	____________
Total Yr. BTUs		**Gross facility ft^2**		**Total BTUs/ft^2/year**

Energy Performance Survey

Date:

Name of Facility:

Location:

	Building Energy Types			
	Electricity		Other Energy Type:	
	kWh Used	Subtotal	Unit Quantity	Subtotal
January		■		▒
February		■		▒
March		■		▒
1st Quarter Subtotal	■		▒	
April		■		▒
May		■		▒
June		■		▒
2nd Quarter Subtotal	■		▒	
July		■		▒
August		■		▒
September		■		▒
3rd Quarter Subtotal	■		▒	
October		■		▒
November		■		▒
December		■		▒
4th Quarter Subtotal	■		▒	
Year's Total	■		▒	

BTU Conversion Formulas

Energy Type	**(Units)**	**Year's Total**		**BTU/Units Bill Coversion**		**BTUs per year**
Electricity	(kWh)		**X**	3,413	=	
Natural Gas	(Therm)		**X**	103,200	=	
	(Mcf)		**X**	10,320,000	=	
Fuel Oil	(#2 grade)		**X**	139,000	=	
Coal	(Ton)		**X**	245,000,000	=	

Total BTUs per year for all energy used ____________

BTUs per square foot per year formula

____________ **/** ____________ = ____________

Total Yr. BTUs **Gross facility ft^2** **Total BTUs/ft^2/year**

SPACE CONDITIONING EQUIPMENT AND SCHEDULES

SIZE, GROSS SQ. FT. ____________________

AREA COOLED____________________ AREA HEATED____________________

TYPE(S) OF OCCUPANCY: (% OR SQ. FT.)

Office____________________ (Other)____________________

Warehouse____________________ (Other)____________________

Manufacturing____________________ (Other)____________________

Retail____________________

Lobbies & Mall____________________
(Enclosed)

BUILDING USE AND OCCUPANCY

Fully Occupied: (50% or more of normal)

Weekdays (Hours)______________ to ______________

Weekends (Hours)______________ to ______________

______________ to ______________ Sunday

______________ to ______________ Holidays

Remarks: Describe below if occupancy differs for different floors, areas, buildings: ____________________

____

____

LIGHTING SURVEY

1. Interior Lighting Type____________________ Watts/Ft² Offices____________________

______________ Other ____________________

Total Install KW____________________

On-Off from Breaker Panel?____________________

Wall Switches?____________________ Control Switching?____________________

Operating Schedule____________________

____

2. Exterior Lighting Type____________________ Sq. Ft. Served: ____________________

Total KW____________________ Foot-Candles:____________________

Remarks: ____________________

____

Operating Schedule: ____________________

____

____

SPACE CONDITIONING EQUIPMENT AND SCHEDULES

SIZE, GROSS SQ. FT. ____________________

AREA COOLED ____________________ AREA HEATED ____________________

TYPE(S) OF OCCUPANCY: (% OR SQ. FT.)

Office ____________________ (Other) ____________________

Warehouse ____________________ (Other) ____________________

Manufacturing ____________________ (Other) ____________________

Retail ____________________

Lobbies & Mall ____________________
(Enclosed)

BUILDING USE AND OCCUPANCY

Fully Occupied: (50% or more of normal)

Weekdays (Hours) ____________ to ____________

Weekends (Hours) ____________ to ____________

____________ to ____________ Sunday

____________ to ____________ Holidays

Remarks: Describe below if occupancy differs for different floors, areas, buildings: ____________________

____

____

LIGHTING SURVEY

1. Interior Lighting Type ____________________ Watts/Ft² Offices ____________________

____________ Other ____________________

Total Install KW ____________________

On-Off from Breaker Panel? ____________________

Wall Switches? ____________________ Control Switching? ____________________

Operating Schedule ____________________

____

2. Exterior Lighting Type ____________________ Sq. Ft. Served: ____________________

Total KW ____________________ Foot-Candles: ____________________

Remarks: ____________________

____

Operating Schedule: ____________________

____

____

☐ **Electricity**

	KWH USED		ELECTRIC DEMAND KW				$ COST				REMARKS
			UNITS		$ COST		UNIT COST		TOTAL COST		
	Current	Base 19___	Current	Base	Current	Base	Current	Base	Current	Base	
JAN											
FEB											
MAR											
APR											
MAY											
JUN											
JUL											
AUG											
SEP											
OCT											
NOV											
DEC											
Total											

☐ **Electricity**

	KWH USED		ELECTRIC DEMAND KW				$ COST				REMARKS
			UNITS		$ COST		UNIT COST		TOTAL COST		
	Current	Base 19___	Current	Base	Current	Base	Current	Base	Current	Base	
JAN											
FEB											
MAR											
APR											
MAY											
JUN											
JUL											
AUG											
SEP											
OCT											
NOV											
DEC											
Total											

☐ **Electricity**

	KWH USED		ELECTRIC DEMAND KW				$ COST				REMARKS
			UNITS		$ COST		UNIT COST		TOTAL COST		
	Current	Base 19___	Current	Base	Current	Base	Current	Base	Current	Base	
JAN											
FEB											
MAR											
APR											
MAY											
JUN											
JUL											
AUG											
SEP											
OCT											
NOV											
DEC											
Total											

☐ **Electricity**

	KWH USED		ELECTRIC DEMAND KW				$ COST				REMARKS
			UNITS		$ COST		UNIT COST		TOTAL COST		
	Current	Base 19___	Current	Base	Current	Base	Current	Base	Current	Base	
JAN											
FEB											
MAR											
APR											
MAY											
JUN											
JUL											
AUG											
SEP											
OCT											
NOV											
DEC											
Total											

☐ **Fuel Oil** ☐ **Propane** **Primary?**__________ **Standby?**__________

Year	Month	FUEL OIL USED		$ COST				REMARKS
				UNIT COST $/GALLON		TOTAL COST		
		Current	Base 19___	Current	Base	Current	Base	
	JAN							
	FEB							
	MAR							
	APR							
	MAY							
	JUN							
	JUL							
	AUG							
	SEP							
	OCT							
	NOV							
	DEC							
Total								

☐ **Fuel Oil** ☐ **Propane** **Primary?**__________ **Standby?**__________

Year	Month	FUEL OIL USED		$ COST				REMARKS
				UNIT COST $/GALLON		TOTAL COST		
		Current	Base 19___	Current	Base	Current	Base	
	JAN							
	FEB							
	MAR							
	APR							
	MAY							
	JUN							
	JUL							
	AUG							
	SEP							
	OCT							
	NOV							
	DEC							
Total								

☐ **Gas** **Firm**____________________ **Interruptible**____________________

		GAS USED ☐ MCF ☐ CCF ☐ CU FT.		$ COST				REMARKS
				UNIT COST ☐ $/MCF ☐ $/CCF		TOTAL COST		
Year	Month	Current	Base 19___	Current	Base	Current	Base	
	JAN							
	FEB							
	MAR							
	APR							
	MAY							
	JUN							
	JUL							
	AUG							
	SEP							
	OCT							
	NOV							
	DEC							
Total								

☐ **Gas** **Firm**____________________ **Interruptible**____________________

		GAS USED ☐ MCF ☐ CCF ☐ CU FT.		$ COST				REMARKS
				UNIT COST ☐ $/MCF ☐ $/CCF		TOTAL COST		
Year	Month	Current	Base 19___	Current	Base	Current	Base	
	JAN							
	FEB							
	MAR							
	APR							
	MAY							
	JUN							
	JUL							
	AUG							
	SEP							
	OCT							
	NOV							
	DEC							
Total								

BUILDING SURVEY – EQUIPMENT LIST

System: ______________ **Date:** ________ **Building/Location:** ______________

	EQUIPMENT			RATED OUTPUT			RATED INPUT			OPERATING SCHEDULE			OUTSIDE AIR CFM	HOURS PER WK SPACE IS OCCUPIED	NIGHT SET BACK?	REMARKS
	Description	Subsystem	Quantity	☐ HP ☐ ___	CFM	☐ TONS ☐ BTU ☐ ___ X ___	☐ KW ☐ ___	☐ TONS ☐ BTU ☐ ___ X ___	Area or ___ Served	Weekdays						
1																
2																
3																
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6																
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30																
31																
		Totals														

BUILDING SURVEY – EQUIPMENT LIST

System: ______ Date: ______ Building/Location: ______

	EQUIPMENT			RATED OUTPUT			RATED INPUT			OPERATING SCHEDULE			OUTSIDE AIR CFM	HOURS PER WK SPACE IS OCCUPIED	NIGHT SET BACK?	REMARKS
	Description	Subsystem	Quantity	☐ HP ☐ ___ CFM		☐ TONS ☐ BTU ☐ ___ X ___	☐ KW ☐ ___	☐ TONS ☐ BTU ☐ ___ X ___	Area or ___ Served	Weekdays						
1																
2																
3																
4																
5																
6																
7																
8																
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27																
28																
29																
30																
31																
		Totals														

Honeywell

Commercial Building HVAC System Energy Upgrade Survey

Job Information

Job / Location: ______________________________

Contact / Telephone: ______________________________

Equipment

Type of Equipment: ☐ Packaged Rooftop ☐ Built-up AHU ☐ Other ____________

Manufacturer: ______________________________

Model ________ CFM ________ Date of mftg. (if known) ________

Heating: ________ BTU/KW ________ Stages

Cooling: ________ Tons ________ Stages

Existing Economizer? ☐ Yes ☐ No General Condition of Equipment ____________

Primary Controller

Stand-alone Space Thermostat (T7300, T87, T874 etc.) Model ______________________________

Stat/Equipment Modules (W973, W945, W7100 etc.) Model ______________________________

Existing night setback? ☐ Yes ☐ No Htg/Cooling stages ______________________________

Existing Economizer

☐ Yes ☐ No

Changeover Control(s)

☐ None

☐ Dry Bulb (Temp. Cntrlr.)

☐ Electronic Enthalpy (H705)

☐ Fixed or Manual Economizer)

☐ Mechanical Enthalpy (H205, etc.)

☐ Differential (H705 & C7400)

Economizer Controller

☐ Electro-Mechanical (W859, etc.)

☐ Electronic Enthalpy (W7459, etc.)

☐ Other ______________________________

Other Existing Sensors ☐ Return Air ☐ IAQ ____________ ☐ Other ____________

Existing Damper Motor(s)

☐ Foot Mount Quantity ________ Mftg./model ______________________________

Signal _____ Torque _____ Voltage _____ Date of mftg. (if known) _______

☐ Direct Coupled Quantity ________ Mftg./model ______________________________

Signal _____ Torque _____ Voltage _____ Date of mftg. (if known) _______

Existing Dampers

__________ Height x __________ Width General Condition ____________

☐ Parallel or ☐ Opposed Edge Seals ? ______________________________

Contractor

Surveyor ______________________________

Date ______________________________

Commercial Building HVAC System Energy Upgrade Survey

Job Information

Job / Location: ______________________________

Contact / Telephone: ______________________________

Equipment

Type of Equipment: ☐ Packaged Rooftop ☐ Built-up AHU ☐ Other ______________

Manufacturer: ______________________________

Model ____________ CFM ____________ Date of mftg. (if known) ________

Heating: ____________ BTU/KW ____________ Stages

Cooling: ____________ Tons ____________ Stages

Existing Economizer? ☐ Yes ☐ No General Condition of Equipment ______________

Primary Controller

Stand-alone Space Thermostat (T7300, T87, T874 etc.) Model ______________________

Stat/Equipment Modules (W973, W945, W7100 etc.) Model ______________________

Existing night setback? ☐ Yes ☐ No Htg/Cooling stages ______________________

Existing Economizer

☐ Yes ☐ No

Changeover Control(s)

☐ None

☐ Dry Bulb (Temp. Cntrlr.)

☐ Electronic Enthalpy (H705)

☐ Fixed or Manual Economizer)

☐ Mechanical Enthalpy (H205, etc.)

☐ Differential (H705 & C7400)

Economizer Controller

☐ Electro-Mechanical (W859, etc.)

☐ Electronic Enthalpy (W7459, etc.)

☐ Other ______________________

Other Existing Sensors ☐ Return Air ☐ IAQ ____________ ☐ Other ____________

Existing Damper Motor(s)

☐ Foot Mount Quantity ________ Mftg./model ______________________

Signal _____ Torque _____ Voltage _____ Date of mftg. (if known) _______

☐ Direct Coupled Quantity ________ Mftg./model ______________________

Signal _____ Torque _____ Voltage _____ Date of mftg. (if known) _______

Existing Dampers

__________ Height x __________ Width General Condition ______________

☐ Parallel or ☐ Opposed Edge Seals ? ______________________

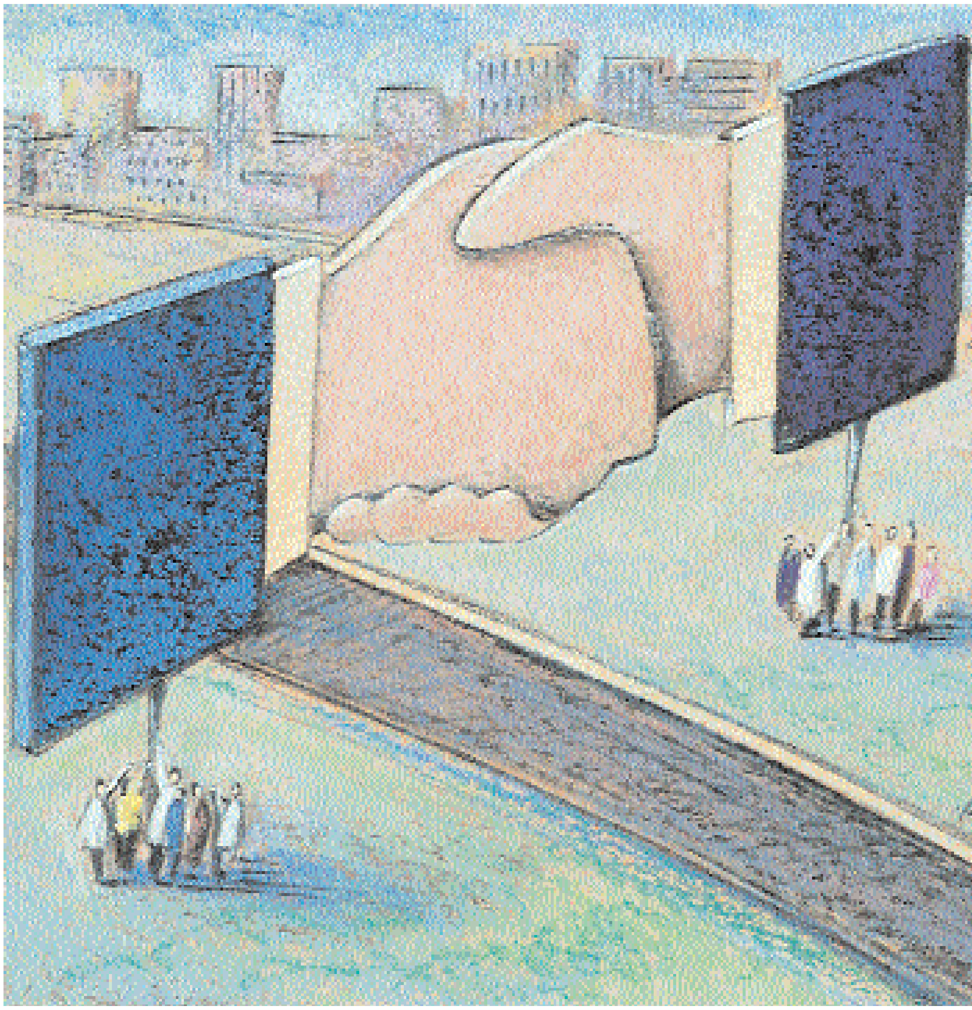
Contractor

Surveyor ______________________

Date ______________________

HONEYWELL:

A TRADITION OF COMFORT



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1-800-345-6770 ext. 435.

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