HVAC DESIGN FROM CLEAN SHEET TO BLUEPRINT

A MECHANICAL DESIGNER'S GUIDE TO SUCCESSFUL DESIGN OF SMALL COMMERCIAL AND INSTITUTIONAL HVAC SYSTEMS

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INTRODUCTION

This book is the distillation of 30 year's experience with HVAC design and construction, as a designer, as a supervisor, as an agency planner and reviewer, and finally, again, as a free lance HVAC designer of small commercial and institutional projects. My HVAC experience followed twenty years in the aerospace industry developing and testing military gas turbines and rocket engines.

The procedures and principles described here will be of value to technicians and graduate engineers with a firm background in thermodynamics and fluid flow – the bedrock fundamentals of HVAC processes. It will be of value to engineers and technicians who are just entering the HVAC design field as members of a consulting engineering firm, or who wish to work independently for local architects and owners designing HVAC systems for small offices, mercantile establishments, churches, and restaurants. The principles and procedures outlined apply to any size job, but specific "how to" instructions are for small projects of limited scope.

In most jurisdictions, a licensed professional engineer must sign and seal plans for submittal to a local building plans reviewer. However, in many engineering firms, much of the actual design work is performed by talented technicians or intern engineers under the supervision of a professional engineer.

This book will also be of interest to architects and licensed HVAC contractors who specialize in small commercial and institutional buildings, and who wish to understand the principles of the designs they contract to implement. In some jurisdictions, such as the State of Florida, licensed mechanical contractors are permitted to perform the design of HVAC systems that fall below specific thresholds of building occupancy and system size and cost. This book will be of use to contractors who wish to take advantage of that provision.

Applying the principles and methods outlined will help the designer avoid the problems that plague many projects with small budgets and unsophisticated owners. The most common of these are moisture and mildew problems exacerbated by code requirements or high density occupancy. On the other hand, these principles will also help control the costs both of design and construction, by guiding the designer to the most cost effective solution commensurate with local codes, indoor air quality, and reasonable energy efficiency.

END

Chapter 1

Procedure Outline - The Clean Sheet

Scope

This book is not intended to teach engineering fundamentals, but to help trained mechanical engineers and technicians understand and undertake the HVAC design of small commercial and institutional buildings. This chapter will outline the tasks that must be executed to arrive at a successful and cost-effective design. Cost-effective from the standpoint of the project cost, but also from the standpoint of the design effort. Clients for small building design, generally owners, architects, or contractors, have legitimate cost constraints, and designers who cannot work within those constraints will soon find themselves out of work.

Important Terms

The following terms will be used throughout this book. They have specific meanings in connection with HVAC systems, ventilation, and indoor air quality.

A **building** is a roofed and walled structure with controlled environment, built for human occupancy and use.

The **thermal envelope** of a building is the physical separation between the conditioned space and the unconditioned environment. It holds the primary insulation layer of the building where resistance to heat transfer is the greatest.

The **pressure envelope** is the primary air barrier of the building, which is sealed to provide the greatest resistance to air leakage from the unconditioned environment.

A **zone** is a group of spaces within the thermal and pressure envelopes which are served by a single air handling system.

A **sub-zone** is a group of spaces within a zone that may be served by a single terminal component such as a variable air volume unit.

A **czone** is a space or group of spaces within a zone having the same **occupancy category** – see Chapter 4.

An **occupancy category** is a designation that defines the activity and use of a space or group of spaces. Examples are office, auditorium, gymnasium, and mercantile.

A **space** is a single room, with or without a ceiling plenum.

A **room** is the part of a space bounded by walls and a ceiling that is usually routinely occupied and served by grilles and registers to supply and recirculate or exhaust conditioned air.

A **ceiling plenum** is a cavity within the pressure envelope that is above a room and that is formed by a dropped lay-in ceiling and floor or roof structure above. Room walls do not necessarily extend above the dropped ceiling to the structure above.

A **return air plenum** is a ceiling plenum with an unobstructed path to an air handler return, and that contains no flammable materials or surfaces.

Supply air is the all of the air delivered by the cooling/heating apparatus to the supply air diffusers in the zone.

Outdoor air, also called **ventilation air**, is air from outdoors that may be mixed with return air before passing into the cooling/heating apparatus, may be introduced to the apparatus directly before entering a zone, or in certain circumstances, may be introduced untempered into a zone.

Return air is the portion of the supply air that is recirculated after being collected by the return grilles in the zone.

Exhaust, or **exhaust air**, is the portion of the supply air that is discharged from the zone to outdoors after passing through the zone.

In general, the **thermal and pressure envelopes** must coincide. However, if they don't, they must be arranged so that outdoor air cannot leak into the space within the thermal envelope – remembering that the thermal envelope generally offers no resistance to air leakage.

Proceeding to the HVAC Blueprint, an Outline

The following outline follows the organization of the chapters in this book.

<u>Gathering Information</u> - This may be the most important part of the designer's job. Chapter 2 discusses the types of information that are needed, what to look for, and the responsibilities of the HVAC designer to review the data supplied and point out problems that may adversely affect the HVAC design or the building operation. The base sheet should be prepared (Chapter 11) as soon as the architect's floor plan is received.

<u>Preliminary Design</u> – Decisions are made about the types of systems to be used, building zoning, equipment locations, and routing of ductwork.

Establishing the Building Air Balance – The requirements for exhaust air and outdoor supply air for each zone must be established prior to calculating cooling and heating loads to ensure that the building remains under positive pressure at all times when occupied, and positive or neutral when unoccupied. Commercial kitchens and assembly occupancies present special problems. Chapter 4.

Estimating Peak Cooling and Heating Loads – In many ways the easiest task, the designer must nonetheless subdivide the building into zones and spaces, and calculate sensible and latent cooling loads and heating loads for each. Chapter 5 is a discussion of building heat gain and loss, and Chapter 6 discusses the cooling load estimating methods that are the most applicable to small commercial buildings.

Energy Efficient Design – Chapter 8. An outline of energy terminology and energy efficient design as prescribed by ASHRAE Standard 90.1 and suggested by good practice.

Selecting the Primary Equipment – This is the most complex task. The cooling equipment must be matched to the sensible and latent peak loads, while at the same time addressing part-load operation as it may affect indoor air quality and moisture problems. Chapter 7 discusses psychrometric considerations and Chapter 9 describes methods for selecting the cooling and heating equipment.

Determining the Optimum Distribution of Supply Air – In chapter 10, a method of determining the air supply to each space for each zone is described. The differences between variable air volume (VAV) and constant volume systems is touched on.

Determining a Control Strategy – Chapter 11. Discusses how all the equipment of a project should be tied together to accomplish the intent of the design.

Design Documents, The Blueprint – Chapter 12. The System Layout – Creating the drawings, schedules, and specifications that fully and accurately describe the HVAC design.

Design Documents – Instructions to the Contractor – Insuring that the design documents convey to the HVAC contractor what the designer wants by clarifying features shown on the drawings and described in the schedules and specifications.

Checking the Work – Quality Control – Chapter 13. Designers working alone are particularly vulnerable to errors and omissions. This Chapter discusses techniques that help avoid mistakes.

Additional useful references are: ASHRAE – *Air Conditioning Systems Design Manual*[1] and Bell – *HVAC Equations, Data, and Rules of Thumb*[2].

END

Chapter 2

Project Information Needed for HVAC Design

The Commission

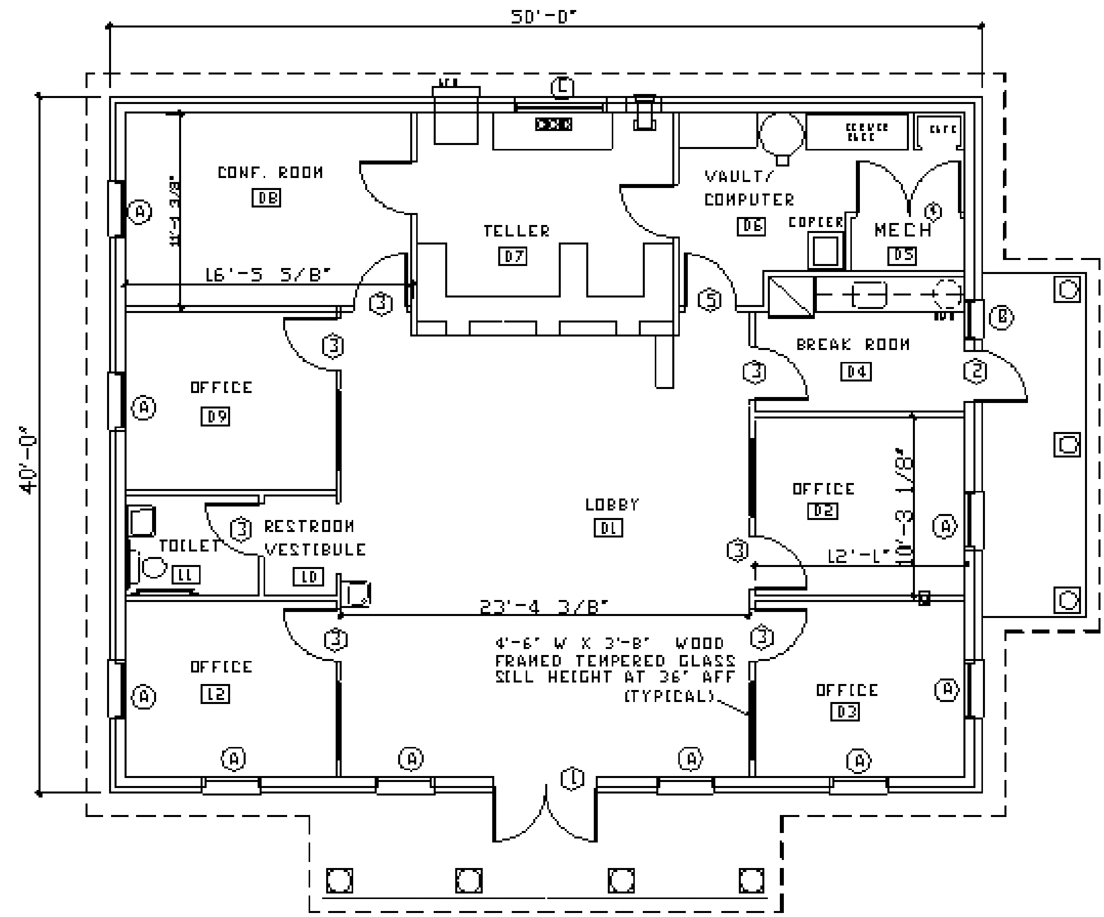
Generally, the projects that are in the scope of this book are initiated by an owner – a professional, merchant, or church group – soliciting a proposal from an architect, who then solicits proposals from his consultants. The mechanical engineer or designer who is responsible for the HVAC will often also be asked to design the plumbing system. However, plumbing design for small projects is largely prescriptive, meaning design parameters are fixed by the local plumbing code. For this reason, the project architect will sometimes assume this responsibility. Fuel gas and fire suppression systems require special training, and may be consigned to specialty engineers.

The architect will contact his consultants, including the HVAC designer, and give them a concept description of the project. The concept description will include as a minimum the building size, location, number of stories, and occupancy type. Other information, such as requirements for special equipment rooms, planned functions, budgetary restraints, etc., may also be provided. The process starts with a phone call (or e-mail message), but the architect may want to meet with the consultants, singly or together to discuss the proposed facility. From this information, the consultants will provide their fee proposals. When the owner is satisfied with the architect's proposal, he will give the architect a notice to proceed with the design. Only after this will the architect prepare the detailed documents needed by the consultants to perform their design work.

Documents

HVAC systems are necessarily applied to enclosed buildings occupied by people or by equipment requiring a controlled environment. The HVAC designer must have detailed information about a proposed building, which may be a stand-alone new building, an addition to an existing building, or renovation/remodeling of an existing building or space. The starting point for obtaining this information is the documents prepared by the architect and other professionals involved. Comprehensively, these documents are the architectural plans, electrical lighting plans, civil site plans, and specialty plans such as commercial kitchen layouts.

Architectural plans include all of the following: floor plans, elevations, wall sections, building sections, window and door schedules, and room schedule. Figures 2-1, 2-2, and 2-3 show a typical architectural plan with schedules, wall sections, and elevations. Floor plans (Figure 2-1) show the building orientation and layout, area of spaces and rooms, basic room use (office, classroom, restroom, etc), and location of windows and doors. Also needed are the locations of heat-producing equipment such

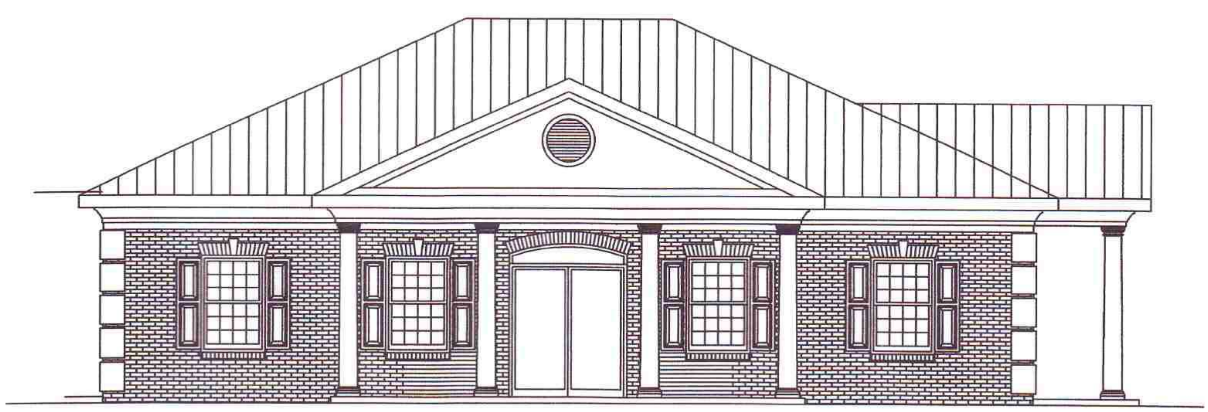


DOOR SCHEDULE					
MARK	TYPE	SIZE	MATERIAL	FINISH	LABEL
1	EXTERIOR STOREFRONT	34x84	ALUM	BRONZE	NONE
2	EXTERIOR PANEL	36x80	WOOD	WOOD STAIN	NONE
3	INTERIOR PANEL	36x80	WOOD	WOOD STAIN	NONE
4	INTERIOR LOUVER	36x80	WOOD	WOOD STAIN	NONE
5	INTERIOR PANEL	36x80	WOOD	WOOD STAIN	B

WINDOW SCHEDULE					
MARK	TYPE	SIZE	GLAZING	FRAME	COMMENTS
A	SINGLE HUNG	36x60	LOW E IG	ALUM	THERMAL BR
B	SINGLE HUNG	16x42	LOW E IG	ALUM	THERMAL BR
C	FIXED	60x36	LOW E IG	ALUM	THERMAL BR

FLOOR PLAN
1/8"=1'-0"
FIGURE 2-1

Architectural plans, elevations, and sections by Tock Ohazama, Architect, Tallahassee, Florida



FRONT ELEVATION (EAST)
1/4" = 1'-0"

Figure 2-2

as vending machines, copiers, and large servers. Floor plans also often include a furniture layout that will help the HVAC designer determine occupancy patterns. Roof plans may be needed for the HVAC designer to coordinate rooftop equipment, piping, and ductwork. Window and door schedules are keyed to the architectural plan and list windows and doors with size, type, and glazing. Room schedules show room wall and ceiling finishes, and ceiling heights for each numbered room.

Architect's plans may already include space for air conditioning air handlers, based on the architect's idea of how much space should be required. The allocated space may be inadequate, and the designer will need to coordinate this with the architect during preliminary design.

Elevations (Figure 2-2) show the relationship between various elements of the building, and can be used to find the size of window and door openings in lieu of a schedule. It is important that the designer have a complete definition of the glazing and fenestration, and this information may have to be requested directly from the architect during preliminary design.

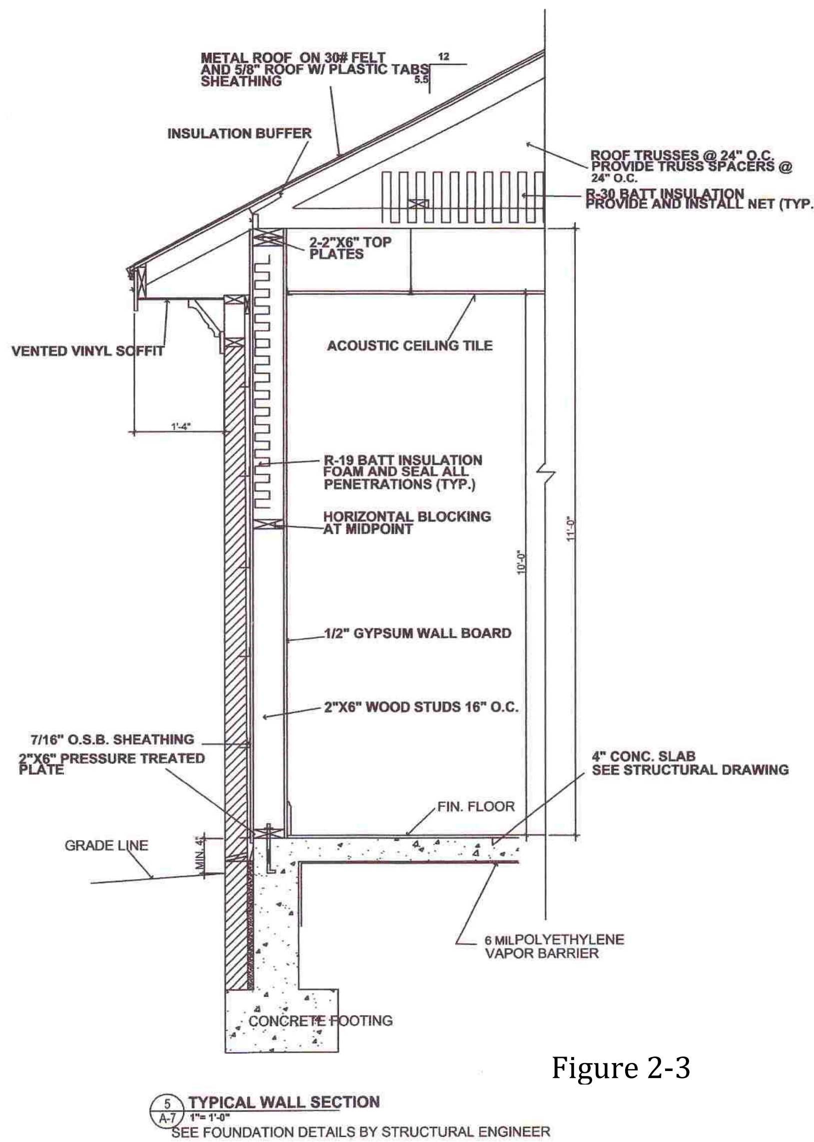


Figure 2-3

Wall sections (Figure 2-3), including sections through the building roof and floor assemblies, provide essential information about the building envelope – thermal conductivity and reflectivity of opaque materials, and location and thickness of building insulation. Window and door schedules give the size of building openings, and the type of window or door (i.e. insulated, storefront, etc).

Building sections such as Figure 2-4 show the designer how the roof-ceiling and floor-ceiling assemblies are constructed, and what clearances and routes w Figure 2-2 e for supply and return ducts or plenums. Combined with wall sections, they also show the boundaries of the thermal and pressure envelope of the building.

Electrical lighting plans will show the location and number of lighting fixtures, coded to an accompanying fixture schedule. With very simple jobs, especially when the HVAC designer and lighting designer have worked together before, the fixture plan and schedule may be dispensed with, and a lighting power density (watts/sq ft) substituted. The designer must also find out the electrical service voltage and phase, using the electrical power plan or directly from the electrical designer.

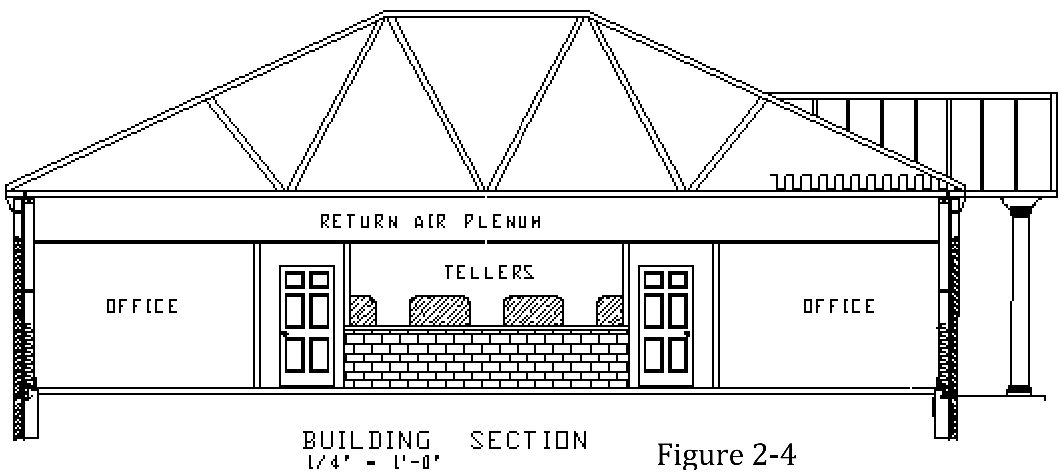


Figure 2-4

If a commercial kitchen is included in the project, the HVAC designer will need to size the grease hood ventilation equipment and the dishwasher and scullery exhaust in the context of the total building ventilation balance, and evaluate the cooling load of the cooking equipment. To do this requires a detailed kitchen equipment layout, with all equipment coded to a schedule describing each item and providing, especially, the fuel type and heat input to that item.

Civil site plans are needed for the designer to coordinate the location of exterior equipment , piping, and ductwork (except for rooftop), and to ensure code compliance with building air intakes and exhausts.

Data Inputs

It was mentioned above that the designer needs to know the location and heat dissipation of heat-producing equipment such as vending machines, copiers, and large servers. This data is often not included on floor plans, and must be solicited directly from the owner or building designer. Also needed is any special ventilation required, either by request of the owner or intrinsic to particular equipment.

Floor plans usually have a generic description of the use of each interior space, but no specific information about occupancy load. Often, standards such as ASHRAE 62 (Chapter 4) may be used to estimate occupancy load, but it is always best to confirm expected peak normal occupancy of each space with the owner or building designer. The HVAC designer must decide what sensible and latent loads to assign to occupants based on the space description and his experience. Tables in the ASHRAE Handbook – Fundamentals, will also give guidance on occupant cooling loads.

Restroom and shower areas are completely exhausted by code in Florida and most other jurisdictions, and thus do not contribute directly to heating or cooling loads. However, the exhaust is inevitably made up by outdoor air introduced into the building either across a cooling coil, by un-tempered mechanical supply, or by infiltration. Of course, it is therefore necessary for the HVAC designer and the HVAC contractor to know where restroom and shower plumbing fixtures are located. This information is usually found on the floor plan, but if it is not, must be solicited from the building designer. Exhaust requirements are discussed in Chapter 4.

The HVAC designer, in collaboration with the building designer and owner, establishes the indoor design conditions for the building – dry bulb temperature and relative humidity. However, outdoor design conditions must be based on local weather data. Weather data is available in many HVAC computer design programs, and in tabular form in the ASHRAE Handbook – Fundamentals. (Beginning with the 2005 Fundamentals detailed weather information for US and worldwide sites is available only on the CD, not in the printed Handbook.) HVAC designers often adjust this data based on local site conditions and personal experience.

Design weather data tables are available for most U.S. Cities, as well as for most cities of any size anywhere in the world. Chapter 6 discusses ways to use the data for calculating cooling, heating, and dehumidification loads.

Review of Information

Upon receipt of the documents and information needed as outlined above, it is the responsibility of the HVAC designer to review the submitted information in detail, and to notify the building designer of possible problem areas. Typical problems are gaps in the pressure envelope (allowing paths for tramp air to leak into the thermal envelope), large uninsulated or un-tinted glass areas, inadequate building insulation, inadequate space to run ductwork or install equipment, configurations that may violate the local mechanical or energy code, and conditions that may compromise indoor air quality.

Codes, Standards, and References

Familiarity with the following documents is essential to execute a successful design.

Local Building Codes : These are the basic codes that the local building officials will use to review proposed projects and inspect work in progress. Most codes are now available on CD through the local building department – examples are the Florida Unified Building Code, the Southern Standard Building Code, and the National Building Code. Each jurisdiction adopts a code that suits it, and often supplements the adopted code with special provisions. All codes include special sections for Buildings, Plumbing, Mechanical, Fuel Gas, and Energy Compliance. The designer should be familiar with the sections of the code applicable to his work, and should have a copy available for reference during the design process. HVAC design will be governed by the Building Code, the Mechanical Code, the Fuel Gas Code, and the Energy Code sections. Of course, the Building Code covers items generally of importance to the architect, but many Building Code sections have a direct effect on HVAC design, such as those pertaining to fire separation and means of egress.

<u>National Fire Protection Association (NFPA) Standards</u> : The HVAC designer will not need a complete set of NFPA standards. For one thing, many of the NFPA standards are incorporated into the local codes. However, a few of the standards should be available for reference by the designer, even though many of the requirements may be incorporated into local codes. These are NFPA 70 "National Electric Code", NFPA 90A and 90B which apply to the installation of HVAC systems, and NFPA 96 *Standard for Ventilation Control and Fire Protection of Commercial Cooking Operations*[3].

<u>Standards of the American Society of Heating, Refrigerating, and Air Conditioning Engineers (ASHRAE)</u> : The ASHRAE Handbook of Fundamentals[4] is referred to often in this book, and the designer should have the latest edition available for ready reference. Important standards to have available in the latest edition for ready reference are Standard 62.1 *Ventilation for Acceptable Indoor Air Quality*[5], Standard 90.1 *Energy Standard for Buildings Except Low Rise Residential Buildings*[6] and the Standard 62.1 *User's Manual*[7]*.* Standard 52.2[8] which defines air filter standards, Standard 55 "Thermal Environmental Conditions for Human Occupancy"[9],

All of the publications referenced are updated on three to five year cycles. The latest update at the time a project is started should be the one used for reference. The latest update at the time of publication of this book is shown in the references.

Summary

Before beginning any HVAC project, the designer must have a comprehensive set of documents and data that completely define the building and the local environment. This usually means a complete and final set of architectural drawings, lighting layouts, civil site plans, internal loads, occupancy patterns, and design points for the spaces and outdoors. In the absence of complete information, and under pressure to deliver a design, the designer may estimate many parameters based generic rules of thumb and on past experience with the building and lighting designers. However, this is risky, and could result in inadequate design, over design, or re-doing many design tasks.

The HVAC designer has the responsibility to review the documents and data for the project, and to inform the building designer of any problems he perceives.

The HVAC designer should maintain a design file stating all the assumptions made for a project, the rationale, and all correspondence with the other designers and with the owner.

END

Chapter 3

Preliminary Design – Planning

Decisions to be Made

Once all of the data about the proposed building project has been obtained and reviewed, the designer is ready to make some basic decisions about how the system will be configured. These decisions must be made in collaboration with the owner and the architect. The owner will have budgetary constraints and will want input into the type of systems to be used. The architect will expect the HVAC system to be compatible with his vision for the building, and will have to provide for spaces for placement of the equipment.

Preliminary design is an essential step in the design process, but does not need to be a detailed or drawn-out task and is generally not formalized for the projects that are the subject of this book. However, it provides an opportunity for the engineer to outline for the owner and architect his response to their requirements, and what he plans to include in his design.

This means that the designer must decide what ambient outdoor conditions to use. He must also decide how the building will be zoned and what kind of core refrigerating/heating equipment, air handlers, and terminal systems will be used before knowing what the building loads will be. Detailed selection of equipment is discussed in Chapter 9, after the cooling and heating loads have been estimated and the room and coil psychrometrics are known. However, the architect, engineer, and owner must agree on the general outline of the HVAC configuration to avoid errors as the project unfolds. This outline should be as detailed as possible and should be submitted to the architect in writing as a basis of design report. The design report should be updated regularly to reflect changes made as the project progresses.

The Design Day and Hour – Outdoor Ambient Design Conditions

Part of the project information is the precise location of the proposed building. The cooling and dehumidification loads will have to be based on either the day with the highest dry bulb temperature or the highest wet bulb temperature. For cooling load calculations, this book will use dry bulb temperature as the design standard. For selecting air pre-treatment systems (chapter 9), most authorities recommend using wet bulb temperature as the standard. The reason for this latter will be found in Chapter 9.

ASHRAE, among other sources has extensive weather data covering all of the United States and much of the rest of the world. A synopsis will be found in the 2009 Fundamentals handbook, Chapter 14, "Climatic Design Information". Complete tables for most U.S. and World Cities is found in the Fundamentals handbook CD, or is available separately from ASHRAE. The reference in handbook Chapter 14 includes a table of complete design conditions for a single location, Atlanta, Georgia, plus basic design conditions for all weather stations in the U.S. and most large world cities. These tables provide the statistical occurrence of the maximum values of the following parameters at .4%, 1%, and 2% of annual hours:

DB	maximum dry bulb temperature
MCWB	mean coincident wet bulb temperature (coincident with the maximum dry bulb temperature
WB	maximum wet bulb temperature
MCDB	mean coincident dry bulb temperature (coincident with WB)

For example, the maximum dry bulb temperature at .4% would be exceeded in only 35 hours of a typical year. In humid climates, the summer condition of WB/MCDB will usually be at a higher enthalpy than the condition of DB/MCWB. As will be discussed in Chapter 9, this is important when selecting outdoor air pre-treatment equipment.

First Cost and Energy Efficiency

The projects that are the subject of this book rarely have the resources for detailed cost-benefit analysis. The architect and engineer will therefore establish the building and systems design based on their experience and judgment, and their understanding of the owner's requirements and budget. More energy efficient systems are often more complex to design, so the fee that the owner is willing to pay will also affect the energy efficiency of the final product.

However, energy efficiency is an important national objective, so it is important for the designer to be familiar with the fundamentals of energy efficiency. Also, most states now have energy efficiency codes, and it is often the responsibility of the HVAC design professional to prepare the energy code compliance document. As such, it is the responsibility of the HVAC designer to not only insure the efficiency and code compliance of the HVAC system, but also to advise the architect and the lighting designer regarding the energy good practice and the energy code requirements of the building envelope, lighting, and equipment.

ASHRAE Standard 90.1 provides the prescriptive and performance standard for energy efficiency in commercial and high-rise residential buildings. Most local energy codes are based on this standard with differences tailored to local conditions or aspirations. The designer should have both Standard 90.1 and the local energy code available for ready reference, and should be familiar with local code sections that may be more restrictive than the Standard.

Elements of Design

Air conditioning systems can be divided into two classes: chilled water and direct expansion (dx). Heating systems come in five classes: steam, hot water, gas or oil, electric, and dx (heat pump). For small commercial systems, chilled water and steam systems are generally impractical. Dx air conditioning with hot water heat or with a gas or oil furnace may be cost effective, especially in cold climates. Electric heat is cost effective in climates with warm winters, such as south Florida. Heat pumps may be an effective solution in climates with mild winters having a limited number of very cold days.

The basic preliminary design decisions will be as follows:

zoning of the building

air source or water source units

split systems or packaged units

type of heating system – furnace, hot water, heat pump, electric

Matched air conditioning units of five tons or less are often designated by equipment manufacturers as "residential". However, such systems are frequently applied to small commercial and institutional buildings.

A typical building project may be designed with a variety of unitary systems which include multiple ducted split systems and packaged units, ductless split systems, gas unit heaters, computer room air conditioning units, etc.

Zoning

As defined in this book, each zone will require a separate air handler. In addition, sub-zones will each require a separate variable air volume terminal unit. Each air handler will require space inside the building unless it is part of a packaged outdoor unit. It is not good practice, and is a code violation in many jurisdictions, to locate air handlers in uninsulated attics or under floor crawl spaces. Zones do not have to be separated by internal walls.

Larger buildings can be zoned so that each air handler serves a different exposure, or a different occupancy type. Buildings with a total peak cooling load of ten tons or less will often be designed with only a single zone.

DX Air Source Systems

These systems are most likely to be chosen for cooling the small commercial systems that are the subject of this book. Air source refers to outdoor air as the source for heat or heat rejection, and air as the heat transfer medium to add or remove heat from the space. Refrigerant transfers the heat from the space to the outdoor air, or in heating mode, from the outdoor air to the space..

DX Water Source Systems

These systems may be an effective option for small buildings. Water source refers to water as the source for heating and cooling, with air as the heat transfer medium to add or remove heat from the space. This type of system has many advantages and is very efficient, but although the heat from the space is transferred to the source water by refrigerant, the source water itself requires a secondary heat source and heat sink.

The secondary heat source/sink will either be "boosted" or "ground water". "Boosted" means a cooling tower for heat sink and a boiler for heat source. "Ground water" means that the system heat source/sink is the ground. Either choice will have significant implications for cost and for space on the grounds outside the building, and so must be coordinated with the owner and architect early in the design phase.

Ground water source systems are of two types, "closed loop" and "open loop". With a closed loop system, the source water is circulated through pipes buried in the grounds outside the building. With an open loop system, the water is pumped directly from the water source – a well into the aquifer, or a pickup in a nearby water body – and then discharged either to a surface water body or back into the local aquifer. Ground water source systems will require coordination with the Civil engineer and may entail special permitting by local water management and utility officials.

Split System or Packaged

Matched split system air conditioners and heat pumps are available in sizes from 1.5 to 25 tons. Packaged systems, with the evaporator and condenser in the same housing, can be used where inside space is unavailable, and can be used on the ground outside, or on the roof. In either case, the architect must provide the space in the context of the interior layout or the visual impact on the building elevations.

Heating Options

Heating may be by electricity, as is dx cooling, but is often by a different fuel source such as natural gas. In south Florida, straight electric heat will have the lowest life cycle cost because it is vastly lower in first cost, and will be activated very little during the life of the building.

Further north, heat pumps are cost effective. However, heat pumps require auxiliary electric heat strips for two reasons:

> first, the outdoor coil must be defrosted periodically, and while this is occurring, the heat pump is delivering refrigerated air to the occupied space. This cold air will cause occupant discomfort if not offset by the operation of the electric heat strips.

second, on very cold days, the heat pump may not be able to heat the occupied space to the heating set point. Usually, if the space temperature falls more than 3° below set point, the electric heat will activate to maintain comfortable conditions, although this option can be disabled if the owner wishes.

Even when operating in heating mode on a mildly cool day, heat pumps will circulate air as cool as 80° to 85°. Air circulated at this temperature will feel uncomfortably cool to many occupants, especially older people. Therefore, heating systems capable of producing warmer air supply temperatures – on the order of 95°, are more suitable for buildings where the occupants will consist largely of senior citizens, or for buildings that are occupied on a 24 hour basis. Such systems are electric strips, circulating hot water, and gas furnaces. High capacity electric strips that occupants can activate as "emergency heat" are sometimes used in conjunction with heat pumps to meet higher circulating temperature requirements.

Consultation With Other Disciplines

During the preliminary design phase, the HVAC designer should advise the architect and lighting designer on the following energy points:

Explain to the architect and lighting designer that reducing lighting power densities below the Standard 90.1 maximums will have a profound effect on the size of the HVAC systems, besides reducing energy costs for the owner.

Review the architectural design and advise the architect of tramp air sources such as gaps in the pressure and thermal envelopes at eves.

Review the proposed building insulation for code compliance, good practice, and ac unit size reduction.

Check that the thermal envelope either coincides with or is inside of the pressure envelope. Notify the designer of problems found.

Advise the building designer regarding glazing – low e, insulated, tinted. Large glass surfaces can cause occupant discomfort if radiation from the glass is not mitigated by multi-pane insulation (winter) and tinting (summer).

Summary

Preliminary design and planning should not be a time-consuming or complex task. Its purpose is to coordinate with the owner and architect to ensure that the HVAC system meets their requirements and can be incorporated into the building and site. It will also help the engineer as the project is developed.

END

Chapter 4

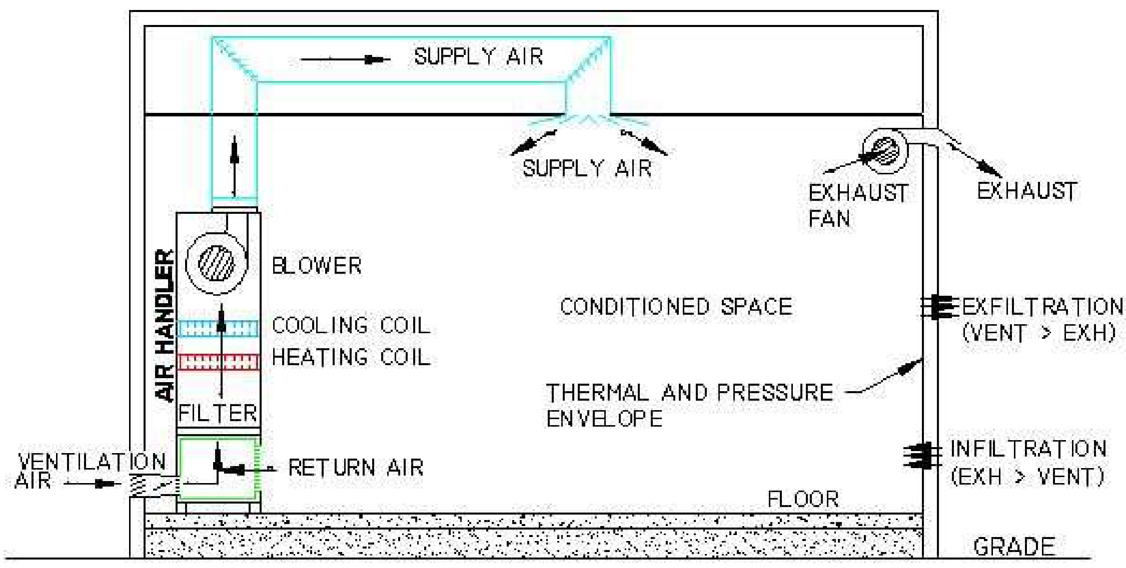
Balancing Outdoor Air and Exhaust Air

Scope

Prior to performing heat loss and heat gain calculations, the minimum required outdoor air to be delivered to the air handling systems must be determined. This is generally determined by local codes, or by good practice where local codes are silent or inadequate. Good practice can be determined by reference to ASHRAE standards, which are also the basis of most code requirements.

What is Ventilation?

Ventilation is the introduction of outdoor air into a conditioned zone after passing through cooling/heating apparatus either mixed with return air or independently. To understand this requires a brief refresher of the process of mechanically heating and cooling a building.



CONDITIONED SPACE - AIR MOTION AND TRANSFER

Figure 4-1

Figure 4-1 illustrates the motion of air within a zone consisting of a single conditioned space, and the exchange of air between the zone and outside (shown) or between the zone and an adjacent zone within the building. During the summer, cool, dry **supply air** passes through the room and picks up heat and moisture. Most of this air is returned to the air handler where it may be mixed with **ventilation air** before passing over a cooling coil to be cooled and dehumidified. Some of the **supply air** is **exhausted** directly from the zone. Air from outdoors or adjacent spaces may **infiltrate** if the rate of ventilation air flow is less than the rate of exhaust air flow. If the ventilation air flow is greater, then the excess air will leak out of the zone, or **exfiltrate.**

During the winter, **supply air** warmed by the heating coil carries heat into the room to replace the heat lost through the building envelope (or ductwork) to outdoors. Otherwise, the air motion and exchange are identical with the cooling mode.

Mathematically, these relationships, shown on Figure 4-1, can be stated as follows:

	supply air – exhaust air = return air + leakage	**(4-1a)**
	ventilation + return air = supply air	**(4-1b)**
therefore	**ventilation + return air – exhaust air = return air + leakage**	**(4-1c)**
and	**ventilation – exhaust air = leakage**	**(4-1d)**
therefore	**if ventilation > exhaust air then leakage is positive (exfiltration)**	**(4-1e)**
	if ventilation < exhaust air the leakage is negative (infiltration)	**(4-1f)**

Infiltration is always undesirable, but it is particularly so in summer, when outside air may be laden with moisture. **Infiltration** does not pass through a cooling coil before entering the room, and so adds directly to the room cooling and dehumidifying loads, and therefore to the **supply air** flow which must remove the room heat and moisture. **Ventilation**, on the other hand, passes through a cooling and dehumidifying coil before being introduced into the room, allowing control of room relative humidity without increasing supply air flow.

Principles of Ventilation

For any **space** within a **zone**, the conditioned air supplied to that zone must equal the air removed by **return**, **exhaust**, and **leakage** (equation **4-1a**). If **exhaust** from the **space** is greater than **supply**, then the **return** will be made up by **leakage** into the space from adjacent spaces. This can occur even if the **zone** ventilation air exceeds the exhaust, and there is no infiltration into the **zone.** This principle also applies to two zones within a building, because each air handler can only supply as much air as is returned to it.

In order to control the zone relative humidity, ventilation air should always be set greater than exhaust so that leakage is always out of the zone, and the zone is thus at greater pressure than outdoors. This principle also applies to spaces within a zone, so that air can be transferred between spaces by adjusting the exhaust proportionately between "downstream" spaces.

There are codes and standards for minimum <u>required</u> ventilation and exhaust. These are dependent on the use and occupancy of the pertinent spaces in the project, and will be explained in more detail presently.

Thus, a building may be divided into different occupancies, each with specific requirements for ventilation and exhaust. For example, building codes prohibit air from restrooms and janitor closets from entering occupied rooms. Another example, explained in detail later in this chapter, is commercial kitchens, where heat and odors from the kitchen must not be carried into the dining area. The direction of air flow within a building or zone can be controlled by regulating the ratio of supply to exhaust for each space within the zone. Exhaust from a space can reduce the pressure in that space relative to other spaces in the zone, while maintaining the entire zone at a positive pressure relative to outdoors.

Basic Rules

In any climate, but most particularly in humid climates such as the Southeast, all buildings must be under positive internal pressure when the cooling systems are operating in order to minimize infiltration. There is no standard for the degree of positive pressurization, but in general, an excess of 25% of outdoor ventilation air over exhaust air is considered good practice.

$$Qoav = 1.25 \times Qexh \qquad (4\text{-}2)$$

An exception to this general rule would be if there is a very small amount of minimum required exhaust air, coupled with a relatively small requirement for ventilation air, as may be the case for a small office, or in a zone with no exhaust required. In such a case, outdoor air could exceed exhaust by multiple factors, since the excess air would readily leak out of the building. This would be up to the designer's judgment. Another exception would be if the amount of exhaust is very large, in which case the ratio of outdoor air to exhaust could be reduced, but never below 1.2. Exhaust in excess of that required by code is only required to limit pressure loading on exterior doors. In order to limit exterior door pressure loading to five pounds or so, pressure differential should not exceed about .05 inches of water.

There are two ways to determine the minimum ventilation air that must be mechanically supplied to the building. One is by adding the required exhausts and multiplying by 1.25 The other is by determining the minimum "Ventilation for Acceptable Indoor Air Quality" as calculated based on use and occupant load using ASHRAE Standard 62.1. The designer should make sure that he or she has the latest issue, since that standard continuously evolves. Both methods must be calculated, and the minimum ventilation air required will be the larger of the two.

The sum of ventilation, exhaust, and leakage must of course equal zero. (equation **4-1d**) Figure 4-2 is a simplified representation because the windward side of a building may experience infiltration while the downwind side has outward leakage. By pressurizing the building with an excess of induced ventilation over mechanical exhaust, infiltration is minimized if not eliminated. Also, the possibility of excess internal pressure causing problems such as heavy door loads is remote, because the excess ventilation air will never exceed .5 air changes per hour (ACH), which would be a tight building.

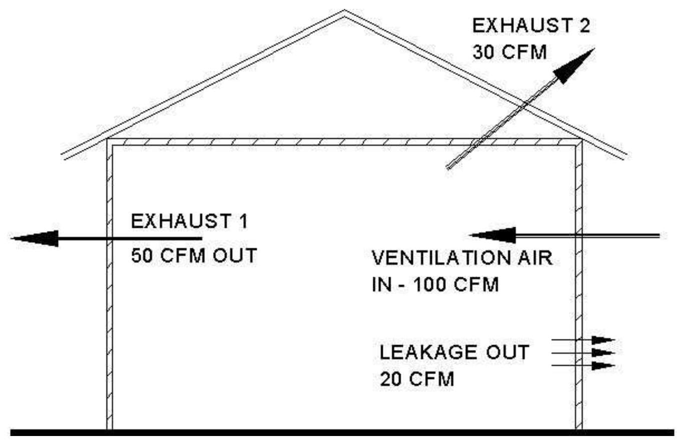


Figure 4-2, Building Air Balance
Mechanical Ventilation and Exhaust

Ventilation Based on Required Exhaust

All local codes have requirements for minimum exhaust from restrooms, lockers, janitor closets, scullery areas, and kitchen fume/grease hoods. Examples from the Florida Uniform Building Code:

Locker Rooms	.5 cfm/sf
Shower Rooms	20 cfm/shower head (continuous)
Toilet Rooms	50 cfm per water closet or urinal
Non-commercial kitchen	100 cfm (intermittent)

Commercial kitchens are a special category that are discussed below. If there is no commercial kitchen, then the minimum outdoor air based on required exhaust is found by adding the total of required exhausts and multiplying by 1.25.

Commercial Kitchens

Commercial cooking appliances are required by NFPA standard 96 and by most local codes to be equipped with a hood. The mechanical designer is usually charged with specifying the hood and showing details on the mechanical plans. This is because the requirements for exhaust and make-up air of the kitchen hood system must be integrated into the ventilation air balance of the entire building.

A commercial kitchen includes at least three exhaust air zones, each of which may have it's own exhaust systems. These are: the cooking lineup with NFPA96 grease hood, the scullery with dishwasher and general exhaust, and a bakery with general and bread warmer exhausts. It is important that odors from kitchen operations not be carried into the dining area or other areas of the building, so the kitchen pressure must be negative with respect to the rest of the building. At the same time, the kitchen pressure should be positive with respect to outdoors to avoid unwanted infiltration of moisture laden outdoor air.

FIGURE 4-3, TYPICAL RESTAURANT VENTILATION AIR BALANCE

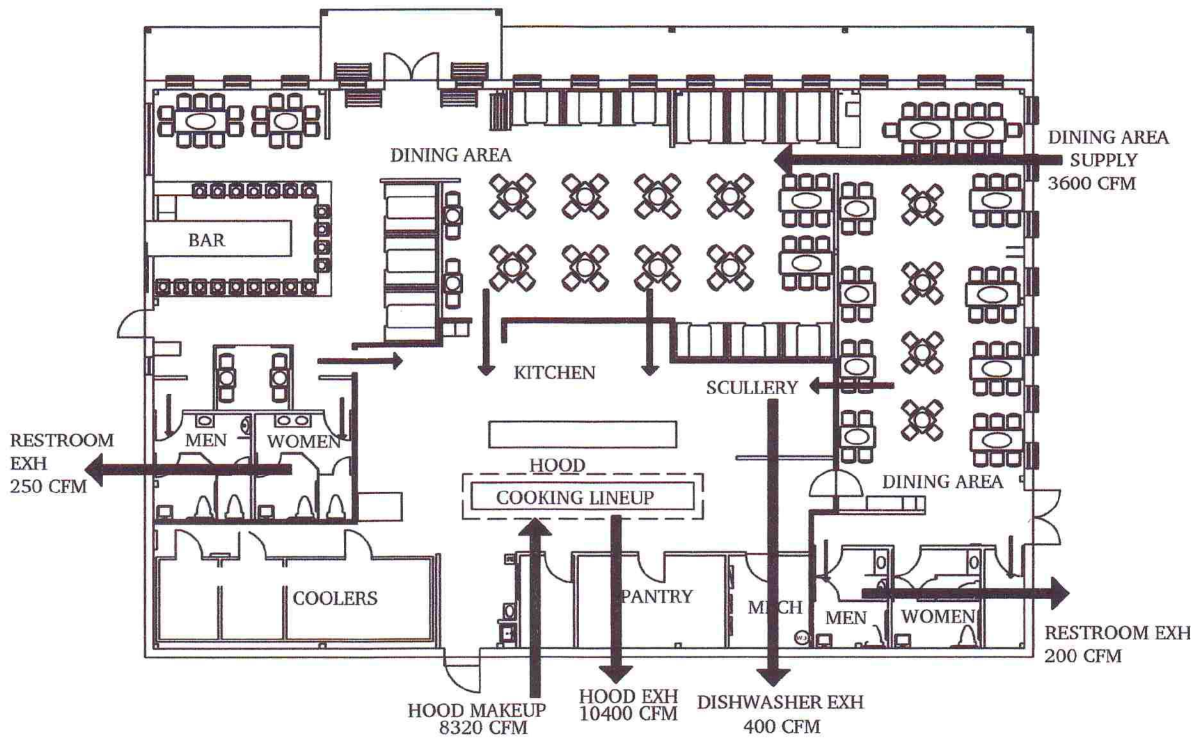


Figure 4-3 is a typical restaurant layout, with the ventilation air balance shown by the large dark arrows. Separate air handling systems are required for the kitchen/food service area and for the rest of the building, which includes the dining areas, bar, and restrooms. The kitchen exhaust consists of the cooking hood and the scullery/dishwasher exhaust. In this case, the kitchen hood is "compensating", meaning that outdoor air to make up a portion of the hood exhaust is introduced integrally through the hood system. The total kitchen exhaust is thus the net hood exhaust plus the scullery exhaust.

$$C\ kit\ exh = C\ hood\ exh - C\ hood\ sup + C\ dw\ exh \qquad (4\text{-}3)$$

The air handler serving the kitchen should have no outdoor air. The kitchen exhaust, plus the minimum restroom exhaust, must be made up entirely by outdoor air introduced through the air handler(s) serving the rest of the building.

This case shows the outdoor air requirement controlled by the net building exhaust. This is because the outdoor air requirement based on ASHRAE standard 62 (occupant load) is less than 1.2 times the total exhaust from the building. The large dark arrows on Figure 4-2 show the ventilation air balance, expressed mathematically as follows:

$$C\ vent = C\ kit\ exh + C\ rest\ exh + C\ leakage \qquad (4\text{-}4)$$

so $C\ leakage = 3600 – 2480 – 450 = 670$ cfm (23% of exhaust)

Note that this air balance also closely satisfies equation 4-2:

$$Cvent = 1.25 \times Cexh$$
$$3600 = 1.23 \times (2480 + 450)$$

Part of the air drawn into the dining area through the dining area air handler(s) is exhausted through the two restrooms. The remainder, as indicated by the small dark arrows, is allowed to flow freely into the kitchen through serving passageways, architectural openings, and transfer ducts, there to be exhausted by the hood and dishwasher fans. Rules for internal transfer of air within a building, such as through restroom doors, or for kitchen makeup, are discussed later in this chapter.

The excess ventilation air causes the building to be under positive pressure relative to outdoor ambient, and introducing the air only into the dining area ensures that the dining area will have positive pressure relative to the kitchen and scullery. The desired pressure relations between the dining area, the kitchen and the outdoor ambient must be maintained under all conditions of operation – for example, when the kitchen hood or scullery exhaust is not operating. Special control features may be necessary to maintain pressure differentials, and these will be covered in Chapter 11.

<u>Special System Exhaust Air Flow Rates</u>

Although local codes establish minimum exhaust air flow rates for many occupancy categories, there may be special process exhausts within a zone for which the air flow rate is established by the need to capture fumes, aerosols such as grease, VOCs (volatile organic compounds), or heat. For example, kitchen hood minimum exhaust is established by each hood manufacturer as a function of the hood type, style, size, and cooking surface temperature. These parameters are defined in the kitchen hood design standard, NFPA 96[3].

In general, the HVAC designer will often be given the required exhaust and maximum make-up supply by the manufacturer's representative of the equipment to be installed, whether it is a kitchen grease hood, manicure station, gun cleaning station or laboratory hood. Guidance for many process exhausts may also be found in the <u>Industrial Ventilation Manual</u>[10] in terms of required capture velocities.

Ventilation Based on Occupancy and Use

Up until the 1970's, while it was considered good practice to ensure positive pressurization of buildings, this was usually limited to ensuring that only enough outdoor air was introduced into the air handling systems to slightly offset exhaust. Some government and institutional agencies had outdoor air requirements tied to occupancy. A few quantified excess air, such the Navy Facilities Command, which specified that outdoor air exceed exhaust by 20% in humid climates.

The energy crisis of the 1970's resulted in codes tightening the envelopes of commercial buildings to reduce infiltration and thus save energy. This reduced the de-facto dilution of contaminants in buildings, and resulted in widespread problems of "sick building syndrome" – where a combination of volatile organic compounds (VOC's) and bio-aerosols (mold and bacteria) reached concentrations in a building's air that caused discomfort and distress to many occupants.

ASHRAE addressed this problem with Standard 62.1, "Ventilation for Acceptable Indoor Air Quality" originally released in 1973. This standard was adopted by most local codes, and has undergone significant changes since its original release. The most recent release as of this writing is 62.1-2010[5]. The stated purpose of the standard, set forth in paragraph 1.1 is "to specify minimum ventilation rates and indoor air quality that will be acceptable to human occupants and are intended to minimize the potential for adverse health effects." The following outline of Standard 62.1 procedure is for illustration only.

Applying Standard 62.1, the designer has the option of using a prescriptive approach or an indoor air quality (IAQ) approach. The prescriptive method specifies minimum "breathing zone" ventilation rates per occupant for various activities, and adds minimum ventilation rates per square foot for various types of spaces. To apply the IAQ approach, the designer may use lower ventilation rates but must show that the levels of indoor air contaminants are held below recommended limits. Most designers using this manual will elect the prescriptive approach, because the resources available for small commercial building design are usually inadequate for the IAQ approach.

OCCUPANCY CATEGORY	Rp	Ra	Dod
Table 1	cfm/person	cfm/sqft	#/1000 sqft
Office Buildings			
office space	5	0.06	5
reception areas	5	0.06	30
telephone/ data entry	5	0.06	60
main entry lobbies	5	0.06	10
Public Assembly Spaces			
auditorium seating area	5	0.06	150
courtrooms	5	0.06	120
libraries	5	0.12	10
museums/galleries	7.5	0.06	40

Examples of prescriptive ventilation rates required for the “breathing zone” (V_{bz}) as found in Standard 62.1 are shown on Table 1 above. The basic formula for the prescriptive ventilation rate is:

$$V_{bz} = Rp^*(\#\ \text{occupants}) + Ra^*\text{Area} \qquad (4\text{-}5)$$

where V_{bz} = required minimum ventilation rate, unadjusted

The Dod column is “default occupant density”. This is the occupant density that must be used unless the designer can verify that a different density is applicable to a particular project.

Actual outdoor air required for the zone air handler must be adjusted to account for **air distribution configuration**, for **multiple occupancy categories** within a zone, for **population diversity** and for **ventilation efficiency.** (Note that in Standard 62.1, the term “system” is used to denote the spaces served by a single air handler, and is equivalent to the term “zone” used in this book. The term “zone” as used in the Standard, means a particular occupancy category within a system served by a single air handler. That will be termed **czone** in this book)

Air Distribution Configuration

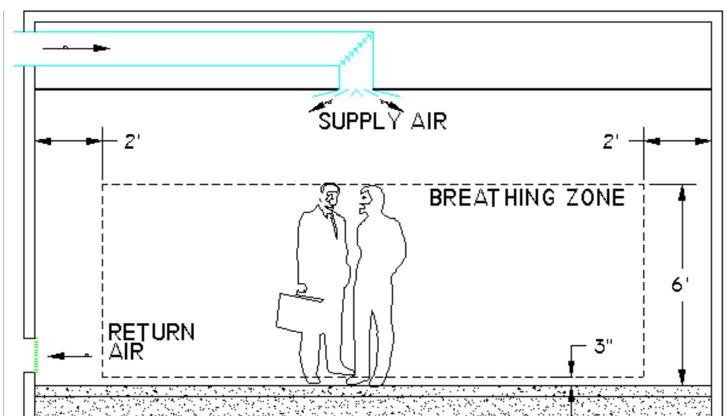


FIGURE 4-4. BREATHING ZONE
EZ = 1 (CEILING SUPPLY AND FLOOR RETURN)

Figure 4-4 is a graphic representation of the “breathing zone” as defined in Standard 62.1. Also shown on Figure 4-4 is one of the ten **air distribution configurations** described in the standard. The parameter $\mathbf{E_z}$ is the “**Czone Air Distribution Effectiveness**”. Applying $\mathbf{E_z}$ results in the Czone Outdoor Air Flow (V_{oz}), modifying V_{bz} as follows”

$$V_{oz} = V_{bz} / E_z \qquad (4\text{-}6)$$

The configuration shown, ceiling supply and floor return of both cooling and heating supply air, results in E_z of 1.0. Other common values of E_z are:

ceiling supply and ceiling return of cool and warm air,	$E_z = .8$
floor supply and floor return of cool and warm air,	$E_z = 1.0$
floor supply and ceiling return of cool and warm air	$E_z = .7$

Equation **4-6** is the ventilation air required for an air handler serving a zone with only one occupancy category, as may be the case for an assembly occupancy such as an auditorium with a dedicated system.

Multiple Occupancy Categories

Most air handler systems will serve spaces with different occupancy categories – called zones in the Standard. For example, an office system may include a conference room, a reception area, and six offices. The Standard designates this as a **Multiple Czone Recirculating System.** The procedure for computing a required minimum ventilation for these systems is intended to provide the needed ventilation to each czone without unnecessary energy use and moisture loads by excessive ventilation. An example calculation will be carried out for the office system cited above. E_z is assumed to be .8, denoting ceiling supply and return.

Diversity Factor and Ventilation Efficiency

The conference room may be designed for a capacity of eight, the reception for six, and each office for one each. However, when the offices are occupied, it may be assumed that the conference room is unoccupied, and it may also be assumed that the normal occupancy of the reception area will only be the receptionist. The Standard allows the designer to compute a **diversity factor** , D, defined as

$$D = P_s/\Sigma P_z \qquad (4\text{-}7)$$

where P_s = The normal population of the zone = 7
P_z = The maximum expected population of a czone
ΣP_z = 17 (three offices unoccupied, conf room occupied, reception full)

(Again, "zone" as used in the Standard means a particular occupancy category within a system served by an air handler.) For the example cited, the diversity factor will be

$$D = 7/17 = .41 \qquad (4\text{-}7ex)$$

In order to compute **ventilation efficiency**, it is necessary to first compute the **primary outdoor air fraction** (Z_p) for each czone.

$$Z_p = V_{oz} / V_{pz} \qquad (4\text{-}8)$$

where V_{oz} is from equation **4-6**
and V_{pz} is the czone primary air flow

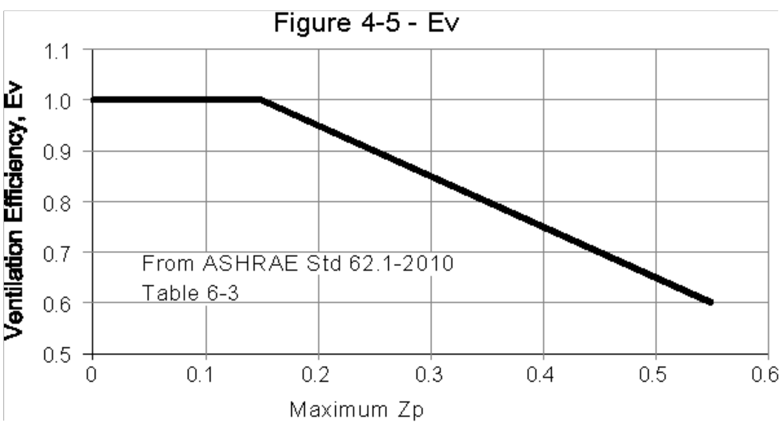
At the beginning of this chapter, it was stated that the required minimum ventilation must be computed for each air handler zone before cooling and heating loads can be calculated. Yet here is a situation where the supply air flow to each czone must be known in order to calculate the required ventilation.

In this case a preliminary load calculation will be needed to estimate the supply air flow (called the primary air flow in the Standard) for each czone. Since in general, the envelope and internal parameters will be known at this point in a project, it is only necessary to estimate a ventilation rate and a primary supply air flow to the zone in order to proceed with a preliminary load calculation. An estimated sensible room load must be calculated for each czone, and an estimated total air flow based on the zone. The preliminary load calculation is beyond the scope of this chapter, but let us assume that it has been performed for the example, with the following results:

Table 2

czone	Pz	Az	Ez	Voz (Eq 4-6)	Vpz	Zp (Eq 4-8)
offices	6	600	0.8	83	450	0.18
reception	6	400	0.8	68	300	0.23
conference	8	315	0.8	74	270	0.27

Ventilation efficiency can now be found using Table 6-3 in Standard 62.1, or by reading from a graphical version of Table 6-3 as shown in Figure 4-5.



From Figure 4-5 we read E_v as .88 for max Z_p of .27 (from Table 2). the ventilation efficiency is applied to the **uncorrected outdoor air intake, V_{ou},** which is calculated using the **Diversity Factor** from equation **4-7.**

$$V_{ou} = D^*\Sigma(R_p{}^*P_z) + \Sigma R_a{}^*A_z \qquad \textbf{(4-9)}$$

for the example, using values from equation **4-7ex** and Tables 1 and 2:

$$V_{ou} = .41^*(5^*6+5^*6+5^*8) + .06^*600+.06^*400+.06^*315 \qquad \textbf{(4-9ex)}$$

$$V_{ou} = 120 \text{ cfm}$$

We're still not done, because we now must modify V_{ou} using E_v to obtain the final minimum design outdoor air intake flow rate, V_{ot}.

$$V_{ot} = V_{ou} / E_v \qquad \textbf{(4-10)}$$

or for the example $V_{ot} = 120/.88 = \underline{136\ cfm}$ (4-10ex)

Earlier it was noted that the "normal" occupancy of the example zone is seven, while the peak occupancy would be 17. So the V_{ot} of 136 cfm is 19 cfm per occupant during normal occupancy, and 8 cfm per occupant during peak occupancy. If equation **4-5** were applied to each czone (V_{oz} in Table 2), and the results added, the total minimum ventilation air would be 225 cfm, or 13 cfm per occupant. Thus, applying the method of Multiple Occupancy Category reduces the ventilation requirement by nearly 40% relative to simply using the ventilation requirements of Table 1.

Applying Standard 62.1

The preceding exercise is intended to acquaint the reader with the fundamentals of applying Standard 62.1. Unless the local code or controlling authority is more restrictive than Standard 62.1, the designer should study and follow the standard to insure a comfortable building and to avoid liability. It is beyond the scope of this course to give comprehensive instructions on application of the Standard, which will vary from project to project. Such details may be found in ASHRAE publications, such as the Standard 62.1 User's Manual[7] which includes a CD with an Excel spreadsheet that can be used to calculate V_{ot}.

Proper application of Standard 62.1 will result in outdoor air rates of 6 to 20 cfm per occupant. In general, rates for assembly buildings such as auditoriums will be on the low end of this range, while rates for classrooms will be near the high end.

Variable Air Volume (VAV) Systems

Variable volume systems are rarely specified for small commercial buildings, because of cost and complexity. Systems for these buildings are usually direct expansion, and special controls such as hot gas bypass are needed to allow these systems to operate properly with large variations in air across the evaporator. However, the designer needs to be familiar with VAV technology and design, so the topic is touched on in this book.

VAV systems have a virtually infinite number of operating points because the air supply is modulated to each sub-zone to follow the sensible cooling load. The largest amount of ventilation air will be required when the "critical" sub-zone is at its lowest load (air flow), because a low sub-zone air flow increases max Z_p – equation (6) - which in return results in a reduced ventilation efficiency (Figure 5). Reference 2 provides methodology to compute the required ventilation for VAV systems.

Another issue with VAV systems is maintaining the minimum required ventilation as zone loads fall and supply air flow is reduced. If the outdoor air is induced by the air handling system, it will decrease as the air demand on the system decreases when cooling load decreases below the design point. To prevent outdoor air from falling below code requirements, and to prevent constant exhaust from causing negative pressurization, a constant, positive outdoor air supply must be provided. To do this requires measuring the outdoor air flow and using the measurement to modulate an outdoor air supply fan or to modulate a damper to throttle the air handler return ahead of the outdoor air intake.

Air Transfer Rules

When outdoor air is brought into a zone, it may be necessary to transfer it to an exhaust zone such as a restroom or kitchen. Adequate free area must be provided across doors and walls to allow unrestricted motion of the zone air into the exhaust area. For restrooms, this is done by undercutting the door, by providing a transfer grille in the door or wall, or by providing a transfer grilles and ducts in the ceiling. Two factors determine which of these choices applies. First, free area must be large enough that the air velocity through the opening never exceeds 500 feet per minute.

The required free area is calculated using the formula:

$$A_f = C/V$$

A_f = free area in square feet
C = air flow rate in cfm
V = air flow velocity in feet per minute

So for a three foot door, a 3/4 inch undercut can only handle 90 cfm of transfer air. Transfer grilles can be selected based on free area to meet this requirement for higher air flows.

The second factor influencing the choice of transfer methods is the fire rating of the wall that the air must travel through or over. Transfer grilles cannot be installed in doors located in fire rated walls. Any solution, while still meeting the 500 fpm maximum velocity requirement, will also have to include a fire damper in the air path at the plane of the fire rated surface.

Introducing Ventilation Air Into the Zone

The discussion so far has assumed that the ventilation air will be induced by the air handler, as shown on figure 1. Actually, this is only one of several ways to bring ventilation air into the space. Others are

forced induction using a damper in the air handler return

dedicated outdoor air (DOA) unit discharging into the air handler return

DOA unit discharging into the zone

energy recovery ventilator that passes all of the zone exhaust through an enthalpy wheel to cool and dehumidify (or heat) incoming ventilation air which is then discharged into the air handler return

Direct Induction

Direct induction, as by Figure 4-1, depends on the pressure in the air handler return being low enough, and the intake louvers and ducts being large enough to admit the required amount of outdoor air. The control damper is located in the outdoor air intake duct, and the flow rate is set by a test and balance technician with the system operating at design supply air flow. The designer sizes the intake louvers and ducts for velocities of <u>less than 500 feet per minute</u> at the required ventilation flow rate.

Forced Induction

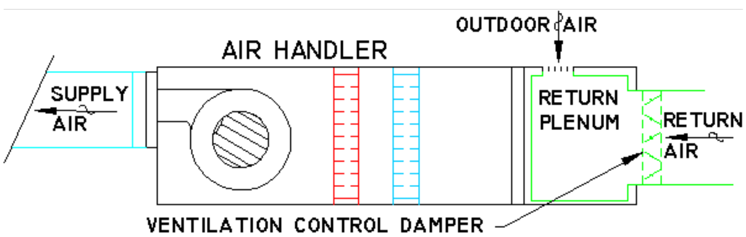


FIGURE 4-6 - FORCED OA INDUCTION

If the outdoor air is a significant percentage – say more than 30% - of the total supply air flow, or if it is impractical to make the ventilation air intake large enough to allow unforced induction, a damper placed in the return ahead of the fresh air intake can be used to drop the pressure as needed. Figure 4-6 shows a typical arrangement. As the damper is closed, the pressure in the return plenum is dropped, forcing the induction of more outdoor air. Of course, in order to maintain return air, the air handler blower speed must be increased, and if the ventilation requirement is constant, the return and outdoor air flow would be set up during test and balance. If the system is variable air volume, then the damper can be modulated by an air flow sensor to maintain constant minimum ventilation air flow, as required by code.

Dedicated Outdoor Air Units

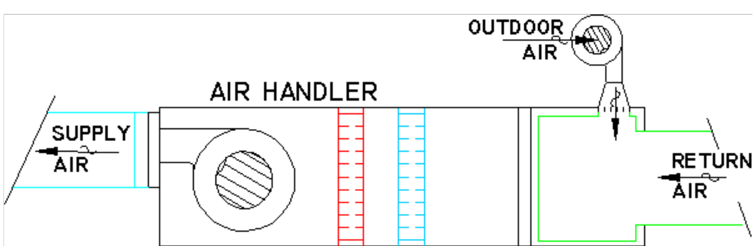


FIGURE 4-7 - DOA BLOWER

Ventilation air can be injected directly into the air handler return by a dedicated outdoor air (DOA) unit either pre-conditioned or un-tempered. An example of an un-tempered DOA unit would be a blower, similar to the system shown in Figure 4-7. In this case the system cooling or heating system must handle the ventilation load in addition to the zone loads, just as in the case of induced ventilation air. The advantage is that the ventilation air flow rate can be set precisely and the air handler blower external static pressure is not adversely affected as in the case of forced induction.

Instead of a simple blower, as shown on Figure 4-7, the DOA unit may have refrigeration, heating, and hot gas reheat to allow the DOA unit to handle all or part of the heating and cooling load of the ventilation air. In some cases, the DOA unit would have sufficient refrigeration for cooling and dehumidifying the ventilation air, and the zone air handler would have cooling capacity sized only for the zone cooling load and heating capacity sized to handle the ventilation heating load as well as the zone heating load.

Ventilation air from a DOA can also be ducted into the zone and discharged through diffusers without passing through the zone air handler. In this case the zone air handler is sized only for the zone loads and the DOA unit must have refrigeration and heat to allow introducing the air into the zone at near the zone design conditions. This method has the disadvantage of requiring a complete second ductwork system, so would be used only if the building configuration or other problems make it impractical to introduce the ventilation into the air handler return.

Energy Recovery Ventilator

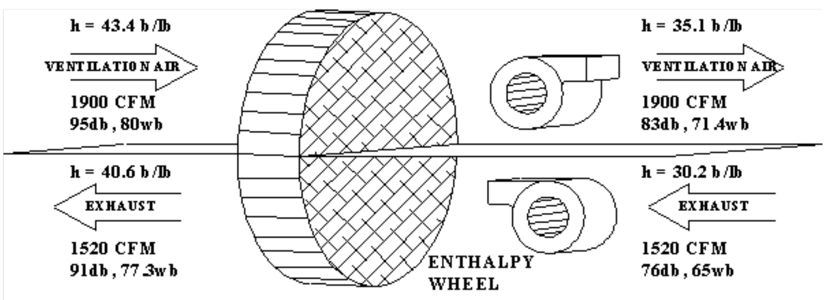


FIGURE 4-8, ENERGY RECOVERY VENTILATOR
crossflow neglected, casing and filters not shown

An energy recovery ventilator consists of a slowly rotating enthalpy wheel and two blowers, one for ventilation air and one for exhaust. The enthalpy wheel transfers heat and moisture from the ventilation air to the exhaust, as shown on Figure 4-8. In cooling mode (shown), the ventilation air is cooled and dehumidified by transferring its heat and moisture to the cool, dry exhaust. The unit shown thus reduces the outdoor air load on the air handler by nearly 80%. Of course, in order to do this, all or most of the non-hazardous exhaust from the zone must be collected and ducted to the exhaust intake of the ERV. The ventilation air discharge is ducted to blend with the return of the zone air handler.

Figure 4-9 shows a typical layout with an ERV tempering the ventilation air for a zone air handler. Ventilation air is induced from outside by the ERV ventilation blower. The zone exhaust, after passing over the ERV enthalpy wheel, is exhausted to outside by the ERV exhaust blower. The ERV in the system shown is larger than the air handler, and this is often the case. ERV's are large, and the designer must ensure that there will be adequate space for installation and for ductwork connections to the air handler.

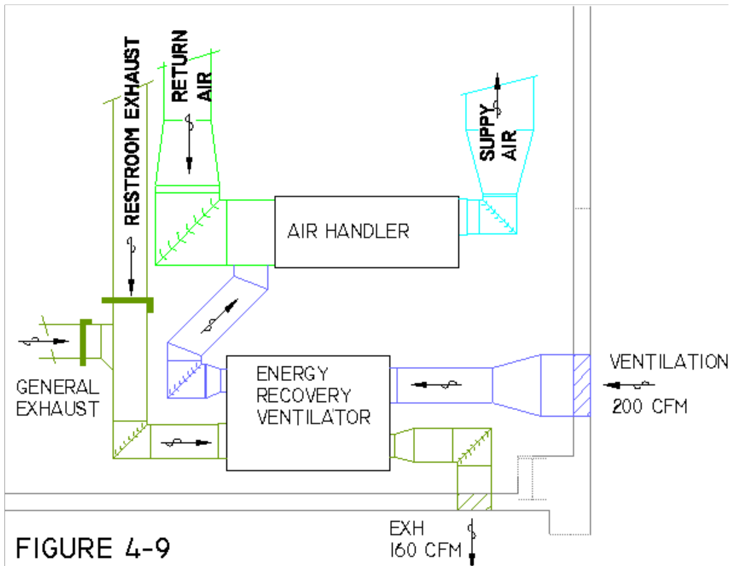


FIGURE 4-9
TYPICAL ERV LAYOUT

Maintenance

Although the subject of this book is design, a brief discussion of maintenance and the responsibility of the owner is justified here. In a misguided effort to save energy, owners sometimes will defeat design elements intended to avoid problems of excessive moisture in their building. This may include sealing the ventilation air intakes and shutting down ventilation pre-treatment systems. (Chapter 9) During the design process and before the final design is submitted, the HVAC professional should report on the moisture control elements of the design, and the importance of maintaining those elements throughout the life of the project. This report should be submitted to the owner, either directly or through the architect.

Summary

Prior to performing heat loss and heat gain calculations, the minimum required outdoor air must be determined. This is generally resolved by local codes, or by good practice where local codes are silent or inadequate. Minimum outdoor air will be the larger of the outdoor air required to offset required exhausts or to meet air quality standards required by local codes or ASHRAE Standard 62.1.

In general, outdoor air must exceed exhaust by a factor of about 1.25 in order to pressurize the building and mitigate infiltration. In some cases, particularly if the building has a commercial kitchen, required exhausts will be large enough that the required offset outdoor air will exceed the outdoor air required to maintain air quality standards. If air quality outdoor air is larger than the required exhaust offset, then supplemental exhaust may be required to prevent over-pressurization of the building.

To make these determinations, the designer should perform an air balance showing all required exhausts, required outdoor air, and supplemental exhaust or outdoor air as needed to provide pressurization. If the building has a commercial kitchen, then air balances should be performed for all expected operational configurations of kitchen exhaust fans. Special controls may be needed to prevent over pressurization or negative pressure. The pressure in the dining areas of a restaurant must always exceed the pressure in the kitchen, and the kitchen pressure must always exceed outside ambient pressure.

END

Chapter 5

Heat Loss and Heat Gain

Scope

This chapter describes the elements that cause a building to lose and gain heat, and how the designer can evaluate these elements. Heat gain includes sensible heat, which serves only to raise the temperature of the space and contents, and latent heat, which is in the form of water vapor introduced into the space by occupants, processes, or outdoor air. Latent heat must be condensed by the air conditioning equipment before it can be removed from the space. Heat loss, of interest only to calculate winter heating loads, consists only of sensible heat that serves to lower the temperature of the space and contents. The term "space" means an occupied enclosed volume that is heated and cooled, and will consist of a room and ceiling plenum, if any. A "room" is the space requiring comfort conditioning which is enclosed by floor, walls, and ceiling . A "plenum" is a cavity between the ceiling and the thermal/pressure envelope which may or not be used for air return to the conditioning equipment. These and other important terms can be found defined in Chapter 1.

In general, the thermal and pressure envelopes must coincide. However, if they don't, then they must be arranged so that outdoor air cannot leak through the thermal envelope to affect the occupied space. That is, if the pressure and thermal envelopes are separated – not contiguous – then the pressure envelope must be outboard of the thermal envelope.

Heat Gain and Heat Loss Elements

Whether performing a hand calculation, or using a computer program, the designer must be familiar with all of the elements that cause heat gain to a space. These are listed below, and covered in some detail in subsequent sections.

Heat Gain

Envelope elements: walls, roof, floor, perimeter slab, windows, skylights.

Internal loads: people, appliances, lights, electric motors

Unscheduled outdoor air intake: building infiltration, duct leakage

Outdoor ventilation air

Heat Loss

Envelope elements: walls, roof, floor, perimeter slab, windows, skylights.

Unscheduled outdoor air intake: building infiltration, duct leakage

Outdoor ventilation air

Obviously, internal heat loads do not contribute to building heat loss, but actually offset losses by other elements.

Envelope Elements – walls, roof, and floor

Heat is transferred through all envelope elements by three thermodynamic mechanisms that depend on temperature difference: radiation, convection, and conduction. On sunny summer days walls, roofs, skylights, and windows gain heat by solar radiation, either direct or reflected from below – ground, water, pavement, etc. Heat is also gained by convection, or transfer to or from ambient air in direct contact with the surface. Surfaces in direct sunlight may become warmer than the surrounding air and lose some of the radiant heat from the sunlight to the air, even on hot days. The combination of air and radiant solar heat warms the material of the surface, which then transfers heat from the exterior surface to the interior surface primarily by conduction through the materials of the wall, roof, or glazing. The heat thus transferred is then transferred to the space and its contents by radiation and convection. These processes are reversed on winter nights.

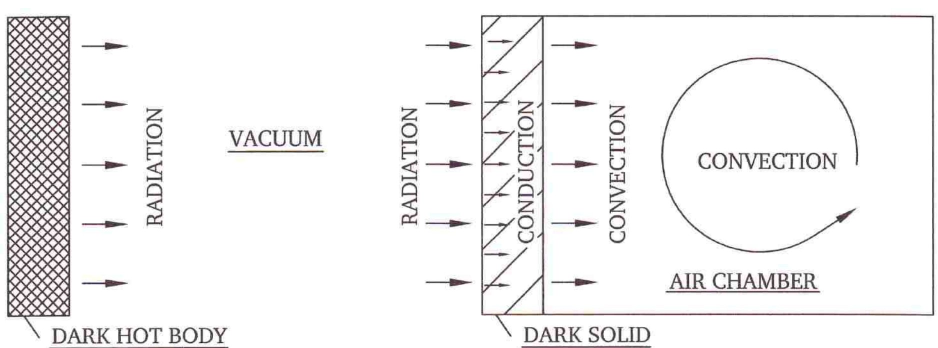


FIGURE 5-1, HEAT TRANSFER PROCESSES
Radiation from a hot body to a solid wall, conduction through the wall, convection from the wall to the air chamber, and distribution of the heat throughout the air chamber primarily by convection

Heat transfer between the exterior and interior of opaque elements is controlled by two constituents, insulation and exterior surface color. Surface color controls radiation, with dark surfaces absorbing more radiation than they reflect, making the surface warmer and increasing the temperature differential. Light surfaces reflect as much or more radiation than they absorb, keeping the surface cooler and decreasing the temperature differential. A light colored or reflective surface, on the other hand, being cooler, may actually increase convective heat transfer to a surface by increasing the temperature difference between the air and the surface itself.

Insulation works by reducing the rate of conductive heat transfer. Most building insulation includes an impermeable reflecting surface to reduce radiant heat transfer across cavities in the wall or roof. This works to reduce both heat gain and heat loss. The position of the impermeable surface can have an important effect on moisture control, since it is important that the impermeable surface dew point be higher than that of the air contacting it. So, for example, in winter, the air in a heated space will be warm, with a dew point probably in the 60's. If the impermeable membrane is on the exterior side of the insulation, it will be near the outdoor temperature, which may be much lower than the indoor dew point. The warm interior air can easily pass through the insulation and be in contact with the membrane, where it will condense, causing moisture problems. Thus, the impermeable membrane is normally on the conditioned side of the insulation. This is discussed in more detail in Chapter 12.

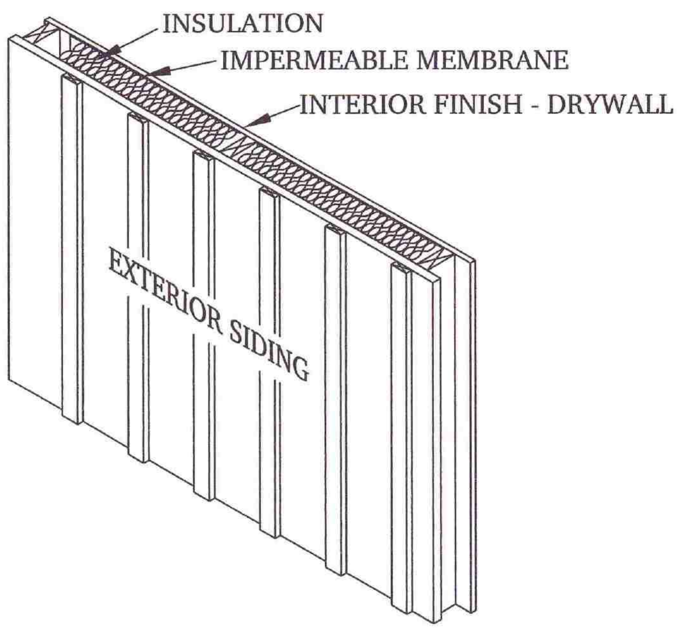


FIGURE 5-2 – INSULATED WALL

In warm, humid climates, such as the southern Gulf and Atlantic states, summer dew points are often in the high 70s. Faced insulation may therefore sometimes experience condensation on the cold room-side surface, especially in occupancies where the zone temperature is kept in the low 70s, such as laboratories, and data centers. The solution is to use un-faced insulation, sometimes called "friction fit" because it doesn't have the facing that can be nailed to the studs. This allows moisture that finds its way into the wall cavity to freely pass into the room, to be removed by the cooling apparatus.

Envelope Elements – Fenestration

The term "fenestration" is defined in ASHRAE fundamentals as "glazing, framing, and in some cases, shading devices and insect screens". Fenestration is therefore windows and skylights, and any associated shading. Heat transfer through windows is essentially the same as through opaque surfaces, except that conduction is generally much lower. Solar radiation, both reflected and direct, passes readily through clear glass and adds heat to interior surfaces such as floors and furniture. At night, heat from the interior may be lost by radiation through the clear glass to a dark sky.

Heat transfer through glazing is controlled by insulating glass, reflective coatings, low emissivity coatings (low e), and shading. Insulating glass will be two or three panes of glazing, separated by a small sealed gap filled with air or an inert gas.

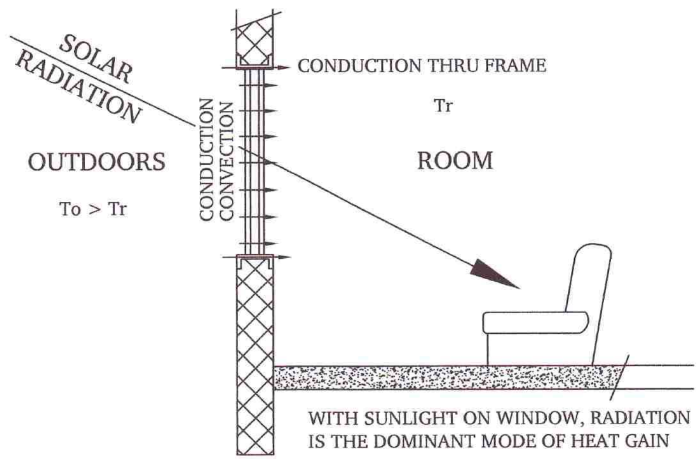


FIGURE 5-3 – HEAT GAIN THROUGH WINDOW

Important factors influencing heat transfer across fenestration are the materials and construction of the window frame. Aluminum window frames readily conduct heat across the frame, unless constructed with thermal breaks, which are low conduction materials separating the exterior and interior parts of the frame. Wood frames will generally allow less heat transfer than either aluminum or plastic frames.

Exterior shading has the effect of blocking direct solar radiation from all or part of the fenestration. Interior shading is more complex, since the radiation heat is already in the space, so is only blocked from radiant heating of interior surfaces and contents. At any rate, it is the recommendation of this writer that interior shading be ignored, since it can usually be easily removed.

Interior Loads

Interior loads consist of people, appliances and computers, lighting, and electric motors.

Occupants of a space release both sensible and latent heat in quantities dependent on age, gender, and activity level. The best reference for occupant heat release is ASHRAE *Fundamentals*.

Appliances include copiers, vending machines, freezers and refrigerators, cooking appliances, computing equipment, tanning beds, water heaters, and performance grade audio equipment. Cooking equipment releases both sensible and latent heat, but when under a hood, latent heat may be ignored and sensible heat released to the room will be significantly reduced. The heat added to a room by appliances may be available from manufacturer's data, or can be estimated from tables in ASHRAE Fundamentals.

Lighting heat is primarily from lamps. Fluorescent and HID lighting also emit heat from the ballasts, and in the case of HID fixtures, this can be significant. When the space has a dropped ceiling with a cavity above that serves as a plenum for return air, some of the heat of recessed fixtures will be released into the plenum, where it will serve to raise the return air temperature without adding to the room sensible heat gain. Lamp and lighting heat can be estimated using manufacturer's data and referring to ASHRAE fundamentals.

<u>Unscheduled Outdoor Air Intake</u>

Unscheduled air enters the conditioned space because of building infiltration and duct leakage. Building infiltration occurs because of doors and windows being opened by the occupants, and by leakage of building structure, including closed windows and doors. Window and door openings are random events influenced by the type of building and its use, and can be mitigated by education of the occupants and controls such as door closers. Leakage through the building envelope is best addressed during construction, and is mitigated during operation by pressurizing the building with outdoor air that is tempered and dehumidified by the air conditioning equipment, as described in Chapters 4 and 7.

The air entering an air handler must equal the air leaving the air handler, absent leaks in the air handler itself. So if ducts are run in the conditioned space, duct leaks will not influence the heat gained or lost by the air handler zone, although they may upset the air distribution desired by the designer. If ducts run outside the conditioned space, then return air leaks will raise the effective amount of outdoor air, increasing the load on the conditioning equipment. Supply air leaks will increase building infiltration, because the room air will be less than required by the return. The only way to prevent duct leakage is to seal all duct seams and terminal connections at the time of construction.

<u>Outdoor ventilation air</u>

This is air that is deliberately brought into the system to meet the requirements discussed in Chapter 4. Because it is designed into the system, its effect on air conditioning and heating loads can be included in the equipment selection process described in Chapter 9.

END

Chapter 6

Cooling and Heating Load Calculations

Scope

This chapter does not provide details of how to calculate cooling and heating loads for buildings. Calculation methods are covered in detail in the ASHRAE Fundamentals Handbooks. They are outlined here, with some comment. The designer should study these methods, and use judgment to settle upon the one best suited to the type and scale of the projects to be designed.

Basic Principles

When starting the heating and cooling load calculations, it's important to remember that this is much more than merely inputting some parameters, obtaining an output from a hand calculation or computer program, and then selecting equipment to match the total loads. A simple heat gain program may not take into account the capabilities of commercially available equipment, so after performing the initial load calculations, the designer must analyze the results as described in Chapter 9 to decide what type and size of equipment will best meet the requirements.

Likewise, the calculated instantaneous heat gain does not reflect the actual cooling load. This is because the building elements – walls, roof, floor, and furnishings must be heated to the average ambient temperature before releasing that heat into the occupied space. Even after the energy source is removed – the sun goes down, for example – the heat it conveyed will continue to appear as cooling load. For this reason, all of the various cooling load calculation methods attempt to account for this delay. Figure 6-1 illustrates the connection between heat gain and cooling load.

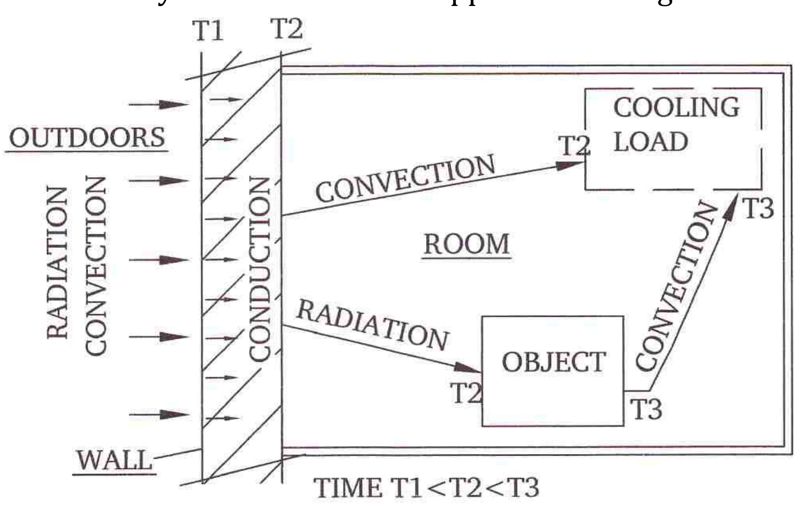


FIGURE 6-1 – HEAT GAIN VS COOLING LOAD

Once the heat absorbed by the exterior walls or roof reaches the interior surface, part of it is retransmitted to the space elements - walls, ceiling, floor, furniture – as radiation and part is transferred to the air in the space by convection. This principle, that sensible heat gain consists of radiation and convection fractions, is a fundamental principle of cooling load determination. The radiation fraction does not immediately appear as cooling load, but the convection fraction does. Likewise, internal sensible loads – lights, appliances, and occupants – have radiation and convection fractions, with the effect of the radiation fraction delayed for a time after the heat source is activated – lights and appliances turned on, occupants entering the space. All latent heat gain, and all ventilation or infiltration heat gain, is seen instantly as cooling load.

Fenestration (window assemblies) convey heat into the space more intuitively, by direct radiation on interior elements, as well as by convection of outside air to the fenestration surfaces, conduction through the fenestration material, and convection of indoor air contacting the fenestration surfaces. See Figure 5-3. Little of the heat conveyed to the interior by fenestration is absorbed by the glass.

Calculation Procedures

Over the years, ASHRAE has published details of a number of different cooling and heating load calculation procedures. These are, in approximate increasing order of complexity, the Cooling Load Temperature Difference/Cooling Load Factor (CLTD/CLF) method, the Total Equivalent Temperature Difference/Time Averaging (TETD/TA) method, the Radiant Time Series (RTS) Method, the Transfer Function Method (TFM), and the Heat Balance (HB) method. All of these methods, with the exception of the CLTD/CLF method, require a computer program.

The 2009 ASHRAE Fundamentals[4] describes only the HB and RTS calculation methods. Thus, these are the methods considered most up-to-date, and preclude many of the subjective decisions required by earlier, simpler methods, such as CLTD/CLF. To quote the 2009 Handbook, "Although the TFM, TETD/TA, and CLTD/CLF procedures are not republished in this chapter, those methods are not invalidated or discredited. Experienced engineers have successfully used them in millions of buildings around the world."

The basic principles of heat loss and heat gain were presented in the previous chapter, and details of cooling and heating load calculation procedures are not within the scope of this chapter. Also, the emphasis will be on cooling load. This is because cooling loads are highest during the day, when buildings are occupied and internal heat gains are in play. Cooling systems can cause serious moisture problems if over sized – especially in humid climates. Heating loads are much simpler to calculate, because they peak during night hours when a building is unoccupied. Also, there is no particular penalty for oversizing heating equipment, because moisture and mildew will not be induced by heating equipment as can happen with over sized cooling systems. (More about this in Chapters 7 and 11.) In fact, heating systems are often deliberately over sized to allow fast warm-up when temperature is cut back to save energy during unoccupied hours, or heating is shut off, as sometimes is the case in southern climates.

Cooling loads are estimated for a month, day, and time of day when the peak load is expected to occur. Weather data for a particular month is assumed to have the same ambient temperature and humidity profile as the 21st day of the month. The designer must use judgment and experience to select the hour of the day when the peak load is expected. It may be necessary to repeat the load estimate for other hours and months in order to find the peak design point. Also, building zones may peak at different hours or even months when zoned by exposure (E, W, etc.).

Finally, in order to select the cooling equipment and to set zone air flow rates, sensible and latent loads must be calculated separately for the occupied zone – usually referred to as room sensible and latent loads, and for the total load on the air conditioning cooling coil, referred to as total (or coil) sensible and latent load. Room load is only the cooling load of the occupied room. It does not include heat gain from outdoor air ventilation or heat gain to a ceiling plenum. It is based strictly on the cooling load that is removed from a room by the air circulating through the space. Other loads not included in room load, but contributing to the coil cooling load are fan heat added ahead of the coil and return duct leakage from an unconditioned space such as an attic.

In this chapter, only the CLTD/CLF and RTS methods will be commented on because the former is the simplest and easiest to use, and the latter is the most up-to-date method that would normally be feasible to apply to small commercial, industrial, and institutional projects.

The CLTD/CLF Method

This method is easily calculated by hand or computer spread sheet, with reference to tables available in ASHRAE Handbooks, ACCA (Air Conditioning Contractors of America) Manual N[11] and other publications that may be found on the internet. The specific ASHRAE handbooks with detailed descriptions of the CLTD/CLF method, including relevant tables, are 1997, 1993, 1989, 1985, 1981, 1977, 1973, and 1969. A copy of one of these is sometimes offered on E-Bay. ACCA Manual N may be ordered from the ACCA web site.

There are several computer programs available that use the CLTD/CLF method. All have the required tables built in, and include weather data and solar data for cities around the world. These programs may be found and purchased on the web.

Programming of the method on Excel or Lotus 123 is relatively straightforward, especially for a local area, where only the relevant portions of the tables need be included. Internal loads are, in general, treated as instantaneous, although there are cooling load factors for lights, equipment, and people to account for the radiation fraction and the time it takes for this to appear as cooling load. However, the Handbook recommends that these heat gains be used with a CLF of 1.0 unless the indoor temperature is maintained for 24 hours – that is, the system is neither shut down nor re-set during unoccupied hours.

Cooling Load Temperature Difference (CLTD) is the temperature difference between the indoor temperature and the effective outdoor temperature, also known as the Sol-

Air temperature. To quote the 1985 ASHRAE Handbook: “Sol-Air Temperature is the temperature of the outdoor air that, in the absence of all radiation exchanges, gives the same rate of heat entry into the surface as would exist with the actual combination of incident solar radiation, radiant energy exchange with the sky and other outdoor surroundings, and convective heat exchange with the outdoor air.”

CLF (Cooling Load Factor) is a dimensionless factor to account for the time delay of the radiation fraction becoming cooling load.

CLTD is presented in tables in the Handbook (or other sources) as a function of solar time and construction type. The tables are computed for the following boundary conditions:

indoor air temperature	78°F
maximum outdoor air temperature	95°F
outdoor mean temperature	85°F
outdoor daily range	21°F
latitude	40°N
day and month	July 21

This method is defined by the following equations:

walls and roofs

$$Q = U * A * CLTD_{CORR} \quad (6\text{-}1)$$
$$CLTD_{CORR} = \{(CLTD + LM) * K + (78\text{-}tr) + (to\text{-}85)\} \quad (6\text{-}2)$$

where Q is space cooling load, Btu/hr
A is surface area of transfer surface, ft^2
U is overall coefficient of heat transfer, Btu/hr/ft^2/Δ˚F
CLTD is obtained from a table as a function of solar time and construction type, ˚F
LM is the correction for month and latitude, from a table, ˚F
K is a factor accounting for surface color
tr is room temperature, ˚F
to is average ambient outdoor temperature, ˚F

solar insolation through fenestration:

$$Q = A * SC * SHGF * CLF + U * A * CLTD_f \quad (6\text{-}3)$$

where SC is shading coefficient
SHGF is solar heat gain factor –from a table as a function of latitude, month, and orientation
CLF Cooling Load Factor – from a table as a function of time of day, orientation and construction weight
U overall heat transfer coefficient of the fenestration

$CLTD_f$ effective temperature difference for fenestration, from a table

partitions, ceilings, floor:

$$Q = U^* A * TD \tag{6-4}$$

where TD is design temperature difference between spaces

internal heat gain from lights, appliances, motors, etc. (designer must account for mitigation of appliance loads due to kitchen hood):

$$Q = W * 3.413 * CLF \text{ or } = q_{INPUT} * CLF \tag{6-5}$$

where W is watts input of lights, appliance, or motor.
CLF is cooling load factor from a table as a function of total hours of operation and hours since turned on.
q_{INPUT} is Btu/hr input of gas appliances
3.413 converts watts to Btu/hr (see Chapter 8)

Heat gain for occupants is broken into sensible and latent heat gain, with a cooling load factor applied to the sensible fraction.

All ventilation and infiltration heat gains are instantly seen as cooling loads, as are all latent loads of any origin.

The terms in equation 6-3 require some explanation. The Solar Heat Gain Factor (SHGF) is the solar heat gain through sunlit DSA (double strength single pane) glass on an average cloudless day at sea level. It is computed based on solar incident angle and intensity, and includes diffuse and reflected radiation. The Shading Coefficient represents the attenuation (or augmentation) of the solar gain through glazing materials different from DSA. So for single pane, 1/4 inch glass, the SC equals one (1.0).

This is important, because more recent cooling load calculations, including the Radiant Time Series, use a slightly different method of calculating solar heat gain, as follows:

$$q = A * ED * SHGC_\theta * (IAC) + A * (Ed + Er) * SHGC_D * (IAC) \tag{6-6}$$

$$Et = ED + Ed + Er \tag{6-6a}$$

where q is local instantaneous heat gain, Btu per hour
ED is direct incident solar radiation on the surface, $Btu/hr/ft^2$
SHGC is Solar Heat Gain Coefficient, subscript θ for direct, subscript D for diffuse and reflected.
Ed is diffuse radiation on the surface, $Btu/hr/ft^2$
Er is reflected solar radiation on the surface, $Btu/hr/ft^2$

IAC interior shading factor

In equation (6-6), ED and (Ed+Er) replace SHGF, and SHGC replaces SC. The difference is that SC is already adjusted to reflect the solar heat gain in the space for a particular glazing – DSA. Therefore, SC modifies glazing only relative to DSA glass, whereas SHGC modifies the incoming radiation for any type of glass relative to an unblocked opening. The reason for this is that the more modern and complex cooling load calculation methods treat direct radiation differently from diffuse and reflected as to their effect on the space to be cooled.

Window manufacturers now list the SHGC for their products, rather than SC. However, to correctly use the CLTD/CLF method, the designer must use SC. Since the type of glass used to compute SHGF is known, the correction from SHGC to SC is a simple constant as follows:

$$SC = SHGC \ /.87 \qquad (6\text{-}7)$$

Radiant Time Series

Radiant Time Series (RTS) is a simplified version of the Heat Balance method. Both are described in the ASHARE Fundamentals Handbooks beginning in 2001. Although it may be possible for an experienced user of Visual Basic to program this method on a spreadsheet, the more practical approach would be to purchase a computer program. Programs are available for purchase on the web.

A basic element of the RTS method is that sensible heat gain is split between a convective portion and a radiant portion. The former becomes cooling load immediately, whereas the latter is stored in room structure, furnishings, and occupants and becomes cooling load at a later time through convection to the inside air. The percentage of sensible heat gain assigned to convection or radiation varies for different elements such as exterior walls and roofs, lights, people, and machinery. A table of these percentages is available in the ASHRAE handbook, and computer programs will have default values built in.

Walls and Roofs

As discussed earlier, heat gain through walls and roofs on a warm sunny day takes time to pass through the structure from the exterior surface to the interior surface. ASHRAE has supplied a table of Conduction Time Series (CTS) values as a function of construction type and elapsed time. The basic equation for heat gain through a solid surface at any particular hour T is the familiar

$$q_T = U * A * (ts_T - tr_T) \qquad (6\text{-}8)$$

where ts is sol-air temperature as defined earlier tr   is room temperature

Calculation of sol-air temperature (ts) for each hour of the day requires an ambient temperature profile applicable to the project site, and calculation of direct, diffuse, and reflected radiation for each hour of sunlight.

$$ts = to + Et * (\alpha/ho) - \varepsilon * \Delta R/ho \qquad (6\text{-}8a)$$

where Et   is defined in equation (6-6a)

α is surface absorption of solar radiation – generally a function of color

ho   is coefficient of heat transfer by long-wave radiation and convection at the surface

The parameter $\varepsilon * \Delta R/ho$ is assumed to be zero for vertical surfaces and 7° F for horizontal surfaces. For glass, ts = to using the RTS method.

To estimate the cooling load for any particular hour of the day, it is necessary to calculate equation (6-8) for every hour of the day. It is obvious that this alone would be a tedious calculation which would have to be repeated for each exterior exposure, making a computer program essential.

The heat that reaches the inside surface of a wall or roof at any particular hour is a fraction of the heat gain calculated by equation (6-8) for every hour of the day. That fraction is determined by the Conduction Time Series factor. For example, at 3:00 PM for a particular construction assembly, the fraction of the current hour's heat gain as computed by equation (6-8) is 18%. This is the CTS value for time zero. The total heat gain reaching the inside surface at that hour is the sum of the heat gains for the 24 previous hours, each multiplied by the conduction time series factor for the elapsed time to the hour of interest. Another example: in the case cited above, the contribution from 1:00 PM would be 20% of the heat gain calculated for 1:00 PM. 20% is the CTS value for elapsed hour 2.

Wait, we aren't done! Once the heat reaches the inside surface, it is still not completely converted into cooling load. Most of that heat enters the room as radiation, so must still be absorbed by floor, ceiling, furniture, and occupants before finally being picked up by the air as convection and adding to the cooling load. From the percentage splits presented in the ASHRAE handbook, Only 37% of wall heat reaching the interior surface and 16% of the roof heat enter the room as convection and add immediately to the cooling load. To estimate the contribution of the radiation portion at any particular hour, requires a particular radiant time series (RTS) value available from a table in the ASHRAE handbook. This is called the Nonsolar Radiant Time Series and is a function of construction, configuration, and location of the interior space, plus elapsed time. To obtain the cooling load from the radiant contribution, the radiant portion of the heat gain calculated for every hour must be multiplied by the RTS value corresponding to the elapsed time from each of the previous 24 hours to the hour of interest.

The final estimate of cooling load for heat gain through walls and roofs at the hour of interest H is therefore:

$$Q_H = Fc^*(c0^* q_H + c1^*q_{H-1} + c2^*q_{H-2} + c3^*q_{H-3} + \ldots + c23^*q_{H+1}) \qquad (6\text{-}9)$$
$$+ \; Fr^*(r0^* q_H + r1^*q_{H-1} + r2^*q_{H-2} + r3^*q_{H-3} + \ldots + r23^*q_{H+1})$$

or $$Q_H = Fc^*\sum(c_i^*q_{H-i}) + Fr^*\sum(c_i^*q_{H-i}) \qquad (i = 0 - 23) \qquad (6\text{-}9')$$

where Q_H is cooling load at the hour of interest H
Fc is the convective fraction of the heat gain
Fr is the radiant fraction of the heat gain
ci is the CTS value for the ith hour preceding hour H
ri is the nonsolar RTS value for the ith hour preceding hour H

Note that in the expanded equation above, the 23rd hour after the hour H is shown as "H+1)". This is because tables in the handbook are from 1 to 24 hours rather than 0 to 23 hours.

Fenestration

Most of the radiation that falls on a window passes directly into the space and heats the interior surfaces and occupants. Only a fraction is lost to reflection and absorption/re-radiation. This fraction is accounted for in RTS by the Solar Heat Gain Coefficient (SHGC). These coefficients are tabulated for the center of the glass and for the total window in the Fenestration Chapter of ASHRAE Fundamentals, and vary according to type of frame, type of glass, type of radiation and, for direct solar radiation, angle of incidence. If the designer is using a computer program, these factors may be built in. They may also be available from manufacturer's data. Where a single value for SHGC is given, it would be applicable to the total of diffuse and direct radiation, with the latter at a nominal incidence of about 60°.

Most of the heat gain through fenestration is, of course, through the glass. The attenuating effect of frames and sashes is usually neglected, except in the case of overall U value, which can be strongly affected by frame and sash design and material. In particular, any continuous metal structure from the outdoor to indoor surface acts as a fin to conduct heat from one surface to the other with little resistance. That said, this effect has more implications for heating load calculations than for cooling loads, where the temperature difference is much smaller.

There are three components of fenestration heat gain, direct radiation, diffuse/reflected radiation, and conduction/convection. The radiation portion is defined for any particular hour as formerly shown in equation (6-6):

direct solar $$q_{DT} = A * ED_T * SHGC_\theta * (IAC) \qquad (6\text{-}10)$$

diffuse/reflected $\quad q_{RT} = A * (Ed_T + Er_T)* SHGC_{DT} * (IAC) \quad$ (6-10a)

Again, the subscript T is any particular hour. θ is the solar radiation angle. The conduction/convection portion is:

$$q_{CT} = U * A * (to_T - tr) \qquad (6\text{-}10b)$$

where to_T is ambient outdoor air temperature at hour T
tr is room (indoor) air temperature

The total heat gain at any given hour T is thus

$$Q_T = q_{DT} + q_{RT} + q_{CT} \qquad (6\text{-}10c)$$

The parameters ED and Ed and Er are functions of latitude, apparent solar time, and surface orientation (exposure angle and tilt). They were calculated in the course of calculating sol-air temperature (equation 6-8a) and so are now available to calculate fenestration cooling load.

The percentages of convective and radiant heat gain may be found in the ASHRAE handbook, and default values are built in to most computer programs. For direct solar radiation through un-shaded glass, all of the heat gain is assumed to be radiant. For diffuse and reflected solar radiation through glass, the radiant portion is assumed to be 63%, and the conduction portion is 37%.

If glass has internal shading, then the indoor attenuation coefficient (IAC), available from tables in the ASHRAE handbook, is used to split the direct solar radiation into "beam" heat gain and "diffuse" heat gain. These continue to be represented by q_{DT} and q_{RT} in equation (6-10c). Both direct and beam heat gain are 100% radiation and are modified by the solar radiant time series rs, while the sum of the diffuse and conductive heat gains are 63% radiation and are modified by the non-solar time series r. The total cooling load at an hour of interest H is thus given by:

$$Q_H = (rs0*q_{DH} + rs1*q_{DH-1} + rs2*q_{DH-2} + rs3*q_{DH-3} + \ldots + rs23*q_{DH+1}) + (r0*(q_{RH}+q_{CH}) + r1*(q_{RH-1}+q_{CH-1}) + r2*(q_{RH-2}+q_{CH-2}) + r3*(q_{RH2}+q_{CH-2}) + \ldots + r23*(q_{RH+1}+q_{CH+1})) \qquad (6\text{-}10d)$$

or $\quad Q_H = \sum(rs_i*q_{DH-i}) + \sum(r_i*(q_{RH-i}+q_{CH-i}) \quad (i = 0 – 23) \quad$ (6-10d')

<u>Internal Loads</u>

Internal loads are estimated using RTS in exactly the same way as the envelope loads outlined above. The basic equations are the same as those used in the CLTD method, but must be calculated for each of the 24 hours of the day. These loads are then split into convection and radiation using the table provided in The Handbook, or as built in to the computer program. The convection portion is immediately included in the cooling load for the hour of interest, and the radiation load is calculated using the non-solar radiant time series as shown in equations (6-9) and (6-10d)

Internal loads are often the largest portion of the cooling load, and are often the most difficult to predict. In order to avoid over-sizing, the designer must insist upon being given detailed information about the average peak number of occupants, power input data and load factor for appliances and machinery, and the planned use of special lighting.

Summary

Cooling load calculations are fundamental to HVAC design, and this book assumes that the reader is familiar with the basic principles. Therefore, this chapter has been no more than an outline of two methods that are effective for small building design, without being overly complicated or time-consuming. Details of both methods are presented in ASHRAE fundamentals as noted earlier, and the reader is directed the relevant volumes for detailed descriptions and methodologies.

The CLTD/CLF method is easily programmed on a spreadsheet for a particular city or location. It is also available in several computer programs which may be found on and ordered through the internet. The RTS method can be programmed in Fortran, Basic, and other languages using compiler software also available on the internet, but this requires some fundamental skill in programming to be feasible for the average engineer. Again, computer programs using RTS and even more sophisticated methods are available from HVAC equipment manufacturers and HVAC software vendors. These too may be found on and ordered through the internet.

This author has used the CLTD/CLF method for 25 years, and is comfortable with it. However, the RTS or Heat Balance methods are recommended for the engineer or designer just entering the field. These methods are now the standard of practice based on their presentation in the ASHRAE Handbook.

Heating and Cooling Load Calculation Notes and Tips

Ref: ASHRAE 2009 Fundamentals, "Nonresidential Cooling and Heating Load Calculations"

Peak cooling or heating load for each space in a zone must be handled by the zone ac system. [for small commercial buildings, there are probably no times when part of the building requires cooling while another part requires heating, unless there are spaces with large areas of south-facing glass]

The two cooling load calculation methods now presented in the reference are the Heat Balance method and the Radiant Time Series Method. The Cooling Load Temperature Difference Method is no longer presented by ASHRAE. For the heat balance method, see the computer program Hbfort which accompanies Cooling and Heating Calculation Procedures, Peterson 1998

Surplus latent load capacity is permissible. Sensible load capacity is required to maintain room dry-bulb temperature at the desired level. Surplus sensible load capacity is undesirable because it reduces the run-time, hence the dehumidification time, of the air conditioning equipment.

The building exterior wall sections will show the building materials used. The designer must determine the heat transfer characteristics of the materials, based on manufacturer's data or generic characteristics, which may often be found in the reference.

Where large fenestrations are used, orientation may not adequately account for solar heat gain reflected from nearby water bodies, parking lots, or reflective buildings.

Floor areas must be measured to the outside of space exterior walls, and to the centerline of interior partitions. Wall areas must include all surfaces that can conduct heat into the interior space. If the space above the ceiling is a return air plenum, then this heat should be added to the plenum heat, which raises the coil entering air dry bulb above the space temperature. If the space above the ceiling is a dead air space within the thermal envelope, it can be ignored, but the wall heat gain to the plenum space should still be included in the building cooling load.

Fenestration areas should include the rough opening, or the overall dimensions of the windows or glass doors.

Recessed florescent fixtures with T8 lamps and electronic ballasts will dump very little heat into the plenum. Ballast factors for HID lamps under 200 watts should be about 1.35. For HID lamps over 200 watts, 1.15.

For dining rooms, estimate the number of meals served per hour, then multiply by 50 Btuh for total heat gain, and 38 Btuh for sensible gain. [For residential kitchens, use 1300 watts total, 850 watts sensible]

Use 150 watts for computers. [One w/sf of floor area is a good approximation for offices, but should be applied only to office space, not to non-office rooms such as corridors, reception, break rooms, or conference rooms]

Use data in the reference for office copiers, vending machines, and mailing equipment and apply according to the specific case. [For a general estimate, assume the copier operates 50% of the time]

END

Chapter 7

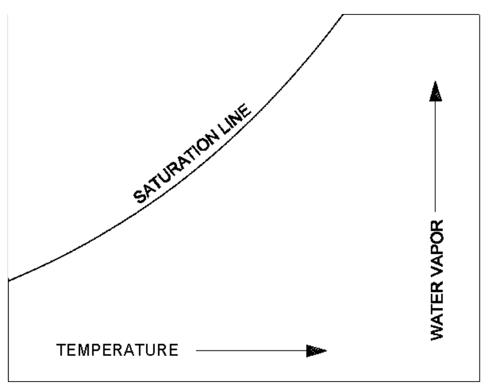
Psychrometric Considerations

Psychrometric Requirements

After the air conditioning cooling load has been determined, a clear picture of the psychrometric requirements of the equipment emerges. This chapter deals with determination of these requirements as a preliminary to actually selecting the equipment for the project. This deals only with air conditioning, where moisture removal is an important part. Heating is not considered because latent heat does not influence the selection of heating equipment. The following brief refresher will hopefully help brush away any cobwebs.

The Psychrometric Chart

The psychrometric chart maps two primary and six secondary properties of air. The primary properties are dry bulb temperature and water vapor concentration. All of the properties are expressed as concentrations in dry air. The psychrometric chart shown below has two principle axes – the horizontal axis is dry bulb temperature, **tdb,** in ˚C or ˚F, and the vertical axis is water vapor content, usually labeled as humidity ratio, **W,** expressed in IP units as pounds of moisture per pound of dry air (lbs H2O/lb) or grains of moisture per pound of dry air (gr/lb). There are 7000 grains to one pound.



On the saturation line, all of the water vapor in the air has condensed to liquid. There is no stable state to the left of the saturation line.

The six secondary properties are:

wet bulb temperature **twb** – The lowest temperature that can be reached by the evaporation of water only.

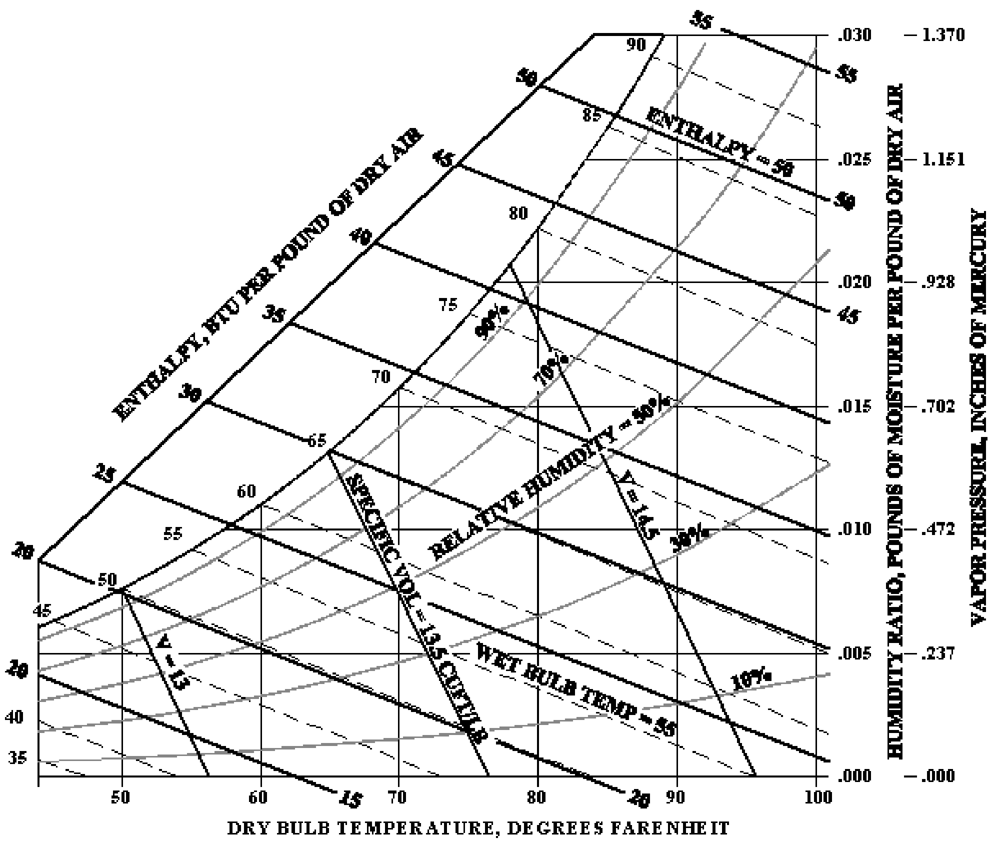
relative humidity, **rh** – the amount of water vapor in air at a given temperature as a percentage of the total amount of water vapor in saturated air at the same dry bulb temperature. Also, the vapor pressure of the air-water vapor mixture as a percentage of the vapor pressure of saturated air at the same dry bulb temperature.

dew point temperature, **tdp** – the temperature to which an air-water vapor mixture must be cooled for the water vapor to condense into liquid water.

enthalpy, **h** – the specific energy of the moist air, expressed in IP units as Btu/lb of dry air.

specific volume, **v** – the total volume of dry air and water vapor in units of cubic feet per pound of dry air (ft^3/lb). This is the reciprocal of the density of the air/water vapor mixture, lb/ft^3.

water vapor pressure, **Vp** - the pressure exerted by the molecules of water vapor in a saturated mixture of water vapor and dry air at a given temperature and at standard pressure for the altitude of the chart.



The Psychrometric Chart, IP Units

Note that these are "secondary" because they are all fully defined by dry bulb temperature and humidity ratio alone. However If any two properties are known, then all of the others can be found by reading from the chart or making a computation. For example, if outdoor air temperature and relative humidity are known, say 90 ˚F and

70% rh, that point can be plotted on the chart above, thus fully defining all of the properties of the outdoor air at that point, including wet bulb temperature, dew point, enthalpy, and specific volume.

Both dew point and vapor pressure are both defined by horizontal lines that intersect the vertical axis, so that for any humidity ratio (vertical axis), there is a unique value of **tdp** and **Vp.**

The curved line bounding the left of the chart is the saturation line. On the saturation line, the dry bulb temperature is the dew point and wet bulb temperature, and relative humidity is 100%. Thus, on the saturation line:

$$\mathbf{tdb} = \mathbf{tdp} = \mathbf{twb} \text{ and } \mathbf{rh} = 100\%$$

Any point on the psychrometric chart is a **state point** because it defines the thermodynamic state – the temperature and humidity ratio - of the air at that point. If the air at a given state is acted upon by an external process such as heating, mechanical cooling, or humidification it will be changed to another state. The path between the initial and final states is a **process line.**

Air conditioning cooling load is defined in terms of **sensible heat load** and **total heat load.** Likewise, a mechanical cooling coil, DX or chilled water, is defined in terms of **sensible heat capacity** and **total heat capacity.** A change in the **sensible heat** of a mass of air creates a measurable change in its dry bulb temperature, **tdb.** A change in the **latent heat** creates a measurable change in the humidity ratio, **W. Total heat** is the sum of the latent heat and sensible heat of a mass of air. A change in the **total heat** of a mass of air is equivalent to a change in its enthalpy, **h.** These changes are quantified by the following formulas:

$$\Delta Q_s = \Delta\mathbf{tdb} * Cc * rho * c_p * \text{ min/hr}$$
$$\Delta Q_l = \Delta W * Cc * rho * Q_{LV} * \text{min/hr}$$
$$\Delta Q_t = \Delta Q_s + \Delta Q_l$$
$$\Delta Q_t = \Delta\mathbf{h} * Cc * rho * \text{min/hr}$$

where (IP units)
ΔQ_s is the change in sensible heat of the flowing air (Btu/hr)
ΔQ_l is the change in latent heat of the flowing air (Btu/hr)
ΔQ_t is the change in total heat of the flowing air (Btu/hr)
Cc is the air flow rate (ft^3/min)
rho is the average air density of the flowing air (lb/ft^3)
c_p is the specific heat of air (Btu/lb/°F)
Q_{LV} is the latent heat of vaporization of water (Btu/lb H2O)

for air at standard conditions:
rho = .075, c_p = .24, Q_{LV} = 1076

$$\Delta Q_s = \Delta\mathbf{tdb} * Cc * 1.08 \qquad (7\text{-}1)$$
$$\Delta Q_l = \Delta W * Cc * 4840 \qquad (7\text{-}2)$$
$$\Delta Q_t = \Delta\mathbf{h} * Cc * 4.5 \qquad (7\text{-}3)$$

In this book, loads and capacity will be expressed in terms of sensible heat and total heat, because that is how most air conditioner system published performance data is expressed.

Paper psychrometric charts suitable for plotting are often available from HVAC manufacturer's representatives. 50-sheet pads of charts similar to the ones used for this course are available from the ASHRAE bookstore. Also, there are numerous software programs for plotting psychrometric analysis which may be found on the web. The examples used in this book were developed using the ASHRAE Psychrometric Analysis Program, available from ASHRAE on CD[12].

Cooling System Psychrometric State Points

In this book, psychrometric state points will be identified as follows:

- 1 – room air
- 1A – maximum supply air temperature and dew point to satisfy room load
- 2 – outdoor air
- 3A – mixed air entering heat pipe or pre-cool coil, where applicable
- 3 – mixed air entering cooling coil
- 4 – air leaving cooling coil
- 4A – air leaving heat pipe or reheat coil, where applicable

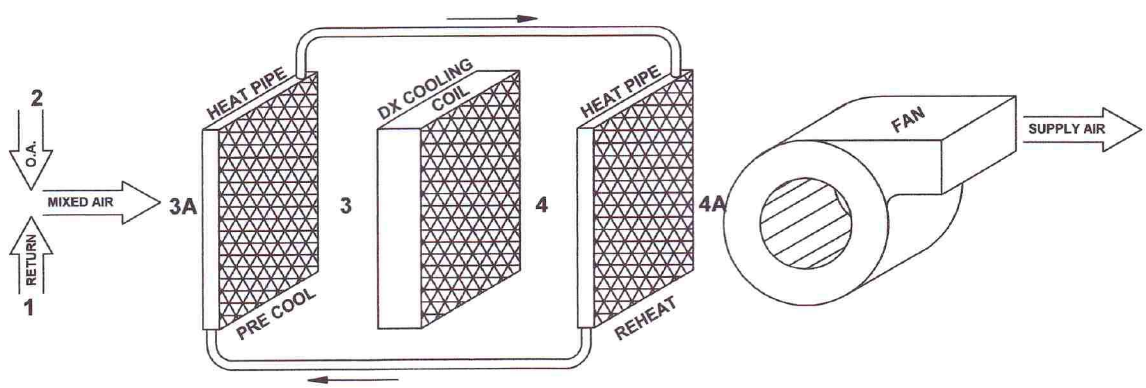


Figure 7-1 – DX Cooling System Schematic

Figure 7-1 is a schematic representation of a cooling system, showing the locations of the state points except for point 1A, which is a calculated, not a physical, point.

A typical psychrometric chart with important parameters labeled is shown on Figure 7-2. The cooling cycle shown is for a five ton rooftop unit serving a small office zone. Room air at 76° and 57% relative humidity is mixed with outdoor air at 94° dry bulb and 78° wet bulb. The mixed air passes over a cooling coil, where it is cooled to 59.6° at 91% rh and then delivered to the room to carry away cooling load heat.

The most important elements of the psychrometric cycle, from the standpoint of the engineer, are room and coil sensible heat ratio, room dew point, and apparatus dew point. Sketching these on a psych chart, will quickly tell you whether a simple system will do the job, or whether additional components and capabilities will be needed.

Process Lines

The lines on a psychrometric chart we will call "process" lines. Referring to figure 3, line **1-1A** is the "room process line". Point **1A** is the **supply air critical state point** and represents the maximum state of the supply air that will satisfy the room load. As the supply air cools and dehumidifies the room, it will follow the slope of the room process line. Thus, the air leaving the cooling apparatus must be at the same or lower temperature and dew point than point **1A** or the desired room temperature and relative humidity cannot be maintained at design conditions. Line **3-4** is the coil process line. Supply air at the flow rate (cfm) of the cooling apparatus enters the coil at state point **3** and leaves the coil at state point **4**, where it is introduced into the room, picks up the sensible and latent load, and then is returned to the cooling apparatus for reprocessing.

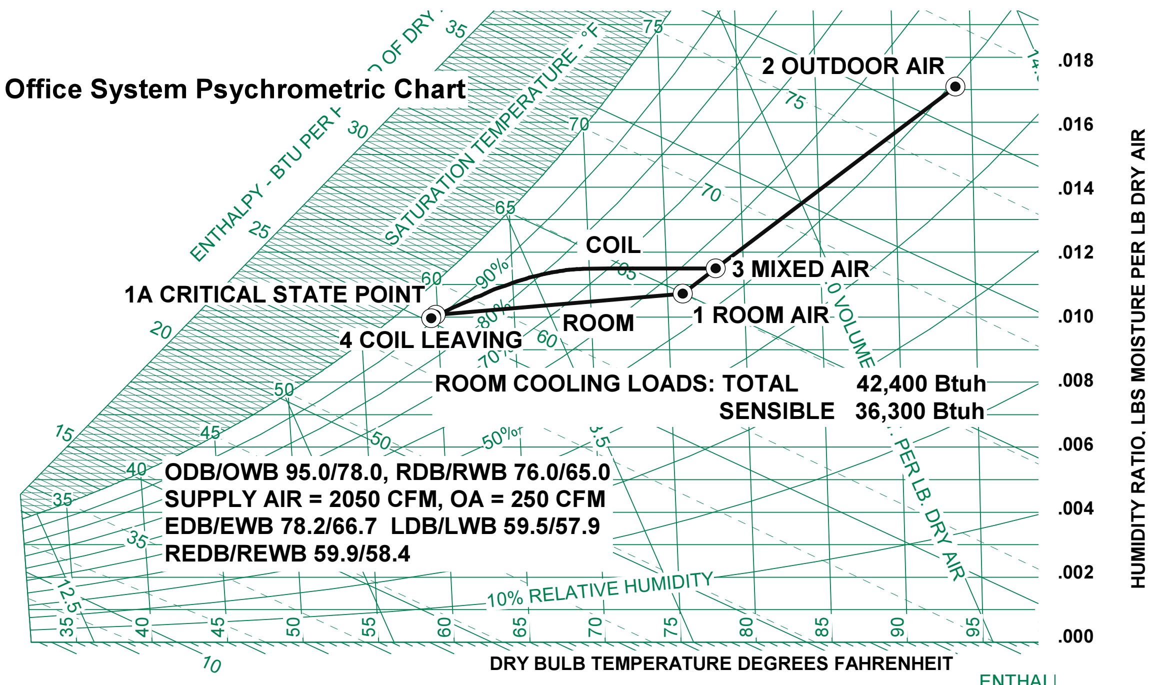


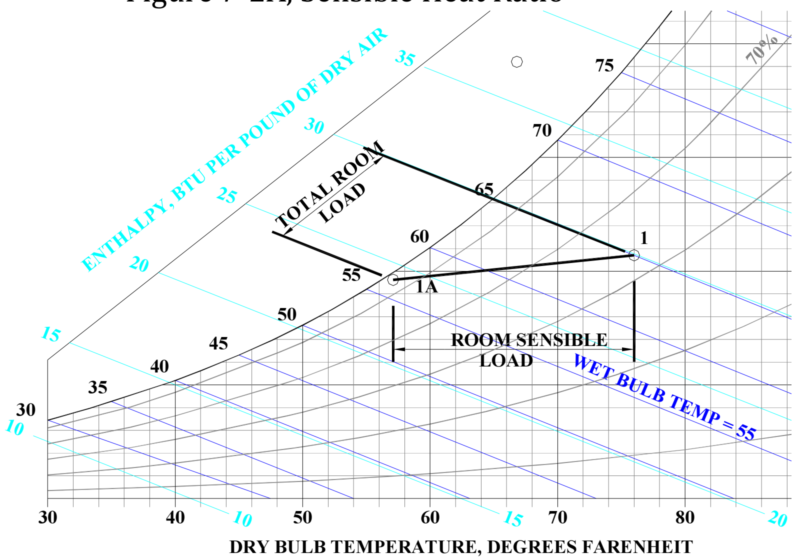
FIGURE 7-2

5 TON PACKAGED AC, SENSIBLE COOLING CAPACITY 42,000 Btuh, TOTAL COOLING CAPACITY 57,500 Btuh COIL COOLING LOADS: SENSIBLE 41,200 Btuh, TOTAL 54,200 Btuh

Sensible Heat Ratio

For a cooling system to be able to satisfy the design conditions of the room and outdoor air loads, the coil process line must satisfy two conditions. First, the coil leaving air temperature, point **4** or **4A**, must be low enough to significantly dehumidify the air, generally under 60°F. Second, the coil process line must cross the room process line so that the supply air dew point and dry bulb temperature is less than point **1A.** That is, point **4** must be cooler and drier than point **1A**, as shown on Figures 3 and 4. These conditions may be difficult to meet with ordinary DX cooling systems. If either condition is not met, then additional processes may be necessary, such as outdoor air pre-treatment, mixed air pre-cooling, or coil leaving air reheat. These processes must be tailored to cause the supply air, point **4** or **4A**, to end up on or below and cooler than point **1A**.

Figure 7-2A, Sensible Heat Ratio



Sensible heat ratio is the ratio of sensible cooling load or capacity to total cooling load or capacity. Room loads are defined in Chapter 6, and include only those loads that originate in the space itself, most notably excluding the load from outdoor air ventilation, duct leakage from or to spaces outside the room, or heat added to a return air plenum.

Sensible loads or capacities are represented on a psychrometric chart by the difference between the high and the low dry bulb temperature of the air stream. Total loads or capacities are represented by the difference between the high and low enthalpy of the air stream. This is shown on Figure 7- 2A for the room loads. It can be seen from this, that the slope of the room process line represents the room sensible heat ratio. Latent loads or capacities are the difference between sensible and total.

The formulas that represent this are (see formulas 7-1, 7-1, and 7-3, pg 57):

$$\mathbf{Qrs = (t_1 - t_{1A}) * Cc * c_p * rho * min/hr} \quad (7\text{-}4)$$
$$\mathbf{Qrt = (h_1 - h_{1A}) * Cc * rho * min/hr} \quad (7\text{-}5)$$
$$\mathbf{SHR = Qrs/Qrt} \quad (7\text{-}5A)$$

where
Cc = coil air flow, ft^3/min
h = enthalpy, Btu/lb
c_p = specific heat, Btu/lb/°F
rho = density, lb/ft^3

For air at standard conditions, c_p = .24 Btu/lb/°F, rho = 0.075 lb/ft^3
thus

$$\mathbf{Qrs = (t_1 - t_{1A}) * Cc * 1.1} \quad (7\text{-}6)$$
$$\mathbf{Qrt = (h_1 - h_{1A}) * Cc * 4.6} \quad (7\text{-}7)$$

The term "load" applies to the cooling coil only in the context of the total computed loads that have to be removed by the cooling apparatus. The coil <u>load</u> includes all loads on the coil, from whatever source. Remember that latent loads will normally come only from people, some food service operations, infiltration, and ventilation air. Infiltration can usually be ignored when a building is properly pressurized, and latent loads from showers or cooking appliances are usually exhausted directly from the building and do not enter the cooling coil. On figure 7-2, the coil "load" would be represented by a process line from point **3** to point **1A**.

The task of the engineer is to select a cooling apparatus with capacity that matches as closely as possible the load represented by the coil process line. After he has selected a system, then the capacity of the coil at a particular air flow with entering conditions at **state point 3** is represented by the coil process line. The convention used here, therefore, is for the sensible heat ratio of the coil to be defined as the sensible heat capacity of the coil divided by its total heat capacity.

Dew Point

The dew point is defined as the temperature at which the water vapor in air will condense. If a room is at a particular humidity ratio, then the dew point of that room will be the intersection of a horizontal line from the room state point to the saturation line on the psych chart. The apparatus dew point (ADP) is the point where a straight line extended from point **3** through point **4** crosses the saturation line. It is the dew point of the coil tube and fin surfaces. The **bypass factor**, sometimes included with performance data, accounts for the fact that not all of the air passing across the coil comes in contact with a cold surface, and is therefore not dehumidified. Bypass factors range from about 4% to as high as 11%.

The sensible and latent cooling capacity of a coil can be found from manufacturer's tables as a function of the psychrometric conditions of the air entering the coil and the air flow rate.

Plotting Points on the Psychrometric Chart

To do psychrometric analysis, it is necessary to use known data to calculate the unknown points. Referring to Figure 7-2, the known data are as follows:

odb = **tdb_2** = outdoor air dry bulb temperature, °F
owb = **twb_2** = outdoor air wet bulb temperature, °F
rdb = **tdb_1** = room air dry bulb temperature, °F
rwb = **twb_1** = room air wet bulb temperature, °F
Qps return air plenum sensible heat gain (if any)
Qrs room sensible cooling load, Btu/hr
Qrt room total cooling load, Btu/hr
Coa outdoor air ventilation rate, cfm
Cc estimated or actual coil air flow rate, cfm
Cr room return (and plenum) air flow rate = **Cc - Coa**
Qscc estimated or actual coil sensible cooling capacity, Btu/hr

Qtcc estimated or actual coil total cooling capacity, Btu/hr

Knowing these parameters, we can find the other points on the chart. Note that it is easiest to find dry bulb and wet bulb temperatures on the chart, although it may be necessary to use enthalpy in the calculation procedure. The cooling apparatus entering state is a mixture of room air and outdoor air, and lies on a line connecting the two states on the chart – see figure 7-2. The mixed dry bulb temperature is a linear function of the two air flows, as follows:

$$tdb_3 = (t_1 * Cr + t_2 * Coa)/(Cr+Coa) \qquad (7\text{-}8)$$
$$twb_3 = f(tdb_3, Cr, Coa,) \qquad (7\text{-}9)$$

the value of twb_3 is most easily found by reading it from the chart at the intersection of tdb_3 and the line connecting points 1 and 2.

To calculate the dry bulb and wet bulb temperatures for points 4 (and 1A) it is necessary to know the enthalpies of points 1 and 3. These can be looked up on the chart, or found using psychrometric software, as a function of dry and wet bulb temperatures.

$$h = f(tdb, twb) \text{ thus } h_1 = f(tdb_1, twb_1) \text{ and } h_3 = f(tdb_3, twb_3)) \qquad (7\text{-}10)$$

and continuing to find points 1A and 4:

from eq (7-1) $\quad tdb_{1A} = tdb_1 - Qrs * Cc * 1.1 \qquad (7\text{-}11)$
from eq (7-3) $\quad h_{1A} = h_1 - Qrt * Cc * 4.6 \qquad (7\text{-}12)$
$\quad twb_{1A} = f(tdb_{1A}, h_{1A}) \qquad (7\text{-}13)$

$$tdb_4 = tdb_3 - Qscc * Cc * 1.1 \qquad (7\text{-}14)$$
$$h_4 = h_3 - Qtcc * Cc * 4.6 \qquad (7\text{-}15)$$
$$twb_4 = f(tdb_4, h_4) \qquad (7\text{-}16)$$

Coil sensible and total capacity, Qscc and Qtcc is determined using performance data published by the manufacturer of the air conditioning equipment. Many manufacturers furnish free software that will compute coil capacity, and manufacturer's representatives will also provide performance data to engineers when requested. This topic is discussed in more detail in Chapter 9, Equipment Selection.

Ceiling Return Air Plenums:

If heat is added to a return air plenum, then the mixed air state point is shifted at constant dew point as shown on figure 7-3 below. The equations then become:

$$tdb_3 = ((t_1+\Delta tp) * Cr + t_2 * Coa)/(Cr+Coa) \qquad (7\text{-}17)$$

$$\Delta tp = Qps/(Cr*1.1) \tag{7-18}$$

Qps is sensible heat gain to the plenum, and **twb_3** is found by reading it from the chart at the intersection of **tdb_3** and the line connecting points **1P** and **2.**

Figure 7-3 is the same building a s represented by Figure 7-2 but with a return air plenum. For this zone, it is estimated that the plenum will receive half the roof cooling load and 30% of the lighting load. Guidelines for determining plenum cooling loads (**Qps**) are vague, and require application of experience and judgment by the designer. The effect of plenum heat is to increase the temperature entering the coil, which increases the sensible capacity of the coil with a small increase in latent capacity. This heat does not appear as "room" load, as it does in Figure 7-2, which is a zone with no return air plenum. Thus, the room loads are decreased by the load picked up in the plenum, and the combination of increased capacity and reduced

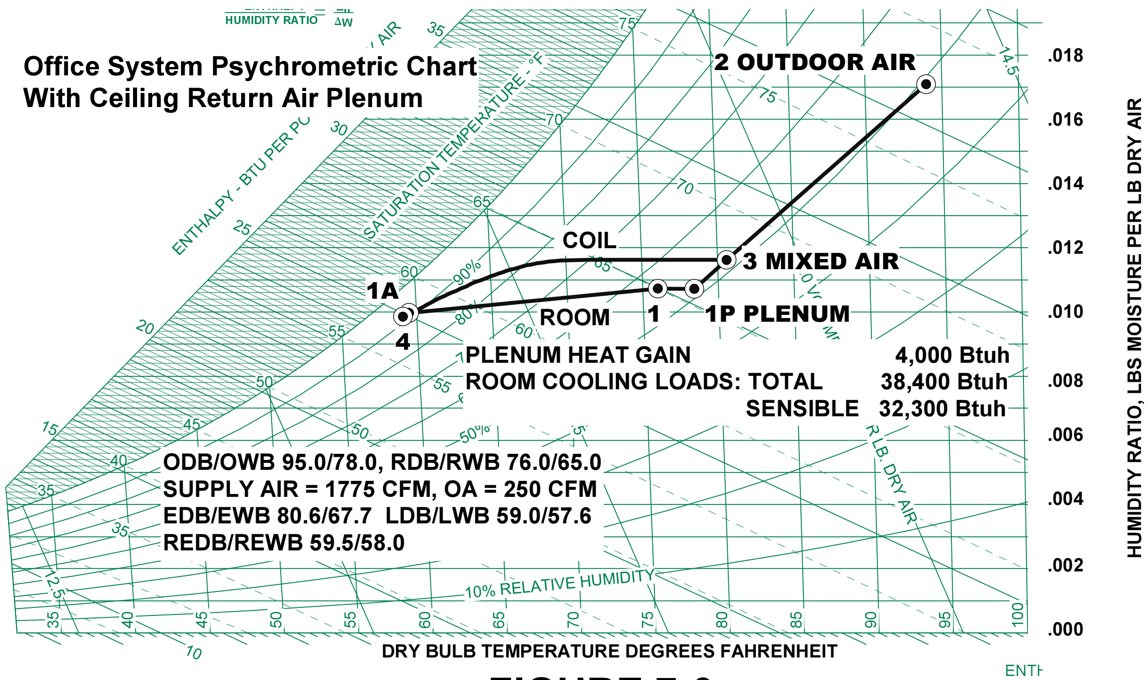


FIGURE 7-3

5 TON PACKAGED AC, SENSIBLE COOLING CAPACITY 42,000 Btuh, TOTAL COOLING CAPACITY 57,500 Btuh COIL COOLING LOADS: SENSIBLE 41,200 Btuh, TOTAL 54,200 Btuh

room load results in a significant reduction in supply air flow. Note the values of room and plenum load and supply air flow on Figures 7-2 and 7-3. Also, note that the separation of points **4** and **1A** improves relative to meeting or bettering the room relative humidity.

<u>Computing State Points and Process Lines for Figure 7-3</u>

Figure 7-3 is an opportunity to review the application of the equations given for computing state points and process lines. First, known parameters:

room dry bulb temperature ,	tdb_1 = 76°F
room wet bulb temperature,	twb_1 = 65°F
outdoor air dry bulb temperature,	tdb_2 = 95°F
outdoor air wet bulb temperature,	twb_2 = 78°F
room sensible heat load,	**Qrs** = 32,300 Btu/hr
room total heat load,	**Qrt** = 38,400 Btu/hr
plenum sensible heat load,	**Qps** = 4,000 Btu/hr
cooling coil sensible heat capacity,	**Qscc** =42,000 Btu/hr
cooling coil total heat capacity,	**Qtcc** = 57,500 Btu/hr
supply air flow,	**Cc** = 1,775 cfm
outdoor air flow,	**Coa** = 250 cfm

These "known" values have been developed as part of the design process (steps 1 through 5 on the design flow diagram, Figure 1 of Chapter 9) which precedes psychrometric analysis. The source of each is briefly described below:

The outdoor air dry bulb and wet bulb temperatures, tdb_2 and twb_2 are determined from weather data for the particular location and design basis month. The "room" or zone temperatures tdb_1 and twb_1 are comfort conditions set by the designer.

The room cooling loads **Qrs** and **Qrt** are all of the heat and moisture gain within the occupied rooms of the zone which must be removed by the cooling system supply air. They do not include ventilation (outdoor air) loads or plenum heat gain. They do include building envelope loads, (heat gain through walls, ceiling, and glass), internal loads (people, lighting, appliances), and infiltration. Infiltration can be ignored if the building is kept under positive pressure by a surplus of ventilation air over exhaust air. **Qps** is sensible heat gain to the plenum already discussed.

Ventilation air **Coa** is outdoor air that is mixed with return air and passes over the cooling coil before being introduced into the zone. Supply air flow equals cooling coil air flow **Cc**. The procedure for determining required ventilation air was described in Chapter 4.

Coil cooling <u>capacities</u> **Qscc** and **Qtcc** are found from manufacturer's data as a function of air flow **Cc** and coil entering dry bulb and wet bulb temperature. They are selected for a specific cooling system to match as closely as possible the coil cooling <u>loads</u>. This is represented by the iteration loop from step 5 back to step 3 of Figure 1, Chapter 9, where air flow is varied until sensible capacity is matched. The coil loads consist of the room or zone loads plus the ventilation load, plus the plenum load. The coil load process line is not shown on a psychrometric chart, but if it were, it would be a line connecting point **3**, which is the coil inlet condition, to point **1A**.

As noted earlier, the room return air flow **Cr** is known because **Coa** and **Cc** are known:

Cr = Cc - Coa = 1,525 cfm

Thus, from equation (7-18) $\Delta tp = Qps/Cr/1.1 = 2.4°F$

On the psych chart, pure heating always occurs at constant dew point, because heating never adds or removes moisture from the air. Since plenum heat is pure sensible heat, the dew point temperature at state point **1P**, tdp_{1P} is plotted at the same dew point temperature as point **1**, tdp_1 plus 2.4° dry bulb. Point **1P** is thus plotted at $tdb_{1P} = 76°F + 2.4°F = 78.4°F$. The dew point temperature $tdp_1 = tdp_{1P} = 59.1°F$. Knowing the dry bulb temperature and the dew point temperature at point **1P**, all other state parameters, such as wet bulb temperature and enthalpy, can be found either by reading from the psychrometric chart or using a psychrometric software program. Thus $twb_{1P} = 65.8°F$.

The mixed dry bulb temperature can now be found using equation (7-8):

$$tdb_3 = ((tdb_1+\Delta tp) * Cr+tdb_2 * Coa)/(Cr+Coa) = 80.7°F$$

As indicated by the discussion of equation (4), the wet bulb temperature at point **3** is located at tdb_3 and a line connecting point **1P** with point **2**. Most psychrometric software programs will also compute a mixed state point. Thus, reading from the chart, $twb_3 = 67.7°F$.

Using equation (7-10), we find the enthalpies of points **1** and **3** as a function of dry bulb and wet bulb temperature at points **1** and **3**, needed to compute the required supply air state, point **1A**, and the coil leaving state, point **4**.

$$h_1 = f(tdb_1, twb_1) = 29.98 \text{ Btu/lb} \qquad h_3 = f(tdb_3, twb_3) = 32.09 \text{ Btu/lb}$$

From Eq. (7-11),(7-12),(7-13)

$$tdb_{1A} = tdb_1 - Qrs/Cc/1.08 = 59.5°F$$
$$h_{1A} = h_1 - Qrt/Cc/4.5 = 25.17 \text{ Btu/lb}$$
$$twb_{1A} = f(tdb_{1A}, h_{1A}) = 58.0 °F$$

From Eq. (7-14),(7-15),(7-16)

$$tdb_4 = tdb_3 - Qscc/Cc / 1.08 = 59.2°F$$
$$h_4 = h_3 - Qtcc/Cc / 4.5 = 24.90 \text{ Btu/lb}$$
$$twb_4 = f(tdb_4, h_4) = 58.0 °F$$

Thus, all of the state points for Figure 7-3 are calculated and plotted.

Load Variations

There are four basic design point load conditions that the designer will encounter when working with small commercial and institutional projects: 1)moderate occupant density and ventilation air; 2)high occupant density with moderate ventilation air; 3)high ventilation air with moderate occupant density; and 4)high occupant density and high ventilation air. Psychrometric analysis will help determine which of these conditions exists, what special processes will be needed when condition 1) is not met, and how to tailor those processes to match the load. A "moderate" density would be fewer than 9 occupants per 1000 sf of conditioned floor area. A moderate amount of ventilation air would be less than 20% of total supply air.

When the SHR of a building or zone falls below the capabilities of standard air conditioners, then additional design elements will be needed to avoid indoor moisture and mildew problems. These may include pre-treatment of outdoor air, or reheat of supply air to false load the system and cause it to run more. Another option to handle both large quantities of outdoor air and high occupant density is to provide a unit with both hot gas bypass and reheat. The key to these choices when selecting equipment is an understanding of the psychrometrics.

In Chapter 9, we will discuss how to select the proper cooling apparatus to most closely match the load. For the remainder of this chapter, we will show the psychrometric characteristics of the basic design point conditions, and how to use the psychrometric chart to select and tailor the appropriate special processes when necessary.

<u>High Occupant Density:</u>

High Occupant density – more than 10 occupants per 1000 sf - results in a low sensible heat ratio because human occupants are a source of latent, or water vapor, cooling load. This is true even if the other primary source of latent load, ventilation air, is moderate.

Figure 7-4 shows the design point condition of high occupant density with moderate ventilation air. The slope of the room process line is very steep, and the temperature at point **1A** is higher than the dew point at the room design condition. It is clear that the mixed air from point **3** cannot be brought directly to point **1A** because it has not been cooled enough to remove the moisture indicated by the dew point at point **1A**. (Just because a line can be drawn on a psychrometric chart, there is no guarantee that there is a physical process that can follow that line.)

The particular project depicted on Figure 7-4 requires a packaged AC unit. Using methods described in Chapter 9, a unit was selected that is large enough to satisfy the mixed air cooling load. Note also that the coil process line **3 – 4** terminates at a higher than usual relative humidity, and that the process line does not cross the saturation line when extended. This is because the manufacturer has included in the unit performance a reheat coil that uses the warm liquid leaving the condenser. The green line shows the approximate actual process line for this unit.

The problem with the coil process line for Figure 7-4 is that the sensible cooling capacity, even with the liquid line reheat, is far too large. Basic air conditioners respond only to zone dry bulb temperature. A low sensible cooling load combined with a low SHR required the ac unit to be over-sized for sensible capacity in order to for the ADP to be lower than the dew point of point **1A**. Over-sizing for sensible capacity will cause zone air temperature to respond rapidly when the air conditioner is running, and then to rise slowly, resulting in long off cycles for the compressor. The air conditioner dehumidifies only when the compressor is running, so a long off cycle allows zone humidity to rise. This can be devastating in jurisdictions where commercial and institutional air handling equipment is required by code to run continuously during occupied periods, thus actually inducing warm, moist outdoor air during the compressor off cycle.

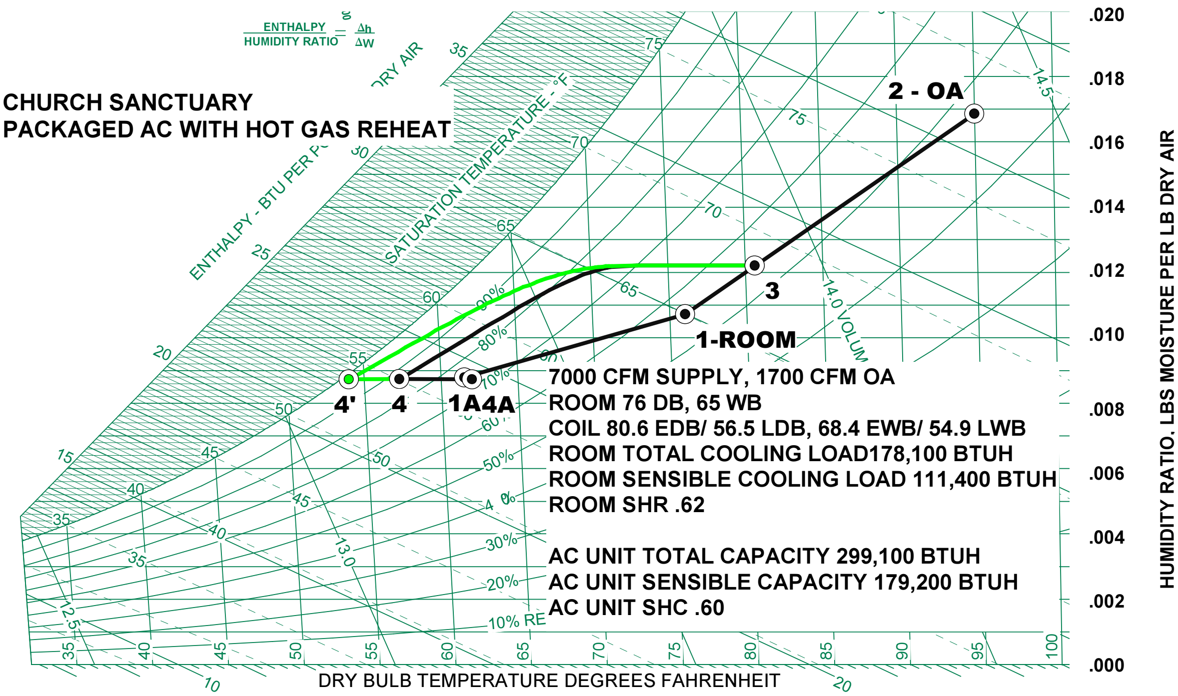


FIGURE 7-4 - LOW ROOM SHR

The solution to this problem is shown on Figure 7-4. The air leaving the cooling apparatus is heated until it intersects the room process line on or near point 1A. This is accomplished by "hot gas reheat" which is an option available on most light commercial packaged AC units and on built-up split DX systems. The effect of reheat is to reduce the sensible heat ratio of a DX system to match the mixed air load and sensible heat ratio.

Another problem when SHR is low, and blowers must run continuously, is that SHR decreases further as cooling load decreases from peak load. At the same time, as outdoor temperature falls, the air conditioner sensible heat capacity increases. Thus, an air conditioner that is satisfactory at peak load on a hot afternoon, may allow indoor humidity to rise unacceptably when outdoor temperatures are low . High room SHR, more than .8, is generally less of a problem than low SHR because the cooling system will run in response to zone dry bulb temperature, at the same time dehumidifying. As outdoor temperature falls, the system with a high peak SHR will stay within the capabilities of the ac unit.

Specialty spaces within a building may be deliberately designed for very high SHR – near 1.0 – both architecturally and mechanically. Large main-frame computer rooms are a good example. These require special air conditioning equipment, and are beyond the scope of this book. Help designing for these cases is available from HVAC manufacture's representatives, and references such as the ASHRAE Handbook.

High Percentage of Outdoor Air

In most applications, occupant density drives outdoor air, and percentages rarely exceed 25%. In general, high occupant density can be handled with reheat or heat pipes, and these processes will also compensate for outdoor air. However, restaurants and laboratories can present special problems of very high percentages of outdoor air, because of the requirements of commercial kitchen hoods and laboratory hoods. Small systems design will most often encounter problems with small restaurants having a limited number of seats. Other cases are small buildings or zones having fingerprint or weapons cleaning stations that require side slot hoods
with high air flows, but otherwise have little sensible load. For lab and industrial hood design, see the publication Industrial Ventilation – A Manual of Recommended Practice by the American Conference of Governmental Industrial Hygienists. Designs for commercial kitchens are readily available through the hood manufacturers' factory representatives.

Figure 7-5 shows the initial psychrometric analysis of a small restaurant with a six foot medium load compensating commercial hood. "Compensating" means that part of the required hood exhaust is made up by outdoor air introduced into the kitchen through the hood system. Hoods smaller than 7' rarely can compensate for more than 70% of the exhaust, as is the case for this example. In Figure 7-5, it is assumed that the make-up air is 25% greater than the exhaust, and is introduced through the dining area ac unit to ensure air flow from the dining area into the kitchen. (See Chapter 4). In this case, because the other loads are small relative to the outdoor air load, the outdoor air is 40% of the total supply air.

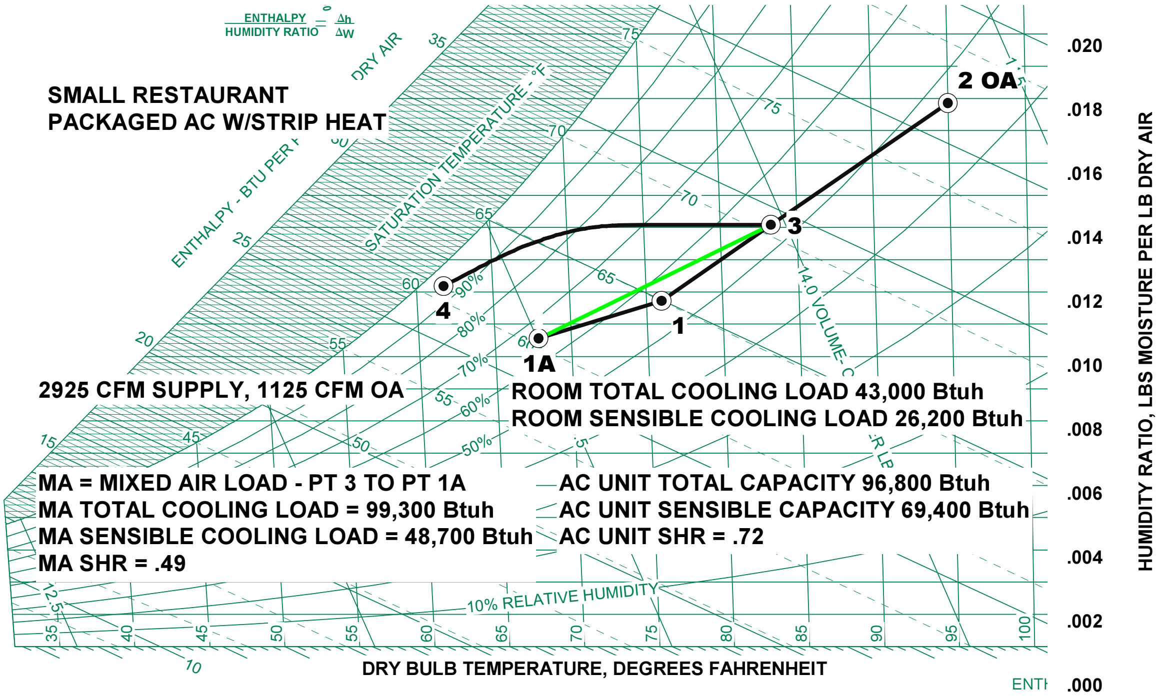


FIGURE 7-5 - Small Restaurant Initial Selection

The system depicted on Figure 7-5 has been selected to satisfy the mixed air total and sensible loads. However, to satisfy the total load, it was necessary to select a system with excessive sensible capacity. The green line from point 3 to point 1A is the process line for the coil design load. It is clear from Figure 7-5 that the selected system cannot ever satisfy the room design conditions, even with massive reheat, because point 4 has a higher dew point than point 1A, which is the maximum dew point that will satisfy the room latent load. Of course, an even larger unit could be tried, but then, if reheat was used, it would actually exceed the mixed air sensible load.

The solution to this situation is to pre-treat the outdoor air before it is introduced into the system. The two most practical ways to do this are with an energy recovery wheel or a 100% outdoor air unit. In the former case, an energy recovery wheel extracts heat and moisture from the incoming ventilation air stream and transfers it to the exhaust stream. In this way, the outdoor air is cooled and dehumidified before being mixed with the room return air, thus reducing significantly the mixed air cooling load. This option is never available with a commercial kitchen or industrial hood, because the exhaust from such hoods is contaminated.

The second option is to cool and dehumidify the outdoor air with a separate air conditioner capable of handling 100% outdoor air. This solution is depicted in Figure 7-6. The selection of pre-treatment equipment is discussed in detail in the Chapter 9, but in general, a 100% outdoor air unit is selected to deliver a dew point to the room or to the zone cooling coil inlet that is lower than the dew point needed to satisfy the room design conditions. In other words, a dew point lower than point 1A at the room supply air flow rate. In addition, since such units usually include hot gas reheat, the outdoor air can be delivered at a dry bulb temperature near or only slightly lower than the room design dry bulb temperature, thus eliminating the outdoor air load on the cooling coil. In the case shown, however, the outdoor air requirement is so dominant that the entire load is best matched by the 100% outdoor air unit with no return air. The hot gas reheat would be regulated to maintain the room dry bulb set point. Electric or gas heat would be used during cold weather.

Large restaurants will generally require only 25% to 30% outdoor air, and so will use systems sized only for the room loads, and will either mix the pre-treated air with the return air from the rooms, or will introduce the pre-treated air directly into the occupied space at the space design state point. Restaurant solutions will usually require the designer to iterate several options to find the optimum match.

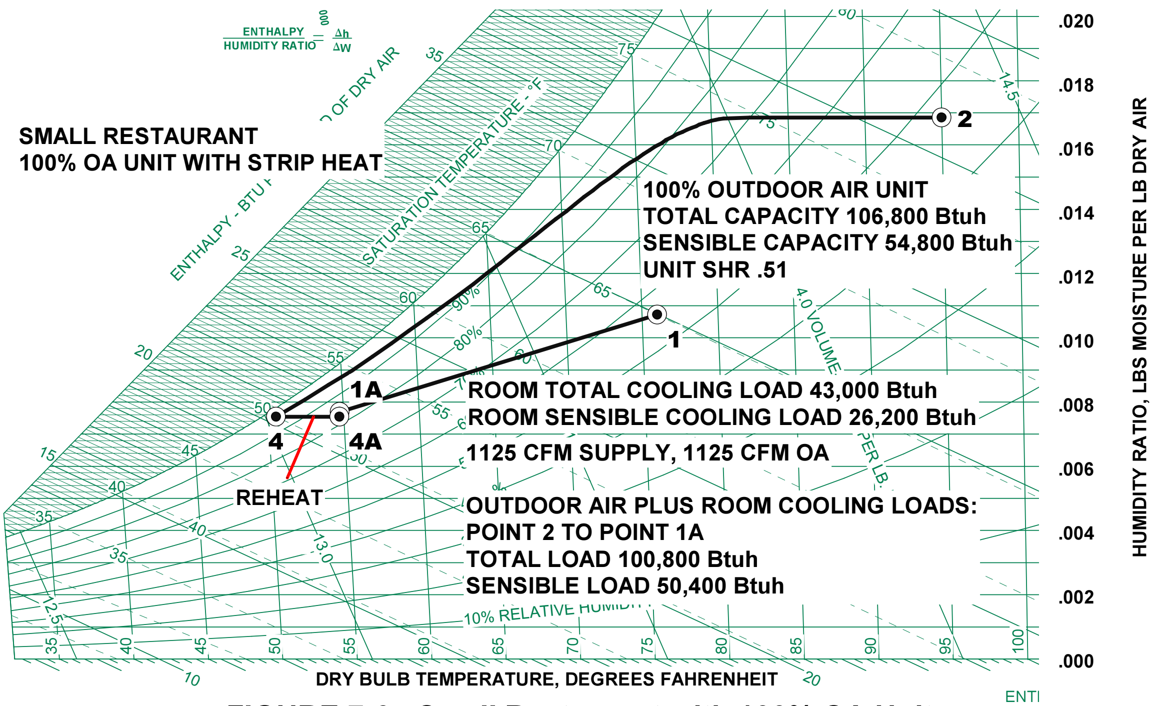


FIGURE 7-6 - Small Restaurant with 100% OA Unit

<u>Heat Pipes</u>

Reheat is generally available as a factory option in light commercial packaged units, and is a simple, cost effective solution for conditions of high occupant density or even moderately high outdoor air percentage. However, light commercial split systems under 25 tons generally are not available with hot gas reheat. Other types of reheat are either prohibited by many codes as is the case with electric reheat, or present special maintenance, safety, and installation problems as is the case with natural gas re-heaters located in the supply duct downstream of the air handler. A frequently viable and cost-effective solution may be a heat pipe. A heat pipe is a coil that wraps around the cooling coil and transfers sensible heat from the entering stream to the coil leaving stream. See Figure 7-1. Because of the characteristics of unitary DX cooling systems, heat pipes can be extremely effective in reducing the sensible heat ratio of the mixed air process line.

Figure 7-7 shows a church vestibule having a high occupant density, with a split system fitted with a three-row heat pipe.

The green process line shown on Figure 7-7 represents the unit cooling coil with no heat pipe. The coil in this case has greatly excessive sensible heat capacity, and has a leaving dew point that is too high to satisfy the room design point. The heat pipe reduces the sensible heat capacity by pre-cooling the mixed air (point 3A to 3). The coil leaving conditions are then brought back to match the room required supply air conditions by reheat. The heat pipe accomplishes this by transferring sensible heat from the warm coil entering air to the coil leaving air. There is no penalty for this, except for somewhat increased pressure drop through the cooling unit. Heat pipe manufacturers call this a "dehumidifying" heat pipe, because as is evident from figure 7-7, the air passing through the coil is much colder than it would be without the heat pipe.

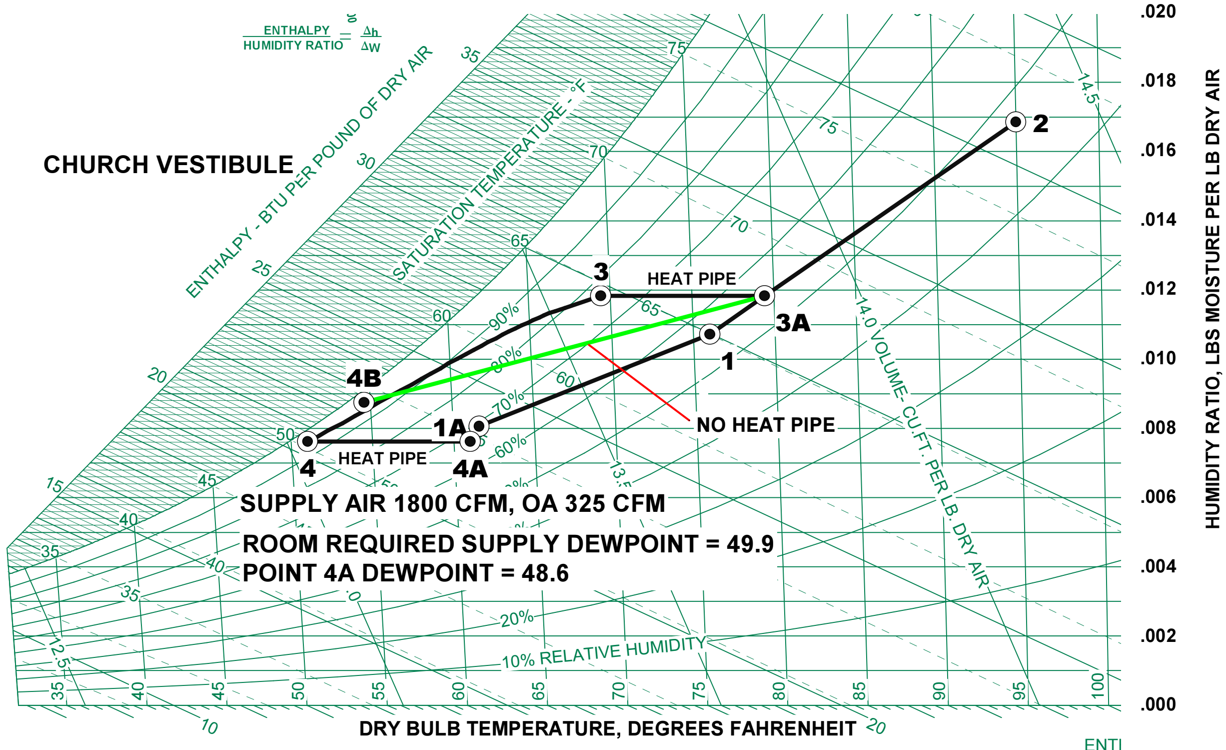


FIGURE 7-7, Heat Pipe Solution

Heat pipe selection and specification will be covered in more detail in the Chapter 9. However, designers are cautioned to work closely with the ac equipment manufacturers' factory representatives and with the heat pipe manufacturers and installers to be certain that the specified heat pipe parameters are viable with the ac equipment available.

Heat pipes are not available as standard equipment from ac manufacturers, but must be fitted to the ac unit after purchase, usually at the heat pipe manufacturer's facility. For small commercial applications, heat pipes are usually applied only to split systems.

This is because most light commercial packaged units are available with liquid line and/or hot gas reheat, which is less costly than heat pipe installation.

Passive – Active Moisture Control

Four moisture control strategies have been discussed in this chapter – excess ventilation air, hot gas reheat (HGR), dedicated outdoor air unit (DOA), and heat pipes. A fifth, energy recovery ventilator, was introduced in Chapter 4, and will be presented in chapter 9 in the context of equipment selection. Two of these, HGR and DOE are "active" strategies because they will operated to control space humidity on signal from a space humidity sensor. The other three are "passive" because they only work when the space air conditioning compressor is running – that is, when there is a call for cooling. It is therefore important to select space cooling systems with adequate capacity reduction, in the form of unloading or multiple compressors, to allow operation at reduced space loads.

Economizer Psychrometrics

See Appendix A "Economizer Psychrometrics and Design Notes"

Rules

Based on the psychrometrics of the room and coil cooling loads, the engineer can apply the following rules to anticipate the type and complexity of the equipment that will be required.

1. The equipment selected must be able to maintain the conditioned space relative humidity below 70% at all times, and below 60% most of the time.

2. If outdoor air is less than 20% of supply air, and occupant density is less than about 7 persons per 1000 sf, then both room and coil sensible heat ratios should be in the range .65 to .8, which small commercial and residential dx systems can accommodate.

3. If outdoor air is more than 20% of supply air, then coil sensible heat ratio may be less than .65, even as the room ratio remains at .7 or more. In this case, pretreatment of the outdoor air, either by energy recovery or 100% outdoor air unit, may be required.

4. If occupant density is more than about 10 persons per 1000 sf, then both room and coil sensible heat ratio may be less than .65, with coil sensible heat ratio being the lower of the two. In this case, pretreatment may be needed to handle the latent load of the outdoor air and in addition, reheat, heat pipe, or other strategy may be needed to handle the latent load due to occupants.

Additional Study

The ASHRAE publication *Psychrometrics, Theory and Practice*[13] includes the latest equations, charts, and tables for thermodynamic properties of moist air, various types of hygrometers and psychrometers for measuring properties and guidelines for the selection of hygrometers for different applications.

END

Chapter 8

Energy Efficient Design

Introduction

The HVAC designer is responsible for providing the building owner an efficient HVAC system that meets of exceeds ASHRAE Standard 90.1 and local energy codes. In the discussion of Preliminary Design (Chapter 3) it was noted that it is also the responsibility of the HVAC designer to advise the architect and lighting designer regarding the energy code requirements of the building envelope, lighting, and equipment. This is because these elements of the building design are addressed in Standard 90.1 and strongly affect the size, type, and configuration of the HVAC system.

ASHRAE Standard 90.1[4] is constantly evolving, so it is important that the designer have the latest issue for reference. This chapter will outline the requirements of 90.1-2010, define critical terms, and discuss the implementation of energy efficient design.

Equipment Performance: AFUE, COP, EER, and SEER

AFUE – <u>Annual Fuel Utilization Efficiency</u>. This is basically the combustion efficiency of fossil fuel fired furnaces and boilers rated as "residential" but likely to be used in small commercial buildings. For conventional combustion equipment, AFUE will range from 75% to 85%. For condensing furnaces, which capture the latent heat of combustion, AFUE will be in the low 90's. Fossil fuel fired heating units classed as "commercial" are rated only by steady-state combustion efficiency at full fire.

COP – <u>Coefficient of Performance</u>. This is the ratio of heat removed or added by the HVAC unit in Btuh to the energy input to the unit in Btuh. It is usually applied only to electric heaters and heat pumps. Electric heaters have a COP of unity because all of the electric energy input appears as heat in the air stream. Reverse cycle heat pumps have a heating COP greater than unity because they transfer heat from outdoors to indoors.

HSPF – <u>Heating Season Performance Factor</u>. Basically, the "average COP" for an entire heating season, based on a standardized test defined by the Air Conditioning and Refrigeration Institute (ARI). It is applied to heat pumps with capacity of less than 65,000 Btuh.

EER – <u>Energy Efficiency Ratio</u>. This is the ratio of cooling capacity in Btuh to input energy in Watts. It is applied to cooling units with capacity of 65,000 Btuh or larger. Note that EER is simply COP multiplied by the factor to convert electric input energy to Btuh. Subscript 0 means output, i means input.

$$\text{Btuh} = \text{Watts} * 3.413$$

$$\text{COP} = \text{Btuh}_o/\text{Btuh}_i$$

$$\text{EER} = \text{Btuh}_o/\text{Watts}_i = \text{Btuh}_o\ /(\text{Btuh}_i/3.413)$$

and $Btuh_o * 3.413/Btuh_i = COP * 3.413 = EER$

SEER – Seasonal Energy Efficiency Ratio. This is the "average" EER for an entire cooling season, based on a standardized test defined by the Air Conditioning and Refrigeration Institute. It is applied to units with a capacity of less than 65,000 Btuh.

IPLV – Integrated Part Load Value. This is a seasonal "average" EER for units rated "commercial" (65,000 Btuh or larger).

LPD – Lighting Power Density. The total wattage of lighting per square foot in a particular space. Maximum lighting power densities are set forth in Standard 90.1 for various space occupancy categories such as offices, auditoriums and lobbies. The lighting power density of a space is required to compute the space design sensible cooling load and assign a supply air flow. The HVAC designer must review the lighting plan for code compliance as well as to perform cooling load calculations.

EUI – Energy Use Index. The total energy in Btu or Watts per unit area used by a building over some defined time period, usually a year. Typical units would be Btu/sf/year. For existing buildings, EUI is easily computed using utility data. For new buildings, EUI can be computed using building simulation software, often based on the code found in the building energy computer simulation DOE-2.

Meeting the Minimum Standard

Minimum requirements are set forth prescriptively in Standard 90.2. Most building codes now incorporate these requirements. Following are the most important requirements. Items in italics are the direct responsibility of the HVAC designer:

Minimum efficiency for air conditioners, heat pumps, and furnaces.

Fan power limitations.

Dual set-point thermostats. Thermostats must be capable of separate and independent set points for heating and cooling.

Seven day time clock control with manual override and night set-up/set-back capability.

Controls to prevent unnecessary activation of heat pump auxiliary heat strips.

Air side economizer. AHSRAE 90.1-2010 mandates economizers throughout the continental U.S. except South Florida, for all equipment larger than 54,000 Btu/hr. This provision may not be adopted in all local jurisdictions. See the Appendix for more information.

CO2 control of outdoor ventilation and exhaust air when high occupant density requires full-occupancy ventilation of 3000 cfm or more.

Minimum insulation requirements for walls, roofs, floors, *piping and ducts*. Maximum assembly U factors for opaque elements and fenestration, and maximum solar heat gain coefficients for fenestration. Because of the impact on the HVAC systems, *the HVAC designer must review the architectural drawings and notify the architect of any non-compliance with minimum insulation requirements.*

Maximum LPDs for indoor and outdoor lighting. *The HVAC designer must review the lighting plan for code compliance and notify the lighting designer of any deficiencies.*

Life Cycle Cost

Competing systems can be evaluated for Life Cycle Cost (LCC) using the following formula:

$$\text{LCC} = \text{First Cost} + \text{PWF} * P_a$$
$$\text{PWF} = ((1 + i)^n - 1)/(i * (1 + i)^n)$$

where P_a = annual operating and energy cost
PWF = Present Worth Factor
i = annual interest rate
n = useful life in years

Estimating LCC requires a computer program that can model the building and calculate annual energy costs. First cost can be estimated from the preliminary design documents. Operating cost savings can usually be manually estimated based on perceived recurrent labor savings, improved equipment life, and reduced equipment maintenance.

As noted in the chapter on Preliminary Design (Chapter 3), LCC calculations are rarely undertaken for the small projects that are the subject of this book. Usually, decisions on energy efficient design must be made based on the knowledge and experience of the designer, in consultation with the owner, architect, lighting designer, and occasionally the contractor. Some jurisdictions, (in particular Florida) use energy compliance software that computes annual energy use based on inputs for building insulation, fenestration, lighting, appliances, and HVAC systems. This software is used to verify energy code compliance when the project is ready for permitting. However it can be also be used by the designer for preliminary design estimates by setting up a mock building similar to that proposed, and then varying elements of the building and comparing the annual cost.

Implementing or Exceeding the Standard

Minimum Efficiency

In general, the air conditioning and heating equipment offered by U.S. manufacturers will meet or exceed the requirements of Standard 90.1. While this should be verified, the designer's challenge will be whether to select a unit that exceeds the minimum. Even though selecting improved efficiency will probably increase first cost, the owner's long term objectives are best met by selecting the highest efficiency unit that will satisfy the project psychrometric requirements. High SEER or EER cooling units may have high sensible heat ratio and poor moisture removal. This will show up with psychrometric analysis, but should also be reviewed for off design performance. Heat pipes can mitigate this problem.

Air Source vs Water Source

Water source heat pumps have inherently higher SEER/EER than air source units. However, the overall system efficiency will be degraded by the energy needed to circulate the water, and by the need for a heat sink and heat source. For boosted systems, the heat sink will be a cooling tower requiring fan energy and the heat source will be a boiler requiring fossil fuel or electric heating. Consider a fifteen ton heat pump with a unit cooling EER of 15 and a unit heating COP of 5. Rated cooling capacity is 175,000 Btuh, and heating capacity is 184,000 Btuh. The unit requires 43 gpm circulated with a .5 hp pump. For cooling, the cooling tower fan will be 1.5 hp. Assuming 80% efficient motors, the system cooling EER will be:

$$\text{EER} = \text{Btuh}_o / \text{Watts}_i = 175{,}000/(175{,}000/15 + (.5+1.5)*915) = 13$$

Assume an 80% efficient boiler to provide the heat pump source heat, remembering that the boiler must provide all of the heat for the system unless part of the building requires heat while part requires cooling. While this condition may occur frequently, most of the time the building will be in one mode or the other, so the heating COP will be:

$$\text{COP} = \text{Btuh}_o/\text{Btuh}_i = 184{,}000/(184{,}000/5 + .5*3125 + 184{,}000/.8) = .69$$

Note that the water source heat pump's huge heating advantage with its COP of 5 evaporates when the boiler is added in the boosted system. Therefore, this system should never be selected where heating may be a significant portion of annual energy use. On the other hand, a ground source water pump system will have superior efficiency over most air source systems in nearly all applications, because the only efficiency loss relative to the unit is the source water pump. See Chapter 3, Preliminary Design, for a brief description of ground water source systems.

Fan Power Limitations

The fan power limits in Standard 90.1 will usually be easily met by following the guidelines outlined in Chapter 12 in the subsection "Ductwork Layout and Sizing". The designer should in addition, where practical after consulting with manufacturer's reps, call for electric motors meeting the requirements of the Federal Energy Management Program (FEMP) to be used in all air moving equipment. The FEMP requirements exceed those in Standard 90.1.

System Controls

The requirements for system controls, thermostats, time-of-day control, and heat pump auxiliary heat, are discussed in Chapter 11, HVAC Controls.

CO2 Control of Outdoor Air Ventilation

CO2 control is only required by the Standard if the ventilation (outdoor air) requirement exceeds 3000 cfm. This limit may occasionally be exceeded in small assembly buildings such as churches, gymnasiums, or dance halls. Even when smaller levels of ventilation air are required, the designer may want to incorporate CO2 control to save energy and costs for the operator. HVAC contractors are familiar with CO2 control of intake air dampers, making it easy for the designer to specify. However, the designer must ensure that as ventilation air is reduced, exhaust air is also reduced, so that the building retains positive pressurization. If there is a large amount of required exhaust, such as may be the case if there are large public restrooms or a large kitchen exhaust, the ability to reduce the amount of ventilation air will be limited.

Economizer

An air-side economizer is a set of louvers and dampers that allow cooling a zone wholly or partly with outdoor air, when outdoor conditions permit. Appendix A is a discussion of economizer design, configuration, and control.

Standard 90.1 establishes eight climate regions, numbered 1 through 8, and a number of sub-regions designated with appended letters such as 2B, 4C, etc. (Appendix B of the Standard lists the climate zones for every state and county in the U.S.) Economizers are required for systems larger than 54,000 Btu/hr in all zones except 1A and 1B, which is basically the entire continental U.S. except south Florida.

Some jurisdictions, especially those in humid climates, may not adopt the economizer mandate. Economizers are not recommended by this writer for small DX projects if not required by local code. Small building owners rarely commission regular competent preventative maintenance, which is essential to maintain proper economizer operation and avoid calamitous malfunction.

Handling Large Latent Loads

In Chapter 7 it was seen that large latent loads will often require special processes such as outdoor air pre-treatment, heat pipes, and reheat. The most energy efficient of these processes will be heat pipes, which should always be considered before either outdoor air pre-treatment or reheat. The preferred method of outdoor air pre-treatment is the energy recovery ventilator, described in Chapter 9.

Off design conditions, such as low occupancy or low outdoor ambient conditions can defeat a moisture control strategy based on the design day if the result is long off times for the cooling apparatus. Dehumidification occurs only when the cooling system is running and the supply air is being passed over a cold coil. Therefore, the designer should try to always make provision for part-load operation either with compressor unloading or multiple compressors.

Hot gas reheat should only be considered if heat pipes or energy recovery cannot meet the design requirements, or if positive control of humidity is required. Fossil or electric reheat should be avoided at all costs.

Consultation With Other Disciplines

During the preliminary design phase, the HVAC designer should advise the architect and lighting designer on the following energy points:

> Explain to the architect and lighting designer that reducing lighting power densities below the Standard 90.1 maximums will have a profound effect on the size of the HVAC systems, besides reducing energy costs for the owner.
>
> Review the architectural design and advise the architect of tramp air sources such as gaps in the pressure and thermal envelopes at eves.
>
> Review the proposed building insulation for code compliance, good practice, and ac unit size reduction.
>
> Check that the thermal envelope either coincides with or is inside of the pressure envelope. Notify the designer of problems found.
> Advise the building designer regarding glazing – low e, insulated, tinted. Large glass surfaces can cause occupant discomfort if radiation from the glass is not mitigated by multi-pane insulation (winter) and tinting (summer).

Additional Reading

Advanced Energy Design Guide for Small Office Buildings, 2004, ASHRAE[14]

END

Chapter 9

Equipment Selection

Selection First Steps

Selection and Sizing of the air conditioning, heating, and ventilating equipment is at once the most difficult and the most important task in the design process. A rational, successful selection depends on the designer's understanding of cooling load, cooling coil capacity, experience and judgment.

The first step in this task was taken in Chapter 3 during the preliminary design phase. At that point, based on the wishes of the owner and architect, and on his own judgment and experience, the designer made a general determination of the type of systems he would use and where the elements would be located. In Chapters 4, 6, and 7 the engineer defined the HVAC characteristics of the building. In this chapter, we will outline the remaining steps.

The Iteration

Selecting the equipment is an iteration. Figure 9-1 is a flow diagram of the selection iteration procedure. Following this procedure for each zone should result in HVAC system design that will match the building characteristics and provide an acceptable environment for the occupants. The remainder of this chapter discusses each of the flow chart elements.

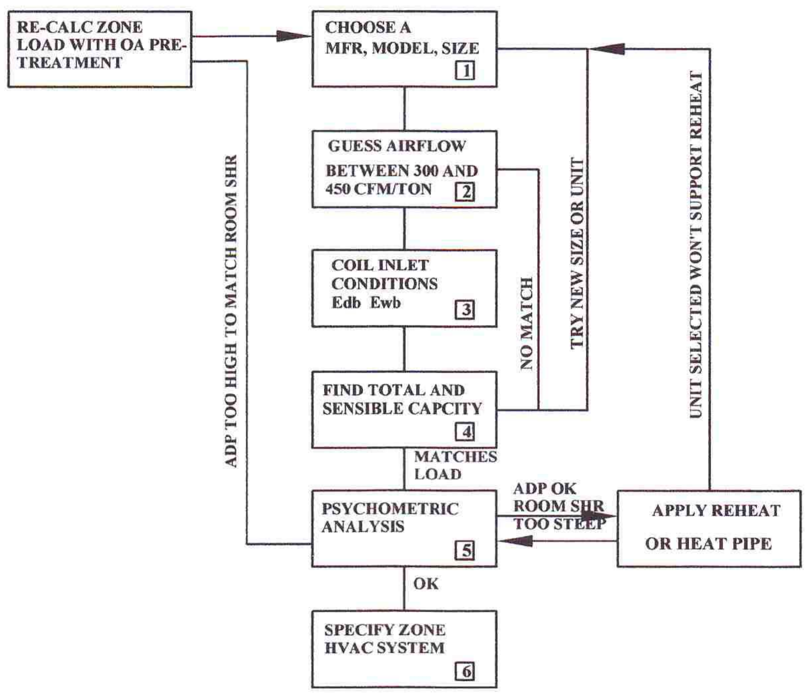


FIGURE 9-1, HVAC EQUIPMENT SELECTION FLOW CHART

Step 1 - Choose a Manufacturer, Model, and Size

The abundance and variety of systems available in the HVAC market will at first seem daunting and bewildering. Where to start? Start with the type of occupancy and the zone conditions you are trying to satisfy. An excellent resource is your local manufacturer's representative. As you gain experience, you'll find that some manufacturer's products are easier to apply to certain types of projects than others. An example is trying to fill the capacity gap between five tons and 7.5 tons. Also, you'll find that you can work better with some manufacturer's reps than with others.

Most manufacturers provide detailed engineering data on their product line. Search on-line for the manufacturer's name, and you'll be led to a site with links to their HVAC product data, contact information for your local representative, local dealers and distributors, and much more. The HVAC product data is almost universally available either on CD or for download as a .pdf file, which you can store on your hard drive for reference. Some manufacturers offer CD or downloadable software that can be used for equipment selection. Your local representatives will be most happy to furnish you with any of the data or software provided by their manufacturer, and will usually run equipment performance points for you based on your zone design conditions.

But whether you plan to do your own performance analysis, or have the work done by a rep, you must narrow your first guess choice of equipment down to a specific model. A rep can help you make this choice, based on the cooling loads, coil and room sensible heat ratios, and any off-design considerations you may want to provide for. For these small projects, the latter will be more qualitative than quantitative – how will the proposed equipment handle reduced loads, much lower occupancies, etc. For example, a projected variation in occupancy load may lead to a choice of a variable air volume unit, a unit with multiple compressors and circuits, or selection of multiple smaller units whose total output will meet the design conditions.

Step 2 - Air flow Iteration, Guess Airflow

Having selected a make and model, the design point performance must be found using manufacturer's software or printed performance data. The next step, using the manufacturer's performance data for the selected make and model, is to guess the air flow. An example of printed data for a five-ton air cooled dx split system heat pump is shown below. Note that the performance - sensible cooling capacity and total cooling capacity - is a function of outdoor air temperature, coil inlet conditions, and air flow. For a given model and size, the range of air flow is limited. Most manufacturers recommend against extrapolating data outside the range of coil inlet conditions and air flow provided. In general, the first guess for air flow must be between 300 and 450 cfm per ton. Below 300 cfm per ton, there is danger of the moisture in the air freezing on the coil, eventually blocking it entirely. Above 450 cfm per ton, dehumidification may be too low to prevent moisture build-up in the zone, and to match the sensible load with a lower air flow, a larger unit will be needed.

For best dehumidification, the smallest unit with an air flow below 350 cfm per ton that will satisfy the coil design sensible heat load is optimum. However, if coil sensible heat load is high, then a smaller unit operating at a higher air flow may be the best solution. This sets up the first two nested iteration loops shown on steps 1 – 4 of figure 9-1. A unit of a given size is tested at several air flows. If a solution matching the coil sensible and latent loads is not found, then the process is repeated with a larger unit of the same model, or possibly by a different model and/or manufacturer. Eventually, a unit and supply air flow that matches the coil design loads will be found. However, this does not mean that the equipment will actually be able to satisfy the room load.

DETAILED COOLING CAPACITIES

EVAPORATOR AIR		CONDENSER ENTERING AIR TEMPERATURE, deg F											
		75			85			95			105		
CFM	EWB	CAPACITY Mbtuh		Total Sys. KW	CAPACITY Mbtuh		Total Sys. KW	CAPACITY Mbtuh		Total Sys. KW	CAPACITY Mbtuh		Total Sys. KW
		Total	Sensible		Total	Sensible		Total	Sensible		Total	Sensible	
FIVE TON HEAT PUMP OUTDOOR UNIT WITH SPECIFIC MATCHING FIVE TON INDOOR UNIT													
1750	72	71.08	36.09	4.41	67.69	34.77	4.86	64.09	33.39	5.36	60.33	31.96	5.90
	67	65.19	45.05	4.36	62.05	43.69	4.81	58.73	42.27	5.30	55.30	40.82	5.85
	63	60.79	43.67	4.31	57.85	42.31	4.76	54.78	40.90	5.26	51.57	39.45	5.81
	62	59.61	53.90	4.30	56.75	52.49	4.75	53.77	51.00	5.25	50.69	49.41	5.80
	57	57.46	57.48	4.28	55.20	55.20	4.74	52.79	52.79	5.24	50.24	50.24	5.79
2000	72	72.12	37.62	4.54	68.55	36.26	4.99	64.87	34.89	5.48	60.98	33.44	6.03
	67	66.24	47.72	4.48	62.96	46.34	4.93	59.50	44.89	5.43	55.93	43.42	5.97
	63	61.83	46.19	4.44	58.77	44.80	4.89	55.56	43.36	5.38	52.22	41.88	5.93
	62	60.73	57.58	4.43	57.79	56.05	4.88	54.77	54.33	5.38	51.88	51.88	5.93
	57	59.63	59.63	4.42	57.18	57.18	4.87	54.60	54.60	5.38	51.89	51.89	5.93
2250	72	72.83	39.05	4.66	69.15	57.69	5.11	65.40	36.31	5.61	61.39	34.85	6.15
	67	66.96	50.26	4.61	63.58	48.86	5.06	60.01	47.39	5.55	56.35	45.89	6.09
	63	62.56	46.57	4.56	59.40	47.15	5.01	56.09	45.69	5.51	52.66	44.18	6.05
	62	61.68	60.83	4.56	58.79	58.79	5.01	56.05	56.05	5.51	53.19	53.19	6.05
	57	61.37	61.37	4.55	58.79	58.79	5.01	56.06	56.06	5.51	53.19	53.19	6.05

Table 9-1, Typical AC Unit Performance

Step 3 - Coil Inlet Conditions

At this point in the selection process, the known parameters are the state of the room air, the state of the outdoor air, the required outdoor air flow, and an estimated supply (coil) air flow. Using these parameters and the method set forth in Chapter 7, equations 7-8 and 7-9, the coil entering dry bulb and entering wet bulb temperatures can be determined. Using a table similar to table 9-1, or using manufacturer's selection software, the coil sensible and total cooling capacity may be found.

Step 4 - Coil Capacity

At this stage in the iteration, the coil capacity is compared to the coil load, as represented on the psychrometric chart (Figure 7-2) by the process line **3 – 1A**. There are three possibilities;

a) The total coil capacity is greater than the total coil load, and the sensible capacity is equal to or less than the sensible load.

b) Both the total and sensible capacities are greater than the load.

c) The total coil capacity is less than the total coil load.

With situation a) above, if the sensible capacity is within 15% of the sensible load, then this may be an acceptable selection, and is ready for psychrometric analysis. A small deficiency of sensible capacity is actually beneficial, since it will improve dehumidification on cooler days, and the unit will simply run longer on extreme days (also enhancing dehumidification). If sensible capacity is more than 15% low, return to step 2 with a higher air flow.

With situation b) when the sensible capacity is greater than the load, this undesirable condition should be remedied by trying a new lower air flow or choosing a smaller unit. Excess sensible capacity will reduce run times, causing humidity in the space to rise during compressor off times. Keep in mind that some of the dehumidification schemes discussed here, low air flow and heat pipes in particular, only work when the compressor is running. Return to step 2 with lower air flow.

Keep in mind that each time the air flow guess is changed, the coil inlet dry bulb and wet bulb temperature change, since ventilation air flow is kept constant.

With c), a low total capacity means that even with the compressor running, dehumidification may be inadequate. If sensible capacity is low, a higher air flow may work, although total capacity is only weakly dependent on air flow. In all probability, a larger unit must be tried, requiring a return to step 1.

It may not be possible to satisfy a requirement of higher total capacity than load, <u>and</u> equal or lower sensible capacity. If that is the case, then a sensible capacity no more than 5% higher than the load can be tolerated, or the unit can be modified by reheat or a heat pipe.

Step 5 – Psychrometric Analysis

When a match that satisfies the criteria of the previous section is found, then psychrometric analysis should be performed to ensure that the unit can satisfy the room set point at the design condition. The ideal condition would be similar to that shown on figure 7-2, where the coil process line crosses the room process line and both terminate at a point near the saturation line at a temperature lower than 60 degrees. This will be the case with a building having a low occupant density and moderate outdoor air flow – less than 20% of supply.

If the occupant density is high, as in an auditorium or classroom, then the room process line may terminate at high dry bulb temperature and a low dew point as shown on figure 7-4. In this case, it would not be possible to match the coil load, as represented by a process line from point 3 to point 1A, with a conventional DX air conditioner. A unit that matches the latent load will have greatly excessive sensible capacity and as a result the total capacity will also be much larger than the load, although, unlike excessive sensible capacity, this is not a problem. One solution is to apply reheat to the ac unit to “false load” the unit and reduce the sensible capacity without affecting the latent capacity.

Reheat

Reheat is the simple process of heating the air leaving the coil at constant dew point before delivering it to the occupied space. Free reheat uses the heat rejected by the cooling process, and therefore does not require additional energy input. However, reheat may be accomplished by any heating process – hot water, electric heat strips, natural gas, or even oil or LP gas. Electric reheat is effectively prohibited by the Florida Energy Code, and possibly by other jurisdictions as well. Gas or oil reheat requires a furnace in the supply duct downstream of the cooling coil, and because the burner sections, being vented, can be damaged by condensation from the cold supply air stream when reheat is off, they must be stainless steel. Also, LP gas, because it is heavier than air, should not be used in unoccupied enclosed spaces such as plenums, basements, or attics, where a gas leak can "pool". Hot water or steam reheat requires a boiler operating during times when no space heating is required, exacerbating the energy penalty.

As indicated on Figure 7-4, there are two methods of "free" reheat. One is to pass the warm liquid condensate through a reheat coil before it is sent to the evaporator. This is rarely used, because there is not a great deal of available heat from the condensate. However, at least one manufacturer offers commercial packaged units with liquid line reheat as an option. Another manufacturer offers a liquid line reheat option on small residential and light commercial split systems. This is a viable solution when the sensible heat ratio of the cooling coil only needs a small reduction to match the system total and sensible loads. Liquid line reheat is an uncontrolled option, and thus the unit performance is published with the reheat included.

The preferred method of reheat is with all or part of the hot gas from the compressor passed through a heat exchanger in the supply air stream before it is sent to the condenser. This requires both a thermostat and a humidistat for control to prevent overheating the space, so manufacturers typically publish performance tables with and without reheat.

Hot gas reheat is typically offered as an option on packaged ac units – never on air cooled heat pumps. It is also offered on most water cooled heat pumps. Temperature/humidity controls are often included by the manufacturer as an additional option. Hot gas systems can usually provide much greater reheat than is needed to match the room conditions, but this can be checked by using the manufacturer's published reheat performance, which uses the same cooling coil entering parameters as the standard performance. In general, the system must be able to provide more reheat than would be indicated by the temperature difference between points **4** and point **1A** on figure 7-4, and the published sensible and total capacities with full reheat should be less than the room sensible and total loads. That is, point **4A** with full reheat should be at a higher temperature than point **1A**, while close to the same dew point. Remember that point **4A** as shown on figure 7-4 assumes that the system control will modulate reheat as needed to fall on the room process line, near point **1A.**

Hot gas reheat can be applied to small commercial split systems, but this is not an option offered by any of the manufacturers. To apply it to a split system, the design engineer must provide a refrigerant piping schematic and control schematics as part of the design documentation. The air conditioning contractor must have experience with hot gas reheat piping and controls to ensure proper installation and operation. Installation of hot gas reheat may affect the manufacturer's warranty. These considerations, the restriction against electric reheat, and the disadvantages of other methods of reheat, make the heat pipe an attractive solution.

Heat Pipes

As shown in figure 7-1, a heat pipe consists of two coils connected in such a way that heat is transferred passively from the entering air stream to the leaving air stream. The heat pipe is very efficient at doing this, and the heat pipe vendors have developed simple calculation procedures to evaluate state points **3**, **4**, and **4A** given conditions at state point **3A** and the **design Δt** of the heat pipe. Figure 7-7 is a psychrometric representation of an actual heat pipe application.

Heat pipes are not a factory option. A heat pipe must be fitted to a unit after purchase, and the installation is usually performed at the heat pipe manufacturer's facility. Heat pipes are defined by the **Δt** and coil face area, which in turn defines the number of rows in each heat pipe coil.

Specifying a heat pipe is basically specifying the **Δt**, shown on Figure 7-7 as the temperature difference between point **3A** and **3** and point **4** and **4A.** When a heat pipe is involved, point **3A** becomes the state point entering the ac unit, while point **3** remains the coil entering state point. Likewise, point **4** remains the coil leaving state, and point **4A** becomes the state of the supply air leaving the ac unit. Refrigerant fluid circulates between the entering heat pipe coil and the leaving heat pipe coil, cooling the entering air stream and re-heating the leaving air stream by the same **Δt.** The **Δt** used in Figure 7-7 is 10°.

To clarify the difference between a heat pipe and conventional reheat, refer again to Figure 7-7. The green line on Figure 7-7 represents the ac unit that will satisfy the room and coil load conditions with a properly selected heat pipe. For reheat, a larger unit would be selected that would have coil leaving air at point **4**, and then reheat would be applied as shown on Figure 7-4. This illustrates that a heat pipe allows selection of a smaller ac unit than would be necessary if reheat alone is used. However, the starting point is similar – a unit that at least satisfies the total heat load, but has excess sensible heat capacity. Suppressing the coil entering air temperature effectively reduces the sensible capacity of the coil and increases the latent capacity, thus decreasing the sensible heat ratio.

Heat Pipe Selection

Selecting the ac unit size and the heat pipe Δt is a trial and error process. A candidate unit will match or exceed the total cooling load, and will also have excess sensible capacity. In order to compute the heat pipe state points using the heat pipe manufacturer's formulas, it is necessary first to compute points **3** and **4** without the heat pipe, using equations 7-8, 7-9, 7-14 thru 7-16.

It is clear that the heat pipe changes the coil entering conditions. The mixed air point is cooled at constant dew point, **tdp,** so the entering dry bulb temperature, enthalpy, and dew point are changed as follows:

$$tdb_3 = tdb_{3A} - \Delta t \tag{9-1}$$
$$twb_3 = f(tdb_3\ ,\ tdp_{3A}) \tag{9-2}$$

Remember that tdb_{3A} and twb_{3A} are the mixed air conditions before the heat pipe or other pre-treatment, and are found by equations 7-8 and 7-9. In this book, point **3** is always the state of the air entering the cooling coil itself.

Continuing, the state points defined by computing points **3** and **4** without the heat pipe will be designated with the prime symbol:

tdb_3', twb_3' = The mixed air condition at the coil inlet without the heat pipe found from equations 7-8 and 7-9.
tdb_4', twb_4' = The cooling coil leaving conditions without the heat pipe found from equations 7-14, 7-15, and 7-16.

Noting that $tdb_3' = tdb_{3A}$ and $twb_3' = twb_{3A}$ and supply air flow is the same with and without the heat pipe.

At this point, we do not have all of the state conditions needed to find the leaving conditions (point **4**) of the cooling coil. It can be seen from figure 7-7 that twb_3 can be found as a function of tdb_3 and the dew point temperature at point **3A**. However, the presence of the heat pipe modifies the performance of the cooling coil, so that tdb_4 and twb_4 must be found using equations provided by the heat pipe manufacturer. Following are functional relations based on the equations of one heat pipe manufacturer, Heat Pipe Technology, Inc., which specializes in servicing the air conditioning and building systems industry. There are others listed in Thomas Register[R] and other industry guides, which may have different functional relationships.

$$twb_4 = f(twb_4', \Delta t) \tag{9-3}$$
$$tdb_4 = f(twb_4\ , tdb_4', twb_4', \Delta t) \tag{9-4}$$

The exact form of these functional relationships is not given here, as it may vary depending on the manufacturer. It is recommended that the designer select a manufacturer and follow the analysis procedure provided in his literature, or have a manufacturer's representative provide the analysis based on input from the designer.

$$tdb_{4A} = tdb_4 + \Delta t \qquad (9\text{-}5)$$
$$twb_{4A} = f(tdb_{4A}, tdp_4) \qquad (9\text{-}6)$$

Referring to figure 7-7 the solution to the heat pipe selection must be that point 4A is at the same or lower dew point than point 1A, and is near the dry bulb temperature of point 1A. To achieve this, it is necessary to iterate on the heat pipe Δt, as well as on the unit selection and supply air flow. An easy way to short-cut this procedure is to use the ac unit manufacturer's data to find the sensible and total cooling capacity of the selected ac unit with varying estimates of Δt until a close match is found to the coil sensible and total cooling loads. This is illustrated by Figure 9-2, which is a flow chart showing the selection procedure graphically. Note that the output of step 5 on figure 9-2 is not used to compute the coil leaving conditions, but instead, steps 4 and 5 provide a short cut to finding a viable heat pipe that will be able to satisfy the design conditions. In fact, once the heat pipe analysis step is entered, the coil sensible and total heat capacity is irrelevant.

Once the coil leaving state has been determined in step 6, the result should be plotted on the psychrometric chart to verify that point 4A can satisfy the room design conditions.

Outdoor Air Pre-Treatment

Outdoor ventilation air pre-treatment may be necessary when ventilation air is a large percentage – 30% or more – of the total supply air. It will be necessary if the designer, having exhausted the methods presented earlier in this chapter, is unable to find a satisfactory match to the building loads and required supply air state points. There are two primary methods of pre-treatment, the enthalpy or total heat wheel, or specialized DX air conditioners designed to handle air at high dry bulb and wet bulb temperatures, and to operate over a wide range of ambient temperature and humidity. A number of manufacturers offer this equipment, often in various combinations of enthalpy wheels and dx coils.

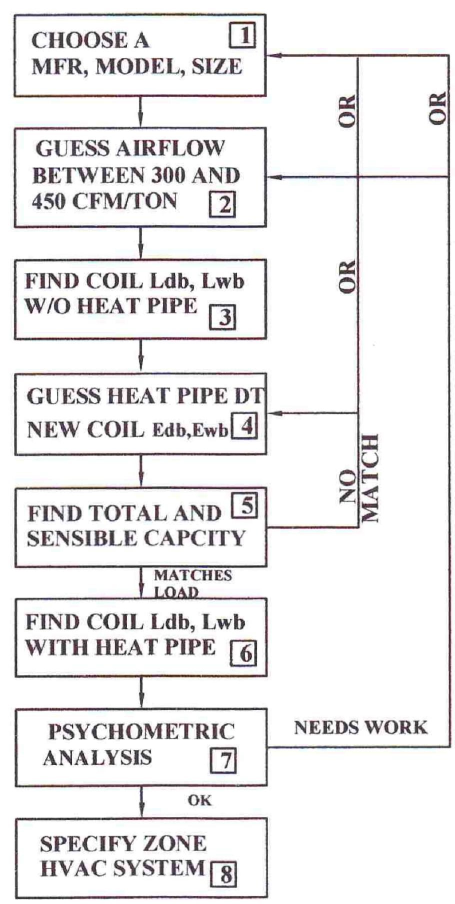


FIGURE 9-2, HVAC EQUIPMENT WITH HEAT PIPE - SELECTION FLOW CHART

Because of the variety of equipment available, this chapter will not go into detail on the selection of pre-treatment systems. Instead, a brief outline of the two principal systems will be presented, along with procedures for integrating the outdoor air pretreatment system with the building space cooling and heating systems.

Outdoor Air Design Condition for Pre-treatment System Selection

At this point is becomes useful to point out that when the minimum outdoor air becomes a large percentage of the supply air, the risk of moisture and mildew problems is greatly increased. Therefore, the designer may want to consider sizing the pre-treatment equipment using the Monthly Design Wet Bulb and Mean Coincident Dry Bulb data from the ASHRAE weather tables. This results in significantly higher latent loads than the Monthly Design Dry Bulb with Mean Coincident Wet Bulb data used to size a space cooling unit. It will therefore result in a larger and more expensive, but possibly more effective, pre-treatment system.

If the outdoor air pre-treatment equipment is sized and selected using the Monthly Design Wet Bulb and Mean Coincident Dry Bulb temperatures, it must be re-analyzed with the cooling load design day Monthly Design Dry Bulb and Mean Coincident Wet Bulb temperatures. This allows consistent application when selecting the building zone cooling system.

Enthalpy Wheels - Energy Recovery Ventilators

Enthalpy wheels transfer heat and moisture from the incoming ventilation air stream which is at high temperature and humidity to the exhaust stream which is at room temperature and humidity, and are generally referred to as energy recovery ventilators. Figure 9-3 is a diagram of an energy recovery ventilator with separate ventilation and exhaust fans. The energy transfer from the outdoor air stream to the exhaust stream is shown by the relative changes in the psychrometric state of the air streams as they pass through the wheel. In effect, the outdoor air is pre-cooled and dehumidified as its heat and humidity is transferred to the cool, dry exhaust air. One aspect of the energy recovery ventilator that may limit its applicability is the requirement to gather all of the zone exhaust streams to pass through the energy wheel. In the example shown, note also that the exhaust is 80% of the ventilation air, as required to properly pressurize the building.

Referring to figures 7-1 and 7-2, the output of the Energy Recovery Ventilator (ERV) becomes point 2 on the psychrometric chart. Because the energy recovery ventilator reduces the enthalpy of the ventilation air, the size of the air conditioning equipment serving the zone is reduced relative to a unit where the return air is mixed directly with the outdoor air. Because the required ventilation air is fixed, the ERV can be selected on the basis of the ventilation air cfm alone, without regard to the room cooling load and without selecting the cooling system.

After selecting an ERV, the designer must re-calculate the cooling coil load and select (or re-select) the cooling system with the reduced ventilation air dry bulb and wet bulb temperatures. Likewise, the winter heating load is reduced because the ERV transfers heat from the warm exhaust air to the colder outdoor air, before the outdoor air is mixed with the return.

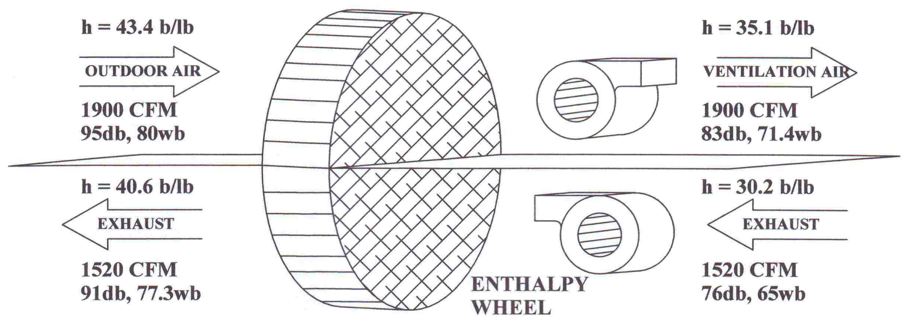


FIGURE 9-3,ENERGY RECOVERY VENTILATOR
crossflow neglected, casing and filters not shown

In selecting an energy recovery ventilator, the designer must take into account not only the ventilation air cfm required, but also the physical size of the ventilator and the space allotted for the zone ac unit. Ventilators must be in close proximity to the air handler or packaged unit. In general, the larger the ventilator, the more efficient, and the more outdoor air cooling load is transferred from the zone ac unit to the ventilator. However, ventilators can be quite bulky, and are sometimes larger than the zone ac unit they serve. The physical dimensions of a unit may preclude its application, and force the designer to consider a DX dedicated outdoor air unit (DOAU).

To summarize, if an energy recovery ventilator is being considered, the following factors will be controlling:

1. The designer has made a preliminary determination that sufficient space is available for the ERV.
2. All of the zone exhausts can be combined for ducting into the ERV.
3. An ERV is selected based on required ventilation cfm and desired efficiency.
4. A zone air conditioner is selected based on the ventilation air output from the ERV.
5. The zone air conditioner/ERV combination results in a satisfactory psychrometric process.

DX Dedicated Outdoor Air Unit (DOAU)

The title of this subsection starts with the term “DX” because this is the type of outdoor air unit most applicable to small HVAC systems that need ventilation air pre-treatment and cannot use ERV systems because of space considerations, dispersed exhaust systems, or lack of adequate dew point suppression. There are systems for 100% outdoor air that use activated or regenerated desiccant, chilled water/steam, or chilled water/hot water components. The desiccant systems are generally needed only if precise control of space temperature and relative humidity to low levels is needed, as for laboratories or operating rooms. Hydronic systems are rarely applied to small commercial projects, and so are not addressed in this book.

DX outdoor air units process the ventilation air by first cooling it to the desired dew point, then re-heating with hot refrigerant gas. In some cases several stages may be used, or units are supplied with hot gas bypass to allow a single cooling stage to be designed for high entering dry and wet bulb temperatures but to still be able to operate at cool, moist outdoor air conditions with adequate head pressure. For cold weather, the designer must decide whether to specify auxiliary heating for the outdoor air unit. Many small HVAC projects use heat pumps as the primary AC equipment, and the heat pump may be able to supply adequate heat when space heating is needed to compensate for unheated outdoor air. If the designer decides that space heating during cold periods could be a problem, he can specify electric, gas, or hot water heat to temper the ventilation air during cold periods.

Outdoor air units are selected to deliver the required minimum ventilation air at a maximum selected dew point. The designer should consider several factors when selecting the dew point. First is the capability of the space cooling unit to handle a ventilation air dew point higher than the room design dew point – point 1 on the psychrometric chart. Also, since reheat is part of the DX outdoor unit package, it is possible to deliver the ventilation air at the same temperature as the room design air, thus relieving the space unit of the need to handle the outdoor air sensible load. Second, the lower the dew point selected for the DX unit to deliver to the space air handler, the larger and more expensive is the outdoor air unit. This book recommends selecting a unit that can supply a dew point that matches or is slightly higher than the room dew point, thus allowing some of the outdoor air moisture to be removed by the space air conditioner.

The designer should specify the outdoor air unit supply air dry bulb temperature. This is generally controlled by a sensor in the supply air stream that will modulate the reheat as needed. By carefully selecting the outdoor unit dew point capability and specifying the outdoor air unit supply dry bulb temperature, the designer can tailor the two systems – outdoor and indoor – to match the capacity of the indoor space air conditioner. END

Chapter 10

Air Distribution

Principles of Air Distribution

The user of this manual will understand that air is the heat transfer fluid that adds or removes heat to or from a building zone. In cooling mode, the air supplied by the air conditioner removes the sensible and latent heat being added to the zone as described in Chapter 5. In Chapter 9, it was noted that zone air flow is determined by the magnitude of the sensible and latent loads, by the ratio of sensible to total load, and by the performance characteristics of the air conditioning system. The distribution of air conditioning supply air to the rooms in a zone is based on the distribution of sensible heat load between the rooms. The sensible loads were determined in Chapter 6, and the zone air flow was established in Chapter 9. This chapter discusses using that information to set the required air conditioning supply air flow to each room or space in a zone at the selected design conditions.

Air Distribution Example

The principles of air distribution can best be determined by example. Figure 10-1 is a hypothetical small single zone office building in the Southeast. The design day and building load characteristics are shown below:

Design Conditions:

outdoor air dry bulb temperature, °F	95
outdoor air wet bulb temperature, °F	78
indoor air dry bulb temperature, °F	76
indoor air wet bulb temperature, °F	65
minimum ventilation (outdoor) air flow	150
latitude, degrees north	32
design day of year	July 21
hour of the design day (apparent solar time)	4:00 PM

Building Cooling Load Characteristics:

exposure	north	east	south	west
wall U value, Btu/hr/ft^2/°F	.05	.05	.05	.05
wall color	med	med	med	med
net wall area, ft^2	475	405	640	800
window U value, Btu/hr/ft^2/°F	.62	.62	.62	.62
window solar heat gain coefficient	.72	.72	.72	.72
window area, ft^2	35	79	45	0
roof/ceiling U value Btu/hr/ft^2/°F	.03	.03	.03	.03

Building Cooling Load Characteristics (continued):

roof area, ft^2	-------------------------1286 ------------------------ -			
door conductance $Btu/hr/ft^2/°F$	.45	.45	.45	45
door area (north only) , ft^2	20	0	0	0

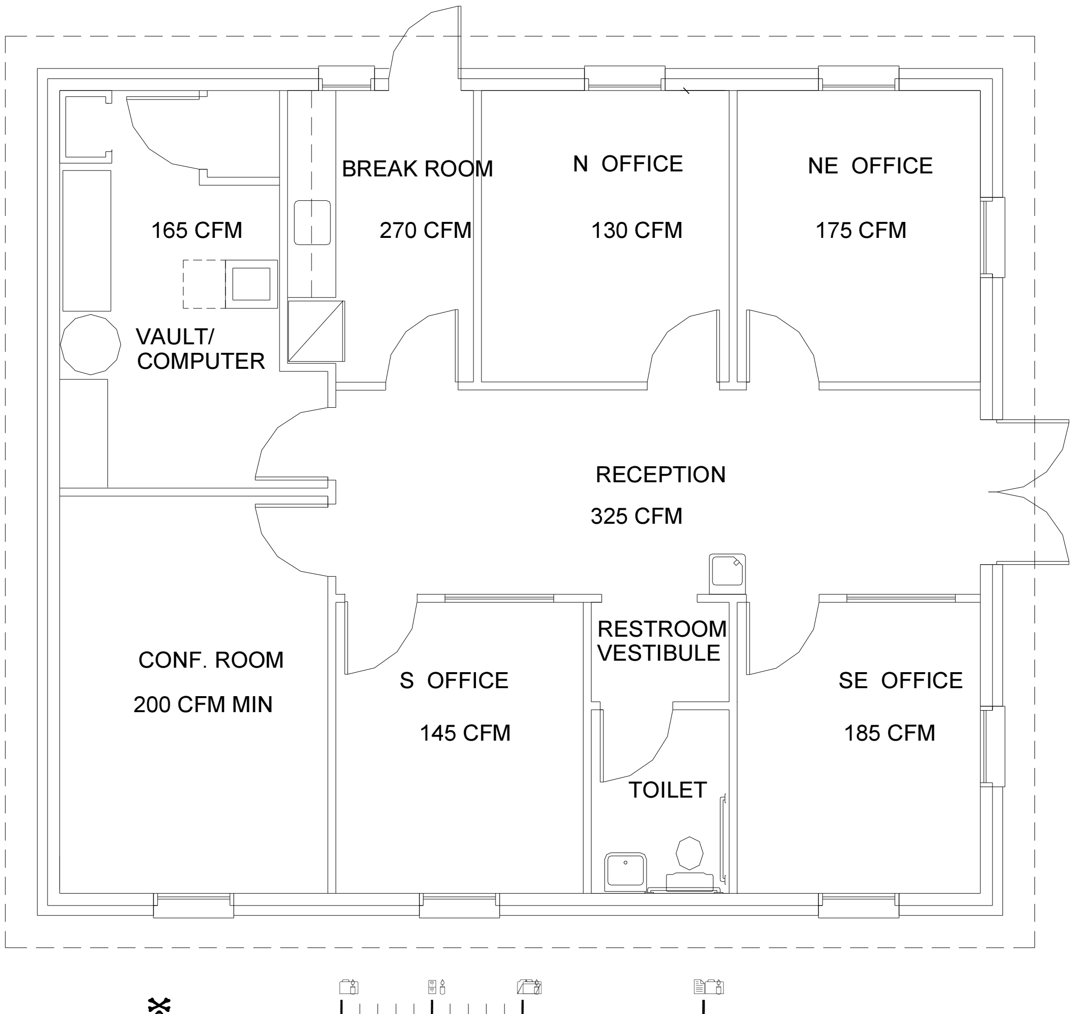


FIGURE 10-1 - SMALL OFFICE

1/8"=1'-0"

EXTERIOR DOOR SCHEDULE					
MARK	TYPE	SIZE	MATERIAL	FINISH	LABEL
1	EXTERIOR STOREFRONT	84x84	ALUM	BRONZE	NONE
2	EXTERIOR PANEL	36x80	WOOD	WOOD STAIN	NONE

WINDOW SCHEDULE					
MARK	TYPE	SIZE	GLAZING	FRAME	COMMENT
A	SINGLE HUNG	36x60	LOW E IG	ALUM	THERMAL BRKS
B	SINGLE HUNG	18x42	LOW E IG	ALUM	THERMAL BRKS

The ceiling height in the building is 9'. The east window area includes the 7'x7' all glass "storefront" that is the main entrance. The only opaque door is the 3' x 6'-8" door in the break room wall. Note that the restroom is not included in the above tabulations. This is because restrooms normally do not require air flow from the air conditioning supply. Most codes prohibit recirculating restroom air, so the only air flow is usually that induced under the door or through a door grille by an exhaust fan. Large restrooms with exterior windows may require cooling air flow, in which case the supplied air flow must be substantially less than the exhaust air flow to preclude recirculation.

<u>Building Sensible Load Elements:</u>

space	offices	lobby	conf	vault	break
design occupant load	4	3	0	0	4
total lighting load, Watts	768	288	288	40	192
misc electrical, Watts	576	125	215	650	895

The sensible portion of an infiltration air load should also be considered an internal load, but since it is assumed that the building is pressurized, infiltration load is assumed to be zero. Note also that the restroom loads are again left out of the tabulation, because no cooling air will be supplied to the restroom other than that induced from the occupied rooms of the building by the restroom exhaust fan.

Note that the conference room occupancy is set as zero for the load calculation. This is because the conference room is occupied only for relatively short periods, always with at least some of the occupants already accounted for in the offices and lobby, and often with no other persons. To handle this variable load - there could at times be up to 8 persons in the conference room for up to four hours – a variable air flow diffuser has been selected for this space. The test and balance technician is directed to set the conference air flow with the variable diffuser valve in the minimum position. The diffuser selected has the capability to respond to sensible heat load and to open to increase air flow as needed. While this air flow will be "robbed" from the rest of the building, this should not be a problem, since at least part of the conference room occupancy comes from other spaces in the building.

On the other hand, the break room occupancy is set at four, even though most of the time those four persons will be occupying other spaces in the building. However, a significant portion of the sensible load in the break room is heat dissipated from appliances and vending machines, and a variable air flow diffuser is not deemed to be necessary. The vault occupancy is set at zero because occupancy of that space is intermittent and infrequent. Most of the sensible load in the vault is, like the break room, heat dissipated from equipment, in this case copiers and the server.

A load calculation was run on the building above (Chapters 6 and 7, including the restroom), and a supply air flow was determined with the selection of an air conditioning unit (Chapter 9) as follows:

sensible cooling load, Btu/hr	34,032
total cooling load, Btu/hr	42,522
sensible cooling capacity of the selected ac unit, Btu/hr	33,292
total cooling capacity of the selected ac unit, Btu/hr	46,043
supply air flow determined by Chapter 9, cfm	1,600

Distributing Air Flow to Each Space

Many, in fact most, canned cooling load calculation programs automatically calculate space sensible cooling loads for each space, and even estimate cooling air flow. The air flow estimates are of little use, because the air flow is seldom close to that calculated during equipment selection, and because the air flows include rest rooms, which generally do not receive cooling air directly. However, the sensible heat loads are very useful, and can be used to proportion the calculated air flow to all of the spaces that need it.

In this example case, the individual space sensible loads were not calculated, although the tabulations of Building Cooling Load Characteristics and Internal Sensible Cooling Load Elements were necessarily made in order to calculate the building zone loads. This building has only one zone, which includes the entire building. This information is therefore available to prepare a spread sheet that can be used to calculate the individual space air flows after further break down, as follows.

Table 10-1

Sensible heat loads -Btuh							
room	sta no	wall/roof	windows	people	lights	misc.	total
n office	1/144	1159	494	250	655	246	2804
ne office	2/144	1159	1360	250	655	246	3670
s office	3/144	1159	805	250	655	246	3115
se office	4/144	1159	1671	250	655	246	3981
lobby	5/250	2012	2830	750	983	426	7001
conf	6/215	1730	805	0	983	734	4252
vault	7/145	1166	0	0	137	2218	3521
break	8/100	804	173	1000	655	3055	5687
Total	1286 sf	10629	8010	2750	5379	7263	34031

In the table above, the miscellaneous electrical power consists of computers, appliances, and office equipment. Estimated computer heat loads are found by estimating the number of computer elements in each space using data in the ASHRAE Fundamentals Handbook. The wall/roof sensible cooling load Qwr is found by subtracting all of the assignable loads – windows(Qw), people(Qp), lights(Ql), and miscellaneous power(Qm) - from the total room sensible cooling load for the zone. This is then apportioned to the various spaces by the ratio of each space area to the total floor area – again excluding restrooms and other areas not receiving cooling supply air.

To express this mathematically: (The subscript “x” refers to the room station numbers in Table 10-1)

$$Qwr_x = A_x/A_{TOT} * (Q_S - Qw - Qp - Ql - Qm) \qquad (10\text{-}1)$$

In the formula above, Q_S is the building sensible load, not the ac unit sensible capacity.

With the tabulation of Table 10-1, it is possible for us now to apportion the cooling supply air flow to each to each of the rooms in the zone. as shown on Table 10-2. The air flow to each room is the same percentage of the total air flow that the sensible load of each room is to the total sensible load on the building. The air flows tabulated below are shown on Figure 10-1.

Table 10-2

Small Office Air Flow Distribution

room	sta no	Qs_x - Btuh	Qs_x/Qs_{tot}	C_x - cfm
n office	1	2804	.082	130
ne office	2	3670	.108	175
s office	3	3115	.092	145
se office	4	3981	.117	185
lobby	5	7001	.206	330
conf	6	4252	.125	200
vault	7	3521	.103	165
break	8	5687	.167	270
total		34031	1.000	1600

Caveats

This air flow distribution is subject to several caveats. First, it is clearly only strictly applicable on the design day, at the design hour – in this case, July 21 at 4:00 PM apparent solar time. Since the largest glass area is on the east side of the building, it can be expected that the morning loads on the spaces with east facing glass will be greater during the morning hours than at the design hour. Also, cooling loads on the east and south sides can be expected to be proportionally lower during the majority of the time when outdoor temperature is less than at the design condition. This is why, if the building is large enough, or if a particular wall has an unusually large expanse of glass, zoning by orientation is a good idea. However, experience indicates that the off-design variations in room loads will not significantly affect occupant comfort if the design principles in the earlier chapters are followed.

Another caveat of table 9-2 is that nothing has been done to guide the contractor in selection of the variable air flow diffuser for the conference room. A reasonable assumption is that the eight persons possibly in the conference room would include four already accounted for. Thus, assuming about 250 Btuh per person, the conference room load might be 2000 Btuh above that assumed in table 9-2, and the building load might be 1000 Btuh load more. Again, it is reasonable to ignore the added building load – probably only for a relatively short duration, but to make provision for the variable diffuser in the conference room to handle an additional 2000 Btuh. This is a nearly 50% increase in the conference room cooling load, so the variable diffuser must be able to open up to at least 300 cfm. This is a very small increase in the design supply air flow, so it can be reasonably assumed that a diffuser capable of opening up to accommodate a full occupant load will not cause the supply duct pressure to fall below the activation pressure of the diffuser.

It can be seen that the designer must exercise considerable judgment when calculating and assigning space air flows within a zone. Large variations of occupant loads or glass loads in spaces must be analyzed to be sure that they will not compromise either the overall zone indoor air quality or occupant comfort. If adverse effects appear likely, the designer must address them. Following are some of the ways this may be done:

Re-zone the building. This may mean starting over.

Modify the zone by providing a dedicated ac unit to the questionable space.

Mitigate the effect by providing for external shading, or other physical change within the questionable space. This requires coordination with the architect and with other building trades.

Provide a simple variable-air-volume (VAV) system, available from most major ac manufacturers. While this is beyond the scope of this book, assistance on design is available from manufacturer's representatives. Understand that even a simple VAV system is substantially more complex than the constant volume systems that are the primary subject of this book.

Implementing the Desired Air Flows

The air flow distribution shown on the plans includes supply air to each diffuser, exhaust from each exhaust register or fan, and outdoor air delivered to each air handler. Implementing the desired distribution is one of the most important requirements for occupant comfort and acceptable indoor air quality. The plans and notes must therefore include instructions to the contractor that will ensure the desired result.

First, the designer must specify that all supply air branch take-offs be equipped with a balance damper, to enable a test and balance technician to set the specified air flow to each supply diffuser. Also, balance dampers must be shown where needed in exhaust ductwork and in outdoor air connections to air handlers. Dampers should never be included in air terminals - supply diffusers, registers, return and exhaust grilles. Ideally, balance dampers will be located in the branch duct where it is connected to the main trunk. This requirement should be part of the notes to the contractor.

Test and Balance is an important part of any HVAC installation. The term refers to testing each system element for its characteristics – pressure, temperature, air velocity, and air flow – and then using system devices such as balance dampers and fan speed controls to balance system air flows and velocities as specified on the drawings, shown on the schedules, and described in the notes. Most contractors can set air flows and balance simple single zone constant volume systems. However, larger systems having outdoor air pre-treatment, reheat, heat pipes, and multiple zones must be tested and balanced by certified professional test and balance technicians. Two organizations presently have test and balance certification programs, the Associated Air Balance Council (AABC) and the National Environmental Balancing Bureau (NEBB). The designer must require that all air flows be set either by the contractor or by a certified technician, and that a report be prepared and given to the owner before final payment.

Fan and blower air flows must also be set as part of the test and balance. Belt driven fans and blowers can be set by adjusting the drive belt, and changing pulleys if necessary. Direct drive fans must be set by throttling the inlet or outlet, or by setting the speed with a manual speed controller. Throttling wastes energy and could cause instability, so the designer should specify manual speed controllers for all direct drive fans. These are simple devices for low and fractional horsepower single phase motors, but may be more costly for three phase motors.

Blowers for air handlers of five tons and under are usually three-speed. After selecting an air flow in Chapter 9, the designer should review the manufacturers blower data – airflow vs external static pressure – and select the lowest speed that will deliver the required air flow at the estimated external static pressure (Chapter 12). Alternatively, the designer may include a note directing the Test and Balance technician to set the air handler speed as low as possible.

Another option available to the designer of systems under 7.5 tons is the “variable speed” air handler. These air handlers can be set to deliver either a “nominal” air flow – such as 1800 cfm for a five ton unit – or an air flow some fixed percent higher or lower than nominal. Often, the low airflow setting will be 300 to 325 cfm/ton, which is a desired range for best moisture removal. The advantage of these systems is that, when properly installed, they will deliver the catalog air flow regardless of the initial or subsequent deviations from the design external static pressure. They are not truly variable speed in the sense of allowing speed variations during operation. True variable speed drives are often applied to air handlers serving Variable Air Volume Systems. Although VAV systems are not described in detail in this book, the designer should be aware of variable speed motor drives and their application to larger air handler blowers, supply fans, and exhaust fans.

Variable Speed Drives (VFD)

Variable speed drives for motors of greater than one horsepower are termed VFD for Variable Frequency Drive. Until fairly recently, cost made VFD impractical for electric motors of less than ten horsepower. However, such drives, are now cost effective and can save considerable energy in variable air flow applications in small systems with long operating times at low air flow. Alternatives to VFD motors are controlled bypass, simple throttling and variable inlet guide vanes.

Controlled bypass systems short circuit air flow from the supply to the return as the system air flow demand falls. This method is applied to DX VAV systems because of coil freeze and refrigerant instability that can occur when air flow across the evaporator drops below about 300 cfm per ton. Fan energy over the range of VAV air flow is constant, because both air flow and external static pressure are held constant.

Simple throttling occurs when the VAV terminal valves are allowed to close and drive the fan operating point up the constant speed line on the fan map. Fan energy decreases somewhat as airflow is reduced, even though esp may be rising. Downsides to this method are that the air flow reduction could drive the fan into unstable operation, and DX systems will require energy-eating hot gas bypass to avoid evaporator freezing and refrigerant system instability.

After VFD, the most energy efficient method of reducing fan air flow is inlet guide vanes, which basically shift the constant speed line on the fan map to allow reducing both air flow and esp as demand is reduced. The downside is that as the inlet guide vanes reduce the fan air flow at its constant speed, fan noise at blade pass frequency can increase to noticeable and unpleasant levels.

For a comprehensive understanding of fans and fan control systems, see reference 5, ASHRAE *Handbook – Systems*, Chapter 20[15].

END

Chapter 11

HVAC Controls and Indoor Air Quality

Basics of Indoor Air Quality Control

This book does not address investigation and mitigation of Indoor Air Quality (IAQ) problems in existing or new buildings. However, good HVAC design can do much to prevent such problems, often manifested as "sick building syndrome", from arising. Acceptable IAQ should follow application of the design methods already described. However, the designer should have explicit knowledge of the basic principles that control air quality and comfort in buildings. In particular, setting up and mandating a good control system is essential.

Reference has been made in earlier chapters to ASHRAE Standard 62 *Ventilation for Acceptable Indoor Air Quality*[5]. Standard 62 defines "Acceptable Indoor Air Quality" as follows: "air in which there are no known contaminants at harmful concentrations as determined by cognizant authorities and with which a substantial majority (80% or more) of the people exposed do not express dissatisfaction." Standard 62 sets criteria not only for outdoor air ventilation, but also for control of contaminant sources, system geometry, and maintenance. Filtration, also an important factor for acceptable indoor air quality, is covered by ASHRAE Standard 52.2 *Method of Testing General Ventilation Air Cleaning Devices for Removal Efficiency by Particle Size*[8]*.* The designer must be familiar with both of these documents because both are incorporated into most state and local building codes.

Acceptable indoor air quality begins with maintaining space dry bulb temperature and relative humidity within the guidelines of ASHRAE Standard 55 *Thermal Environmental Conditions for Human Occupancy*[9]*.* Beyond this, the following principles apply for humid climates:

1. Maintaining positive pressure in the building at all times during occupancy. See Chapter 4.

2. Selecting HVAC systems that can maintain average relative humidity below 60% under all conditions. See Chapters 7 and 9.

3. Ensuring that the building pressure envelope and thermal envelope coincide, or that the pressure envelope is outboard of the thermal envelope, and that both are continuous. This requires review of architectural plans and coordination with the architect. See Chapter 2.

4. Ensuring that cabinets or closets containing volatile materials are properly exhausted to the outside. This is a requirement of many codes, but where no code requirement exists, the responsibility lies with the design team.

5. Ensuring that commercial cooking and dishwashing operations have properly designed hoods in accordance with NFPA 96.

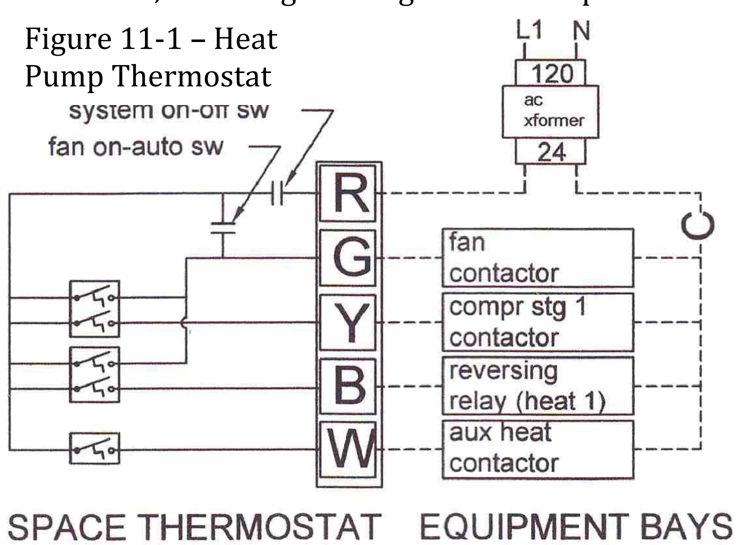
6. Ensuring that processes that may be conducted within a zone, such as lab operations, cosmetic services, and gun cleaning stations have properly designed exhaust hoods, as described in the *Industrial Ventilation Manual*.

7. Ensuring that no mechanical exhaust or outdoor air intake is operating when a zone is unoccupied and ac equipment is off. This is a control function.

The Basic HVAC Control

Most readers are familiar with the low voltage ac unit thermostat. However, for those who are not, following is a diagrammatic representation of a single stage cool, single stage heat thermostat for a heat pump.

Figure 11-1 – Heat Pump Thermostat



The thermostat represented here is controlling a heat pump. Control power is a 120vac/24vac transformer located in the control bay, which in a split system may be either in the air handler or the outdoor unit. The solid lines represent thermostat internal circuitry, and the dashed lines are field wiring from the thermostat located in the zone to the various components in the ac unit cabinets. The letters represent wire colors, and are an industry convention.

R – Red, this is system control power.
G – Green, power to ac unit indoor blower
Y – Yellow, power to the compressor contactor
B or O –Blue or Orange, power to the heat pump reversing relay – in this case energize to cool. (On some systems, the default is cool, and either B or O will energize the reversing valve to heat.)
W – White, auxiliary electric heat
C - Cream, transformer common

The above are the standard colors, except that with conventional cooling/heating systems (electric or gas), white would be the wire to activate the system heat, rather than blue or orange, and there would be no reversing relay. Beyond the colors noted above, there is some departure from standardization, so the designer must refer to wiring diagrams by the thermostat manufacturer and by the ac unit manufacturer. In general, however, for more complex thermostats and systems, the following practice prevails for terminal identification:

- Y2 – 2nd stage cooling
- W2 – 2nd stage heating
- E - emergency heat relay, heat pump
- B – transformer common on some older systems
- T – outdoor anticipator reset

A comprehensive discussion of thermostat color codes is available on the web at the URL http://www.toad.net/~jsmeenen/wiring.html .

Because the thermostat is basic to HVAC control, the designer has no need to do more than show the location of each zone thermostat on the HVAC plan (Chapter 12). However, it is usually desirable, even if only thermostat control is needed, to specify for the contractor certain features that should be included. These include dual control set point, system on-off switch, and fan on-auto switch. The dual control set point allows the user to set a dead band between the heating and cooling set points – for example, the heating set point may be 70°, and the cooling set point may be 76°. If dual control set point is not specified, the contractor will tend to use a cheaper alternative with only a single set point.

Besides the thermostat, the simplest systems must still provide for two additional functions: Outdoor air temperature sensor for heat pumps, and shutoff of exhaust fans and ac blowers during unoccupied periods, when the cooling system is shut down.

Heat pumps require auxiliary electric heat strips to temper supply air during the defrost cycle. When the heat pump is in defrost cycle, the system is put into normal cooling mode to de-ice the outdoor coil. However, another function of auxiliary heat is to augment reverse cycle heat when the outdoor temperature falls very low, or when the system is in warm-up mode after being off or set back overnight or for the weekend. Energy Standard 90.1 requires that an outdoor temperature sensor be used to lock out this function, saving energy, when outdoor temperature is relatively warm – say above 40° F.

Automatic Time of Day Control

Referring to Chapter 8, One of the requirements of Standard 90.1 is automatic time of day control with seven different day schedules. This can be done with programmable thermostats or with time clocks interlocked with HVAC equipment .

The previous chapters of this book have emphasized the importance of pressurizing a building to prevent infiltration from causing moisture build-up. A building is pressurized by inducing outdoor air with the ac unit air handler and ensuring that mechanical exhaust is smaller than the induced fresh air. It is very important that neither of these functions is operative when the ac system is not active – as when it may be shut down for nights and weekends. The simplest way to do this is to interlock each fan serving a zone with the zone air handler, and locking out the air handler when the ac system is off – that is, disabling the fan "on" switch when the ac system switch is "off". Without these precautions, cool moist night air can be continuously drawn into the building, allowing humidity to build up.

An element of the automatic time of day control as required by Standard 90.1 is the requirement for set-up and set-back of the thermostat during unoccupied periods. In most zones, simply shutting down a system at night or on weekends may allow unacceptable low temperatures in Winter, or high indoor humidities during Summer. Therefore, thermostats should be selected to allow unoccupied set back to some reasonable temperature, say 60°, during the heating season and set up to 80-85° during summer. The latter will cause occasional operation of the ac system, hence dehumidification, during extended unoccupied periods. An even better way to ensure dehumidification is to use a time clock to run the system at occupied temperature for one hour every morning and one hour every afternoon – a method this writer has used to control humidity in school libraries during the long summer-school shut down.

Humidity Control

In Chapter 7 it was explained that air conditioners control zone humidity by running zone air across a coil capable of cooling the air below the dew point needed to allow the supply air to condense all of the moisture added by occupants, processes, and outdoor air. In many cases, a simple ac unit with thermostat control does not have this capability. When this is the case, the designer must incorporate processes and equipment to enhance humidity control, as discussed in Chapters 7 and 9:

heat pipe
liquid line reheat
energy recovery ventilator - ERV
hot gas reheat - HGR
dedicated outdoor air DX unit - DOAU

The first two above are entirely passive, and require no control instructions or interlocks beyond a thermostat. The energy recovery ventilator is also essentially passive, but must be interlocked to run only when the ac unit system is running. The last two both require active humidity control using a zone humidistat and a zone or ventilation air thermostat. The designer must instruct the contractor on how to incorporate these devices.

A zone humidistat is a sensor with a contact that makes or breaks at a relative humidity setting. Here, we are concerned with dehumidification, so the contact will usually make on relative humidity rise above set point. Humidistats are also available to activate a humidifier on fall of relative humidity, or to both dehumidify and humidify with a dead band middle range. This is rarely required for the projects that are the subject of this book.

Reheat

Reheat requires a zone humidistat and a zone thermostat. The function of the humidistat is to activate cooling. If the zone temperature is below the cooling set point and the relative humidity is above set point, the humidistat will operate the system compressor in order to dehumidify the supply air, and if the zone is over cooled, the zone thermostat will bring on the system heat to maintain the zone cooling set point. This is illustrated by the following matrix:

cooling set point = 76°, heating set point = 70°, humidity set point (max) = 60%

temperature	rel humidity	cooling	reheat	heat
<70°	<60%	off	off	on
<70°	>60%	on	on	on
70° - 76°	<60%	off	off	off
70° - 76°	>60%	on	on	off
>76°	<60%	on	off	off
>76°	>60%	on	off	off

It can be seen that in one case, both the cooling and heating systems will be simultaneously activated. This is because, when reheat is on, the compressor is running regardless of space temperature, so heat is required to prevent overcooling. It is very unlikely that a normal progression of temperature falling below the heating set point – as on a cold day - would occur while zone relative humidity remains high, so this condition should almost never occur.

Reheat is activated any time that the relative humidity exceeds set point, except when the thermostat is calling for cooling. Referring back to Chapter 7, it can be seen that this is necessary if the room operating line has a very steep sensible heat ratio, and the unit selected is thus oversized to allow the coil to dehumidify. If the compressor runs only on call for cooling, it will quickly cool the zone to the set point, and then be off for a long period, allowing the humidity in the zone to rise.

The simplest way for the designer to specify reheat controls is with a “Sequence of Operations” statement on the HVAC plan. Following is an example of a Sequence of Operations for a packaged AC unit with reheat.

> **Exhaust fans shall be interlocked to operate only when the thermostat system is in occupied mode.**
>
> **The indoor blower shall operate continuously when the thermostat system switch is “on”, and shall be locked out when the system switch is “off”.** [Continuous

blower operation during occupancy is required in many jurisdictions to satisfy code requirements for outdoor air induction.]

When the thermostat is calling for cooling and relative humidity is below set point, the compressor shall operate and reheat shall be off.

When the thermostat is calling for cooling and relative humidity is above set point, the compressor shall operate and reheat shall be off.

When the zone temperature is below the cooling set point set point and relative humidity is above set point, the compressor shall operate, and reheat shall be on.

When the zone temperature is below the cooling set point set point and relative humidity is below set point, the compressor shall be off, and reheat shall be off.

Space heat shall operate at any time that the thermostat calls for heat.

Dedicated DX Outdoor Air Unit

A dedicated outdoor air unit (DOAU) requires a zone thermostat, a zone humidistat, and a supply air temperature sensor. The unit itself has one or more refrigeration compressors for dehumidification, reheat to deliver a constant temperature to the space air conditioner, and hot gas bypass and condenser fan control to automatically maintain head pressure when outdoor ambient temperature is low – below 50 ˚F.

The zone thermostat controls the zone space air conditioning unit, not the DOAU. The DOAU blower is typically interlocked to run at any time that the space air conditioner blower runs. The zone humidistat controls the DOAU refrigeration compressor(s), and the supply air temperature sensor controls DOAU reheat. Following is a typical Sequence of Operations for a zone air conditioner with an DOAU.

The DOAU blower shall be interlocked or operate whenever the zone space unit blower operates, except during pre-conditioning prior to occupancy.

When relative humidity is below set point, the DOAU compressor shall be off.

When relative humidity is above set point and supply air temperature is below set point, the DOAU compressor shall operate and reheat shall be on.

When relative humidity is above set point and supply air temperature is above set point, the DOAU compressor shall operate and reheat shall be off.

Outdoor air units are available with gas or electric heat to temper outdoor air during winter months. Remember, the DOAU must operate whenever the space unit blower operates during occupancy in order to supply the minimum required outdoor air. The designer can provide the space air conditioner with only enough heat to handle the zone heat loss, and specify enough heat in the DOAU to handle the design day outdoor air heating load. However, commercial buildings have substantial internal heat gain when occupied, and warm-up can be performed prior to occupancy with the DOAU off. To do this requires on-off control of HVAC equipment using a time clock. In warm climates, this should eliminate the requirement for heat in the DOAU.

Additional Reading

From ASHRAE *Humidity Control Design Guide for Commercial and Institutional Buildings*[16]. From the EPA: *An Introduction to Indoor Air Quality* www.**epa**.gov/iaq/ia-**intro**.html.

END

Chapter 12

Design Drawings, Specifications, Notes, and Schedules

Document Plan

Objective

The design documents must describe the project in sufficient detail for the contractor to perform the work in the manner intended by the designer. Since the contractor will be a licensed and presumably competent HVAC installer, the documents do not need to describe procedures or methods unless the designer wishes to have something done in a particular way, or wants to ensure a level of quality. For example, the designer should show locations for the zone thermostats, but it is not necessary to define the mounting method or height, since that is a routine function of HVAC installation. Likewise, a thermostat field wiring diagram is not needed unless the designer wants to show a non-standard function, such as the interface with a time clock. In general, as much discretion as possible should be left to the contractor, since over-specifying will inevitably increase the contractor's proposal price.

Medium

It is assumed that the reader will be using one of the many fine drafting programs available. Three of these are, in order of my preference: AutoCAD, FelixCAD, and DataCAD. All of these are reasonably easy to learn – FelixCAD is virtually identical to AutoCAD LT, at a much lower price. In my experience, most engineers use AutoCAD, either the full version, or "LT" which is much lower in cost but does not include many features, the most important of which is 3D drafting capability. Nevertheless, both AutoCAD LT and FelixCAD are suitable for preparing HVAC drawings for most projects. DataCAD is primarily used by architects. Any of the programs listed have features to allow conversion to either of the other two. This is helpful when the HVAC designer is using AutoCAD, say, and needs to work with an architect using DataCAD.

Important features of all of the listed programs are layering, block definition, line style and line weight. Layering allows the designer to define different "layers" for various HVAC elements, such as supply ducts, return ducts, equipment, exhaust ducts, etc. Each of these can be assigned a color to make it easy to differentiate elements while working on the drawing. However at present, it is not practical to try to print the documents in color for construction sets, so the designer must remember that the contractor will see only black lines on white paper. Layers can be turned on and off while working and for printing.

Block definition allows the designer to draw a feature that will be standard, such as a fire damper detail, and then to save this as a "block" that can be used over and over on other projects. Examples of elements that I have found useful as blocks are outdoor condensing units, turning vanes, ceiling fans, diffusers and grilles, etc. Practically any element that is generic in nature and is likely to be used on other projects. I have defined hundreds of blocks, including thirty that I use on virtually every project.

The primary need for line styles is to depict center lines and hidden lines. Line weight can be used to make the base elements (walls, windows, doors, plumbing) lighter so they do not interfere with the HVAC elements, and to make certain HVAC elements darker if that will enhance the clarity of the drawing.

<u>Base sheet</u>

The base sheet is nothing more than a modification of the project floor plan, furnished by the architect as described in Chapter 2. It is usually the responsibility of the HVAC designer to modify the floor plan so that it will be suitable for the HVAC layout. The plan must be stripped of all of the information that will be of no interest to the HVAC contractor. This generally means dimensions, door and window codes, special finish notes, furniture, crosshatching of certain elements, etc. Elements to remain are room names, room numbers, plumbing fixtures, doors and door swings, and kitchen layouts. The designer must use his judgment to decide which objects on the architectural plan to retain and which to remove.

The modified plan must be "screened" or given a unique light line weight so that it will appear as a light background with the darker, normal weight HVAC elements overlaid. It is good practice to put the room names, room numbers, and plumbing fixtures on separate layers from the base plan to allow them to be turned off if they interfere with HVAC elements. It is also good practice to include the interior lighting plan as a separate layer of the base plan. The lighting layer can be turned on when it is necessary to coordinate with HVAC ceiling elements, such as diffusers and grilles. It will of course be turned off in the final contract version.

<u>Elements of the final documents</u>

The basic elements of the final documents are plans, specifications, and schedules. Plans include the basic floor plan, often with a portion of the site shown to locate outdoor equipment. In addition to the floor plan, at the discretion of the designer, there may be elevation details showing equipment installation, building sections, piping schematics, and wiring diagrams. Flag notes may be used to explain desired features that cannot easily be depicted graphically, and to provide clarifying instructions to the contractor. Specifications are basically quality control notes and are usually, for small commercial projects, shown on the plans rather than as a separate document.

For Legend, see Figure 12-3

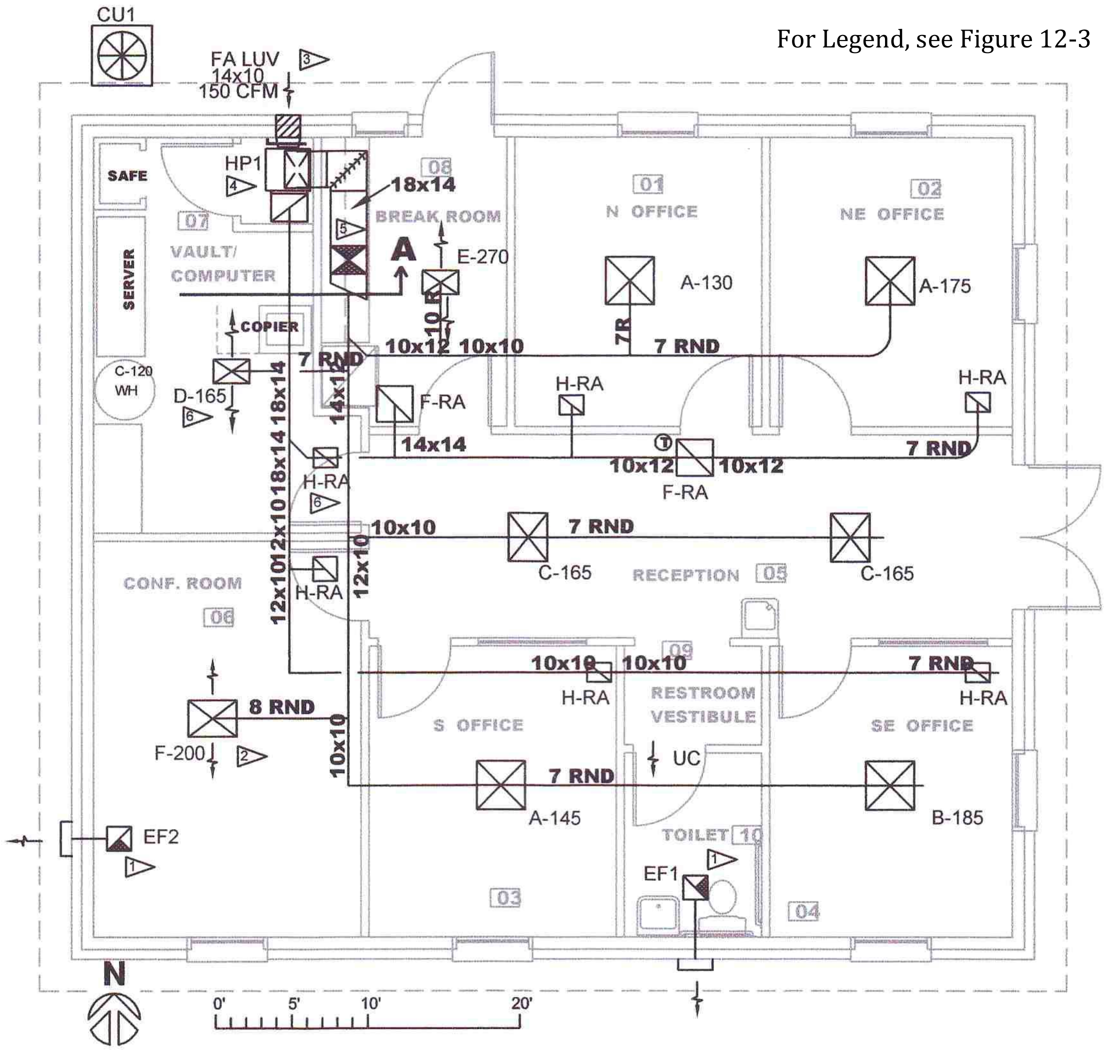


FIGURE 12-1 HVAC LAYOUT

Building Dimensions 40' Wide and 35' Long

FLAG NOTES

1. *DUCT FAN DISCHARGE TO WALL JACK, SAME SIZE AS FAN OUTLET.*
2. *THERMAFUSER VAV DIFFUSER – DUAL SET POINTS THERMOSTATS FOR HEATING AND COOLING. SEE TEST AND BALANCE NOTE.*
3. *FRESH AIR INTAKE LOUVER LEVEL WITH HP1 RETURN AIR PLENUM. SLEEVE WALL, TRANSITION TO 10x10 AT PLENUM CONNECTION.*
4. *MOUNT HEAT PUMP AIR HANDLER ON 20" HIGH ANGLE FRAME WITH EXTERNALLY INSULATED SHEET METAL ON ALL SIDES AND BOTTOM. SUPPORT ON NEOPRENE PADS. CONNECT RETURN DUCT AND OUTDOOR AIR DUCT TO PLENUM AS SHOWN. PROVIDE 12x12 DOUBLE CAM INSULATED INSPECTION DOOR IN FRONT OF PLENUM.*
5. *ROUTE DUCT BELOW ROOF TRUSSES IN CHASE OVER CABINETS. RISE AS SHOWN INTO ATTIC WHEN CLEARANCE IS AVAILABLE. ROUTE SUPPLY DUCT MAINS ABOVE RETURN MAINS. BOX TRUSSES AS NECESSARY. COORDINATE WITH STRUCTURAL AND FINISH TRADES. SEE ELEVATION DETAIL.*
6. *DIFFUSER AND FIRE DAMPER IN RATED CEILING. SEE DETAIL.*

Finally, equipment schedules are used to define in detail the features and performance of the HVAC equipment which is the basis of the design. Following is a description of the required elements of the HVAC plan and documentation, in the approximate order of creation. The HVAC floor plan for a small office building is shown on Figure 12-1. This is the building first presented in Chapter 10 to illustrate air distribution. The schedules, notes, and details that go with it are discussed in the remainder of this Chapter.

Creating the Documents

Base Sheet

The base sheet is described in the preceding section, and is the first element of the documents to be created. It should be created in the preliminary design phase, but may need later modification as a result of design requirements – for example, the space allotted by the architect for an air handler may not be large enough. The HVAC designer must coordinate modifications with the architect, and should provide a recommended solution to any problem he identifies.

Equipment Schedules

Equipment Schedules are created as the design progresses, because each step of the design must be based on selection of equipment, beginning with the air conditioner as described in Chapter 9. As a minimum, each design must include schedules for the following equipment: air conditioners, diffusers and grilles, exhaust fans. Additional schedules may be required, such as air intake or exhaust gravity vents, louvers, energy recovery ventilators, and dedicated outdoor air units. Figure 12-2 presents the equipment schedules needed for the HVAC plan shown on Figure 12-1, plus a few generic schedules for fans and miscellaneous equipment.

Equipment Layout

The next step is to lay out the equipment on the base sheet. All equipment whose size is important to insure compatibility with the building architecture and structure should be shown to scale. This would always include the air conditioners, air handlers, energy recovery ventilators, and outdoor air units, even if they are outside the building, since there may be issues relating to the site plan such as walkways, parking areas, property line setbacks, esthetics, and noise.

Note the use of flag note 4 to describe the installation of the air handler, HP1. This note can obviate the need for an elevation drawing of the installation. However, an elevation drawing may be desired to clarify the installation note, or to show other features that may be difficult to visualize with a narrative.

HEAT PUMPS

MANUFACTURER	CARRIER
INDOOR UNIT	HP1
MODEL	FY4AN048
AIRFLOW, cfm	1600
OUTDOOR AIR, cfm	150
STATIC PRESSURE, iwg	0.4
FAN SPEED	HIGH
VOLTS/PHASE	208/1
FLA	4.3
FAN HP	3/4
DIMENSIONS (LxWxH)	22x21x54
WEIGHT, lbs	147
FILTER	
TYPE	Perm
NO/SIZE	1
AUX ELECTRIC HEAT	
VOLTS/PHASE	208/1
CARRIER P/N CAELHEAT	0201N05
KW @ 240 VOLTS	8
STAGES	
OUTDOOR UNIT	CU1
MODEL	25HBA348
DIMENSIONS (LxWxH)	35x35x33
WEIGHT, lbs	263
VOLTS/PHASE	208/240/1
COMPRESSOR RLA	117
FAN FLA	1.2
APPLICATION RATING	
ARI cfm	1600
TOTAL CAPA CITY, BTUH	46500
SENSIBLE CAP., BTUH	
SEER	13
HEATING CAP, BTUH	48000
COP	3.4
HPSF	8.3
REFRIGERANT	R410A(1)
COMPRESSOR(S)	SCROLL

NOTES (1) EQUIPMENT USING CFC OR HCFC REFRIGERANT MAY BE SUBSTITUTED ONLY WITH THE APPROVAL OF THE OWNER

SCHEDULES FOR FIGURE 11-1

FAN SCHEDULE

CODE	EF1,2
MANUFACTURER	L. COOK
MODEL	LIL GEM I
TYPE	CENT
MOUNTING	CEIL
DIMENSIONS L,W,H	12x14x8
DRIVE	DIRECT
AIRFLOW, cfm	50
SP, iwg	0.1
SPEED, rpm	SET
SONES	4.0
VOLTS/PHASE	110/1
WATTS OR HP	60
ACCESSORIES	
LOCATION	6,10
CONTROL	(1), (4), (3)
AIR HANDLER(S)	HP1

NOTE: (1) PROVIDE SPEED CONTROL ON FAN BODY
(3) SET FAN AIRFLOW WITH SPEED CONTROL
(4) INTERLOCK FAN TO RUN WITH AIR HANDLER SERVIING AREA

REGISTER SCHEDULE

CODE	MANUFACTURER	MODEL	TYPE	SIZE
A	METALAIRE	M1S4	CEILING SUPPLY	6x6
B	METALAIRE	M1S4	CEILING SUPPLY	9x9
C	METALAIRE	M1R4	CEILING SUPPLY	6x9
D	METALAIRE	M1R2L	CEILING SUPPLY	6x9
E	METALAIRE	M1R2L	CEILING SUPPLY	6x15
F	THERMAFUSER	TF-HC	CEILING RETURN	10"
G	METALAIRE	RH	CEILING RETURN	12x12
H	METALAIRE	RH	CEILING RET/TR	10x8

CODE: A-250
CODE - CFM

EXAMPLE FAN SCHEDULE

CODE	EF1	EF3A	TR	EF3	EF
MANUFACTURER	L.COOK	LCOOK	BROAN	L. COOK	L COOK
MODEL	GN620	GC320	510	90C10DH	120C2B
TYPE	CENT	CENT	PROP	CENT	CENT
MOUNTING	IN LINE	CEIL	IN WALL	ROOF	ROOF
DIMENSIONS L,W,H	17,12,12	12,11,11	11x11	18OD,7D	28D,8D
DRIVE	DIRECT	DIRECT	DIRECT	DIRECT	BELT
AIRFLOW, cfm	400	150	380	300	1000
SP, iwg	0.25	0.2	NA	0.125	0.25
SPEED, rpm	950	1360	1550	1050	1200
SONES	1.2	3.3	6.5	6.0	8.0
VOLTS/PHASE	110/1	110/1	110/1	110	110
WATTS OR HP	245	77	1/4	91	1/6
CONTROL	(1),(3),(4)	(1),(3),(2)	(5)	(1),(3),(4)	(4)
LOCATION	5	22	3-4	ROOF	ROOF
AIR HANDLER(S)	HP1	HP3	N/A	EXHP1	PAC2
ACCESSORIES	BD	BD	GRILLES	CURB/BD	CURB

NOTE: (1) PROVIDE SPEED CONTROL ON FAN BODY
(2) OPERATE FAN WITH LIGHTS
(3) SET FAN AIRFLOW WITH SPEED CONTROL
(4) INTERLOCK FAN TO RUN WITH AIR HANDLER SERVIING AREA
(5) OPERATE FROM WALL SWITCH

ENERGY RECOVERY VENTILATOR - EXAMPLE

CODE	ERV-1
MANUFACTURER	GREENHECK
MODEL	ERV251H
OUTDOOR AIRFLOW	1000
DRY BULB/WET BULB TEMP	95/80
EXHAUST AIRFLOW	750
EXHAUST AIR DB/WB TEMP	76/65
SUPPLY AIR TO HP DB/WB	83.3/71.5
SUPPLY FAN HP	1/2
EXHAUST FAN HP	1/2
OUTDOOR AIR WHEEL EFF	61.8%
EXH AIR WHEEL EFF	82.4%
ELECTRICAL	
VOLTS/PH	115/1
MCA	26
DIMENSIONS LxWxH	46x34x27
WEIGHT	300

NOTES
(1) INTERLOCK TO RUN WITH HEAT PUMP
(2) FOR DIRECT DRIVE UNITS, PROVIDE SPEED CONTROLLER FOR EACH MOTOR
(3) SET DESIGN AIRFLOWS USING SPEED CONTROLLERS OR BELTS/SHEAVES

FIGURE 12-2
Equipment Schedules

Diffusers and Grilles

In Chapter 10, the principles of air distribution were presented. Each space in a zone is assigned an air flow based on the sensible cooling load of that space at the design condition. However, in order to introduce this air flow, the designer must carefully select the proper type and number of supply air diffusers for each space.

The principles of diffuser selection are laid out in the Chapter titled "Space Air Diffusion" in the ASHRAE Handbook *Fundamentals.* These principles should be studied and understood before undertaking the diffuser selection process, because improper diffuser selection will have serious detrimental effects on occupant comfort.

The basic parameters that must be considered when selecting diffusers for a space are position of introduction, type of mounting surface, room geometry, noise characteristics, and throw. Position of introduction refers to floor, wall, or ceiling diffuser location. Type of mounting surface may refer, for example, to lay-in vs drywall ceilings. Diffuser noise is obviously important, and is generally expressed in terms of NC level. Throw is the distance from the diffuser where the air jet has decelerated to 50 feet per minute. This influences diffuser performance relative to spacing of diffusers and distance (in the direction of throw) from a wall. Manufacturer's catalogs, available on line or from manufacturer's representatives, are the source of the data needed to make a proper selection. Supply diffuser locations are shown on Figure 12-1 along with a code to identify each diffuser on the register schedule, and the air flow to be set for that diffuser by the contractor or Test and Balance technician. The code is shown below the register schedule on Figure 12-2.

Grilles are return air or air transfer devices. They do not diffuse the air, and the location of returns in a space is only important if they are located in the path of a diffuser supply jet. However, returns and transfer devices must be selected to minimize noise and system pressure loss. Return grilles are shown on Figure 12-1 with a schedule identification code only, and the notation RA to denote return air. No air flow is shown on returns because they can only return air flow that was introduced into the space by the air handler.

It is recommended that neither supply diffusers nor grilles be equipped with integral balance dampers, to avoid noise problems. As explained in the section on duct layout and sizing, to avoid noise problems system balance should be accomplished with dampers located in the supply ductwork as remote from diffusers as possible. Balance dampers are used in return ductwork only for the purpose of forcibly inducing outdoor air, as was explained in Chapter 4.

Diffuser and grille model numbers are important because they usually denote the mounting method. The designer must be careful to specify models that will mount in the wall or ceiling type provided, or should direct the contractor to determine the mounting type before ordering the devices.

In laying out diffusers and grilles, it is not necessary to show these at actual size, which would often be too small to show up clearly on the drawing. Generally, diffusers and grilles are shown 24x24 or 18x18. The actual neck size of a diffuser or grille is shown on the register schedule. See figure 12-2.

<u>Ductwork Layout and Sizing</u>

Referring to Figure 12-1, the following elements of the HVAC layout are complete: air conditioner HP1 and CU1, exhaust fans EF1 and EF2, supply diffusers, and return air grilles. The next step is to lay out ducts to connect HP1 with the supply and return diffusers, and to show ducting for the exhaust fans and outdoor air intake.

Layout of duct routing is representational only – the contractor has the ability and the freedom to make changes as dictated by the actual conditions at the site. In doing so, he must retain the equivalent duct sizes shown on the drawings, and must not add complexities that will significantly increase the system pressure drop. That is, much of the ductwork detail may be left to the contractor, within restraints dictated in the quality control notes on the drawing and by industry accepted good practice. Specifications and quality control notes are covered in a subsequent section of this chapter.

Figure 12-1 shows ductwork drawn both in two dimensions and as single lines. Two dimensional ducts should be shown when the designer wants to clarify a complex or unusual routing or configuration. In this case, a portion of the ductwork must be run below the ceiling because the air handler is located on an outside wall, and the hipped roof has insufficient clearance near the wall to allow the supply duct to rise into the attic. The designer has elected to solve this problem in the design stage, rather than risk a change order at the job site that may be deleterious to the owner.

When single line ducts are used, the designer may depict various types of branches or tees using an equivalence diagram similar to that shown on Figure 12-3. Single line drawings handled in this manner save drafting time and make the drawing less cluttered, therefore easier to read. However, the designer must keep in mind possible interference with lighting, sprinkler piping and heads, and other ducts. If there is a question, double line ducts, showing the duct width or height to scale, should be used.

The objective of the designer is to route and size the ducts for a reasonable external static pressure loss (e.s.p.) on the air handler. This is the total of the static pressure lost by the duct system from the air handler discharge to the most remote diffuser, and from the room and outdoor air inlet back to the air handler blower return inlet. A reasonable e.s.p. loss is one that is within the capabilities of the air handler and does not result in excessive ductwork costs.

The external static pressure includes the pressure loss due to air friction in the duct system, the pressure loss as the air stream passes through fixtures, and the "system effect" which is the unique loss caused by the configuration of the duct attachments to the air handler or fan.

Duct sizing fundamentals are covered in detail in the ASHRAE Handbook *Fundamentals*, and sizing software is ubiquitous. The two basic methods are equal friction and static regain. The former is the simplest and easiest to apply using manual methods, and is the method used to size the ducts in Figure 12-1. The friction factor used is .095 inches H20 per 100 feet, which gives satisfactory results of external static pressure and duct noise for most small HVAC projects. To achieve especially low pressure loss, as for fresh air intakes or transfer ducts, a friction factor of .055 will give satisfactory results.

Fitting losses are either defined as static pressure loss from catalog data, or is computed from loss coefficients found in publications such as the *Handbook of Hydraulic Resistance*[17] and the *ASHRAE Duct Fitting Database*. The loss coefficients, designated C_0, are functions of fitting geometry and air flow ratios. Pressure loss is computed by multiplying velocity pressure, v_{dp}, by C_0.

$$v_{dp} = rho*(V/1097)^2$$

and $$\Delta p = C_0 * v_{dp}$$

where V = air flow velocity
1097 = conversion factor (don't ask)

The equal friction sizing method is not self balancing, as is the case with static regain. Therefore, it is essential that each terminal runout to a diffuser incorporate a damper to allow setting design air flow rates to each room. These dampers should be as remote as possible from the diffuser to attenuate noise that can be generated when the damper is near the closed position, as may be the case for diffusers near the air handler. It is not good practice to specify diffusers or grilles with built-in dampers, as this ensures that any noise generated will be transmitted directly into the room, and also encourages occupants to tamper with the settings, thus upsetting the results of the Test and Balance.

As a general rule, there should be no dampers in the return system. Because the air handler blower must induce exactly what it discharges, the only effect of a damper anywhere in the return system is to increase the e.s.p. on the blower. An exception to this may be a damper in a return just ahead of the outdoor air intake. If the intake path is very long, or if the outdoor air flow is very large, such a damper may be needed to allow the air handler to induce the required outdoor air.

Finally, note the use of flag notes on figure 12-1 for the exhaust ducts (flag note 1) and for the part of the supply duct that is routed below the ceiling (flag note 5).

Exhaust and Outdoor Air Intake

The requirements for exhaust and outdoor air intake are set forth in Chapter 4, “Ventilation and Air Balance”. These requirements must be depicted on the drawings in a way that ensures the system ability to deliver the required air flows. Bathroom exhaust, for example, requires provision for air to be drawn from the adjacent occupied spaces into the bathroom, which often has no independent supply air from the air handling system. To avoid excessive pressure loss, it is good practice to size the transfer path for a maximum velocity through the throat of 500 feet per minute (fpm). In figure 12-1, the door undercut (uc) with dimension provides the make-up air for the bathroom fan EF1. The arrow next to the “uc” in the door opening denotes the direction of flow. In this case, a 1/2” undercut in a 36” door will provide a “throat” velocity under the door of 400 fpm for the 50 cfm required by the fan. If a larger undercut is needed, the dimension should be noted next to the “uc” on the drawing. Air flows that require more than a 1” undercut (125 cfm in a 3’ door) should be handled by a door grille, using the symbol “dg” with a direction arrow, and a size. Door grille data is available in the catalogs of most manufacturers of diffusers and grilles. Door undercuts and door grilles are the responsibility of the HVAC contractor, who must coordinate these with the general contractor and affected trades. Door grilles cannot be installed in fire rated doors or paneled wood doors.

Discharge of exhaust from a building is either by a roof or wall “jack” or a louver. Wall jacks are simple hoods designed to block the entrance of rain, and are often available from the fan manufacturer. A backdraft damper should be specified if the exhaust fan called out does not have one built in.

The required outdoor air can usually be induced by the air handler without requiring either a return duct damper (see the preceding section) or a dedicated outdoor intake fan. However, either of these should be considered if there is doubt about the suction capability of the air handler.

Outdoor air is introduced either through a wall mounted louver or a roof mounted gravity intake. A gravity intake is a housing mounted on the roof, either flashed or curb mounted. They should be represented by a schedule, and are needed where a large amount of outdoor air is needed, often for several air handlers. Louvers need not be scheduled, but the size must be shown, and the basis of design should be included in the specifications or quality control notes. Important data for louvers is free area, pressure drop, and rain induction velocity. Louvers used for air intake should be sized for a free area that will yield a maximum intake air velocity of 500 fpm at the scheduled air flow, so rain induction (velocities of 800 fpm or higher) will not be a factor. Exhaust louvers can be sized for a higher pressure drop to match the fan static pressure capability at the scheduled air flow, and hence may be smaller than the intake louvers for a given air flow.

Figure 12-1 shows an intake louver sized for the required 150 cfm using the rule of 3 – that is, sizing the louver at three times the catalog dimensions to provide the free area required to achieve a maximum intake air velocity of 500 fpm. Alternatively, using the manufacturer's data for free area vs nominal size, and specifying a make and model as basis for design, will usually result in a somewhat smaller size. Flag note 3 directs the contractor to provide a sheet metal sleeve to duct the outdoor air through the wall. Without this requirement, some contractors will simply cut a hole in the wall, install the louver on the outside, and allow the return to pull the outside air from the wall cavity.

Elevations and Details

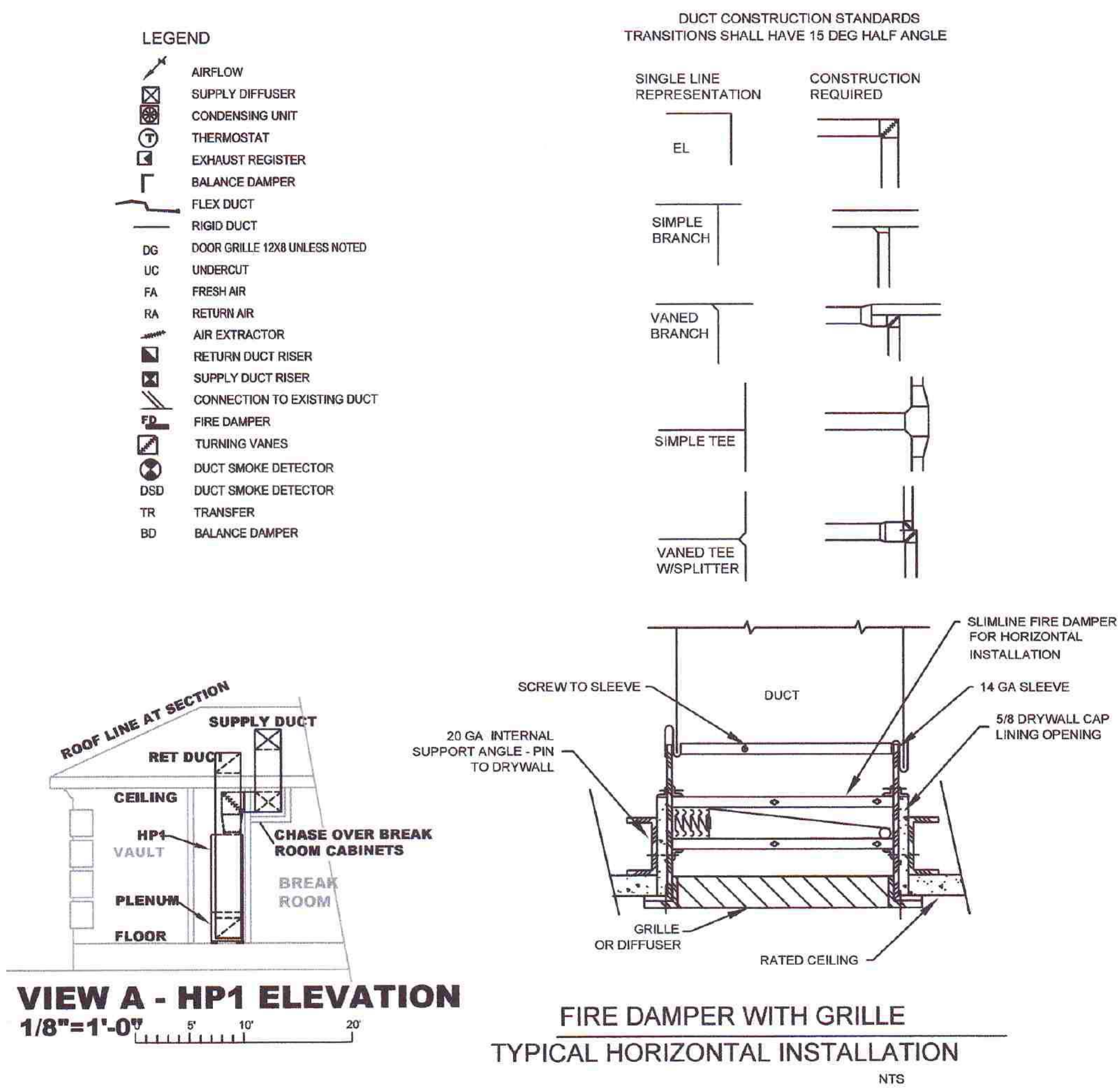


FIGURE 12-3
Legend, Details, Duct Drafting Equivalence

There are two classes of details that may be needed to clarify design intent or just to satisfy a requirement of local codes or rules. Building related details, such as installation elevations for ducts or equipment, must be created by the designer for the particular job. Generic details are those that generally apply and can be used on any document. Examples of the latter are fire damper installation details, piping schematics, and wiring diagrams. These are often available from web sites operated by manufacturers or HVAC trade organizations.

Figure 12-3 shows the elevation "A" from figure 12-1, and a detail of a diffuser and fire damper mounted in a fire rated ceiling. In this case, flag note 4 is probably adequate to describe the installation of HP1, but elevation "A" is shown here to clarify the routing of the supply air ductwork where it must run below the ceiling. Elevation "A" is an example of a building related detail.

The diffuser with fire damper detail is needed to ensure that the installation meets the requirements of the designer and of local codes. It describes the diffuser/fire damper in the vault ceiling (Flag note 6). This detail is a modification of a generic detail found on the web.

The use of elevations and details is generally at the discretion of the designer. However, Florida imposes on professional engineers some specific requirements for details in the design documents, and this may also be true of local jurisdictions and of other states.

Quality Control and Operational Notes

As stated at the beginning of this chapter, the design documents must describe the project in sufficient detail for the contractor to perform the work in the manner intended by the designer. The contractor must also know what level of quality is expected of him in order to arrive at a price for the job, and in order that multiple bidders are bidding on the same thing. Large "bid and spec" jobs convey this information with a book of specifications. On small commercial projects, this is conveyed by notes on the drawing that set the requirements for materials, performance, and configuration. For example, if there is no minimum requirement for duct materials, one contractor may decide to save money by using duct board while another decides to bid with internally insulated sheet metal and a third with externally insulated sheet metal meeting the standards of SMACNA, the Sheet Metal and Air Conditioning Contractor's National Association. Thus, the owner will receive three different prices that represent entirely different final outcomes.

A contractor once told me, after I had complained about an aspect of his work, "I didn't know you wanted that. If you want something done a certain way, you have to tell me!" The notes that follow are ones that I have evolved to tell contractors what I want. The general notes cover areas of duct materials, duct configurations, routing, refrigerant piping, test and balance, etc. Without these notes, contractors are forced to choose what I consider to be inferior products and materials if they expect to be the low bidder. The control notes, often supplemented by wiring or logic diagrams, explain to the contractor the sequence of operations and the interlocks I want to be sure that the system performs as I have designed it. Without the control note, the contractor would have no way to know what I want and expect.

For small projects, test and balance is usually just a matter of setting the diffuser, exhaust, and fresh air intake air flows. However, in some cases a special variable volume box or diffuser may be used for a conference room, as is the case with the project shown on figure 12-1. In that case, the contractor or the test and balance technician needs specific instructions on how to set up the system so that air flows will be according to the designer's desire for both the case when the conference room is empty, and when it is fully occupied.

Examples of Notes:

HVAC NOTES

1. ALL DUCTWORK SHALL BE IN COMPLIANCE WITH THE REQUIREMENTS OF THE 2004 FLORIDA MECHANICAL CODE.

2. INSULATED FLEXIBLE DUCTING SHALL BE CONNECTED TO RIGID DUCTWORK WITH SUITABLE TAPS WITH AIR EXTRACTORS AND BALANCE DAMPERS. FLEXIBLE DUCT SHALL BE NO LONGER THAN 6' UNLESS SHOWN LONGER ON THE PLANS. FLEXIBLE DUCT SHALL BE SIZED AS FOLLOWS:

5" UP TO 60 CFM	6" UP TO 100 CFM
7" UP TO 160 CFM	8" UP TO 230 CFM
10" UP TO 420 CFM	12" UP TO 690 CFM
14" UP TO 800 CFM	

3. DRAWINGS ARE DIAGRAMATIC, AND DO NOT NECESSARILY REPRESENT ACTUAL SIZES OR POSSIBLE ROUTINGS OF DUCTS, WIRING, OR PIPING. THE CONTRACTOR SHALL VERIFY CONDITIONS AT THE JOB SITE BEFORE ANY COMPONENT FABRICATION OR ORDERING OF EQUIPMENT THAT IS SUBSTITUTED FOR THAT SCHEDULED.

4. DUCT SIZES SHOWN ARE CLEAR INSIDE DIMENSIONS. ALL DUCTWORK SHALL BE EXTERNALLY INSULATED GALVANIZED SHEET METAL CONFORMING TO SMACNA STANDARDS. DIMENSIONS MAY BE VARIED TO FACILITATE INSTALLATION PROVIDED THAT THE HYDRAULIC DIAMETER IS NOT DIMINISHED. TRANSITIONS FROM ONE SIZE TO ANOTHER SHALL BE WITH SMOOTH TAPERED SECTIONS HAVING A HALF ANGLE OF 15 DEGREES MAXIMUM. ALL ELBOWS SHALL BE MITERED WITH SINGLE THICKNESS TURNING VANES OR CURVED WITH CENTERLINE RADIUS NOT LESS THAN 1.5 TIMES THE DIMENSION OF THE DUCT IN THE PLANE OF THE TURN.

5. MAKE AND MODEL OF COMPONENTS ARE SHOWN AS BASIS OF DESIGN. SUBSTITUTIONS SHALL BE EQUAL TO OR BETTER THAN THE EQUIPMENT SCHEDULED IN QUALITY, APPEARANCE, AND PERFORMANCE. SUPPLY REGISTERS SHALL MATCH OR BETTER THE SPECIFIED UNIT IN THROW, NOISE (NC), APPEARANCE, AND QUALITY OF CONSTRUCTION.

6. ALL BRANCH TAKE-0FFS TO SUPPLY, FRESH AIR, OR EXHAUST TERMINALS SHALL HAVE AN ACCESSIBLE BALANCE DAMPER.

7. DOOR GRILLES SHALL BE METALAIRE MODEL DG-SF 12X6 OR OF THE SIZE INDICATED. SUBSTITUTES SHALL HAVE EQUAL OR GREATER FREE AREA.

8. ALL DUCTWORK SHALL BE SEALED WITH HARDCAST DUCT SEALANT, BOTH TRANSVERSE AND LONGITUDINAL SEAMS. ALL DUCTWORK SHALL BE INSULATED WITH R-6 MINIMUM (2.2" DUCT BLANKET). CLOSURE SHALL BE WITH ALUMINUM TAPE.

9. EXTERIOR DUCTWORK SHALL BE INSULATED TO R-8 USING TWO LAYERS OF 1.5" UNICELLULAR FOAM. EXTERIOR OF INSULATION SHALL BE PAINTED WHITE USING VINYL PAINT AS RECOMMENDED BY THE INSULATION MANUFACTURER. EXTERIOR INSULATION SHALL BE CARRIED THROUGH WALLS AND EXTEND 6" INTO BUILDING SPACE BEFORE TRANSITION TO INTERIOR INSULATION.

10. SYSTEMS SHALL BE FITTED WITH DUAL CONTROL TWO STAGE OR SINGLE STAGE PROGRAMMABLE HEAT PUMP THERMOSTATS HAVING SEPARATE HEATING AND COOLING SETPOINTS, AND EQUIPPED WITH FAN ON-AUTO AND SYSTEM OFF SWITCH. THERMOSTATS SHALL, AS A MINIMUM, ALLOW USERS TO SHUT DOWN SYSTEMS DURING UNOCCUPIED DAILY PERIODS.

11. ALL MITERED ELBOWS SHALL HAVE TURNING VANES. FLEXIBLE DUCT AND ROUND BRANCH TAKE-OFFS FOR SUPPLY DIFFUSERS SHALL BE SPIN-IN FITTINGS WITH AIR SCOOP AND DAMPER. FLEXIBLE DUCT CONNECTIONS FOR RETURN GRILLES SHALL NOT HAVE DAMPERS OR SCOOPS.PROVIDE SQUARE TO ROUND TRANSITION AT DIFFUSERS AND GRILLES. INSULATE ATTIC SIDE OF DIFFUSER TO PREVENT CONDENSATE FORMATION.

12. CONDENSATE LINES SHALL HAVE 2" DEEP TRAP AND BE INSULATED INSIDE THE BUILDING. ROUTE CONDENSATE TO DRY SUMP OUTSIDE BUILDING NEAR CONDENSING UNITS. DRY SUMP SHALL BE 8 CUBIC FEET (2’x2’x2’) FOR EACH 6 TONS OF A/C, AND FILLED WITH GRAVEL OR PEA ROCK. DISCHARGE CONDESATE 6" ABOVE GRADE OVER SUMP.

13. PROVIDE DUCT SMOKE DETECTORS IN SUPPLY AND RETURN AT ALL AIR HANDLERS OF 2000 CFM OR OVER. COORDINATE WITH ELECTRICAL.

14. AIRFLOWS SHALL BE SET BY A TEST AND BALANCE FIRM CERTIFIED BY AABC OR NEBB, AND A TEST AND BALANCE REPORT SHALL BE PROVIDED TO THE OWNER BEFORE FINAL PAYMENT.

15. AIRFLOWS SHALL BE SET BY THE CONTRACTOR, AND A TEST AND BALANCE REPORT SHALL BE PROVIDED TO THE OWNER BEFORE FINAL PAYMENT.

16. ISOLATE AIR HANDLERS AND OTHER ROTATING EQUIPMENT FROM DUCTWORK AND PIPING USING FLEXIBLE CONNECTIONS, FLEXIBLE PIPE JOINTS, AND HOSE KITS.

17. SET CONDENSING UNITS ON 4" REINF CONC SLAB EXTENDING 6" BEYOND UNIT ON ALL SIDES. ANCHOR AT EACH CORNER USING 1/4 x 2-3/4 WEDGE ANCHOR OR EQUAL FASTENER WITH MINIMUM RATING OF 1900 LBS TENSION AND 1200 LBS SHEAR.

18. PROVIDE TYPE II FULL FLOW FIRE DAMPERS IN ALL DUCTS AT PENETRATIONS OF FIRE RATED WALLS OR FLOORS. SEE ARCHITECTURAL FOR RATED WALLS/FLOORS.

19. AIR HANDLERS SUSPENDED OR INSTALLED ABOVE OCCUPIED SPACES SHALL HAVE CONDENSATE OVERFLOW PROTECTION. PROVIDE OVERFLOW PAN UNDER EACH UNIT WITH FLOAT SWITCH TO SHUT DOWN BOTH OUTDOOR AND INDOOR UNIT IF PAN BECOMES FULL.

20. HEAT PUMPS SHALL HAVE OUTDOOR THERMOSTAT TO PREVENT OPERATION OF AUXILIARY ELECTRIC HEAT, EXCEPT FOR DEFROST,WHEN OUTDOOR TEMPERATURE IS ABOVE 40 DEG F. AUXILIARY HEAT SHALL ENERGIZE FOR DEFROST OR WHEN INDOOR TEMPERATURE IS MORE THAN 3 DEG BELOW SETPOINT. AUX HEAT OF MORE THAN 10 KW SHALL HAVE TWO STAGES. THE SECOND STAGE SHALL ENERGIZE WHEN INDOOR TEMPERATURE IS MORE THAN 5 DEG BELOW SETPOINT. BOTH STAGES SHALL ENERGIZE FOR DEFROST.

21. PROVIDE CONDENSATE PUMP WITH OVERFLOW ALARM AND SWITCH FOR GROUND FLOOR UNITS, AND ROUTE CONDENSATE OVERHEAD TO DRY SUMP NEAR CONDENSING UNITS. PROVIDE OVERFLOW RELIEF OUTSIDE AT A POINT LOWER THAN THE BUILDING SLAB. OVERFLOW ALARM SWITCH SHALL SHUT DOWN AC SYSTEM IF RESERVOIR BECOMES FULL.

22. REFRIGERANT PIPING SHALL BE SIZED FOR PRESSURE DROP CORRESPONDING TO NOT MORE THAN 2.5 DEG F, USING CARRIER OR TRANE REFRIGERANT SIZING HANDBOOKS.

23. LOUVERS SHALL BE ALUMINUM DRAINABLE BLADE EQUAL TO RUSKIN ELF 375.

Notes 14 and 15 above cover the same subject. Note 14 allows the contractor to set the air flows, and is acceptable for very small jobs such as that shown on Figure 12-1. Note 15 should be used for larger projects with multiple zones and more complex control instructions. Any generic notes developed by the designer must be edited for applicability to each project.

CONTROL NOTES

PAC1/OAU1/EF1/EF2/EF3/EF4:

THE PAC1 BLOWER SHALL OPERATE CONTINUOUSLY WHEN THE SYSTEM IS IN OCCUPIED MODE. THE PAC1 COMPRESSOR SHALL OPERATE ON CALL FOR COOLING OR HEATING BY THE THERMOSTAT.

OAU1 SHALL BE INTERLOCKED WITH PAC1 AND THE BLOWER SHALL OPERATE CONTINUOUSLY WHEN THE PAC1 BLOWER IS ON. OPERATE COMPRESSORS TO MAINTAIN RELATIVE HUMIDITY AT HUMIDISTAT SET POINT OF 60%. OAU1 SHALL MAINTAIN DELIVERED AIR TO PAC1 AT 75 DEGREES USING MECHANICAL COOLING AND HOT GAS REHEAT, SUPPLEMENTED WITH HOT GAS BYPASS TO MAINTAIN EVAPORATOR LEAVING TEMPERATURE IN THE RANGE 52 TO 56 DEGREES.

EF1, EF2, AND EF4 SHALL BE INTERLOCKED WITH OAU1 AND SHALL OPERATE WITH OAU1 IS ON.

HP2/MAU1

THE HP2 BLOWER SHALL OPERATE CONTINUOUSLY WHEN THE SYSTEM IS IN OCCUPIED MODE. THE HP2 COMPRESSOR SHALL OPERATE ON CALL FOR COOLING OR HEATING BY THE THERMOSTAT.

MAU1 SHALL BE INTERLOCKED WITH HP2 AND SHALL RUN WHEN HP2 IS ON.

TEST AND BALANCE NOTE

SET ALL AIR FLOWS, INCLUDING THE CONFERENCE ROOM AIR FLOW, WITH THE CONFERENCE ROOM VAV DIFFUSER DRIVEN TO THE MINIMUM AIR FLOW POSITION.

These are just a few examples of clarifying notes. The term “system in occupied mode” is used instead of “system switch on” or “system is on” because if the system is turned on and off by a time control, then the thermostat system switch may be “on” even when the system is disabled by time of day control.

END

Chapter 13

Checking Your Work

The Designer's Dilemma

How can errors and omissions be avoided, especially for the practitioner working alone? Errors can be expensive, if they lead a contractor to perform work that later has to be undone. Omissions of needed equipment are classified as "first cost", meaning that the owner would have had to pay for the equipment if it had been included in the original contractor's bid. Thus, omissions are not usually costly to the designer, but can be very costly to the owner, who pays a non-competitive price to the contractor for providing the omitted equipment.

Design engineers working in a firm with colleagues, or working under a more senior engineer, have the advantage of a second or third set of competent eyes to check their work. But even this can be detrimental, because the knowledge that others are checking after him may cause the designer to take less care in his own reviews. Following are a few common pitfalls, and a procedure to avoid most errors and omissions.

Common Pitfalls

Failure to properly edit generic schedules: Most designers and design offices necessarily use "generic" equipment schedules and notes – basically re-using a schedule or note set used on a previous project. Referring to Figure 12-2, it can be seen that the data list on a schedule can apply to any manufacturer or size of a particular type of device. So when a schedule is imported from a previous project, it may have to be edited for applicability to the current project. The most common error is to use a schedule that matched the make and model of the new job, but has a different voltage. If the air conditioning contractor, who may not be familiar with the electrical service to be used on the job, orders the wrong voltage because of the schedule, correcting this problem can be costly to the designer. Likewise, it can be seen by reviewing the example "HVAC Notes" in Chapter 12 that several of the notes do not apply to the HVAC plan of Figure 12-1.

Using the wrong ID code: The schedules show an ID code that identifies the device on the HVAC drawing. This obviously must be edited for each project, and failure to do so can result in confusion, delay, and incorrect orders by the contractor.

Failure to re-check schedules, notes, and duct sizes after a design change: Changes to the design are relatively common during the design process. Even minor changes may involve elements such as duct and louver sizes, diffuser and grille schedules, flag notes, etc.

Incomplete or inaccurate data about the project. Owners are usually in a great rush for the design, after they have spent months in planning and obtaining financing. There is great pressure on the designer to begin work before all of the design information (described in Chapter 2) is available. Starting before the project is completely defined is often called "fast track". While the HVAC designer will be entitled to additional fees if architectural changes or owner's instructions are changed in a way to require re-design of parts of the job, the pitfall here is for errors and omissions to creep in as a result of the changes.

Reducing or Eliminating Errors and Omissions

HVAC Project checklist:

Project:	Example Office		
Date:	6/30/2007		
ITEM	TASK	DONE	CHECKED
1	exhaust fans and ducting		
2	fresh air intake		
3	building air balance		
4	duct standards		
5	duct sizing		
6	condensing units		
7	roof equipment code statement		
8	outside duct insulation		
9	smoke detectors		
10	secure ductwork outside		
11	condensate disposal		
12	control interlocks		
13	Thermostats		
14	balance dampers		
15	hvac legend		
16	fan schedule		
17	register schedule		
18	hp/ac schedules		
19	outdoor air unit schedule		
20	heaters, furnaces, etc		
21	energy recovery		
22	hvac notes		
23	flag notes		
23	details - fire dampers, sections, etc.		

To reduce errors, first follow the procedure outlined in Chapter 1. Complete each task before beginning the next, as each task is generally a prerequisite to the next.

At the completion of each task, review the work, and apply applicable items of the check list.

The most important element of design quality control is not the review of a peer or supervisor, or a careful scan of the project by the designer. It is the check list. The check list currently used by the author is shown on the preceding page.

Note that there are two columns for each element of the check list. It is important to follow this protocol – when satisfied that an item is done, check the "done" column. Then, when the job is complete, go back to each item and check that it is complete and accurate. If working in a large organization, there should be two "check" columns, one for the designer, and one for a colleague or supervisor. The second checker should be given a set of design documents, to allow checking simultaneously with the designer. If one or the other knows that the job has already been checked, the second checker may relax vigilance.

END

SYMBOLS

A	heat transfer surface area, sq ft
C	air flow, cfm
Cc	coil air flow rate, cfm
CLF	Cooling load factor
CLTD	cooling load temperature difference, °F
Coa	ventilation (outdoor air) air flow rate, cfm
c_p	specific heat, Btu/lbm/˚F
Cr	return air flow rate, cfm
Ed	diffuse solar radiation on a surface
ED	direct solar radiation on a surface
eh	coil entering enthalpy, btu/lb dry air
Er	reflected solar radiation on a surface
h	enthalpy, btu/lb dry air
hent	coil entering air enthalpy, btu/lb dry air
hlv	coil leaving enthalpy, btu/lb dry air
hr	room air enthalpy, btu/lb dry air
hrrs	room required supply air enthalpy, btu/lb dry air
K	factor to correct CLTD for color
LM	correction to CLTD for month and latitude, °F
OAT	outdoor air temperature
odb	outdoor dry bulb temperature, °F
owb	outdoor wet bulb temperature, °F
q	local instantaneous heat gain or loss, Btu per hour
Q	space cooling load, Btu per hour
Qrs	room sensible cooling load
Qrt	room total cooling load
Qscc	Sensible Cooling Capacity of an air conditioning System
Qtcc	Total Cooling Capacity of an air conditioning System
rdb	room dry bulb temperature, °F
renth	room enthalpy, Btu per pound of dry air
rho	density, lbm/˚F
rrsh	room required supply enthalpy
RSCL	room sensible cooling load
RTCL	room total cooling load
rwb	room wet bulb temperature, °F
SC	shading coefficient, modifies solar heat gain factor
SCC	Sensible Cooling Capacity of an air conditioning System
SCL	Sensible Cooling Load
SHGC	solar heat gain coefficient
SHGF	solar heat gain factor, function of latitude, month, orientation
TCC	total cooling capacity of an air conditioning system
TCL	total cooling load
TD	temperature difference between rooms or spaces.

tdp	dew point temperature, °F
tedb, edb	coil entering dry bulb temperature, °F
tewb, ewb	coil entering wet bulb temperature, °F
tldb, ldb	coil leaving dry bulb temperature, °F
tlwb, lwb	coil leaving wet bulb temperature, °F
to	average outdoor ambient temperature, °F
tr, trdb	return air dry bulb temperature, °F (usually = trm)
trm	room dry bulb temperature, °F
trsdb	room required supply dry bulb
trswb	room required supply wet bulb
trwb	return air wet bulb temperature, °F (also room wet bulb)
ts	sol-air temperature, °F
U	overall coefficient of heat transfer, Btu/(hr*ft^2 *°F)

END

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APENDIX

ECONOMIZER PSYCHROMETRICS AND DESIGN NOTES

1. Foreword

The purpose of an air-side economizer in an HVAC system is to provide cooling energy savings by shifting cooling load from the air conditioning system to a regulated flow of relatively cool and dry outdoor air using a combination of sensors, dampers, and actuators. To quote from reference 6: "Unfortunately, the economizer's collection of dampers, actuators, linkages, sensors, and controllers rarely achieves its savings potential. Estimates indicate that only about one in four economizers works properly, with the remaining three providing sub-par performance or, worse yet, wasting prodigious amounts of energy."

2. Scope

This appendix provides air side economizer design guidance to help engineers and technicians avoid the errors that lead to economizer failure and resultant occupant discomfort and wasted energy. The most common problem is failed damper systems, which can be avoided by specifying high quality components and employing the principles embodied in ASHRAE *Guideline 16*.[1] Occupant discomfort can also result from humidity and moisture problems that occur during economizer operation if operating limits are set too aggressively. The use of psychrometric analysis to set rational operating limits will be demonstrated using two examples – an office and an assembly occupancy. Finally, the characteristics of various control systems will be discussed, with examples of how they should be specified.

3. ASHRAE Standards

The current energy standard of the American Society of Heating Refrigeration and Air Conditioning Engineers (ASHRAE) is Standard 90.1-2010.[2] Standard 90.1 establishes eight climate regions, numbered 1 through 8, and a number of sub-regions designated with appended letters such as 2b, 4c, etc. Standard 90.1-2007 [3] mandated economizers in all zones except 1a, 2a, 3a, and 4a, which are basically all of the counties east of the Mississippi and south of the Mason-Dixon line plus western Louisiana and eastern Texas. (Appendix B of the Standard lists the climate zones for every state and county in the U.S.) The recently released Standard 90.1-2010 added zones 2a, 3a and 4a to the mandate, leaving only zone 1 exempt. In all zones, systems under 58,000 Btuh capacity are exempt.

The student who intends to design and specify economizer systems must obtain and be familiar with the latest issues of Standard 90.1 and Guideline 16, in order to comply with details specific to his climate zone area and economizer configuration. This appendix will outline the requirements in those publications, and will provide details where relevant to examples.

4. Important Terms

The following terms will be used throughout this appendix. They have specific meanings in connection with HVAC systems, ventilation, and indoor air quality.

A **building** is a roofed and walled structure with controlled environment, built for human occupation and use.

A **ceiling plenum** is a cavity within the pressure envelope that is above a room and that is formed by a dropped lay-in ceiling and floor or roof structure above. Room walls do not necessarily extend above the dropped ceiling to the structure above.

An **air side economizer** is a coordinated set of louvers, dampers, sensors, actuators, and controls that can allow relatively cool, dry outdoor air to partially or fully cool a zone in lieu of mechanical cooling equipment.

The **economizer high limit shutoff control** is the state or states of outdoor air that define the locus of temperature and humidity below which the economizer may be activated.

Exfiltration is the portion of supply air that leaks out of a conditioned zone through breaches in the pressure envelope when zone pressure exceeds outdoor ambient pressure. **Infiltration** is the opposite.

Exhaust, or **exhaust air**, is the portion of the supply air that is intentionally discharged from the zone to outdoors after passing through the zone.

The **pressure envelope** is the primary air barrier of the building, which is sealed to provide the greatest resistance to air leakage from the unconditioned environment. One or more **zones** may be within a building pressure envelope.

Return air is the portion of the supply air that is recirculated to the cooling/heating apparatus after being collected by the return grilles in the zone.

A **return air plenum** is a ceiling plenum with an unobstructed path to an air handler return, and that contains no flammable materials or surfaces.

A **room** is the part of a space bounded by walls and a ceiling that is usually routinely occupied and served by grilles and registers to supply and recirculate or to exhaust conditioned air.

A **space** is a single room, with or without a ceiling plenum

Supply air is the all of the air delivered by the cooling/heating apparatus to the supply air diffusers in the zone.

The **supply air critical state** is the maximum temperature and dew point of the supply air that will satisfy the zone design temperature and relative humidity at design air flow and cooling load.

The **thermal envelope** of a building is the physical separation between the conditioned space and the unconditioned environment. It holds the primary insulation layer of the building where resistance to heat transfer is the greatest. One or more **zones** may be within the thermal envelope.

Ventilation air, also called **outdoor air,** is air from outdoors that may be mixed with return air before passing into the cooling/heating apparatus, may be introduced to the apparatus directly before entering a zone, or in certain rare circumstances, may be introduced un-tempered into a zone.

Minimum **ventilation air** is the minimum outdoor air flow that must be introduced into a zone as make-up for code required exhaust or to meet indoor air quality standards. The calculation of minimum ventilation air flow is explained in Chapter 4 of this book.

A **zone** is a group of spaces within the thermal and pressure envelopes which are served by a single air handling system.

A **sub-zone** is a group of spaces within a zone that may be served by a single terminal component such as a variable air volume unit.

5. Basic Configuration

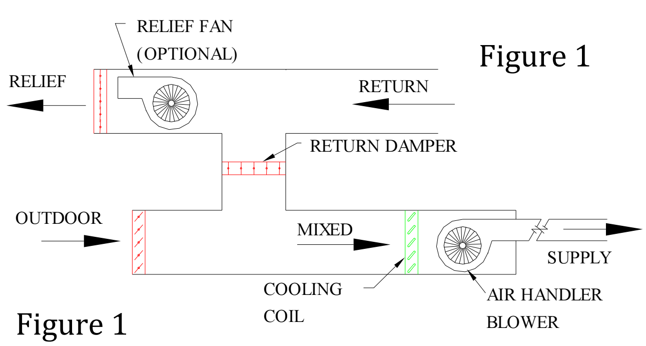

Figure 1

Figure 1

ECONOMIZER SCHEMATIC

Figure 1 is a schematic representation of an air side economizer. The outdoor air and relief dampers are sized to pass the full required cooling supply air flow and the return damper is sized to provide the supply air flow less the required minimum outdoor air flow. When the zone requires heat, or when outdoor conditions are warm and humid, the relief damper is fully closed, the return damper is open, and the outdoor air damper is fixed in the minimum ventilation position, as established by Standard 62.1-2010 and building pressurization requirements. See Chapter 4. If the zone requires cooling and outdoor air is cool and dry, the economizer can modulate the outdoor, return, and relief dampers to either provide all of the cooling needed, or to take some of the cooling load off of the zone mechanical cooling system.

Figure 2 shows a schematic configuration of an air handler zone with an economizer. This is to put figure 1 in the context of an actual system and to illustrate terms. The system is shown in “normal” (not economizer) mode with the return damper open, the relief damper closed, and the outdoor air damper (O.A. Damper) in the minimum ventilation position.

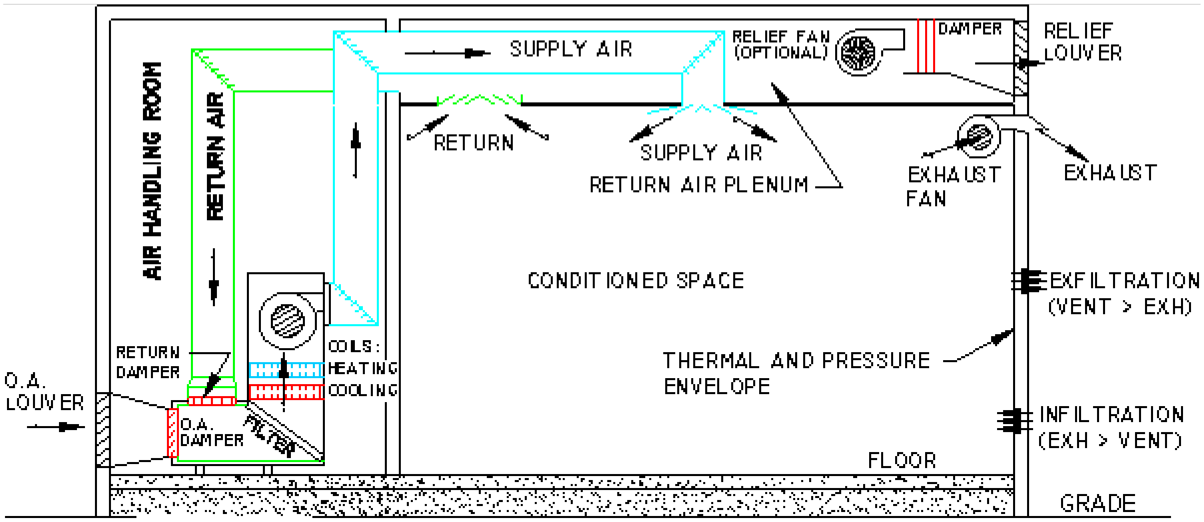

Figure 2 AIR HANDLER ZONE WITH ECONOMIZER

The outdoor air intake may be divided into two dampers, an economizer damper and a small damper to set minimum air flow and allow the large economizer damper to be tightly closed when the economizer is not operating. This configuration is good practice when the minimum ventilation air is a relatively small percentage of design supply air flow, as is often the case for office zones. (If the minimum flow is small and the damper is large, precise control of minimum air flow is virtually impossible because a small variation in the damper minimum flow blade angle will cause a large error in minimum air flow.)

The object of the mandatory economizer requirement is of course to save energy without compromising comfort conditions in the occupied spaces. If occupant comfort is compromised because of damper failure or moisture introduced by the economizer itself, then the benefits of any energy saving will be lost.

It is assumed that all of the allowed high limit shutoff controls, as described in section 8, can achieve the space design temperature. However, it will be shown that certain of the allowed control methods can result in space relative humidity exceeding the base design, which will usually be between 50% and 60%. It is therefore essential that the engineer perform psychrometric analysis of his selected economizer high limits.

6. Louver and Damper Sizing and Quality Control

The basic economizer configuration shown on figure 2 includes a constant volume air handler, modulating return and outdoor air dampers, and either a modulating relief damper or a variable speed relief fan with two-position relief damper. The relief and outdoor air dampers are not in themselves weather proof, and so must be ducted to louvers located on the building exterior weather envelope.

Louvers

Louvers should be good quality aluminum or galvanized steel. To preclude rain entrainment, the outdoor air intake louver should be sized for a free area flow velocity of not more than 500 feet per minute (fpm) at the maximum design air flow. The relief louver should be designed for a maximum combined pressure drop with the relief damper of not more than 0.05 inches of water at the maximum design relief air flow, to prevent excessive building exterior door loads (caused by excessive building internal pressure) when the modulating relief damper is fully open. Louvers should be specified with bird screens, but not with insect screens that can become clogged with airborne dust. Any insects will be interdicted by the AC system filters.

Damper Sizing

As noted above, the combined pressure drop of the fully open relief damper and the relief louver should not exceed .05 inches of water at the maximum design relief air flow. However, in order to ensure stable control over the full range of travel, both the outdoor air damper and the return damper should be designed with a slightly higher fully open pressure drop of about 0.1 inches of water at their respective maximum design air flows.

This rule applies strictly only to the configuration shown on figure 2. Other configurations, which may include return air fans and variable volume air handler systems, may call for different sizing criteria. The objective is to ensure seamless transfer from normal to economizer operation, without pressure surges or noticeable changes in space air flow. Several alternative configurations are discussed in reference 1, with recommendations for sizing and controlling dampers.

Quality Control

The most common economizer failures are jammed, broken, or bent dampers. A few days with a relief or outdoor air damper that is partly or fully open on warm, moist days can easily offset a year of energy saving by a properly functioning economizer. Occupant reaction to discomfort caused by damper failure can lead to demands to disable the economizer controls and permanently seal the damper openings. The engineer can diminish these problems with a good damper specification.

Since these dampers are expected to be continually working and modulating, and because they represent a large heat transfer and potential leak area directly from outdoors into the air conditioning ductwork, they must be insulated, low leakage, and very high quality. They should include the following properties:

- extruded aluminum or galvanized steel
- airfoil blades
- extruded silicone or similar rubber blade seals
- stainless steel blade shafts
- bronze sleeve shaft bearings
- compression type stainless steel jamb seals.

Leakage should be less than 10 cfm/ft^2 at 1" H2O pressure differential, or 4 cfm/ft^2 where required by Standard 90.1. Outdoor air, return, and relief dampers should be operated by direct acting actuators having sufficient torque to operate the damper as recommended by the damper manufacturer.

These features will minimize failures, but do not remove the requirement for annual or bi-annual inspections by qualified technicians.

7. Operation and Control

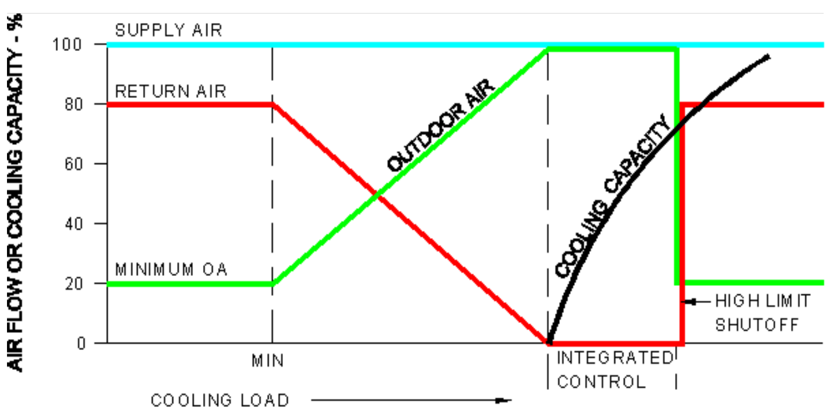

Figure 3 - Typical Economizer Sequence

Figure 3 is a graphic representation of an economizer cooling cycle. At some low cooling load, when outdoor conditions permit economizer operation, all of the cooling can be provided with the outdoor air damper in the minimum ventilation position, the return damper fully open, and the relief damper closed.

As the cooling load increases, the outdoor air damper modulates open to maintain the space temperature cooling set point, the relief damper modulates open to maintain constant zone positive pressure, and the return damper modulates to maintain constant supply air flow. Eventually the cooling load increases to the point that the return damper is closed and all of the zone air flow is supplied through the outdoor air damper.

As cooling load continues to increase, Standard 90.1-2010 mandates "integrated" economizer operation, where the economizer is fully open (providing all of the supply air) and the zone cooling system is cycling to maintain the space temperature set point.

This mode of operation will continue until the outdoor conditions exceed the high limit set point, when the dampers will return to "normal" cooling operation – economizer damper closed or in minimum ventilation position, relief damper closed, and return damper open. (Standard 90.1-2007 mandates integrated economizer operation only in zones 2B, 3B, 3C, 4B and 4C, and some jurisdictions may continue to allow the 2007 exemptions for some period into the future.) There are problems with integrated operation when cooling is provided by DX systems, which must be addressed by the engineer.

8. Psychrometric Fundamentals

This Appendix is about applying psychrometrics to air side economizer operation. It is assumed that the student is already familiar with psychrometric processes. The student is directed to Chapter 7 of this book for a brief refresher in psychrometrics, with details and examples of plotting points on the psychrometric chart.

9. Cooling System Psychrometric State Points

As established in Chapter 7, psychrometric state points will be identified as follows:

1 – room air
1A – **supply air critical state** point – not a physical point
2 – outdoor air
3A – mixed air entering heat pipe or pre-cool coil, where applicable
3 – mixed air entering cooling coil
4 – air leaving cooling coil
4A – air leaving heat pipe or reheat coil, where applicable

Figure 4 is a schematic representation of a cooling system, repeated from Chapter 7. It shows the physical location of the state points except for point 1A, which is a calculated, not a physical, point.

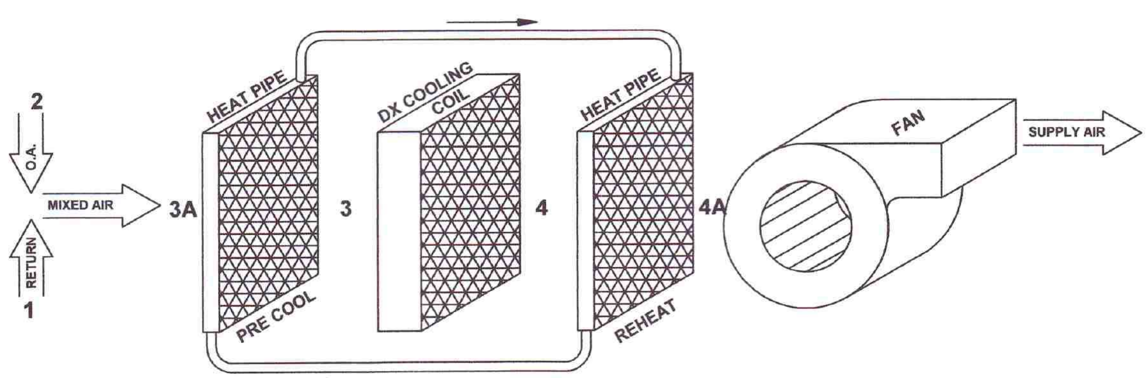

Figure 4 – DX Cooling System Schematic

10. High Limit Shutoff Controls

Standard 90.1 prescribes a limited number of allowable high limit shutoff controls:

- fixed dry bulb
- differential dry bulb
- fixed enthalpy
- electronic enthalpy
- differential enthalpy
- dew-point/dry bulb

Each of these systems defines the locus of an outdoor temperature/humidity condition below which “free” cooling using outdoor air may be enabled.

Fixed dry bulb is a fixed temperature below which economizer operation may be permitted. Standard 90.1-2010 prohibits fixed dry bulb in climate zones 1a, 2a, 3a, 4a, and sets minimum allowable values for other zones. The minimum allowable fixed dry bulb temperature for climate zones 5a and 6a is 70˚F, but is 75˚F in all others where allowed. Under certain conditions, especially in zones where the 75˚ minimum is mandated, this control will allow economizer operation with outdoor air that is excessively humid.

Fixed enthalpy permits economizer operation below a set value of enthalpy. It is permitted only in zones 1a, 2a, 3a, 4a, 5a, and 6a. At sea level, it must be set to 28 Btu/lb or higher.

Electronic enthalpy controllers are packaged devices that use a combination of humidity and dry-bulb temperature to establish a shutoff curve on the psychrometric chart. All such devices have limit curves from A to D. The standard requires that the “A” limit curve be used. See reference 4.

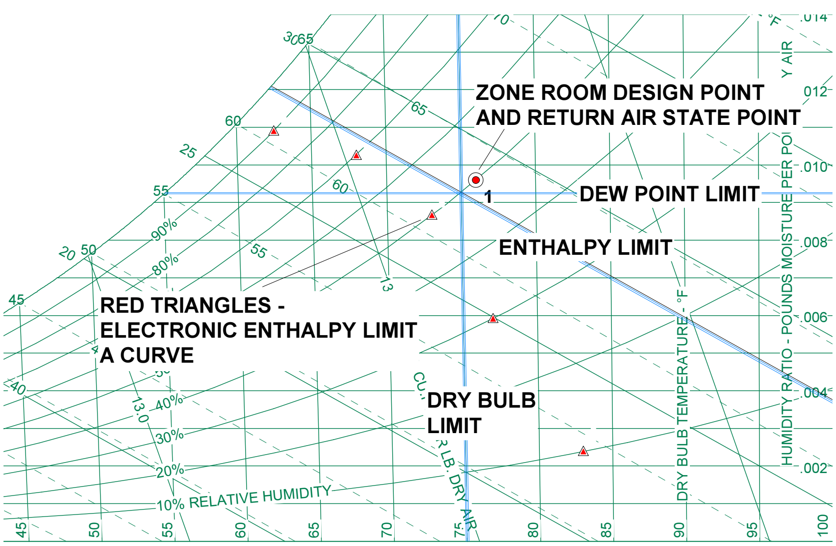

Figure 5 – Economizer High Limit Controls

Differential dry bulb requires only that the outdoor air temperature be lower than the return air temperature. This control is prohibited in climate zones 1a, 2a, 3a, and 4a. This control can allow economizer operation with excessively humid air and that uses more energy than mechanical cooling.

Differential Enthalpy requires only that the outdoor air enthalpy be lower than the return air enthalpy. While this will ensure that energy use is lower than with mechanical cooling, it does not prevent operation with high outdoor air humidity.

Dew point/dry bulb sets a minimum dew point limit of 55˚F and a minimum dry bulb limit of 75˚ for all climate zones.

Of the six allowed cutoff control schemes, only the dew point/dry bulb provides iron-clad assurance that the economizer will operate without inducing excessive moisture. However, both enthalpy control and electronic enthalpy control provide reasonable assurance against excessive humidity. It will be shown that in warm, dry climates enthalpy control can allow economizer operation when the outdoor temperature is too high to maintain the zone design set point. This point is also made in reference 5.

Figure 5 shows the allowable high limits plotted on a psychrometric chart with a typical zone design point. The differential enthalpy and differential dry bulb points would run just to the left of the design point.

11. Psychrometric Analysis

So when should the economizer be allowed to operate? As noted before, this question will be addressed for the occupancy types of office and church sanctuary. The control Schemes to be examined are fixed dry bulb, electronic enthalpy, and dew point/dry bulb. The location is Minneapolis, Minnesota, in climate zone 6a. This climate zone was chosen because it is a zone with relatively moist conditions, though no really hot days. Reference 5 discusses psychrometric conditions in several other regions of the U.S.

Office

First it is necessary to review the psychrometric charts of the two systems operating at their design points. This is shown on Figure 6 for the office and represents the operation of a 5 ton AC unit with a supply air flow of 2000 cfm and an outdoor air flow of 330 cfm, at design outdoor conditions of 91˚F dry bulb and 73.5˚ wet bulb -

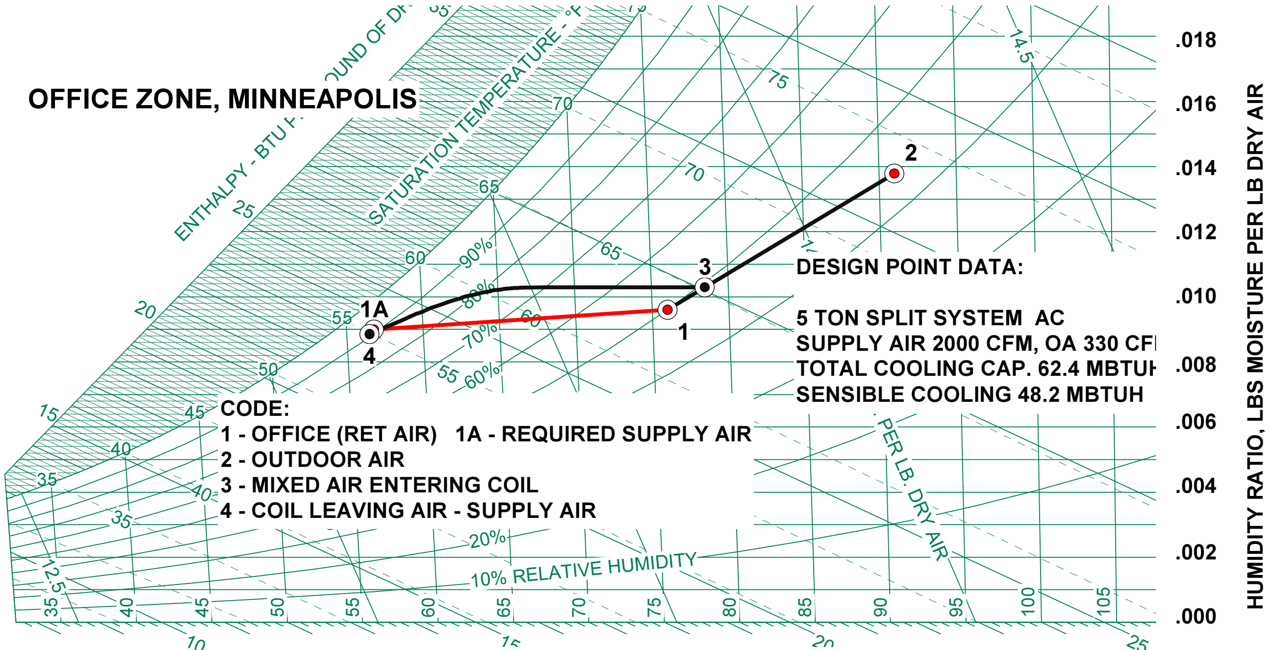

FIGURE 6 MINNESOTA OFFICE AIR CONDITIONING DESIGN POINT

point 2 on the chart. The design room condition is point **1**, 76˚db and 50% RH. The coil entering conditions are point **3**.

The **room process line**, **1-1A** represents the room sensible and total air conditioning load at the design condition and airflow. The **coil process line 3-4** represents the capacity of the air conditioning system – specifically the evaporator coil. Point **4** is the state point of the cooling/dehumidifying air being delivered to the room air diffusers.

Since the supply air will follow the slope of the room process line as it picks up heat and moisture, it is essential that point **4** be at a lower dew point and dry bulb temperature than point **1A**. If point **4** is at a higher temperature and dew point than point **1A**, then the system will not hold the desired design condition, and the room will be warmer and more humid than point **1**. Thus, point **1A** is designated the **critical supply air state** point.

Figure 6 represents a hot, humid design day, not a day when the economizer could be used. Figure 7 shows the room process line for two conditions, both cool, dry days in late September or October. As noted before, no outdoor air point with a higher temperature than point **1A** can satisfy the design space temperature without mechanical refrigeration supplement, and no outdoor air state point that lies above the room process line **1-1A** can satisfy the design room humidity ratio. The blue lines on figure 7 show the mandatory climate zone 6A minimums of economizer operation for dry-bulb and enthalpy control, respectively.

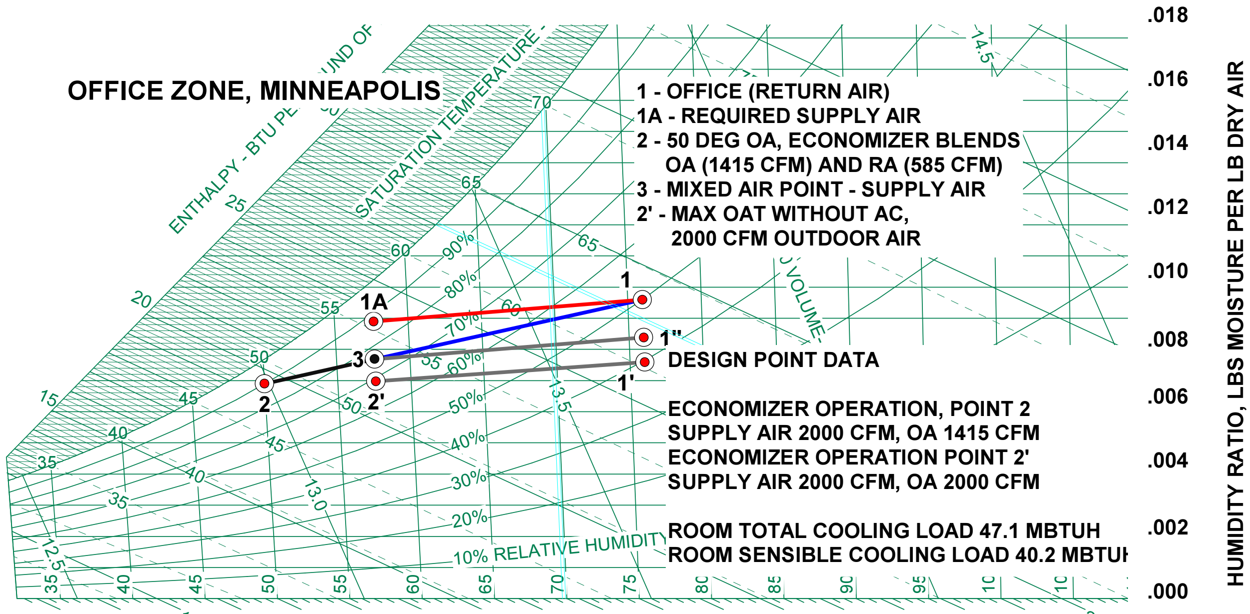

FIGURE 7 PURE ECONOMIZER, TWO CONDITIONS

In the case of point **2**, outdoor air at 50° is blended with return air to deliver mixed air at point **3**, exactly the temperature needed to achieve the design room set point. As outdoor air temperature rises, economizer air flow increases and return air flow decreases until at point **2'** the outdoor air temperature is the same as the required supply air temperature (point **1A**) and supply air is 100% outdoor air. Going in the other direction, as outdoor air becomes cooler and cooler, the economizer damper, responding to room temperature, will close more and more, until only the minimum outdoor air, blended with return air, is needed to meet the cooling load. If minimum ventilation is supplied through a separate damper, the economizer and relief dampers will be fully closed at this point. The system is basically back to "normal" operation as the room temperature falls through the dead band to the heating set point.

Note that point **1A** on Figure 7 is at a higher dry bulb temperature than on Figure 6, the design condition. This is caused by the reduced envelope loads at the cooler outdoor temperatures. Point **1'** on the gray process line shows the room condition that will occur with the economizer operating at point **2'**.

Under 90.1-2007, integrated control is not required, so the economizer high limit could be set at about the temperature of point 1A, and at higher temperatures, the system would be operating with minimum ventilation. If this is the case, a simple dry-bulb limit will work in all cases for the office zone, since there is little space above the room process line for any significant increase in humidity. However, under 90.1-2010 integrated control is mandated, and a dry bulb limit alone may not be the best choice. Also, it should be pointed out that in the previously exempted southeast zones 2a, 3a, and 4a both fixed and differential dry bulb control is prohibited.

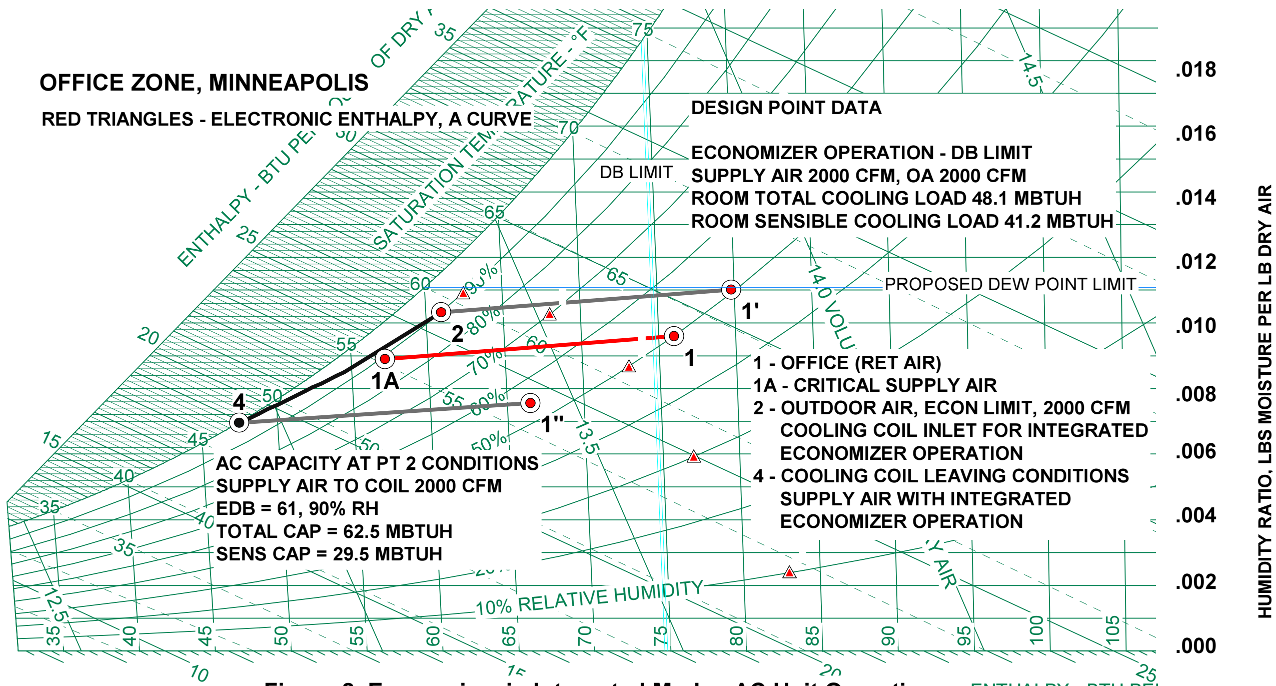

Figure 8, Economizer in Integrated Mode - AC Unit Operation

Figure 8 shows the office of figure 5 operating under integrated economizer control at an outdoor temperature of 61˚and 90% rh, not unusual conditions for Minneapolis in late September. Since operation is now "integrated", point **2** is also the inlet condition to the DX coil. The leaving state point of the 5 ton evaporator coil, assuming a single stage compressor, is represented by point **4**[i]. The reader must study this chart to understand its implications.

Air supplied to the diffusers at the conditions of state point **1A** will satisfy the design room conditions of 76˚and 50% RH. Since the air entering at point **4** is cooler and dryer than point **1A** the single stage AC unit compressor will cycle as needed to maintain the room set point. During off times, the relatively moist air of point **2** will be brought into the building, raising its humidity, as represented by the gray process line and point **1'**.

While the moisture gain caused by cycling is difficult to quantify, and is dependent on the part load capabilities of the AC unit, it can be seen that operation in the large triangle above the dew point of 60˚F and to the left of the 75˚F limit could introduce significant moisture during compressor off times. To avoid this, the engineer could choose a dry-bulb/dew point limit, such as indicated by the blue lines on Figure 8.

Another solution is to specify a two speed unit or multiple condensing units with a face split evaporator. Two stage and multi-speed compressors are offered in units as small

[i] DX system performance is not directly available at 61˚ outdoor air in manufacturer's performance data. Therefore, the coil performance shown is extrapolated from 75˚condenser air and 80˚ coil entering data, using curve fit. A better solution would be to request that the manufacturer provide performance data at the coil inlet conditions to be studied.

as five tons by some manufacturers, and further development of such systems is ongoing.

The red triangles represent the "A" curve of an electronic enthalpy module[4] – one of the permitted high limit shutoff controls permitted by Stanbdard 90.1-2010[2]. This is the control used on many packaged ac units, and has the advantage of being a single pre-engineered module, while the dry bulb-dew point limit will be most easily implemented as part of a direct digital control system. A consideration when electing the electronic enthalpy is the risk in dry climates of operation at outdoor temperatures that exceed the design temperature when relative humidity is very low – less than 30%. In such a case, the air conditioner may not be able to hold space temperature.

There can be problems with DX systems acting in integrated economizer mode with low temperature on the condenser and at the evaporator inlet, so the engineer should consult with manufacturers when specifying this mode.

Church Sanctuary

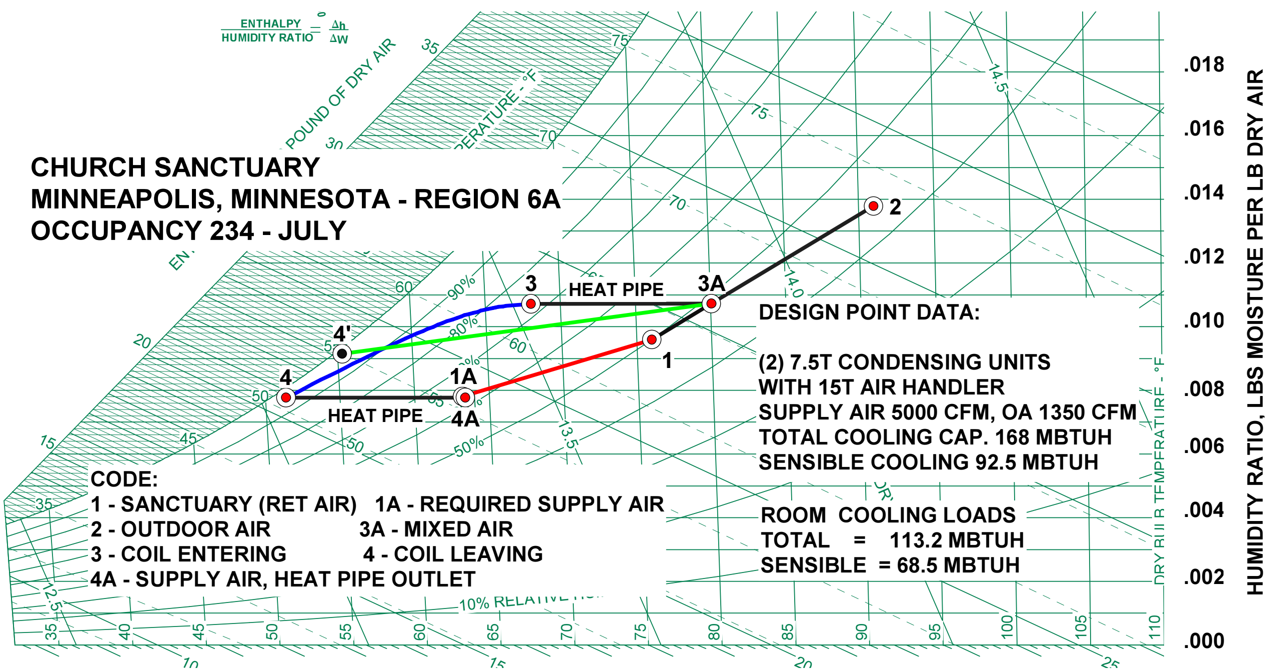

Figure 9, Assembly Occupancy Design Point

Economizer design for buildings with low occupant density, such as offices, is straightforward and without significant risk of problems because the room load sensible heat ratio is relatively large, with a small triangle of outdoor state points lying above the room load process line. This is not true of zones with high occupant density, such as church sanctuaries, classrooms, and auditoriums. To illustrate, a church sanctuary in Minneapolis will be analyzed.

Figure 9 is a chart of the cooling system design point for a church sanctuary served by a 15 ton DX air handler on two 7.5 ton condensing units. As before, the occupied zone design point is 76˚at 50% RH. Outdoor ambient is the ASHRAE .4% cooling design point for Minneapolis, 91˚DB at 73.5˚MCWB. Note that the sanctuary (room) process line, **1 – 1A**, is very steep, and does not intersect the saturation line. The steep slope is caused by the latent load of 234 occupants. Also, the temperature at point **1A**, which is the maximum temperature that will achieve the room design point, is 63˚. There is no DX air conditioning system that can follow this process line, and if there were, the coil leaving temperature of 63˚ would be too high for effective dehumidification.

The green line from **3A** to **4'** on Figure 9 is the process line for the 15 ton cooling coil at coil inlet conditions of mixed air point **3A**. It is evident that supply air delivered to the sanctuary at the conditions of point **4'** cannot satisfy the moisture removal requirements, and the large sensible capacity of the unit will result in short run times and long off times, exacerbating the moisture conditions in the sanctuary.

The solution selected here is a heat pipe, which allows both a low coil leaving temperature (point **4**) and a supply air temperature and dew point near the ideal (point **4A**). (A system with hot gas reheat was also considered, but would

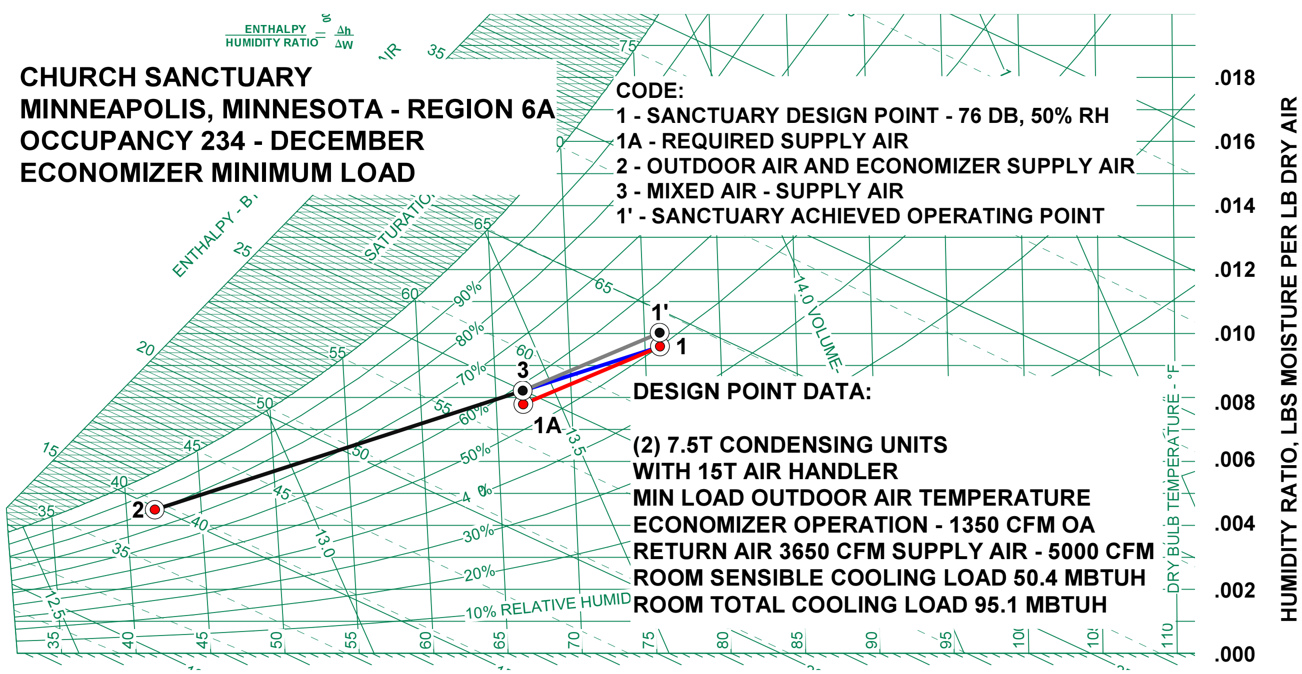

Figure 10, Economizer Operation at Minimum Load

have required the next larger size unit – 20 tons – and would result in longer run times for dehumidification.) As shown on the diagram of Figure 4, a heat pipe is a refrigerant coil wrapped around the cooling coil which transfers sensible heat from the air entering the coil to the air leaving the coil, thus cooling the entering air and re-heating the air leaving the coil by the same Δt. For this example, The heat pipe selected temperature difference (Δt) is 12˚F. Heat pipes are examined in detail in Chapters 4 and 9.

Figure 10 represents the same system with an economizer operating at minimum load. Even though the outdoor temperature is a cold 42˚, the sanctuary space load is only about 25% below the design load in July, because it is dominated by internal loads. In this case, minimum load means the cooling load that results in the economizer delivering the minimum allowable outdoor air. If OAT falls further, the space temperature will simply fall through the thermostat dead band until the heating set point is reached.

The gray line on Figure 10, from point **3** to point **1'**, shows that at the outdoor condition shown, the economizer cannot maintain the design RH of 50%[ii]. However, the achieved point of 76˚ at less than 60% RH is acceptable based on ASHRAE standards.

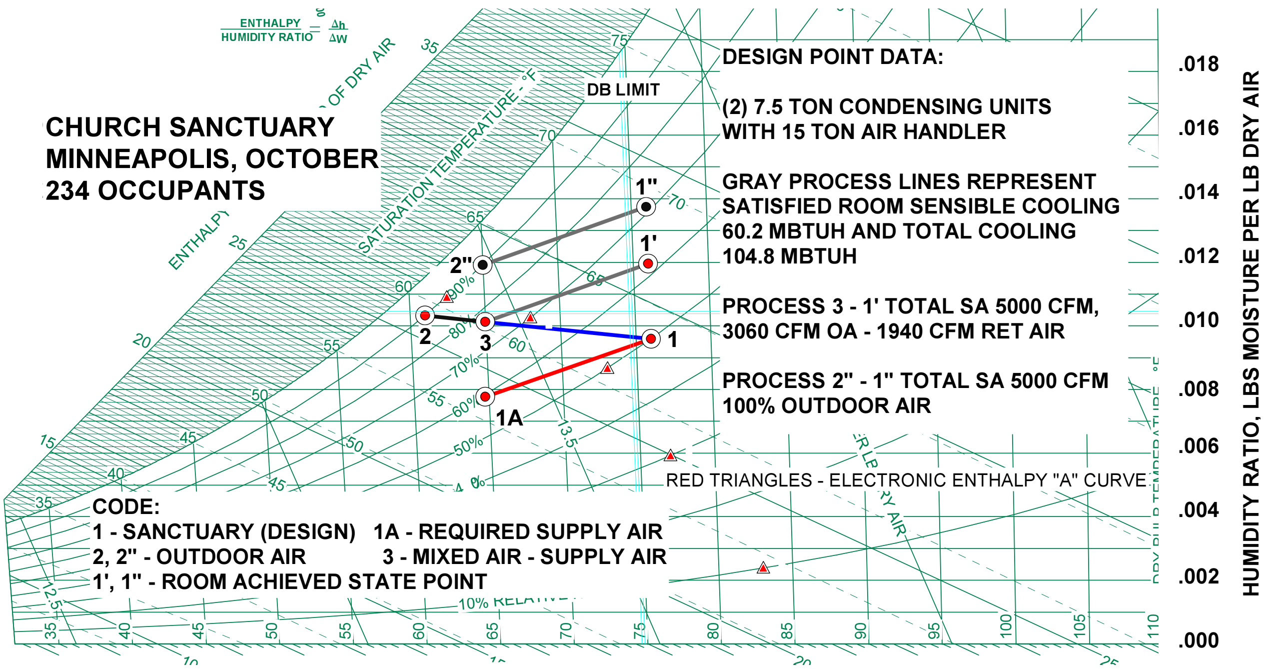

Figure 11, Economizer Operation – Assembly Occupancy

Figure 11 shows an economizer operating at two points. Point **2** is the same as used in the office example, 61˚ with 90% RH. The mixed air blend of outdoor and return air, point **3**, is exactly at the temperature needed to maintain our room set point of 76˚, since our control, which is modulating the economizer, relief, and return dampers , is perfect. The operating RH, shown by point **1'**, is higher than design, but still an acceptable 62%. (See footnote ii)

Point **2"** is at the maximum temperature at which the economizer can operate without mechanical cooling – the same temperature as point **1A**, but at 90% RH. The dew point

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[ii] Point 1' is the lowest humidity level possible in this example. Point 3 is the blended return and outdoor air, but the return used in the example is the design return. The actual return is point 1' which in turn moves point 3 up on the chart. The final achieved room condition will be at a higher humidity than is shown by 1', but cannot be determined without iteration.

temperature at point **2"** is nearly 12˚ higher than at point **1A**, which is needed to maintain the design point of 50% RH. The sanctuary RH will exceed 70%, as shown by the gray process line **2" – 1"**.

The gray lines on Figure 11 illustrate that the removal of heat and moisture from the room by the 5000 cfm supplied by the economizer will follow the slope of the calculated load process, **1A – 1**. This illustrates the pitfall of using dry bulb economizer limit alone when the room sensible heat ratio is steep. In this example, a dew point limit of 58˚ (passing through point **2**) would avoid exceeding 65% RH in the sanctuary, as would using the electronic enthalpy "A" curve (red triangles), with the caution mentioned before.

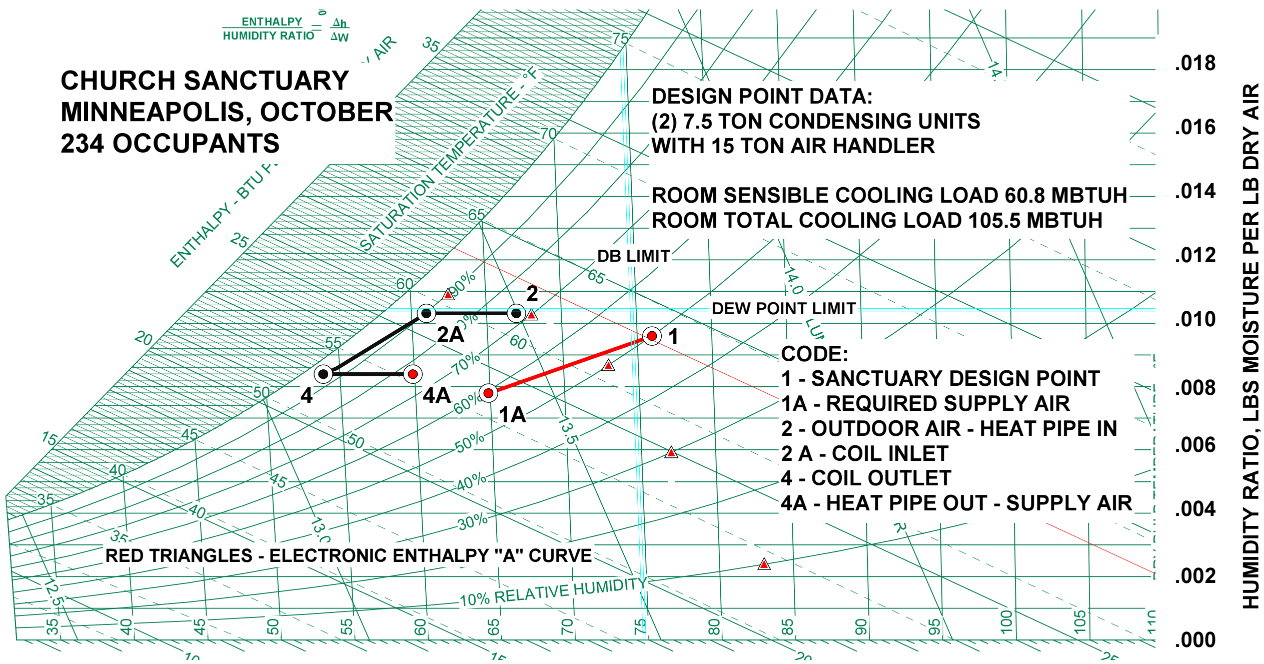
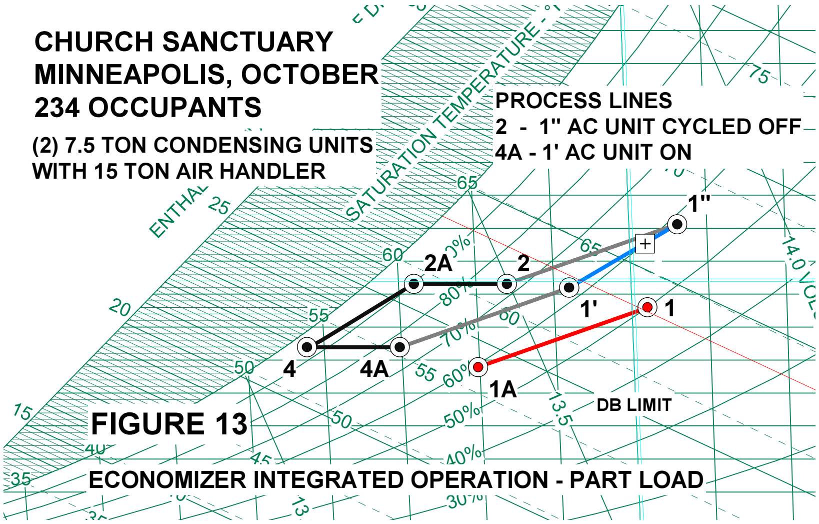


Figure 12, Integrated Operation – Part Load

Figure 12 shows integrated operation with the economizer delivering 100% outdoor air and the AC system operating to control space temperature. Outdoor conditions are 67˚F at the suggested dew point limit of 58˚. The AC system is assumed to be operating at half load on one of the two condensing units and the heat pipe ΔT is assumed to be half of design, or 6˚. Point **4A** is the delivered supply air, based on the assumed heat pipe ΔT and manufacturer's performance software run at the coil inlet condition, point **2A**. The process line **2A – 4** is the coil capacity at half load.[iii]

[iii] To estimate the part load performance of a 5000 cfm face split air handler operating with one of two 7.5 ton condensing units, the software input was the conditions at point 2A, a hypothetical smaller air handler at 2500 cfm air flow, and one 7.5 ton condensing unit. The resulting total and sensible heat capacity was then applied to the chart using the full 5000 cfm supply air.



Depending on the outdoor temperature, the AC unit will cycle at part or full load to maintain the design room set point. Figure 13 shows the room conditions with the system running, **line 4A – 1'**, and cycled off, line **2 – 1"**. It seems likely that the achieved room relative humidity will be about 60% as shown by the X on the line connecting points **1'** and **1"**.

Selecting a dew point limit above 58˚, or using a fixed enthalpy limit, may risk economizer operation with marginal or unacceptable high humidity in the sanctuary.

<u>Conclusion</u>

These examples demonstrate the need for the engineer to examine room or zone loads at the outdoor conditions where economizer operation is permitted, with the internal loads – people, lights, etc – expected under these ambient conditions. The room load process line can then be plotted on the psychrometric chart, as shown in the examples. Supply air conditions under economizer operation will fit one of three cases:

Blended outdoor and return air, as shown on figures 7, 10, and 11.

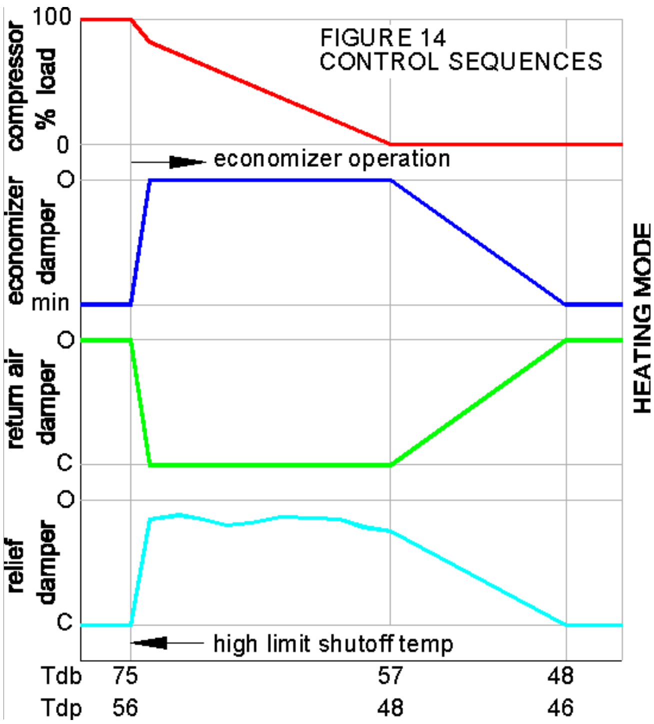
100% outdoor air when the room load equals the capacity of the economizer alone, Figures 7 and 11.

100% outdoor air with ac unit operation (chilled water or DX) to achieve room temperature set point, Figures 8, 12, and 13.

All of these can be plotted on the psychrometric chart, and the final room conditions can be estimated by assuming that the supply air will follow the slope of the room load process line as it picks up heat and moisture.

12. Economizer Controls and Sequences

A Basic Control Sequence



Economizer controls are necessarily complex in comparison to the cooling system zone space temperature control. In its simplest form, the zone space temperature control consists of a single sensor that signals on-off controls, usually relays, to operate the mechanical cooling compressor, the condenser fan, and the air handler blower. The same sensor can also signal heating apparatus, and circuitry in the sensor housing can regulate the operation of heating and cooling as a function of zone space temperature.

The simplest economizer control sequence requires at least four additional sensors: outdoor air temperature, outdoor air relative humidity, supply air flow volume, and zone /outdoor pressure differential. When the outdoor air sensors detect outdoor conditions falling below the high limit shutoff levels, the following occurs, as depicted by Figure 14 for DX cooling:

- The economizer OA damper will immediately ramp to full open, reducing the load on the mechanical cooling compressor which will either unload or cycle to maintain zone space temperature. The economizer damper will remain open as long as mechanical cooling assistance is needed to maintain the space temperature thermostat set point. As outdoor temperature falls, cooler outdoor air blended with return air can continue to maintain set point without assistance from the mechanical cooling system.

- The return air damper will ramp closed to maintain supply air flow, as measured by an air flow rake in the supply or return air duct near the air handler, Alternatively, the return air damper is linked to the OA damper and will close as the OA damper opens to maintain approximately constant supply air flow.

- The relief damper will modulate open to maintain a constant building internal pressure. It is important to maintain the internal pressure higher than outdoor ambient to prevent infiltration. See Chapter 4. On the other hand, too high an internal pressure can cause problems with loads on external doors. Maintaining a pressure differential of approximately .05 inches of water will insure positive internal building pressure while keeping loads on exterior doors under five pounds.

- As outdoor conditions fall further below the shutoff limits, the air conditioning compressor further unloads (or cycles off for longer periods) until the zone space temperature can be maintained with 100% outdoor air alone (57˚ on Figure 14), or outdoor air blended with return air, and the compressor does not run.

- As the outdoor temperature continues to fall (below 57˚ on Figure 14), the zone space temperature will begin to fall below the set point. The economizer controller will then begin to close the economizer outdoor air damper to maintain the zone space temperature set point. The return damper will modulate open to maintain the supply air flow rate, and the relief damper will continue to modulate to maintain a constant building internal pressure.

- When the outdoor air temperature falls to the point that the economizer damper is at minimum ventilation position ,or fully closed if there is a separate ventilation damper (48˚ on Figure 14), zone space temperature begins to fall and continues to fall until it reaches the heating set point and signals the heat to come on.

Understand that the only actual control point on Figure 14 is the high limit shutoff temperature of 75˚. The temperatures shown for other transitions, such as the point where compressor operation is no longer needed, are the result of decreasing sensible and total zone load and will occur automatically with the control system functioning as described.

Economizer controls are complex and subject to instability, and should be furnished and installed only as factory-furnished components of the specified zone cooling system, or field furnished and installed by one of the major control system manufacturers.

In addition to specifying the size and quality of the economizer louvers and dampers, the job of the economizer engineer-of-record is to establish the high limit shutoff parameters, using the techniques outlined in section 11, and to define a sequence of operation compatible with the climate zone and the economizer configuration he has designed. Design variations include return air blowers, relief fans, outdoor air fans, and dedicated minimum outdoor air intakes. Reference 1 provides guidance to help the engineer configure and devise sequences for these variations.

13. Filtration

Economizers introduce a large amount of outside air, laden with very small particulates associated with construction, road and highway traffic, industrial processes, and other outdoor pollutants. The MERV 7 or 8 filters normally specified for commercial and institutional occupancies are intended to remove only the lighter loading of particulates introduced with minimum ventilation air and generated mostly within the space. When an economizer is specified, the engineer should consider increasing the MERV rating to 10 or 11 (equivalent to 85% efficient based on dust spot method) to handle the increased loading of fine respirable particles from outdoors.

14. Afterword

Since the economizer mandate first appeared in ASHRAE Standard 90.1 in 2004, there has been very little guidance for designers in the professional literature. In particular, there is virtually no guidance on selecting the economizer cutoff high limits, other than a few articles in professional and commercial journals that emphasize energy saving and dismiss occupant comfort considerations. This appendix provides the first comprehensive design recommendations that cover damper quality control as well as setting high limits for energy saving without compromising occupant comfort.

The material on louver and damper sizing and quality is applicable to all economizer configurations. The psychrometric examples are based on actual projects with DX constant volume cooling systems. The approach to chilled water systems will be somewhat different, as will the approach to variable volume systems. The designer can use the principles set forth in this course as a basis for design, with appropriate adjustments for these systems. Some discussion of economizers operating with chilled water systems is found in reference 5. Reference 1 includes guidance for damper systems and control sequences applicable to variable volume HVAC systems.

References

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4. Honeywell H705A Solid State Enthalpy Controller. http:customer.honeywell.com/techlit/pdf/PackedLit/63-2144.pdf , page 8, figure

5. Zhou, Wei, Turner, and Claridge, "Air Side Economizer – Different Control Strategies and Common Misconceptions", Proceedings of the Eighth International

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6. Liescheidt, Stephen, "Economizers in Air Handling Systems", Continuing Education and Development, Inc., Course No. M02-014 - http://www.cedengineering.com/upload/Economizers%20In%20Air%20Handling%20Systems.pdf