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Design Guidance for Command, Control, Communications, and Intelligence (C^3I) Facility Cooling Systems

by
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V. Alexander Moore
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Design guidance has been developed for cooling systems in command, control, communications, and intelligence (C^3I) facilities. This report describes design and equipment alternatives available with special attention to operation, principal applications, cost, advantages, and disadvantages. The alternatives are evaluated and ranked according to their ability to fulfill each of the following criteria: flexibility in meeting load redistribution and load expansion, humidity control, particulate matter filtering ability, static electricity control, backup cooling, reliability, lowest cost for shielding against electromagnetic pulse (EMP) and related phenomena, and lowest life-cycle cost.

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Design guidance has been developed for cooling systems in command, control, communications, and intelligence (C^3I) facilities. This report describes design and equipment alternatives available with special attention to operation, principal applications, cost, advantages, and disadvantages. The alternatives are evaluated and ranked according to their ability to fulfill each of the following criteria: flexibility in meeting load redistribution and load expansion, humidity control, particulate matter filtering ability, static electricity control, backup cooling, reliability, lowest cost for shielding against electromagnetic pulse (EMP) and related phenomena, and lowest life-cycle cost.

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FOREWORD

This research was conducted for the U.S. Army Engineering and Housing Support Center (USAEHSC) by the Energy Systems Division (ES) of the U.S. Army Construction Engineering Research Laboratory (USACERL) under Project Order E87870241, "Studies in Optimizing Heating, Ventilating and Air-Conditioning (HVAC) Hardware for C^3I Facilities" dated 8 July 1987. The study was prepared for USACERL by Science Applications International Corp. (SAIC), McLean, VA, under delivery order #0001, contract No. DACA 88-88-D-0026. Dr. Harold Hollis, CEHSC-PREP, was the USAEHSC Technical Monitor. Glen A. Chamberlin was the USACERL Principal Investigator.

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1 INTRODUCTION

1-1. Background

a. Command, control, communications, and intelligence (C^3I) facilities often house computer and communications equipment, radar systems, printers, disk and tape drives, monitors, and system consoles--all of which generate substantial amounts of heat. To optimize performance and ensure continued operation, the environment containing the electronic equipment must be maintained within stringent temperature and humidity levels. In addition, the comfort needs of personnel required to operate the equipment must be met and airborne particulate matter effectively filtered from entering the C^3I room airstream. The electronic equipment and personnel must also be protected from electromagnetic pulse (EMP) phenomena, biological contamination, and radiation. In the event of a power failure, a backup system must be available to provide chilled water and cool air to the water- and air-cooled electronic equipment, respectively, for at least 15 min.

b. With the rapid advancements in state-of-the-art electronics, existing equipment within C^3I facilities is constantly being replaced by new, efficient, and more powerful models. As a result, electronic equipment within these C^3I facilities is continually relocated to enhance the performance of particular C^3I functions. In addition to equipment relocation, the volume of equipment inside a C^3I building often increases over time. The cooling systems used to maintain environmental conditions within individual C^3I rooms must be designed to adapt to these changing circumstances, while still maintaining adequate filtration, EMP protection, and backup.

c. The Department of Defense (DOD) owns and operates many C^3I facilities across the nation and abroad, ranging from small computer rooms to large radar facilities. The diversity of C^3I applications and specialized requirements, coupled with the need for flexibility, has given rise to a large number of different design solutions. For this reason, DOD needs simplified design guidance that addresses the criteria most critical to decisions supporting the military mission.

1-2. Objective

The objective of this study is to provide simple, easy-to-use design alternatives for C^3I cooling systems that meet the specific technical criteria of greatest consequence in design decisions. Part of this objective is to present advantages and disadvantages for each alternative.

1-3. Approach

a. Information for this study was collected from the product literature, manufacturers, and field operating agencies. (Published sources are listed in the **Bibliography**.) Subsystem alternatives were evaluated on the basis of:

(1) Operation: physical characteristics and mode of operation for each subsystem alternative. Included is information on common components, control strategies, and performance.

(2) Principal applications: those for which each subsystem alternative is best suited.

(3) Size: the available ranges.

(4) Cost: normalized on the basis of either "per square foot"* of C^3I room floor area, "per cubic foot per minute (CFM)" of air flow, or "per ton" cooling capacity, depending on the subsystem.

(5) Advantages/disadvantages: described for each subsystem alternative in various applications.

b. The subsystems were then evaluated according to applicable criteria (listed in **Section I** of each chapter. Evaluation tables were developed to summarize the results of the qualitative ranking procedure used. These results were analyzed to explain why certain subsystem alternatives ranked higher or lower for particular criteria.

c. When ranking each subsystem alternative, certain assumptions were made regarding the system configuration. These assumptions were based on the most common system configuration in which each subsystem alternative is used. The summary evaluation in each chapter lists these assumptions; it is important to read these sections before viewing the evaluation tables in order to understand the results in proper context.

1-4. Scope

a. This report is intended to be used by designers as an aid in evaluating the relative merits of alternative cooling subsystems with respect to specific criteria or design considerations. It is not intended to be a C^3I cooling system design manual or handbook. It is the individual engineer's responsibility to design a system that performs the required air-conditioning functions effectively in compliance with DOD and other applicable codes.

b. The equipment and installation cost estimates given in this report are for comparison only and should be used with caution. The estimates do not replace the need to conduct detailed cost estimates that consider locally available resources.

c. The guidelines present information on four main subsystems that comprise all cooling systems: (1) supply air distribution systems, (2) chilled liquid distribution

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*Metric conversion factors are listed on p 111.

systems, (3) air-handling units, and (4) liquid chilling systems. The rationale for evaluating subsystems and the specific criteria addressed are described below.

1-5. Ranking Criteria

This study evaluated subsystems rather than complete systems to provide the designer with maximum flexibility in combining subsystem options. The various subsystems are evaluated against a standard set of criteria, not all of which apply in every case. The specific criteria addressed for each subsystem are presented in **Section I** of each chapter. The interpretation of the evaluation criteria varies by subsystem. The operation and function of different cooling subsystems dictates how they are evaluated under the various criteria. The standard set of criteria is as follows:

a. *Flexibility in Meeting Load Redistribution.* This criterion is used to judge the cooling subsystem's ability to accommodate the relocation of electronic equipment (and therefore the location of concentrated cooling loads) within the C^3I room.

b. *Flexibility in Meeting Load Expansion.* This criterion addresses the cooling subsystem's ability to handle the increased cooling load resulting from the installation of additional electronic equipment.

c. *Precise Humidity Control.* This criterion addresses the subsystem's ability to control the humidity level in the C^3I room within prescribed limits.

d. *Particulate Matter Filtering.* This criterion involves the subsystem's ability to prevent the introduction of particulate matter into the C^3I room air stream.

e. *Static Electricity Control.* This criterion addresses the subsystem's ability to prevent the buildup of static electricity within the C^3I room, avoiding possible damage to the electronic equipment. Static electricity control often is a function of the humidity level in the C^3I room.

f. *Backup Cooling.* This criterion is used to assess the need for air- and water-cooled electronic equipment to have an uninterrupted supply of chilled water. In the event of a power failure, the backup system must supply chilled water to the equipment for at least 15 min, in which time the emergency generators can be started and placed in service.

g. *Reliability.* This criterion addresses the subsystem's ability to provide reliable, low-maintenance operation.

h. *Least Cost for EMP Shielding.* This criterion is considered in judging the relative cost of protecting EMP shielding when various subsystem alternatives are used.

i. *Lowest Life-Cycle Cost.* This criterion addresses the total life-cycle cost of the subsystems, including first cost, energy operating cost, maintenance cost, replacement cost, and any salvage value benefits. It indicates the true owning and operating costs of each subsystem alternative.

(1) Where indicated, the life-cycle cost analyses were conducted in accordance with National Bureau of Standards Handbook 135.[1] Total life-cycle cost was selected as the economic measure over other modes of analysis.

(2) National average base-year energy prices and uniform present worth (UPW) factors, adjusted for energy price escalation, were used in the analyses. The UPW factors for determining the present value of future annually recurring (nonfuel) maintenance costs were based on a 7 percent discount rate.

j. Other Evaluation Areas. Where applicable, other considerations such as nuclear, biological, and chemical (NBC) contamination noise, energy conservation, and space utilization efficiency were addressed in the evaluations.

[1]National Bureau of Standards Handbook 135, *Life-Cycle Cost Manual for the Federal Energy Management Program* (December 1980).

2 AIR DISTRIBUTION SYSTEMS

Section I: GENERAL

This chapter evaluates design approaches for supply and return air distribution systems serving electronic equipment rooms. The following criteria are used to evaluate each system component:

1. Flexibility in meeting load redistribution.
2. Flexibility in meeting load expansion.
3. Precise humidity control.
4. Filtering of particulate matter.
5. Static electricity control.
6. Capacity to provide 15 min of backup cooling.
7. Reliability.
8. Least cost for EMP shielding.
9. Lowest life-cycle cost.

Some criteria do not apply to all components of air distribution systems. In these cases, "N/A" appears in the table displaying the results of the ranking procedure. Components of air distribution systems for electronic equipment rooms fall into two main categories: (1) plenums and ducts, and (2) air-diffusing equipment. Supply air plenum and duct configurations typically used in electronic equipment rooms are described first, in addition to common variations (e.g., variable-air-volume [VAV] boxes and constant-volume [CV] mixing boxes). The common methods of handling return air are then addressed. Finally, air-diffusing equipment used to supply conditioned air to the space, and return air to air-handling units, are described. The following air distribution system components are evaluated in this report:

Plenums and Ducts	Diffusers, Grilles, and Registers
Underfloor Plenum Supply	Perforated Floor Panels
Ceiling Plenum Supply	Perforated Ceiling Diffusers
Overhead Ducted Supply	Grilles and Registers
- VAV Boxes	Louvered Ceiling Diffusers
- CV Mixing Boxes	Linear Diffusers
Free Return	Light Troffers
Ceiling Plenum Return	Return Air Inlet Coverings
Ducted Return	

Section II: SYSTEM DESCRIPTION

Electronic equipment rooms have unique air distribution requirements. The system must deliver air in a way that effectively dissipates the concentrated cooling loads of electronic equipment, while providing comfort to occupants working in the area. The system must satisfy minimum ventilation for the introduction of outside air. Also, the air distribution system must maintain positive pressure to prevent the introduction of air into the C^3I room from adjacent areas. Frequently, C^3I rooms undergo constant change due to the relocation, addition, and updating of electronic equipment. The air distribution system should be able to adapt to the changing environment.

2-1. Plenums and Ducts

As mentioned in Section I above, air distribution systems have two major components. This paragraph describes plenums and ducts, whereas paragraph 2-2 describes air-diffusing equipment. The function of plenums and ducts is to transport supply air from the air handler to the air-diffusing equipment and to return air from the return inlets back to the air handler. The supply and return air circuits can be configured in numerous ways. The methods most often used in C^3I rooms are as described below.

a. Underfloor Plenum Supply.

(1) Operation.

(a) Connecting electronic equipment to system consoles, printers, disk drives, monitors, and other peripheral equipment requires large amounts of electric cable. For this reason, electronic equipment is usually placed on a false, or raised, floor under which the interconnecting cables are concealed. Frequently, the cavity between the building floor and the raised floor is also used as a supply air plenum (Figure 2-1).

(b) The standard raised floor system consists of an understructure and floor panels. The understructure uses pedestals to support the corners of the floor panels. The understructure for most electronic equipment rooms also has a stringer system in which the pedestals are connected by structural members. A stringer understructure provides lateral stability and is available as snap-on or bolted construction.

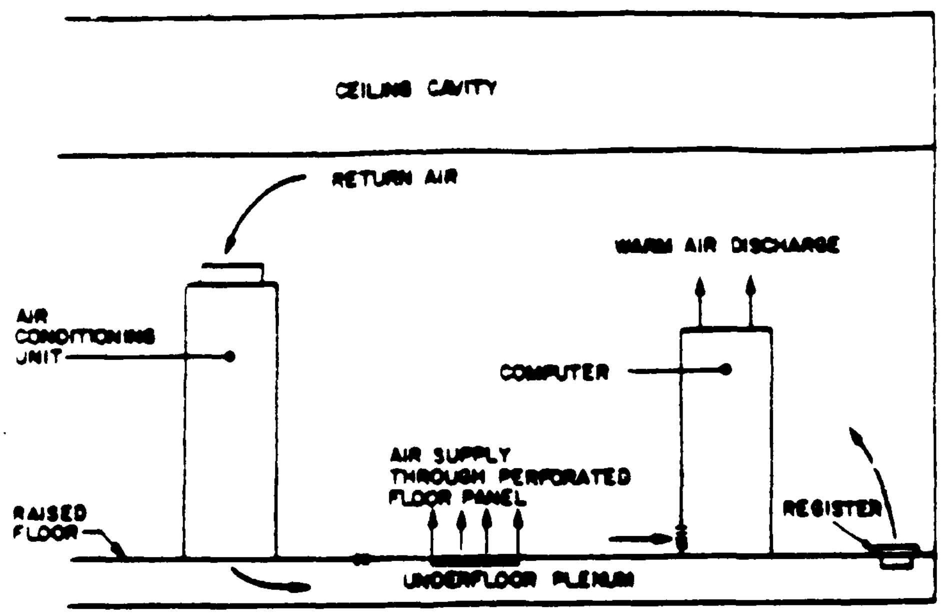

Figure 2-1. Underfloor plenum supply. (Source: *American Society of Heating, Refrigerating, and Air-Conditioning Engineers [ASHRAE] Handbook* [1987], Chapter 33. Used with permission.)

(c) To ensure adequate supply air to the electronic equipment room, a clearance of 12 in. between the building floor and the raised floor is desired, with 10 in. being the minimum accepted clearance. In electronic equipment rooms that require extensive cabling or large volumes of supply air, additional clearance may be needed.

(d) When the air-handling units serving the underfloor plenum are located within the C^3I room, the supply connection to the plenum should be as close as possible to the center of the area served. Dividing baffles for zoning in the underfloor plenum may be unnecessary with this configuration, since air flowing through the path of least resistance results in a self-zoned system. Air turbulence at the supply connection to the underfloor plenum should be minimized by the use of turning vanes.

(e) If supply air temperatures are low, insulation or vapor barriers may be required on the plenum surfaces. Underfloor plenum surfaces should have a smooth finish and be completely clean and airtight to prevent the introduction of particulate matter into the air stream.

(2) Principal Applications. The underfloor plenum supply system can be used with remote air handlers or, more commonly, with air handlers located within the C^3I room. Air handlers located within the C^3I room may be the chilled-water type served by a liquid chilling system or may be self-contained, packaged, direct-expansion (DX) units. According to several raised floor system manufacturers, 95 percent of all computer rooms use underfloor plenum supply systems.

(3) Sizes. Both the standard and perforated floor panels measure 24 by 24 in. A typical perforated floor panel has 25 percent free area (air passage area divided by total panel area) and can handle 550 to 1400 CFM, depending on air velocity, static pressure, and the presence of a volume control damper.

(4) Cost. A typical budget figure for a raised floor system not used as a plenum is $8/sq ft. This cost includes the raised floor understructure, standard floor panels, and installation (including contractor's overhead and profit). A perforated floor panel averages $20 more in cost than a standard floor panel. In most applications, an underfloor plenum supply system has a premium cost of 1 to 3 percent ($0.08 to $0.24/sq ft) more than a raised floor not used as a plenum. Thus, if the C^3I room has or is designed to have a raised floor, the additional cost of using it as an underfloor plenum is between $0.08 and $0.24/sq ft. If the C^3I room does not already have a raised floor, the underfloor plenum costs between $8.08 and $8.24/sq ft. It should be noted that these costs do not include EMP protection. The principles and costs of EMP protection for air distribution systems are addressed in paragraph 2-3 below.

(5) Advantages. Since most electronic equipment rooms use a raised floor system, the underfloor plenum supply system can be incorporated at a low additional cost. Perforated floor panels are usually used with the underfloor plenum, providing good flexibility in accommodating the constant relocation of equipment experienced in today's C^3I rooms. By simply exchanging standard floor panels with perforated floor panels, the air distribution pattern can be modified to satisfy the changing locations of concentrated cooling loads.

(b) If the air handlers serving the underfloor plenum are located within the C^3I room, the air distribution system can handle load expansion by accommodating additional air handlers. With this configuration, air handlers can be added without interrupting the operation of existing air handlers. The use of multiple air handlers provides the redundancy required by C^3I rooms. In critical C^3I facility applications, a rule of thumb for redundancy often cited is that sometimes referred to as "N+2." This convention states that, for mission-critical equipment, two additional units should be provided. If one unit fails while another is down for maintenance, the third unit can satisfy mission requirements. If the mission is less critical, one extra unit (100 percent redundancy) may be adequate. Three factors dictate the level of redundancy required: (1) critical nature of the mission, (2) equipment reliability, and (3) equipment cost. It is the design engineer's responsibility to assess these three factors and determine the level of redundancy required for a specific facility.

(c) Controls in the air handler are primarily responsible for regulating the C^3I room air temperature and humidity. However, the air distribution system affects the variation of air temperature and humidity within the space. The perforated floor panels usually used with the underfloor plenum supply system have a higher entrainment ratio (total room air divided by discharge air) near the point of discharge than many types of diffusers, grilles, and registers. The higher the entrainment ratio, the faster differences in temperature and humidity between room air and discharge air are equalized. This means that perforated floor panels can be located closer to electronic equipment with less fear of condensation inside the equipment than with many other types of air-diffusing equipment.

(d) The raised floor systems that comprise an underfloor plenum are available with antistatic and conductive coatings to control the discharge of static electricity to electronic equipment. In addition, the stringer system used with raised floors in most C^3I rooms provides an effective path for the dissipation of static electricity.

(7) Disadvantages.

(a) Underfloor plenums must have adequate clearance to deliver the required volume of air to the C^3I room. If the cabling between electronic equipment is extensive, raised floor heights of up to 36 in. may be needed to provide the minimum plenum clearance. If the floor-to-ceiling height in the room is limited, the use of an underfloor plenum may be precluded. While locating the air handlers within the space provides flexibility in meeting load expansion, the air handlers occupy space that is often quite valuable in the C^3I room. If the underfloor plenum is served by a remotely located air handler, load expansion must be accommodated by more traditional methods. The volume of air discharged by the air handler must be increased by increasing fan speed. If a substantial cooling load is added, the cooling coil of the air handler may have to be replaced with one of increased capacity. Both of these methods require that the air handler be shut down for modification.

(b) If properly constructed, underfloor plenums should allow minimal entrainment of particulates into the discharge air stream. However, underfloor plenums are more likely to accumulate dust and dirt than ceiling plenums and ducted supply systems.

(c) The panels in raised floor systems are designed to withstand rolling loads of up to 2000 lb and dynamic impact loads of up to 200 lb. The panels are constructed to support concentrated static loads of up to 2000 lb with a 0.08-in. top deflection; they have ultimate load ratings of up to 6000 lb. Despite the structural integrity shown by these static and dynamic load ratings, as well as the durable floor coverings available--both of which suggest that raised floor systems generally have good reliability--wear and tear are more of a concern for underfloor plenum supply systems than for ceiling plenum or ducted supply systems.

b. Ceiling Plenum Supply.

(1) Operation.

(a) Similar to the underfloor plenum supply system, this system uses the cavity between the building roof and the ceiling of the electronic equipment room as a supply air plenum. Supply air from the ceiling plenum flows through perforated ceiling panels to the C^3I room. Ceiling plenum supply systems are not considered to be quite as flexible as underfloor systems in handling load expansion since they are typically served by remotely located air handlers, which require upgrading to increase cooling capacity.

(b) Distribution ductwork is sometimes used with a ceiling plenum to equalize the air distribution throughout the plenum (Figure 2-2).

(2) Principal Applications. Ceiling plenums are typically used with remotely located air handlers. If the return air also passes through the plenum, return air ductwork must be installed, as shown in Figure 2-2.

(3) Sizes. Ceiling plenums can be designed to handle a wide range of air volume flow rates. The combination of the plenum size, the size of the duct serving the plenum, and the CFM rating of the air handler dictates the maximum flow rate possible. The maximum acceptable noise rating of the air-diffusing equipment also impacts allowable air flow rates.

(4) Cost. A typical budget figure for a suspended ceiling is approximately $3/sq ft. This cost includes acoustic ceiling panels, T-bar carrier channel suspension system, and installation (including contractor's overhead and profit). The additional cost of installing perforated ceiling diffusers increases the budget figure by about 2 percent ($0.06/sq ft). If distribution ductwork is used, the installed cost approaches that of an overhead ducted system. Thus, if the C^3I room already has a suspended ceiling, the additional cost of using it as a ceiling plenum is about $0.06/sq ft. If the room does not have a suspended ceiling, the cost of a ceiling plenum is about $3.06/sq ft. It should be noted that these costs do not include EMP protection. The principles and costs of EMP protection for air distribution systems are addressed in paragraph 2-3 below.

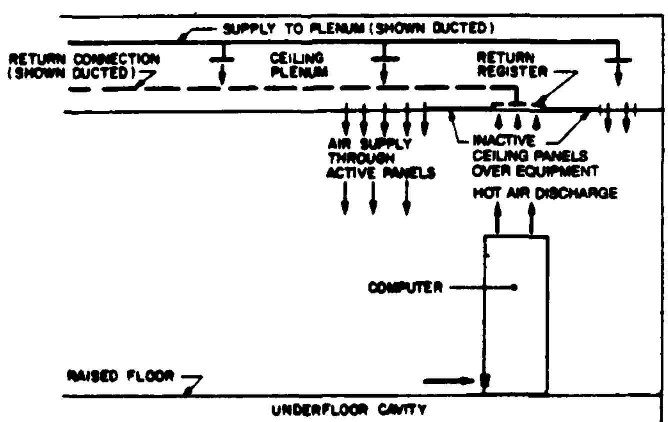

Figure 2-2. Ceiling plenum supply. (Source: *ASHRAE Handbook* [1987], Chapter 33. Used with permission.)

(5) Advantages. Ceiling plenum supply systems have many of the same attributes of the underfloor plenum supply systems. Load redistribution within the C^3I room can be handled easily by relocating the perforated ceiling diffusers. Similar to the underfloor plenum, the ceiling plenum must have proper clearance to provide adequate air flow to the C^3I room. However, with the ceiling plenum, no allowance for cabling is necessary. The ceiling plenum is also more likely to stay clean than an underfloor plenum. In addition, the installed cost of a ceiling plenum supply system is less than that of an underfloor plenum supply system.

(6) Disadvantages. Since ceiling plenums are typically served by remotely located air handlers, load expansion must be accommodated by upgrading the air handler as described above. In most cases, this work would require an interruption of air flow to the plenum. Also, in some cases, supply air must flow up through the electronic equipment. This requirement is better satisfied with an underfloor plenum supply system.

c. *Overhead Ducted Supply.*

(1) Operation. In limited applications, electronic equipment rooms may be supplied by overhead ducted distribution systems like those used in most heating, ventilation, and air-conditioning (HVAC) systems. Overhead ducted supply systems have little flexibility for handling load redistribution or expansion due to the difficulty in relocating air outlets in C^3I rooms with continuous duty requirements. Flexible ducting can be used to increase the ability to handle load redistribution. However, the length of flexible duct runs is normally limited to 6 ft or less due to the high pressure loss associated with flexible ducting that incorporates steel springs. When cooling loads are high, overhead diffusers may cause drafts.

(2) Principal Applications. Overhead ducted supply systems are restricted to applications where large volumes of supply air are not needed and relocation of electronic equipment is not anticipated. Typically, the overhead ducted supply system is served by a remotely located air handler that introduces fresh air for occupants in the space and for positive pressurization.

(3) Sizes. Overhead ducted supply systems can be designed to handle a wide range of air volume flow rates experienced in C^3I rooms.

(4) Cost. For galvanized steel ductwork, the material cost ranges from $0.70 to $2.50/sq ft. Labor, overhead, and profit bring the installed cost to a range of $1.45 to $4.00/sq ft. The use of flexible ducting reduces these costs some 20 percent. Again, these costs do not include EMP protection. The principles and costs of EMP protection for air distribution systems are addressed in paragraph 2-3.

(5) Advantages. An overhead ducted supply system will stay cleaner than either the underfloor or ceiling plenum. Ducted supply systems have a proven track record of reliability.

(6) Disadvantages. Overhead ducted supply systems are restricted to applications not requiring large volumes of supply air or flexibility in meeting load redistribution or expansion. The installed cost of an overhead ducted supply system is more than either the underfloor or ceiling plenum system, assuming that the C^3I room already has a raised floor or suspended ceiling.

d. Constant Volume (CV) Mixing Boxes.

(1) Operation.

(a) Terminal boxes are used in ducted systems to regulate air from the air-handling unit before it reaches the air-diffusing equipment. There are several different kinds of terminal boxes that control various characteristics of the supply air stream. This discussion addresses a commonly used terminal box called the "CV mixing box" (Figure 2-3).

(b) The CV mixing box maintains a constant-volume flow rate of air to the air-diffusing equipment and varies the air temperature to satisfy changing cooling or heating loads. The CV box is typically connected to both a hot air duct and a cold air duct. By regulating the mixture of hot and cold air, the air temperature to the air-diffusing equipment can be controlled effectively. Each CV mixing box is usually controlled by a separate thermostat located in the space it serves.

(c) Another type of CV mixing box is connected only to a cold air duct. Return air provides the warm air for temperature blending and is drawn into the box by a separate, integral fan.

(2) Principal Applications. CV mixing boxes are used exclusively with ducted supply systems. They are normally used to condition a space where cooling and heating loads vary substantially throughout the day or from zone to zone and where a constant flow of air is required by equipment or occupants.

(3) Sizes. CV mixing boxes typically range in size from 100 to 3000 CFM.

(4) Cost. The installed cost of CV mixing boxes ranges from $0.30 to $2.00/CFM. The cost of the boxes must be added to that of an overhead ducted system to arrive at a realistic system cost. If both hot and cold ducts serve the boxes, the cost of the ducted system increases dramatically.

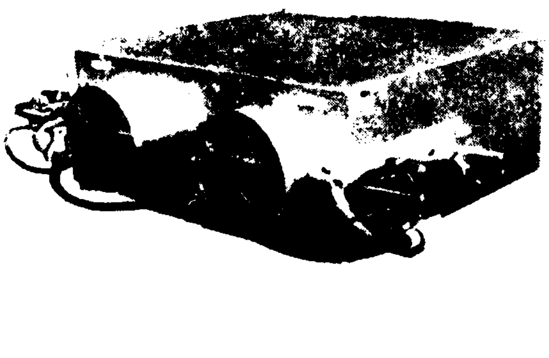

Figure 2-3. Constant-volume mixing box.

(5) Advantages. CV mixing boxes are appropriate when the cooling and heating loads vary significantly throughout the day or from zone to zone and when a constant flow of air is required by equipment or occupants. CV mixing boxes increase the ability of a ducted supply system to accommodate the relocation and expansion of load within the C^3I room.

(6) Disadvantages. In addition to the high cost of materials and installation, a supply system that uses CV mixing boxes is inherently wasteful from an energy standpoint. The energy used to heat and cool air is wasted at the box when the two air streams are mixed. Furthermore, C^3I rooms typically have a constant cooling load, negating the benefit of the CV box in handling varied loads.

e. Variable-Air Volume Boxes.

(1) Operation. Another variation of the standard overhead ducted supply system uses terminal boxes similar in appearance to CV mixing boxes. Unlike the CV mixing box, the VAV box regulates the *volume* of air supplied to the air-diffusing equipment rather than the temperature. Each VAV box reads the space temperature sensed by a dedicated thermostat and adjusts the volume flow rate of constant-temperature air based on the need for cooling. When the cooling load is high, the VAV box delivers the maximum CFM of cool air to the air-diffusing equipment. When the cooling load subsides, the volume of air is reduced.

(2) Principal Applications. VAV boxes are used exclusively with ducted systems. Like the CV mixing boxes, they are used to condition spaces where the cooling or heating load varies throughout the day or from zone to zone.

(3) Sizes. VAV boxes are available with air flows ranging from 100 to 3000 CFM.

(4) Cost. VAV boxes' installed cost ranges from $0.20 to $1.75/CFM. The cost of the boxes must be added to that of an overhead ducted system to obtain a realistic estimate of system cost.

(5) Advantages. VAV mixing boxes are appropriate when the cooling or heating loads vary significantly throughout the day or from zone to zone. A VAV system increases the ability of a ducted supply system to accommodate the relocation and expansion of load within the C^3I room. A VAV system matches the fan load to the cooling load. In this way, energy consumption for air distribution is optimized.

(6) Disadvantages. Although VAV boxes increase the flexibility of a ducted supply system to handle the relocation or addition of electronic equipment in a C^3I room, the added cost to the standard ducted system is considerable.

f. Free Return.

(1) Operation. When the air handler is located within the electronic equipment room, return air can be drawn directly back into the air handler, eliminating the need for return air plenums or ducts. Such a configuration is called a "free return system" and is shown in Figure 2-1. With a free return system, the air handler should be located near the concentrated cooling loads. Outside air is not introduced and mixed with return air in a free return configuration. It must be introduced into the C^3I room by a separate system (such as a central station air handler) to satisfy minimum ventilation requirements and maintain positive pressure within the electronic equipment room.

(2) Principal Applications. A free return system can be used only in applications where the air handler is located within the C^3I room, such as with an underfloor plenum supply system, since return air is drawn directly from the room into the suction side of the air handler.

(3) Sizes. The air volume flow rates achievable with a free return system are solely dependent on the supply CFM of the air handlers and the static and velocity pressures of the air-diffusing equipment.

(4) Cost. Since there are no plenums, ducts, grilles, or registers associated with a free return system, direct material and labor costs are minimal. However, the cost of the C^3I room floor space occupied by the air handlers must be taken into account. Also, if not already present, the cost of a separate system to introduce and filter outside air for minimum ventilation and positive pressurization must be considered in estimating the cost of a free return system.

(5) Advantages. The free return system is the most flexible and least expensive of the return systems evaluated. Load redistribution and expansion are automatically accommodated by the number and location of air handlers within the C^3I room.

(6) Disadvantages. The free return system does carry an indirect cost in that the air handlers must be located within the C^3I room, where space is often limited. Special provisions are necessary for introducing outside air to the C^3I room.

g. Ceiling Plenum Return.

(1) Operation. Similar to the ceiling plenum supply system, a ceiling plenum return system uses the cavity between the building roof and the ceiling of the C^3I room as a return air plenum. With this arrangement, heat from the electronic equipment, as well as some of the heat from the lights and people, is drawn into the ceiling plenum. Frequently, air is returned to air handlers located within the C^3I room through duct collars.

(2) Principal Applications. The ceiling plenum return system can be used with remotely located air handlers or with air handlers located within the C^3I room. With the latter configuration, return air often flows from the ceiling plenum through a duct collar to the air handler, where it is conditioned and discharged to an underfloor plenum supply.

(3) Sizes. The air volume flow rates achievable with a ceiling plenum return system depend on the size and quantity of return air inlets and on the supply CFM of the air handler.

(4) Cost. If the air handlers are located within the C^3I room, a ceiling plenum return system has an incremental material cost of about $0.02/sq ft more than that of a standard suspended ceiling. The installed cost is approximately $0.03/sq ft. If ductwork is required to return air to a remotely located air handler, the installed cost is about 60 percent more than that for a standard suspended ceiling.

(5) Advantages.

(a) Ceiling plenum return systems provide good flexibility in accommodating the relocation of electronic equipment within the C^3I room. Perforated return panels can be interchanged with standard ceiling panels to easily adapt the return air distribution system to changing equipment locations.

(b) Ceiling plenum return systems are frequently used in conjunction with the underfloor plenum supply system served by air handlers located within the C^3I room. Air flows from the ceiling plenum, through duct collars, and back to the air handler, where it is conditioned and fed to the underfloor plenum. With this configuration, load expansion is handled in the same way as with the underfloor plenum supply: by installing additional air handlers.

(6) Disadvantages. While locating the air handler within the C^3I room offers flexibility for the ceiling plenum return system, the air handlers occupy space that is often limited. If the air handler is located outside the electronic equipment room, load expansion becomes a concern. Also, the plenum may not be airtight and may draw unwanted air from adjacent areas.

h. Ducted Return.

(1) Operation. In rooms where a ceiling plenum or free return system is not practical, a ducted return system can be used. Return air is drawn from the C^3I room, through the return inlets, and transported back to the air handler through ductwork.

(2) Principal Applications. The ducted return system is usually used when the air handler is remotely located from the C^3I room; however, ceiling-hung fan coil units located inside the C^3I room sometimes use a ducted supply and return. A ducted return system can be used with the ceiling plenum or overhead ducted supply systems.

(3) Sizes. Ducted return systems can be designed to handle a wide range of air volume flow rates experienced in C^3I rooms.

(4) Cost. The installed cost of the ducted return system is relatively high compared with the ceiling plenum and free return systems. Even with the lower cost of return grilles versus supply diffusers, the cost of a ducted return system approaches that of an overhead ducted supply system.

(5) Advantages. In cases for which a ceiling plenum return system is not airtight and draws unwanted air from adjacent areas, a ducted return system can be used with good results. A ducted return system is also well suited for applications that require 100 percent outside air. When used for this purpose, the ducted return becomes an exhaust system.

(6) Disadvantages. Return inlets should be placed near areas of concentrated heat gain. When electronic equipment is redistributed within the C^3I room, the return inlets should be relocated. With a ducted return system, relocating the return inlets requires the demolition of existing ductwork and the installation of new ductwork. This requirement makes the ducted return system inflexible in meeting the needs of load redistribution. In addition, the air handler must sometimes be shut down during duct modifications; however, careful design and placement of dampers may allow isolation of the duct branches to be modified.

2-2. Diffusers, Grilles, and Registers

a. General.

(1) Supply air is usually discharged at velocities much greater than that acceptable to occupants in the space. Also, in a C^3I room, supply air discharge temperatures

are normally below that of the room air temperature. Thus, a well designed air distribution system must entrain room air into the discharge air stream outside the area of occupancy and C^3I equipment air intakes to reduce air velocity and temperature differences to levels acceptable to both occupants and equipment. Failure of the distribution system to accomplish this condition can result in occupant discomfort and possible equipment damage (if air supplied to equipment inlets is below the dew point).

(2) Air-diffusing equipment is normally divided into three main categories: diffusers, grilles, and registers. A diffuser is an outlet that discharges supply air in various directions and planes. A grille is simply a covering for any opening through which air flows. A register is a grille equipped with a volume control damper. Outlets have been classified according to their location and air direction/plane geometry. Air-diffusing equipment is also categorized by air direction/plane geometry, in addition to physical characteristics.

(3) The location and selection of supply outlets are dictated by the location of heat-producing electronic equipment. The localized heat gain associated with C^3I equipment can be handled by directing cool air toward the equipment or by placing return grilles near the equipment. The latter is more efficient since heat is expelled at the source instead of being dissipated into the conditioned space. The return air should be taken from as high a point as possible to take advantage of thermal stratification.

(4) Depending on the size of the air-diffusing equipment and the air velocity in the plenum or duct, each type of diffuser, grille, and register has a different noise criteria level. Recommended noise criteria levels are generally stated according to the function of the space. A category is not normally listed for C^3I rooms, but the criteria listed for tabulation and computation areas of general office buildings approximate the conditions found in most C^3I rooms. The recommended maximum noise levels for these spaces range from 40 to 60 dB.

(5) Most air-diffusing equipment is designed to provide occupant comfort in a normal office environment. Because of the unique conditions in a C^3I room, a compromise must be made between occupant comfort and the cooling requirements of electronic equipment. For this reason, not all air-diffusing equipment is suitable for use in C^3I rooms.

b. Perforated Floor Panels.

(1) Operation.

(a) A common diffuser used in C^3I rooms with raised floors is the perforated floor panel shown in Figure 2-4. As its name implies, this diffuser is a raised floor panel into which many holes have been drilled to serve as the air passages. The perforated floor panels are interchangeable with the standard panels in a typical raised floor. This feature allows easy modification of supply air distribution to accommodate the relocation or addition of electronic equipment. Because they mount flush with the raised floor, perforated floor panels are ideal for use in normal traffic areas.

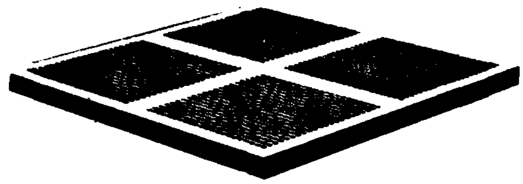

Figure 2-4. Perforated floor panel.

(b) Though the free area and performance vary, perforated floor panels generally have a high entrainment ratio (total air divided by discharge air) near the point of discharge compared with standard grilles and registers. Because of the relatively high mixture of room air with discharge air, perforated floor panels can be placed closer to electronic equipment than registers or grilles with less fear of delivering air that is below the dew point.

(2) Principal Applications. Perforated floor panels are used exclusively with raised floors. They are the air-diffusing equipment most often selected for underfloor plenum supply systems.

(3) Sizes. Perforated floor panels are available with free area ratios (air passage area over total panel area) of 25 to 50 percent. The panels are designed to handle air flows in the range of 550 to 1400 CFM.

(4) Cost. Depending on whether particle board or metal is specified and the quantity ordered, perforated floor panels cost approximately $0.04/CFM. The installed cost runs about $0.05/CFM.

(5) Advantages. As mentioned in the evaluation of underfloor plenum supply systems, perforated floor panels provide good flexibility in accommodating load redistribution and expansion. Because they are interchangeable with standard raised floor panels, perforated panels can be added or relocated easily. Their high entrainment ratio quickly equalizes differences in temperature and humidity between room air and discharge air.

(b) As with standard raised floor panels, perforated floor panels are available with antistatic and conductive coverings. The installed cost per CFM of perforated floor panels is among the lowest of the air-diffusing equipment evaluated.

(6) Disadvantages. As mentioned in describing the underfloor plenum supply system, perforated floor panels typically discharge air supplied by air handlers located within the C^3I room. However, floor space in C^3I rooms is often limited. Also, the air flow of the perforated floor panel could be obstructed by improper placement of furniture or equipment.

c. *Perforated Ceiling Diffusers.*

(1) Operation. One of the most commonly used ceiling diffusers is the perforated diffuser. These diffusers are available with fixed or adjustable air patterns (Figure 2-5). The perforated faces are easily removed to provide access to the pattern deflectors and dampers. Some perforated ceiling diffusers consist of only the perforated

face and generally are used in ceiling plenum supply systems. Most perforated ceiling diffusers have a noise level of less than 20 dB at plenum/duct air velocities up to 700 feet per minute (FPM). At 1000 FPM, the noise level can reach 60 dB, less 10 dB for room attenuation.

(2) Principal Applications. Perforated ceiling diffusers can be used with overhead ducted and ceiling plenum supply systems. For occupant comfort, they are usually used with ducted systems and installed with adjustable deflectors. When installed in ceiling plenums, they are typically used with just the perforated face.

(3) Sizes. Perforated diffusers are available with faces ranging from 12 by 12 in. to 24 by 24 in. Round neck sizes range from 5 to 16 in. Square neck sizes range from 6 by 6 in. to 18 by 18 in. Air volume rates range from 27 to 2250 CFM, depending on neck size and air velocity.

(4) Cost. Perforated ceiling diffusers range in cost from \$0.05 to \$0.33/CFM, with the installed cost ranging from \$0.07 to \$0.47/CFM. Without deflectors, the material cost ranges from \$0.007 to \$0.03/CFM and the installed cost ranges from \$0.008 to \$0.04/CFM.

(5) Advantages. When used with a ceiling plenum supply system, the perforated ceiling diffuser provides good flexibility in meeting the needs of load relocation and expansion. The entrainment ratio is relatively high, approaching that of a perforated floor panel. The installed cost per CFM (without deflectors) is the lowest of the air-diffusing equipment evaluated.

(6) Disadvantages. Some electronic equipment requires that conditioned air flow up through the equipment from the bottom. This requirement is better satisfied with an underfloor plenum supply system and associated air-diffusing equipment.

d. Grilles and Registers.

(1) Operation.

(a) By definition, a grille is a covering for any opening through which air flows. A register is a grille equipped with a volume control damper. There are three main types of supply air grilles: adjustable bar, fixed bar, and stamped grilles. The adjustable bar grille has a set of parallel blades that are adjustable to vary the direction of air flow (Figure 2-6). Some adjustable bar grilles have two sets of blades--one set to vary air flow in the horizontal plane and another to adjust flow in the vertical plane.

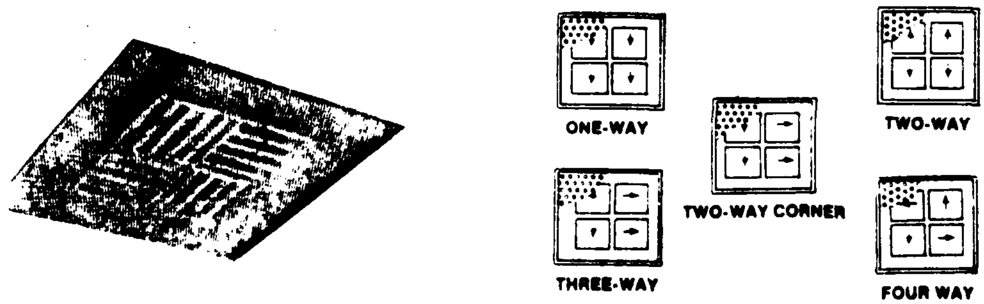

Figure 2-5. Perforated ceiling diffuser.

(b) In a fixed bar grille, the blades' deflection is preset at the factory (Figure 2-7). A stamped grille is similar in function to a fixed bar grille but is stamped from a single piece of metal.

(c) A fourth grille type, commonly called a "variable area grille," is actually a register in that it incorporates a volume control device. Any of the three main grille types can be made into a register by adding a volume control device, such as a set of opposed blade dampers.

(d) The noise criteria level of most grilles and registers is less than 20 dB at a plenum/duct air velocity of 300 FPM. The maximum noise level is about 60 dB at 1200 FPM, less 10 dB for room attenuation.

(2) Principal Applications. Grilles and registers can be used in high side wall, floor, perimeter, or ceiling applications. However, a high side wall location is most suitable for the throw and diffusion characteristics of grilles and registers. In tiered control rooms, grilles and registers are sometimes placed in the vertical face of stairs leading from one tier to another. Grilles and registers are usually used with ducted systems; however, they can also be installed with plenum systems. They are used for both supply and return systems.

(3) Sizes. Grilles and registers are commonly available in sizes ranging from 8 by 4 in. to 36 by 36 in. Larger outlet sizes can be accommodated by using more than one grille.

(4) Cost.

(a) Material costs for aluminum, single-deflection, adjustable, supply air grilles range from \$0.06 to \$0.20/CFM. Installed costs range from \$0.08 to \$0.40/CFM. For double-deflecting grilles, the installed cost is between \$0.14 and \$0.68/CFM, depending on size. In general, steel grilles cost about 10 percent less than aluminum.

(b) Single-deflecting registers with adjustable vertical or horizontal face bars cost from \$0.07 to \$0.38/CFM, with the installed cost ranging from \$0.10 to \$0.62/CFM. Single- or double-deflecting registers with adjustable, curved face bars cost from \$0.08 to \$0.46/CFM, with an installed cost range from \$0.10 to \$0.72/CFM.

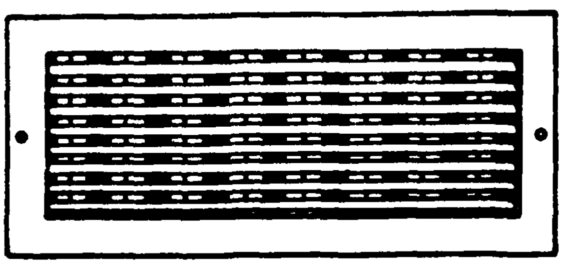

Figure 2-6. Adjustable bar grille.

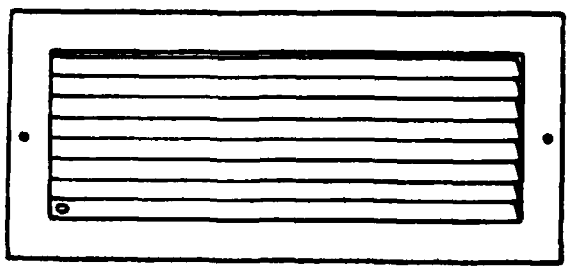

Figure 2-7. Fixed bar grille.

(5) Advantages. Grilles and registers are occasionally mounted in raised floors to provide a more directional air flow pattern than is possible with perforated floor panels. If the grille or register is mounted in a single floor panel, its flexibility for relocation is greatly increased. In terms of lowest installed cost per CFM, grilles and registers are second only to perforated ceiling diffusers and perforated floor panels.

(6) Disadvantages. In general, grilles and registers are more suitable for comfort applications than for conditioning electronic equipment. Grilles and registers are designed to be used primarily with ducted systems. This constraint reduces their flexibility in handling load redistribution and expansion.

e. Louvered Ceiling Diffusers.

(1) Operation. One of the most basic types of ceiling diffusers is the louvered diffuser. It consists of a set of concentric square or circular louvers that direct the air flow pattern (Figure 2-8). The concentric louvers are sometimes arranged flush with the surrounding ceiling, though some have a "stepped down" configuration that protrudes below the ceiling surface.

(2) Principal Applications. The louvered ceiling diffusers used in most buildings provide an air pattern parallel to the ceiling. The parallel pattern provides good mixing of the incoming cooled air with the warm air near the ceiling. Thus, the air moving around the room has adequate velocity and the temperature has been equalized. Ceiling diffusers with a perpendicular pattern are effective for dissipating concentrated cooling loads, such as those from electronic equipment. Louvered ceiling diffusers are typically used in ducted systems to provide comfort to occupants in the space.

(3) Sizes.

(a) Louvered ceiling diffusers are available in many different sizes. An example of a smaller size is a louvered diffuser with a 12 by 12 in. louvered area, 5-in. neck, total pressure drop of 0.04 to 0.41 in. of water (depending on air velocity), and a range of 54 to 190 CFM.

(b) The larger sizes feature a 24 by 24 in. louvered area, 14-in. neck, total pressure drop of 0.011 to 0.21 in. of water, and air volume rate ranging from 428 to 1500 CFM.

Figure 2-8. Square and round louvered diffusers.

(4) Cost. Rectangular louvered ceiling diffusers range in cost from $0.48 to $0.96/CFM, with an installed cost ranging from $0.74 to $1.28/CFM.

(5) Advantages. Louvered diffusers provide a good surface effect with the ceiling (i.e., discharge air tends to flow in the parallel plane close to the ceiling). Diffusers with a good surface effect are appropriate for conditioning occupant areas.

(6) Disadvantages. Louvered ceiling diffusers, like most ceiling diffusers, are designed to be used with ducted systems and to provide human comfort; therefore, their flexibility and suitability for C^3I room applications are limited. The installed cost per CFM of the louvered ceiling diffuser is relatively high when compared with the other air-diffusing equipment evaluated.

f. Linear Diffusers.

(1) Operation. Linear diffusers are long, narrow diffusers usually installed end-to-end to give the appearance of one continuous slot running the length of a ceiling, wall, or floor. A typical linear diffuser is the slot diffuser (Figure 2-9). The air flow pattern of many linear diffusers can be adjusted to provide perpendicular or parallel flow (Figure 2-10). At 140 CFM/lin ft, slot diffusers have a noise level of less than 20 dB. At 500 CFM/lin ft, they have a noise level of 57 dB, less 10 dB for room attenuation.

(2) Principal Applications.

(a) Linear diffusers are used primarily for occupant comfort. They can be installed in high sidewalls, ceilings, or around the perimeter of a space in sills or floors.

(b) With perpendicular flow, high side wall and perimeter applications are most suitable. Linear diffusers installed in the ceiling can provide parallel flow.

(3) Sizes. Linear diffusers are typically available with 1/2-, 3/4-, or 1-in.-wide slots and may be up to eight slots wide. Depending on the number of slots, these diffusers can supply from 10 to 500 CFM/lin ft.

(4) Cost. Depending on the number of slots, these diffusers cost between $15 and $47/lin ft. Including contractor overhead and profit, the installed cost ranges between $25 and $66/lin ft. This equates to an installed cost of $0.13 to $2.50/CFM.

(5) Advantages. Linear diffusers are appropriate for perimeter applications in occupant areas.

(6) Disadvantages. Linear diffusers, designed to provide human comfort, are rarely (if ever) used in electronic equipment rooms. Since they are used mainly with ducted systems and require complex installation techniques, linear diffusers are severely restricted in ability to meet the changing demands of load redistribution and expansion. The installed cost per CFM of linear diffusers is the highest of all air-diffusing equipment evaluated.

g. Light Troffers.

(1) Operation. Another type of diffuser used in overhead ducted supply systems is called a "light troffer." It usually consists of a 2 by 4 ft light fixture with an opening around the perimeter that functions as the diffuser. This opening is served by a cavity on the back of the light fixture that connects to the supply duct.

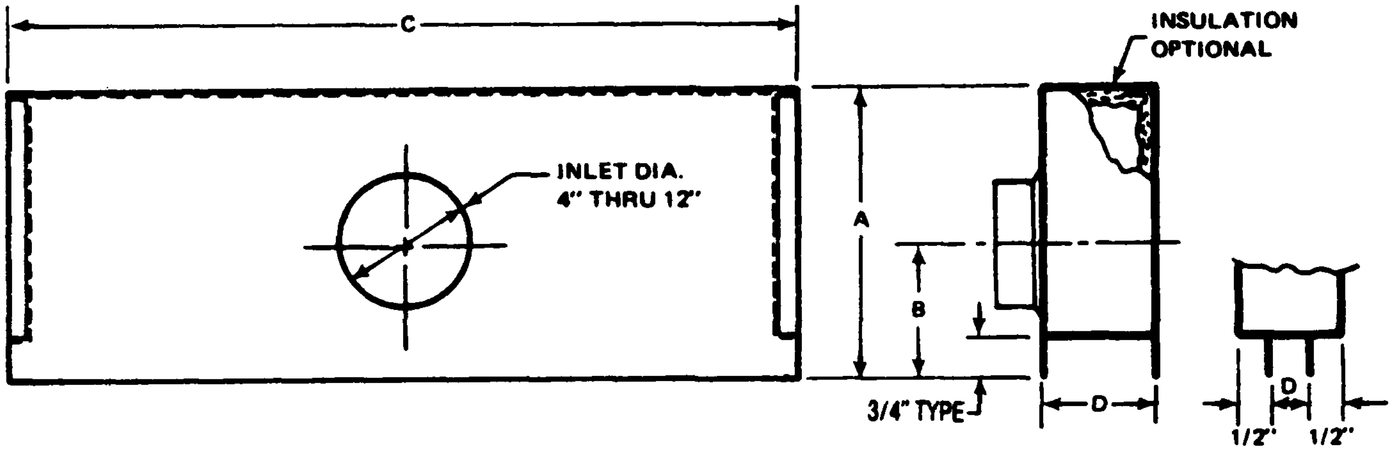

Figure 2-9. Slot diffusers.

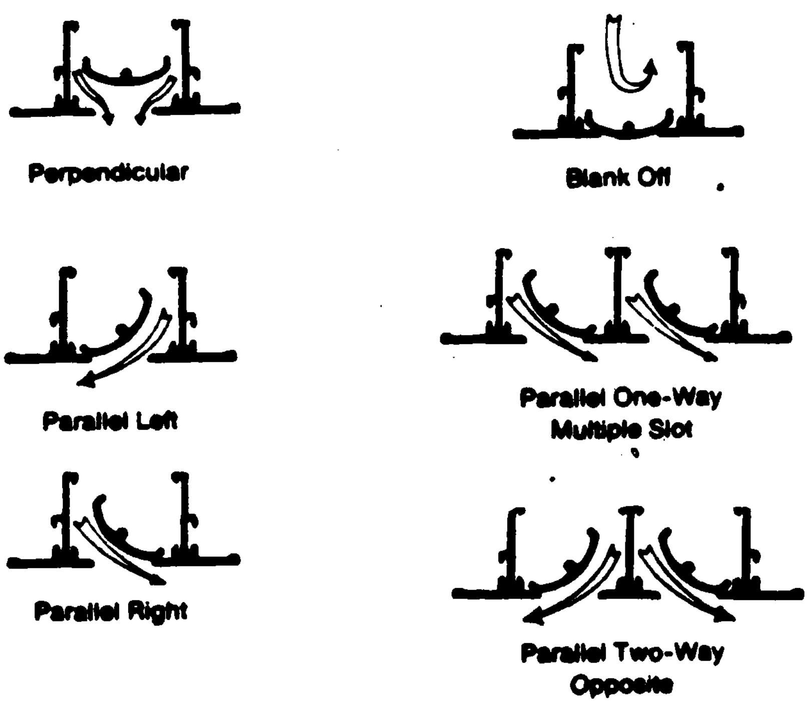

Figure 2-10. Adjustable air flow pattern.

(2) Principal Applications. Light troffers are used mainly for occupant comfort. The maximum flow rate of approximately 150 CFM/troffer is usually inadequate for use in C^3I rooms.

(3) Sizes. Air flow rates for light troffers range from 50 to 150 CFM/troffer.

(4) Cost. Light troffers have an installed cost ranging from $0.20 to $0.60/CFM. These figures reflect the added cost of a light troffer over that of a standard 2 by 4 ft light fixture.

(5) Advantages. Light troffers are effective for conditioning small individual offices since they have low CFM rates.

(6) Disadvantages. Light troffers are rarely used in C^3I room applications because of their low air flow rates. If used, they will be relatively inflexible in accommodating the relocation and addition of electronic equipment in the C^3I room.

h. Return Air Coverings

(1) Operation.

(a) Return air inlets of ducted return systems are typically covered with the standard grilles or registers described earlier. Since deflectors have very little effect on return air flow patterns, their use is rarely warranted. Simple fixed-bar or stamped grilles are used most commonly for return air inlets. Registers are sometimes used to balance return air volume flow rates. In ceiling plenum return systems, the perforated face of the ceiling diffuser is commonly used for the return air inlet covering.

(b) Return grilles and registers generally have noise levels of about 15 dB at a duct/plenum velocity of 400 FPM. At 1000 FPM, the noise level can reach 48 dB. The noise level of the perforated ceiling diffuser, used as a return air inlet cover, is typically less than 20 dB at plenum velocities of up to 1000 FPM.

(2) Principal Applications.

(a) Grilles and registers can be used as return air inlet covers for both ducted and plenum return systems. The same is true for perforated ceiling diffusers.

(b) In addition to standard grilles, transfer grilles are used to deliver return air from the conditioned space to a remote space where the return inlet is located. Transfer grilles are installed in walls or doors. A common transfer grille is the V-bar type consisting of a set of V-shaped louvers stacked on top of each other. This configuration eliminates direct line-of-sight vision through the transfer grille.

(3) Sizes. Return air grilles are commonly available in sizes ranging from 6 by 6 in. to 48 by 48 in. Perforated ceiling return inlets typically measure 24 by 24 in., enabling easy substitution for standard suspended ceiling panels. Transfer grilles range in size from 8 by 4 in. to 30 by 30 in.

(4) Cost. For a 6 by 6 in. return air grille rated at 125 CFM, the material cost is approximately $0.05/CFM and the installed cost is about $0.14/CFM. A 48 by 48 in. grille at 8000 CFM costs $0.02/CFM, with an installed cost of $0.03/CFM. Perforated ceiling diffusers without deflectors used as return inlet covers range in cost from $0.007 to $0.03/CFM, with the installed cost ranging from $0.008 to $0.04/CFM.

(5) Advantages/Disadvantages. Grilles, registers, and perforated ceiling return inlet covers have different negative static pressure properties. The cost per CFM of a particular type of return air inlet cover must be balanced against the effect of the static pressure curve on fan power requirements when evaluating the life-cycle cost of an overall cooling system.

2-3. Other Considerations

a. Static Pressure Impact. Another consideration is that the static pressure properties of air-diffusing equipment will have an impact on fan energy requirements. Figure 2-11 is a plot of static pressure versus CFM for several different types and sizes of air-diffusing equipment. The plot indicates that, for a flow rate of 800 CFM, the perforated ceiling diffuser has the highest static pressure. Note that the louvered ceiling and linear diffusers have the lowest static pressures for any given CFM. While perforated floor panels and perforated ceiling diffusers have a low first cost, they also have high fan energy requirements.

b. EMP Protection. The penetrations to EMP shielding caused by air distribution systems are among the largest in the C^3I facility. The protection parameters for air distribution penetrations are the same as those for piping penetrations (see Chapter 3, **Chilled Liquid Distribution Systems**). In both cases, the waveguide-beyond-cutoff technique is used. The waveguide must be made of a conductive material and must be continuously welded or soldered to the primary EMP shield so that current flowing on the waveguide can be discharged to the primary EMP shield. The maximum inside diameter of the penetration must be 4 in. or less to achieve a cutoff frequency of 1.47 GHz for a rectangular penetration and 1.73 GHz for a cylindrical penetration. The unbroken length of conducting material adjacent to the penetration must be a minimum of five times the diameter of the conducting material (i.e., pipe, duct) to attenuate by at least 100 dB at the required frequencies. Since, in general, ventilation duct dimensions cannot reasonably be limited within the 4-in. restriction, the duct itself cannot be used as the waveguide.

(1) EMP protection for air distribution penetrations uses a "honeycomb" waveguide air vent panel (Figure 2-12). As the term "honeycomb" implies, the cross sectional area of the panel is divided into a number of cells (Figure 2-13), each of which complies with the 4-in. maximum diameter and five-diameter length requirements. The grid structure must be metallic and all joints must be continuously bonded.

(2) Honeycomb panels are commercially available in dimensions up to 18 by 18 in. Larger panels can be made by soldering the seams of multiple panels. The honeycomb material is normally cadmium-coated steel or tinned brass. This material is soldered into a steel frame, which is subsequently welded into the penetration plate (Figure 2-14) or attached with RFI gasketed and bolted seals.

(3) The material retail cost of a honeycomb air vent panel is about $50/sq ft. The cost to weld the frame to the penetration plate results in an installed cost of $70 to $80/sq ft of penetration area. If multiple panels are required to protect a large penetration, the cost of soldering together the seams of the panels will increase the costs significantly.

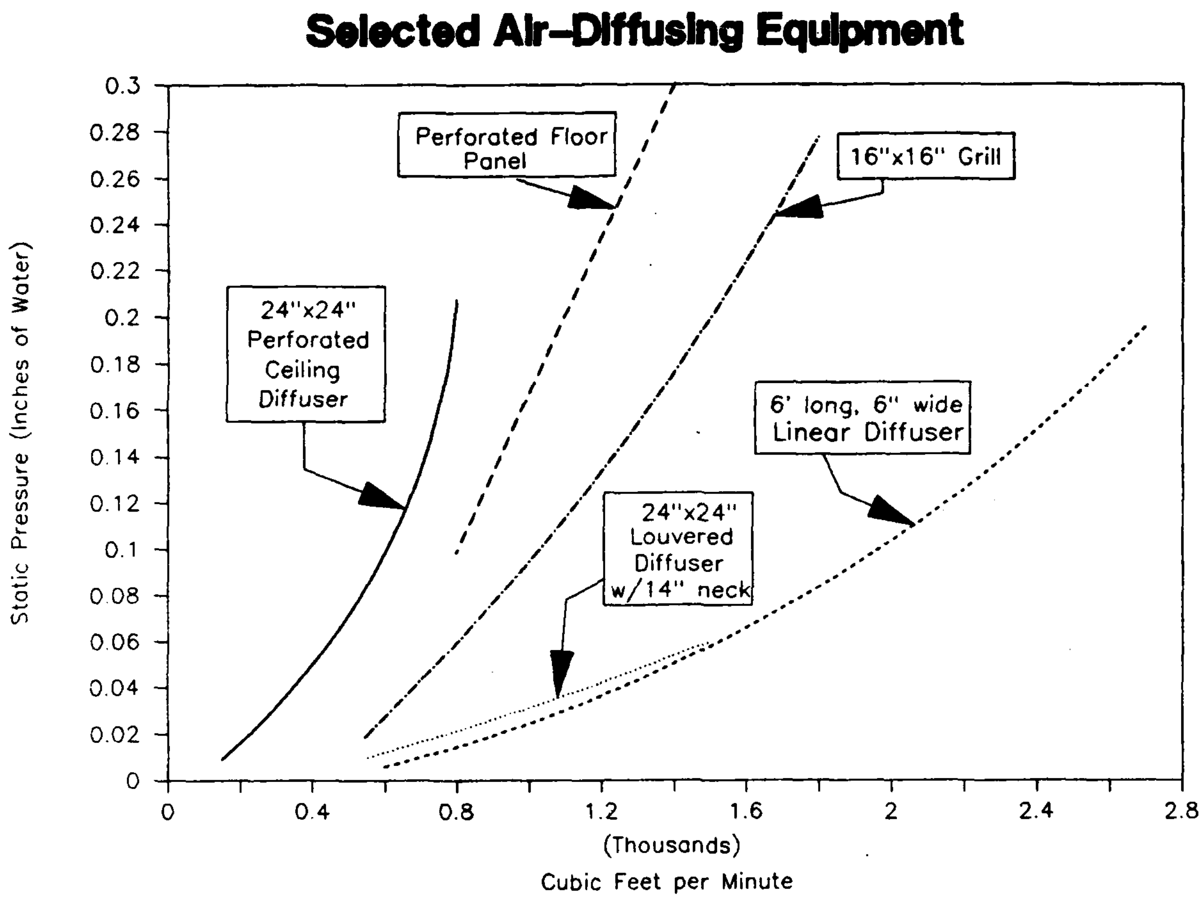

Figure 2-11. Static pressure vs. volume flow rate.

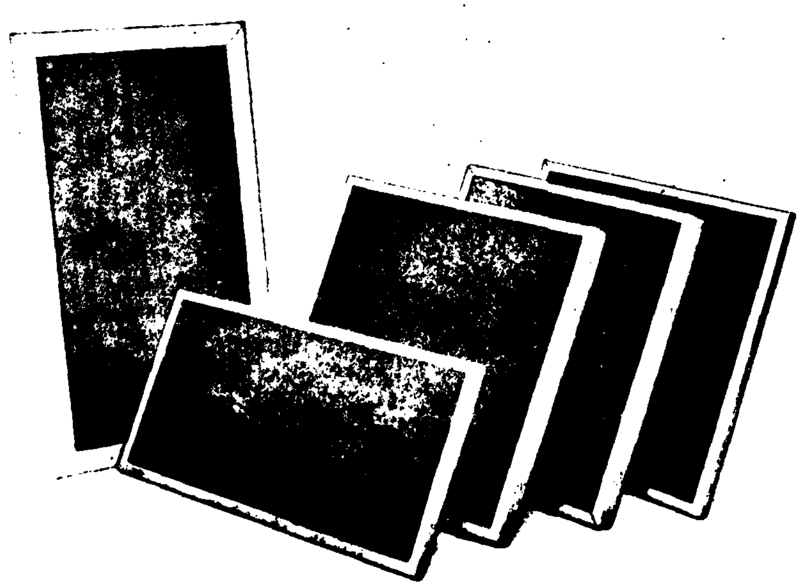

Figure 2-12. Honeycomb waveguide air vent panels. (Source: *USAF Handbook for the Design and Construction of HEMP/TEMPEST Shielded Facilities* [1986].)

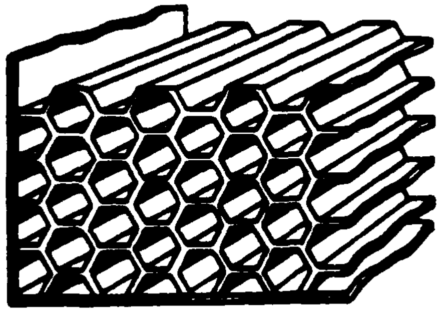

Figure 2-13. Cell construction. (Source: *USAF Handbook for the Design and Construction of HEMP/TEMPEST Shielded Facilities* [1986].)

(4) The pressure drop across the honeycomb panel must be taken into account when designing the air distribution system and determining fan power requirements. Figure 2-15 shows the static pressure drop versus air flow for a honeycomb panel with individual cell widths of 1/8th. Because of the small cell size, the panels tend to clog with dirt easily. Thus, filters and access for changing them should be incorporated into the design.

(5) Another consideration is the EMP protection of controls for air distribution systems. Whenever possible, damper actuators and other air distribution controls should be located inside the C^3I room shield. Controls located outside the shield must be protected separately and any penetrations by control wiring must be treated with the various filter and surge-limiting strategies described in the EMP literature.[2] For air distribution controls located outside the C^3I room shield, consideration should be given to installing pneumatic controls, which are not as susceptible to an EMP event as direct digital controls.

c. *NBC Protection*. The C^3I room also must be protected against NBC contamination. "Collective protection" is the most common method used. This approach involves two main principles: (1) maintaining positive pressure within the C^3I room to increase NBC protection from sources other than the outside air intake, and (2) using filters to prevent introduction of NBC into the outside air intake. The filters typically are the high-efficiency particulate matter type used with charcoal or other absorptive filters.

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[2]*EMP Engineering Handbook for Ground Based Facilities, Vol II: Design and Engineering* (Defense Nuclear Agency, 1986); MIL-HDBK-419, *Grounding, Bonding and Shielding for Electronic Equipments and Facilities* (Department of Defense [DOD], 1982); MIL-HDBK-253, *Guidance for Design and Test of Systems Protected Against the Effects of Electromagnetic Energy* (DOD, 1978); *USAF Handbook for the Design and Construction of HEMP/TEMPEST Shielded Facilities*, Sections IV and VI (U.S. Air Force, 1986); M. McInerney, et al., *The Effect of Fluids on Waveguides Below Cutoff Penetrations as Related to Electromagnetic Shielding Effectiveness*, Technical Report M-354/ADA144685 (USACERL, July 1984).

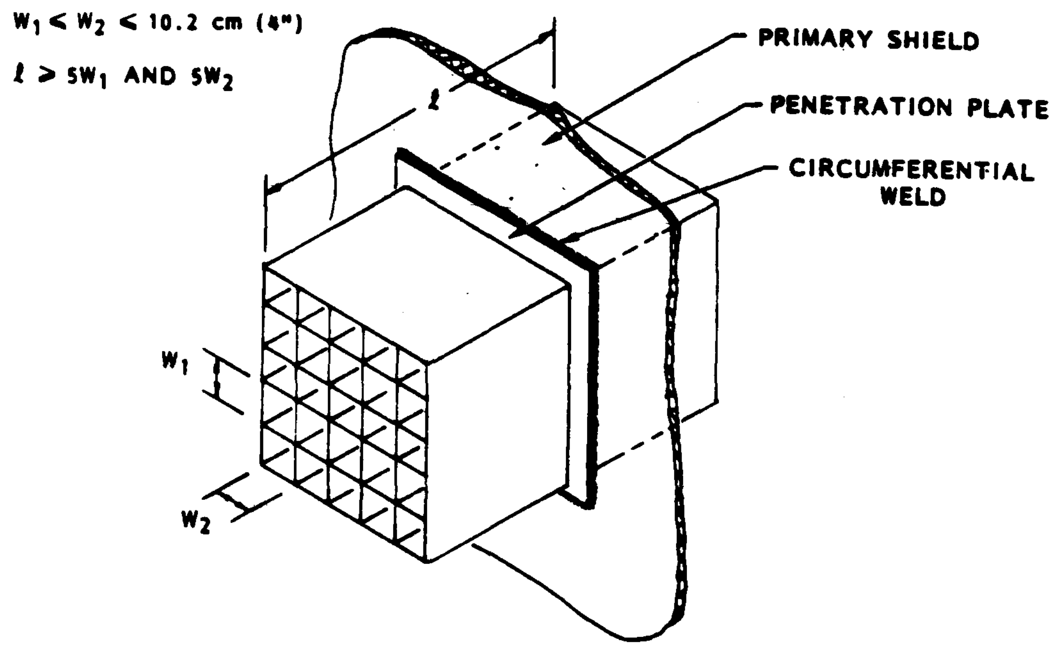

Figure 2-14. **Honeycomb waveguide panel attached to primary shield. (Source:** *USAF Handbook for the Design and Construction of HEMP/TEMPEST Shielded Facilities* **[1986].)**

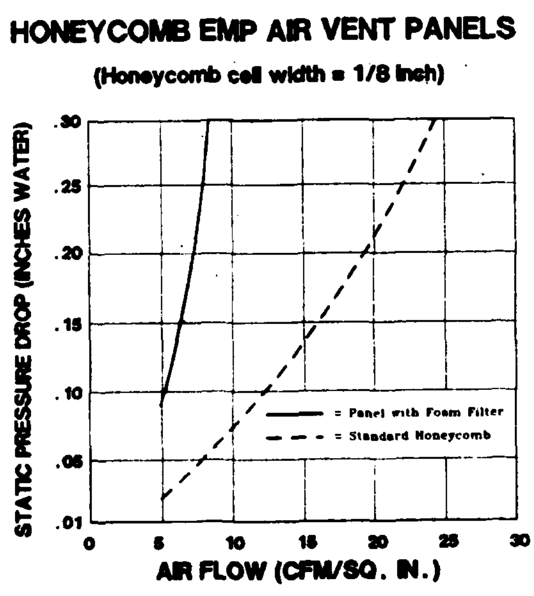

Figure 2-15. Air flow characteristics of a honeycomb panel.

(1) NBC filters produce a large drop in pressure in the air distribution system. Therefore, fans and fan motors must be sized approximately 20 to 30 percent larger to overcome this pressure drop. Due to the increased fan energy costs, the system is usually designed to bypass NBC filters under normal operation. NBC detection sensors and alarms will signal the bypass damper to direct outside air through the filters in the event of NBC contamination. Provisions must be made in the design to enable personnel to change the NBC filters without exposure to NBC contamination.

(2) NBC filters are quite expensive. Collective protection can add 10 to 15 percent to the construction cost of a facility.

Section III: SUMMARY EVALUATION

2-4. How To Use the Evaluation Table

a. Table 2-1 ranks each air distribution system component on the criteria presented in Section I. These rankings are qualitative, with a black circle representing a high ranking, a gray circle denoting a medium ranking, and a white circle indicating a low rating. It should be noted that a low rating simply means that other components are more suitable for a specific criterion, and not necessarily that the component is a poor choice in general.

b. For a particular criterion, only the rankings of components in the same category should be compared. For example, when the primary consideration is life-cycle cost, the ceiling plenum supply, free return, and perforated floor panels all have a high rating. However, a free return and perforated floor panels cannot be used with a ceiling plenum supply system. Instead, the ranking of the ceiling plenum supply system should be compared only with that of the underfloor plenum and overhead ducted supply systems. Similarly, a certain return system should be compared with other return systems and a particular diffuser should be compared with other air-diffusing equipment.

c. When ranking each component, certain assumptions were made regarding the system configuration. These assumptions were based on the most typical system configuration in which each component is used. It was assumed that the underfloor plenum supply system is served by air handlers located within the C^3I room. The ceiling plenum and overhead ducted supply systems were assumed to be served by a remotely located air handler. Similarly, it was assumed that the ceiling plenum and ducted return systems serve a remote air handler and that the free return system serves air handlers inside the electronic equipment room.

(1) It was also assumed that VAV and CV mixing boxes are used with overhead ducted supply systems. Furthermore, it was assumed that the boxes are used with either the ceiling plenum or ducted return system, but not with the free return system.

(2) In most cases, air-diffusing equipment was assumed to be used with the supply system for which it is most suitable. For example, it was assumed that perforated floor panels are used with the underfloor plenum supply system and that louvered and linear diffusers are used with the overhead ducted supply system. While it was assumed that perforated ceiling diffusers are used with the ceiling plenum supply system, they can also be installed with ducted systems. Grilles and registers can also be used with plenum and ducted systems.

Table 2-1. Summary Evaluation--Air Distribution Systems

AIR DISTRIBUTION EQUIPMENT	Load redistribution	Load expansion	Humidity control	Filtering	Static electricity	Backup cooling	Reliability	* EMP shielding cost	* Life-cycle cost
Supply:									
1) Underfloor Plenum Supply	●	●	●	○	●	N/A	●	●	◍
2) Ceiling Plenum Supply	◍	◍	●	◍	◍	N/A	●	◍	●
3) Overhead Ducted Supply	○	○	◍	●	◍	N/A	●	◍	○
A) with VAV Boxes	◍	◍	○	●	◍	N/A	◍	◍	○
B) with CV Boxes	◍	◍	◍	●	◍	N/A	◍	◍	○
Return:									
4) Free Return	●	●	◍	○	◍	N/A	●	●	●
5) Ceiling Plenum	◍	◍	◍	◍	◍	N/A	●	◍	◍
6) Ducted Return	○	○	◍	●	◍	N/A	●	◍	○
AIR-DIFFUSING EQUIPMENT									
1) Perforated Floor Panels	●	●	●	N/A	●	N/A	●	N/A	●
2) Perforated Ceiling Diffusers	●	●	●	N/A	◍	N/A	●	N/A	●
3) Grilles and Registers	◍	◍	◍	N/A	◍	N/A	◍	N/A	◍
4) Louvered Ceiling Diffusers	○	○	◍	N/A	◍	N/A	●	N/A	○
5) Linear Diffusers	○	○	◍	N/A	◍	N/A	●	N/A	○
6) Light Troffers	○	○	◍	N/A	◍	N/A	●	N/A	○

● = High ◍ = Medium ○ = Low

* A high ranking for cost-related criteria indicates a low relative cost. A low ranking indicates a high relative cost.

2-5. Evaluation Results

a. Load Redistribution. When load redistribution is the primary consideration, the underfloor plenum supply system has the best ranking. As mentioned, it is assumed that air handlers (either chilled water or DX) located within the C^3I room serve the underfloor plenum. Though not listed as one of the criteria, the availability of space within the C^3I room is often a critical factor. In this case, the space within the C^3I room occupied by the air handlers may preclude the use of the underfloor plenum supply system even though load redistribution and expansion are also important considerations. The same reasoning applies to the free return system, since it can only be used with an underfloor supply plenum based on the assumed configurations.

b. Load Expansion. When load expansion is the primary consideration, the underfloor plenum supply system has the best ranking. Again, it is assumed that air handlers located within the C^3I room serve the underfloor plenum. Load expansion is accommodated by installing additional air handlers to satisfy the extra cooling load. This modification can often be done without shutting down the existing air handlers. With a remotely located air handler, the supply CFM must be increased to satisfy the additional cooling load. In addition, it may be necessary to increase the size cf the cooling coil to increase the cooling capacity. These types of modifications require that the air handler be shut down.

c. Precise Control of Humidity.

(1) Chiller and air-handler controls are primarily responsible for regulating the humidity level of the C^3I room within required limits. The rankings for the precise control of humidity criterion are based on the ability of the air distribution system components to quickly equalize the differences in humidity between room air and discharge air within the C^3I room.

(2) Because of their high entrainment ratio, perforated floor panels and perforated ceiling diffusers received high ratings. Since this equipment is typically used with certain supply systems, the underfloor plenum and ceiling plenum supply systems were also ranked highly.

(3) The remaining air-diffusing equipment, and the supply systems used with it, had almost the same ability to quickly equalize the differences in humidity between room air and discharge air within the C^3I room. Therefore, they all received a medium rating. However, the VAV box system received a low rating since, during periods of low cooling load, air flow is reduced significantly.

d. Filtering of Particulate Matter.

(1) The ability to filter particulate matter from ventilation makeup air is fully evaluated in Chapter 4. In terms of air distribution systems, rankings for the filtering criterion are based on the system's ability to minimize the introduction of dust and debris into the recirculated air stream.

(2) Since the overhead ducted supply system is more likely to stay clean than a ceiling plenum supply system, the overhead ducted supply system, together with the VAV and CV box variations, received high ratings. The ceiling plenum supply system was given a medium ranking since it is more likely to stay clean than an underfloor plenum supply system. By this reasoning, the underfloor plenum supply system was given a low rating. It should be stressed that none of the three supply systems compromise the cooling system's ability to filter particulate matter.

(3) Similarly, although none of the return systems compromise filtering ability, the ducted return system was given a high rating, the ceiling plenum return system was rated medium, and the free return system received a low rating.

(4) Filtering, as evaluated for air distribution systems, was considered to be a function of the supply and return systems and not applicable to air-diffusing equipment. Table 2-1 indicates this approach by noting "N/A" for the air-diffusing equipment on this criterion.

e. Static Electricity Control. For cooling systems, a major factor in controlling static electricity is the regulation of humidity within the C^3I room. As mentioned, chiller and air-handler controls are primarily responsible for humidity control. For air distribution systems, the fact that raised floor and perforated floor panels are available with antistatic and conductive coatings gave the underfloor plenum supply and perforated floor panels high ratings. All other system components were considered equal in terms of static electricity control and, therefore, were given a medium ranking.

f. Backup Cooling Capacity. The capacity of a cooling system to provide 15 min of backup cooling in the event of a power failure depends on the configuration of the chilled water distribution system. For this reason, ranking air distribution system components for the backup cooling criterion is not applicable and is so indicated by "N/A" in Table 2-1.

g. Reliability.

(1) Reliability is fairly equal among the different system components. If comparisons must be made, components with moving parts are more susceptible to malfunction than those with no moving parts. Malfunction is defined here to mean either breakage, sticking, or improper adjustment by C^3I room occupants. Since the VAV and CV mixing box variations of the overhead ducted supply system have moving parts, they received a medium rating. The remaining supply and return systems generally have no moving parts; all received a high ranking.

(2) While all of the air-diffusing equipment evaluated is reliable, rankings were based on the relative susceptibility for malfunction as just discussed. For the evaluation, it was assumed that none of the air-diffusing equipment is fitted with opposed-blade dampers for air volume control. One exception is the register, which by definition includes a volume control device. Perforated floor panels, louvered ceiling diffusers, perforated ceiling diffusers, and linear diffusers all received a high reliability rating since they typically have no moving parts. Grilles and registers received a medium ranking since they often have movable deflectors that can be adjusted by occupants in the space.

h. Least Cost for EMP Shielding.

(1) Component ranking for the criterion "least cost for EMP shielding" was based on the relative size of the shielding penetrations associated with the typical configuration for the supply and return components. It was assumed that the C^3I room itself is EMP-shielded. For the underfloor plenum supply system, air handlers serving the plenum are typically located inside the C^3I room and a free return system is used. This configuration requires relatively small fresh air ducts compared with the ducts sized for total supply air associated with remotely located air handlers.

(2) With the air handlers located within the space, the only other penetrations are those made by the air-handler piping (i.e., chilled water, condenser, condensate, and humidifier water). Since the relative size of all penetrations to the EMP shield are smaller for systems with air handlers located within the C^3I room than those with remotely located air handlers, the underfloor plenum supply and free return systems received high ratings.

(3) Because the typical configurations of the remaining supply and return systems involve penetrating the C^3I room EMP shield with ductwork sized for total supply air, they received a medium rating. It should be noted that the difference in cost for EMP shielding among the supply and return systems is marginal and that the rankings for this criterion merely reflect the considerations just described.

(4) Ranking air-diffusing equipment in terms of least cost for EMP shielding is not applicable since these components are always located inside the shield.

i. Lowest Life-Cycle Cost.

(1) A high ranking for the life-cycle cost criterion indicates a relatively low life-cycle cost. Conversely, a low rating denotes a relatively high life-cycle cost. Since no direct costs and negligible maintenance costs are generally associated with air distribution systems, the life-cycle cost criterion is basically an issue of first cost. Of the five supply systems evaluated, the ceiling plenum supply system has the lowest life-cycle cost per unit of C^3I room floor area and was given a high rating. The underfloor plenum supply system was next lowest in cost and was, therefore, ranked as medium. The overhead ducted supply system, together with the VAV and CV mixing box variations, are the most expensive and received a low ranking. It should be noted that the difference in cost between the ceiling and underfloor plenum supply systems is small compared with the substantial cost premium of the overhead ducted supply system and its variations.

(2) The free return system has the lowest life-cycle cost and, therefore, received a high ranking. The ceiling plenum return system received a medium ranking and the ducted return system was ranked low. It is important to note that use of the free return or ceiling plenum return system with air handlers located inside the C^3I room may require that a separate system be installed to introduce outside air. The costs of each system could thus become higher than that of a ducted return system, which is typically used with a central station air handler that mixes outside air with return air.

(3) The average installed cost per CFM was determined for each diffuser, grille, and register type. These average costs were then divided into three groups corresponding to the three degrees of rank. The perforated ceiling diffuser and perforated floor panel have the lowest average cost and, therefore, were ranked high. Grilles, registers and light troffers are the next most expensive and received a medium ranking. The average cost of the louvered ceiling diffuser and the linear diffuser are both high, so they both were ranked low.

(4) As mentioned, no direct costs and negligible maintenance costs are associated with air distribution systems, so the life-cycle cost criterion is basically that of first cost. However, there are some indirect operating costs associated with some system components. For instance, the cost of the C^3I room floor space occupied by air handlers in the underfloor plenum supply system was not considered during the ranking process, but should be taken into account when designing the overall cooling system.

3 CHILLED LIQUID DISTRIBUTION SYSTEMS

Section I: GENERAL

This chapter evaluates design approaches for chilled liquid distribution systems serving electronic equipment rooms. The following criteria are used to evaluate each distribution system:

1. Flexibility in meeting load redistribution.
2. Flexibility in meeting load expansion.
3. Capacity to provide 15 min of backup cooling.
4. Reliability.
5. Least cost for EMP shielding.
6. Lowest life-cycle cost.

Most criteria do not apply to distribution system components. The evaluation was done for completeness only.

3-1. System Types

a. Although there are many distribution arrangements for supplying chilled water to C^3I rooms, three "generic" ones that represent the major variations are described and evaluated in this chapter. These are:

(1) A central chiller serving the C^3I room, C^3I building, and other buildings.

(2) A chiller serving the C^3I room and C^3I building.

(3) One chiller serving the C^3I room and another chiller serving the C^3I building.

b. There are several ways to provide chilled water to the air handlers and electronic equipment within the C^3I room. These approaches are described and evaluated along with a piping configuration for meeting the C^3I requirement of 15 min backup cooling.

c. In addition to piping configurations outside and inside the C^3I room, the major piping system components are described and evaluated. These are piping, valves, and pumps. The discussion of the these components includes information on sizes and costs. However, the size and cost of the configurations depend on the specific application and, therefore, are not discussed in detail.

d. After each piping configuration or component is fully described, the configurations are evaluated according to the criteria listed in **Section** I above. The evaluation table summarizes the results of the qualitative ranking procedure (see **Section III** of this chapter).

Section II: SYSTEM DESCRIPTIONS

3-2. System Configurations

Chilled liquid distribution systems are discussed in two main parts: (1) piping configurations and (2) piping components. The first part of this section describes typical chilled liquid piping configurations outside and inside the C^3I room. The second part describes the main chilled liquid piping components (piping, valves, and pumps) used in these configurations.

a. Typical piping configurations outside the C^3I room are as described in paragraph 3-1*a* above.

b. Piping configurations within the C^3I room include:

(1) Piping methods to air handlers and C^3I equipment.

(2) The 15-min backup cooling capacity.

c. The first part of this discussion deals with moving the chilled water to the C^3I room and the second part presents information on possible configurations once the chilled water is in the C^3I room--including arrangements for meeting the 15-min backup cooling requirement.

d. Even though one chiller, pump, etc., may be discussed in terms of the configurations listed above, it is understood that multiple equipment will be connected to meet the redundancy requirements of C^3I facilities. Typically, the "N + 2" rule of thumb is used to provide backup. Using N + 2, if a unit breaks down while another is under maintenance (or also broken down), complete backup is provided without interrupting operations. However, more or less backup may be desired depending on:

(1) The importance of the mission.

(2) The reliability of the equipment.

(3) The cost of the equipment.

3-3. Piping Configurations—Outside the C^3I Room

Possible piping configurations outside the C^3I room are quite numerous. This section describes the possible combinations of the three common configurations mentioned previously.

a. A Central Chiller Serving C^3I Room, C^3I Building, and Other Buildings.

(1) Operation. One common configuration is a central chiller serving multiple buildings. Figure 3-1 shows a typical central chiller piping layout. Chilled water is piped from the chiller to various buildings, including the C^3I building. The chilled water entering the C^3I building serves both the C^3I building and the C^3I room.

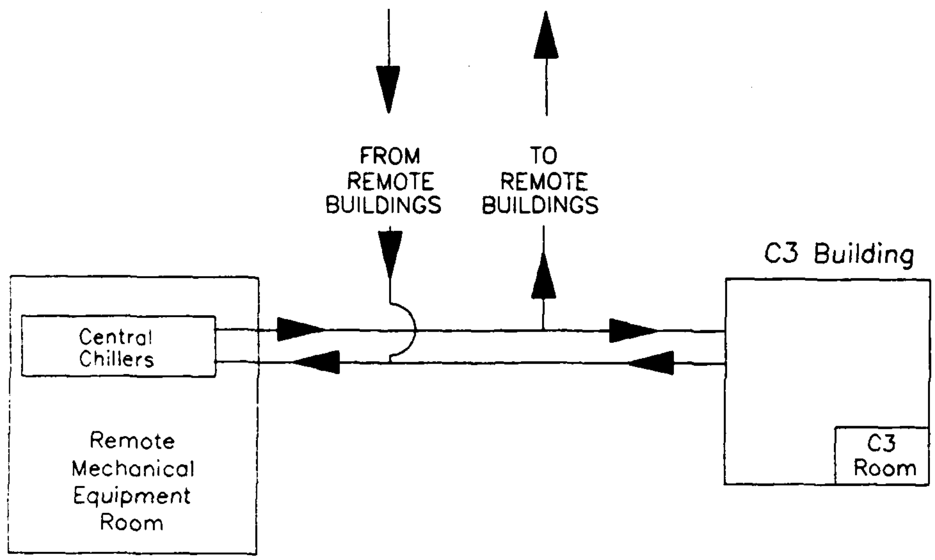

Figure 3-1. Central chiller.

(2) Principal Applications. This configuration is used when there are many buildings located relatively close to each other and/or when it is not economically feasible to locate a chiller in each building.

(3) Advantages. The first cost of one large chiller is less than that of many small chillers. The coefficient of performance (COP) is higher for a large chiller than for a small chiller. Therefore, the operating cost of one large chiller is also generally less than that of many small chillers. Maintenance costs are also lower for one large chiller compared to that for many small units.

(4) Disadvantages.

(a) With the chiller located outside the C^3I room, longer piping runs are necessary. This requires larger pipe diameters and extra insulation. Precise temperature regulation is more difficult due to the longer distance from the C^3I room load. Also, it is inefficient and expensive to run a large central chiller during off-hours to cool or back up a relatively small load.

(b) An additional consideration is water pressure and flow rate. A chiller located in a distant area is designed to pump water over long distances; consequently, water pressures far beyond the allowable specifications of computer equipment may be encountered. Adjusting the flow rate and water pressure requires additional valving and controls.

(c) EMP protection is also a concern. If many of the buildings served by this central chiller are EMP-protected, more penetration points are required. Since the chiller serves an EMP-protected building or room, it also must be protected. In contrast, if the chiller is located within a shielded building, it does not require additional protection.

(d) Finally, if the chiller is located far away from the critical facility, the chilled water piping will cover long distances. If the piping has to travel through an unsecured/unprotected area, it will be exposed to possible sabotage or attack from someone trying to deter operations. Also, longer pipe runs increase the possibility of undetected leaks since the entire length of pipe is more difficult to inspect.

b. Chiller Serving C^3I Room and C^3I Building.

(1) Operation. Figure 3-2 is a schematic of a C^3I building chiller configuration. This chiller serves the building air-handling units and/or the chilled water loop. In addition, the C^3I room is served by the building chiller.

(2) Principal Applications. This configuration is commonly used when there is a dedicated building chiller and the C^3I room loads are relatively small and do not require a dedicated chiller.

(3) Advantages. If the building containing the chiller is EMP-shielded, the chiller does not need to be shielded. Also, the chiller will be more responsive to changing loads than the first configuration since it will be closer in size and proximity to the C^3I room loads.

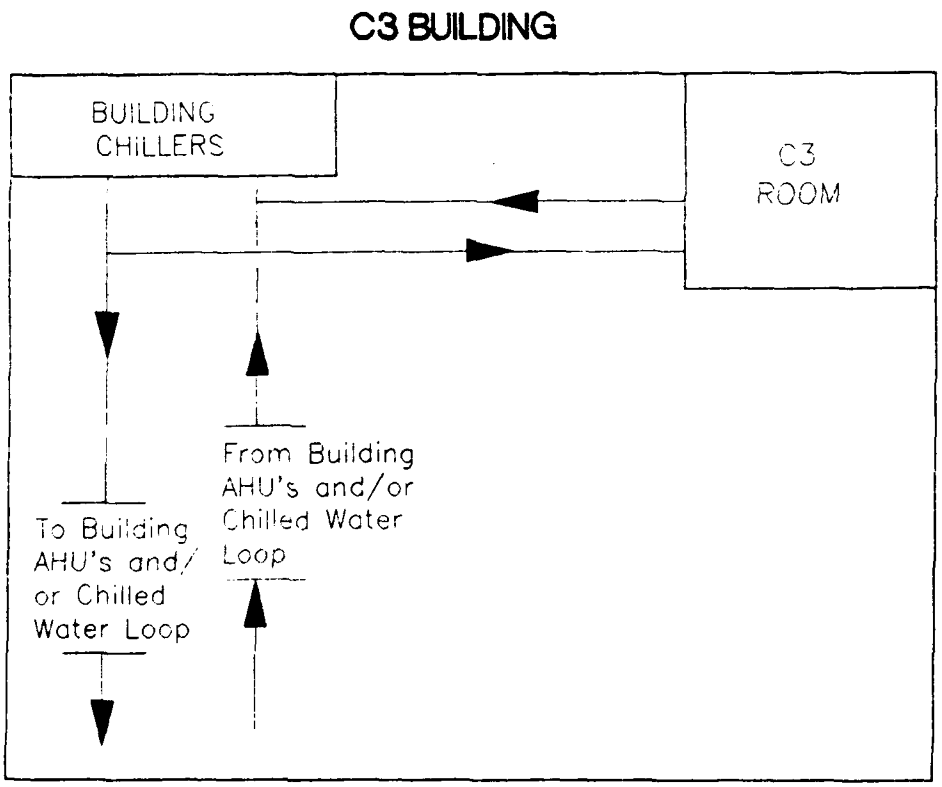

Figure 3-2. C^3I building chiller.

(4) Disadvantages.

(a) Since the building chiller is usually larger than the size needed for the C^{3}I room, the initial and operating costs will be higher than if a smaller dedicated chiller is used for the C^{3}I room. However, if the building chiller runs 24 hr/day for other operations, then this would not be a disadvantage.

(b) If the building chiller is a substantial distance from the C^{3}I room, larger pipe diameters and extra insulation will increase the cost considerably.

c. One Dedicated Chiller Serving the C^{3}I Room, Another Serving the Rest of the Building.

(1) Operation. Figure 3-3 shows a common dedicated C^{3}I chiller system. The C^{3}I room is supplied by a dedicated chiller while the rest of the building is supplied by a different chiller. The dashed lines between chillers denotes a possible connection which is often used if additional backup and redundancy are desired. These cross-connections can be used should the C^{3}I chillers fail, assuming the building chiller's capacity is sufficient. In addition, if the cross-connections are piped for two-way flow, should the building chillers fail, the C^{3}I chillers could be used as backup. Of course, the C^{3}I chillers must be sized to accommodate the additional load.

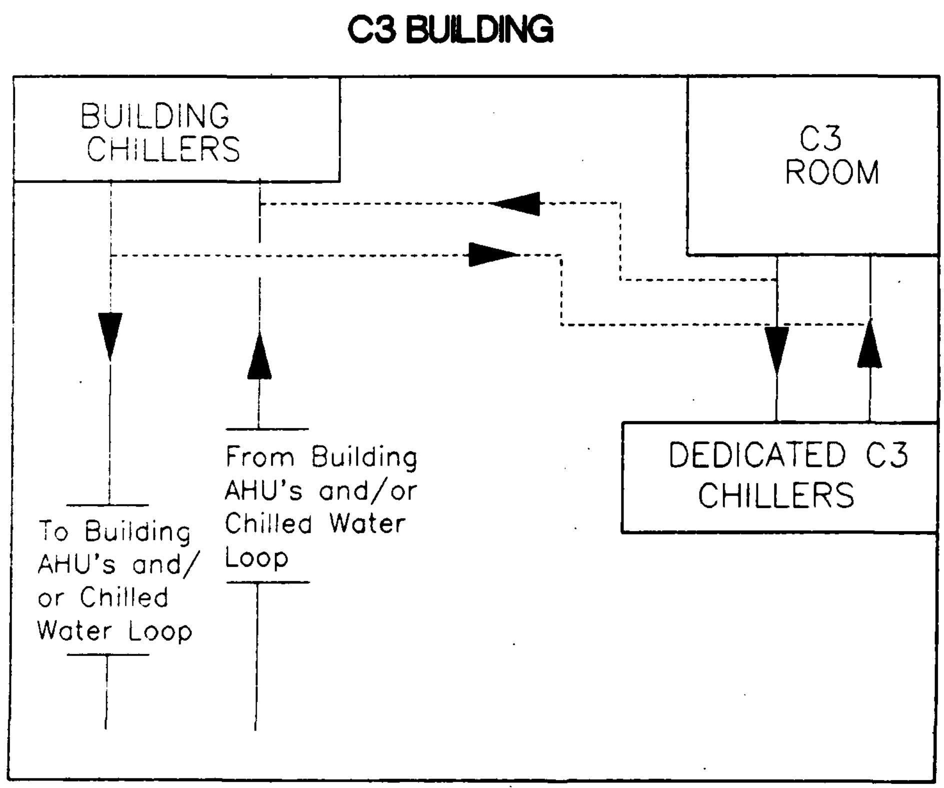

Figure 3-3. Dedicated C^{3}I chiller.

(2) Principal Applications. Dedicated C^3I room chillers are used when continuous operation of the C^3I room is necessary and it is not economically feasible to run the building chillers or central chillers for 24 hr/day. If the C^3I room load is smaller than 40 percent of the chiller's rated capacity, then a dedicated chiller is usually used. Many centrifugal chillers cannot operate below 40 percent rated capacity due to surging. This condition is discussed further in Chapter 5.

(3) Advantages.

(a) The chiller is responsive to changing loads and expansion. Since many C^3I rooms require 24-hr operation, it is more cost-effective to run the size of chiller needed rather than run a large chiller for a comparatively small load.

(b) If the cross-connection between the dedicated chiller system and the building chiller is used, this arrangement would provide an additional level of redundancy. That is, not only would there be redundancy with the dedicated C^3I room chillers, but if they were to fail, the building chillers could be used.

(4) Disadvantages. With more chillers and piping configurations, the costs for maintenance, monitoring, and control increase.

3-4. Piping Configurations—Inside the C^3I Room

This paragraph discusses various ways of piping chilled water inside the C^3I room. In chilled-water cooling systems, chilled water is commonly piped from a chiller to a remote air-handling unit and then directly to items in the C^3I room. If chilled water is piped to a remote air-handling unit, conditioned air is then supplied to the C^3I room using one or a combination of the air-supply systems discussed Chapter 2. If chilled water is piped to the C^3I room, one or a combination of the following may be served: (1) room air-handling units, (2) fan coil units, and (3) water-cooled electronic equipment.

a. Interior C^3I Room Piping.

(1) Operation. Figure 3-4 shows a common piping configuration used when chilled water is piped to the C^3I room.

(a) The branches could serve air handlers, fan coils, or water-cooled electronic equipment as noted above. The sectional valves and multiple valved branch connections permit modifications such as adding an air handler without requiring complete shutdown. Valve mains can also be added for cross-connection or expansion. Adding extra valves and branch connections will increase the life-cycle cost, but load redistribution and expansion capabilities are also increased.

(b) The piping could be above the C^3I room ceiling, but it is more commonly run under the floor when the C^3I room has a raised floor system.

(2) Principal Applications. This piping method is used when:

(a) There is not an abundance of computer wiring or other obstructions under the C^3I flooring.

(b) The depth of the raised flooring allows for underfloor piping without obstructing the computer room air-handler unit's air flow or interfering with computer wiring.

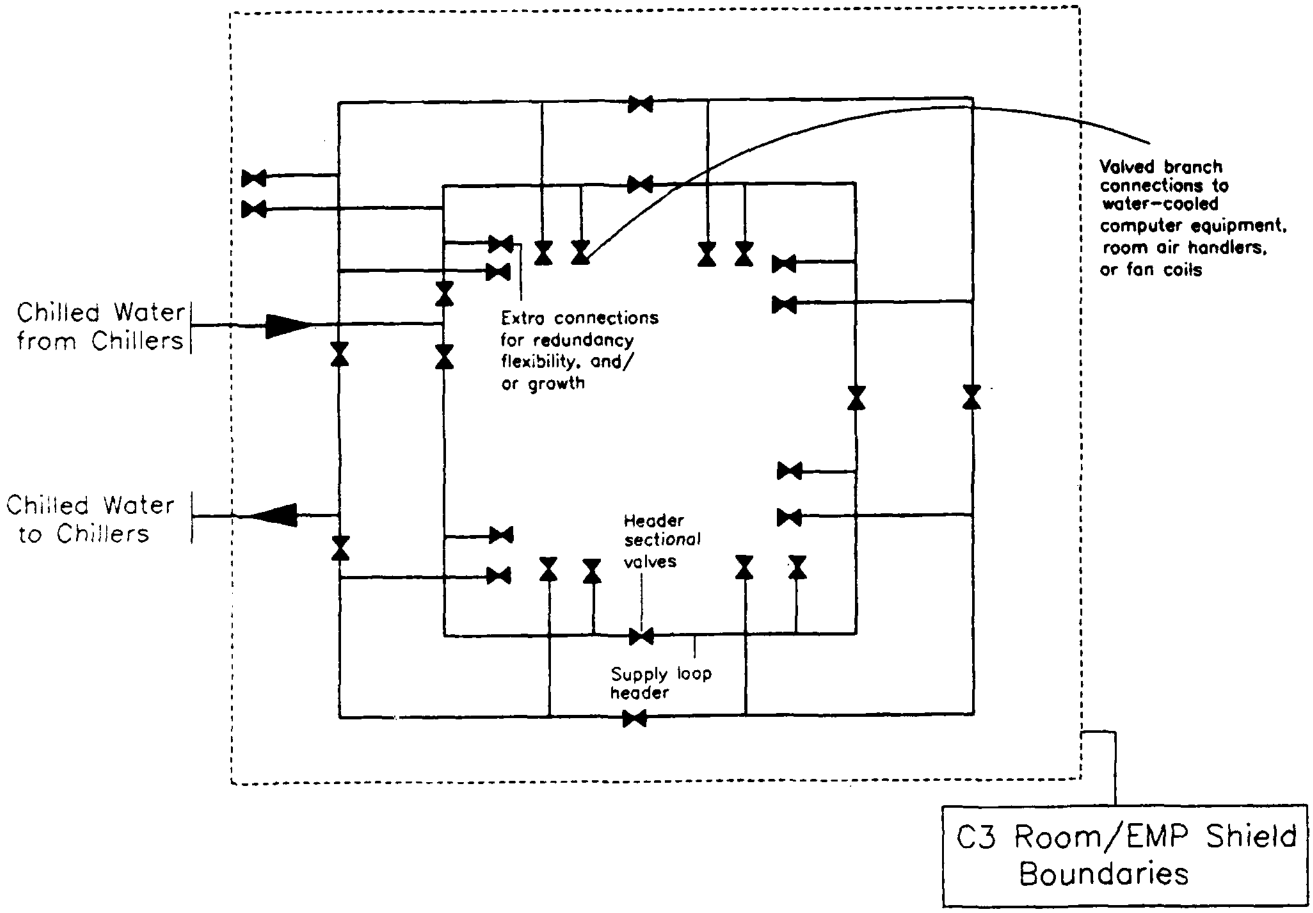

Figure 3-4. Interior C^3I room piping.

(3) Advantages. An advantage of this configuration is that the piping is under the floor and is easily accessible for hook-up of additional equipment.

(4) Disadvantages. If the C^3I room flooring is not deep enough, the piping may obstruct the air handler's air flow and interfere with the computer wiring. Also, locating water pipes under the floor increases the chance of water damage to the computer cables. If a leaky pipe or valve is not detected immediately, considerable damage could occur.

b. Perimeter C^3I Room Piping.

(1) Operation. A variation of this configuration is shown in Figure 3-5. Instead of the piping loop being located within the C^3I room, it is placed outside the room around the perimeter. The chilled-water lines penetrate the C^3I room walls where needed. All valves and branch connections are the same as in the previous case (para *a* above).

(2) Principal Applications. This piping arrangement is used when the raised floor is not deep enough to accommodate chilled-water piping loops. It is also used when the C^3I room is shielded and a perimeter must be placed around the room to allow inspections.

(3) Advantages. One advantage to this configuration is that there will be less piping under the raised floor. Less piping decreases the obstruction to air flow and improves underfloor air distribution if an underfloor plenum is used. Also, with less internal C^3I room piping, there is less chance for water to flood the room should a pipe or valve start leaking. Since many computer room air handlers are located around the perimeter of the room, a very short piping run is needed to reach the unit.

(4) Disadvantages. A disadvantage to perimeter piping is that it requires additional space. Also, if the room is EMP-shielded, this arrangement requires either a larger shielded area or EMP treatment for all of the penetration points. Both situations increase costs. This is not a disadvantage if the entire C^3I building is shielded since the room may not need to be shielded again.

c. Oversized Pipe.

(1) Operation. An option for C^3I room piping is to oversize the pipe to accommodate future growth.

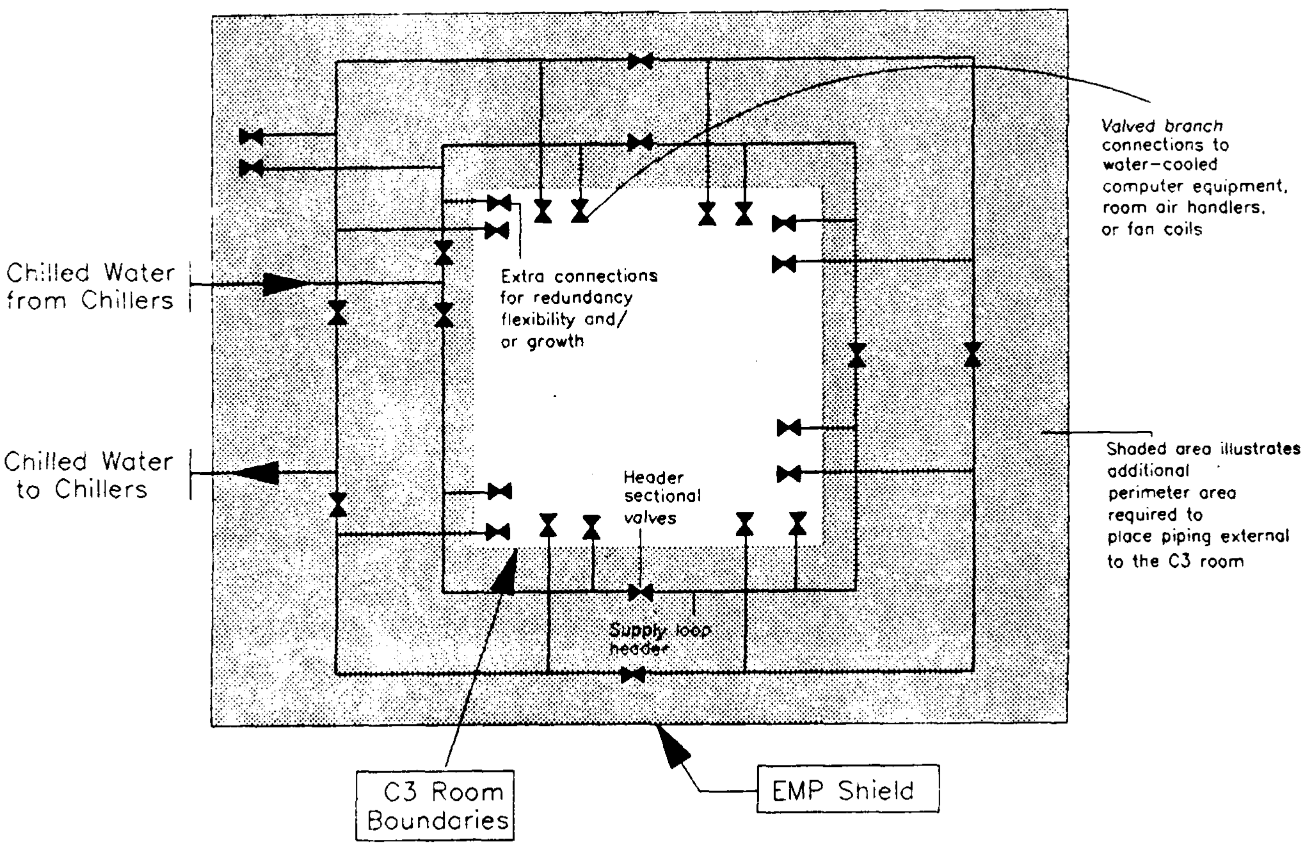

Figure 3-5. Perimeter C^3I room piping.

(2) Principal Applications. Oversized piping is used when larger loads are anticipated but are not yet present.

(3) Advantage. Additional piping is not needed should the load increase.

(4) Disadvantages.

(a) The first cost for the oversized piping is higher than for design load piping. Until the load is expanded to require the full capacity of the oversized pipe, throttling valves will be needed to reduce flow. These valves introduce additional pressure losses and add to electricity costs for pumping.

(b) For EMP protection, there is a 4-in. limit on pipe size unless a honeycomb filter is installed inside the pipe. Larger pipes will increase the cost and the pressure drop and thus pump size. Details on this subject are discussed in paragraph 3-7 below.

d. *Redundant Piping.*

(1) Operation. In contrast to larger piping, a dual-pipe arrangement can be used. The piping that leaves the chiller is divided into two equally sized pipes, each capable of handling the required C^3I room flow rates. The piping would enter the C^3I room and run parallel throughout the room.

(2) Principal Applications. This piping arrangement can be used if future growth is anticipated or when redundancy is required for a very critical operation.

(3) Advantages.

(a) If something should happen to an existing pipe or valve and the piping system needed to be shut down, the alternate piping arrangement could be used until the other is fixed. This redundancy affords little or no disruption to operations.

(b) Also, the additional piping system could be used when the first system is at full capacity and additional chilled water is needed to supply the C^3I loads.

(4) Disadvantages. A clear disadvantage is the extra cost for the redundant pipe. Another drawback is the extra space occupied up by the additional pipe. Also, if both pipes are used, there will be more pressure loss than in one oversized system for the same flow rate.

e. *Dedicated Equipment Chiller.*

(1) Operation. Most direct-cooled electronic equipment requires chilled water at a higher temperature than an air handler or fan coil. If a dedicated chiller is used, one chiller can be used for direct cooling of the C^3I electronic equipment while another is used for the air-handling equipment.

(2) Principal Applications. This arrangement can be used if there is an abundance of directly cooled electronic equipment and a tempering loop cannot accommodate the C^3I load.

(3) Advantages. If a dedicated chiller is used for the equipment and another chiller for the air-handling equipment, a tempering loop will not be necessary. Also, a chiller running at a higher temperature reduces operating cost.

(4) <u>Disadvantages</u>. An additional chiller would require more space and would also be an added expense.

f. Chiller Tempering Loop.

(1) <u>Operation</u>.

(a) If a dedicated chiller is not used, and the equipment requires elevated temperatures, a tempering loop can be used. Figure 3-6 shows a typical chilled-water tempering loop.

(b) The chilled water supply is blended with return water until the desired water temperature is met. Use of three-way valves to achieve mixing is not recommended since this would place the primary and secondary pumps in series. When pumps are in series, pressure changes in the primary loop affect the pressure of the computer room loop. These pressure changes are due to diversity or load fluctuations and will make it difficult to attain the setpoint temperature in a loop with a constant load.

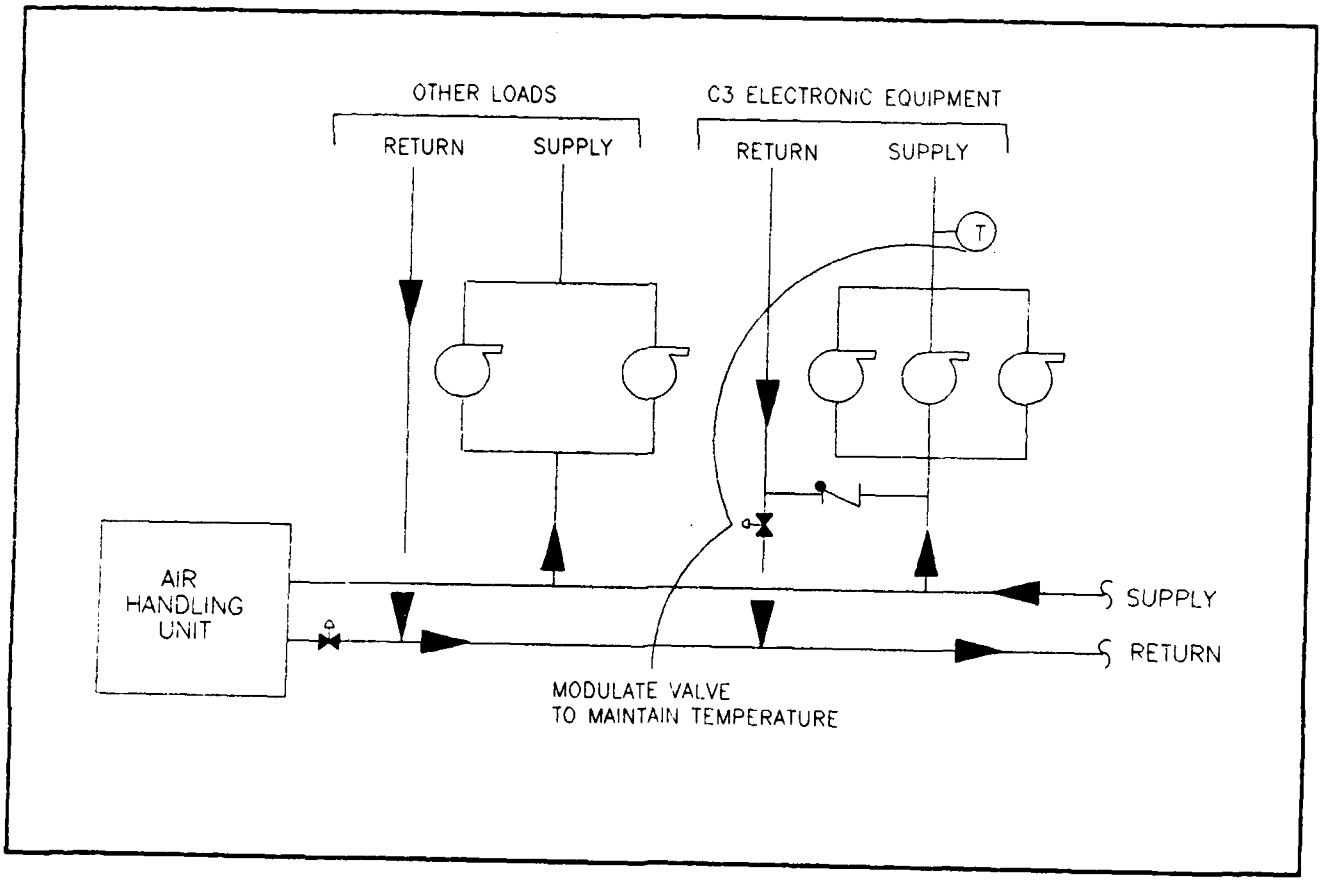

Figure 3-6. Chiller tempering loop.

(2) Principal Applications.

(a) This configuration is used when the C^3I equipment load is small and does not warrant a dedicated chiller. The tempering loop can be used for direct electronic equipment cooling and/or for raising the temperature of the chilled water for a "dew point control" system.

(b) A dew point control system is used to avoid the common problem of "fighting" between room floor-mounted air handlers due to each having its own reheat and moisture addition system for humidity control. One unit may be cooling for dehumidification and reheating to maintain room temperature while another is delivering full humidification. This wastes a tremendous amount of energy and reduces the lifespan of the equipment.

(c) The floor-mounted air-handling units are set for sensible cooling only while a central overhead unit delivers conditioned air based on the room dew point temperature. This method is effective since the computer room has a mostly sensible load.

(3) Advantages.

(a) When used to temper chilled water for the equipment, an advantage to this configuration is the savings from not having to run a dedicated chiller. Also, the space that the chiller would have occupied is freed.

(b) When this system is used for a dew point control system, wasted "fighting" among room air handlers is eliminated and the room temperature and humidity are maintained better. Also, if the room air handlers can be purchased without humidity and reheat controls, there will be a substantial savings in first cost as well as O&M costs.

(4) Disadvantages. If the amount of directly cooled equipment increases, the tempering loop may not accommodate the increased load.

3-5. 15-Min Backup Cooling Capacity

In the event of a power failure, critical water-cooled electronic equipment and air handlers serving the C^3I room must be provided with chilled water by a backup system for at least 15 min. During this time, diesel-fired emergency generators will be started and the chillers brought back online. Several strategies can be used for the backup cooling system. A gas engine-driven chiller (see para 5-7 in Chapter 5), powered by fossil fuel, could be used. However, some time is required to start the engine and bring the driven chiller online. The cost of the additional engine/chiller system compared with that of a chilled water storage system makes this alternative less attractive than others. The backup cooling system should provide an uninterrupted flow of chilled water to equipment and air handlers. The chilled water storage system described in this paragraph affords this capability. It provides 15 min of backup cooling quickly and cost-effectively.

a. Operation.

(1) Figure 3-7 shows a 15-min backup cooling system. Under normal operation, chilled water directly from the chillers or from a tempering loop (depending on equipment requirements) enters near the top of the coolant storage tank and exits near the bottom. The chilled water then bypasses a circulating pump by going straight through a diverting valve to the equipment. After cooling the equipment, the chilled water then

passes straight through another diverting valve and back to the chiller. If the power should fail, the diverting valves would cause flow to bypass the chillers and the circulating pump would circulate chilled water from the backup system through the tank to the equipment.

(2) The coolant storage tank is insulated and can withstand system water pressure. Having the chilled water enter at the top and exit at the bottom of the tank creates thermal stratification of the chilled liquid. The storage tank is sized to deliver 15 min of chilled water to the equipment should the power fail.

(3) The storage tank is usually sized for the maximum rise in chilled water temperature that the electronic equipment can tolerate. For example, an IBM 3090 mainframe computer requires chilled water at a temperature of 52 °F and can tolerate a temperature rise of no more than 4 °F. This means that after the 15 min of backup cooling has been provided, the tank outlet temperature must be no more than 4 °F above the normal chilled water supply temperature. When power is restored and the chillers come back online, chilled water will be supplied to electronic equipment and air handlers at this elevated temperature until the storage tank has been recharged. However, the equipment mentioned above can tolerate the 4 °F rise in chilled water temperature while maintaining satisfactory operation.

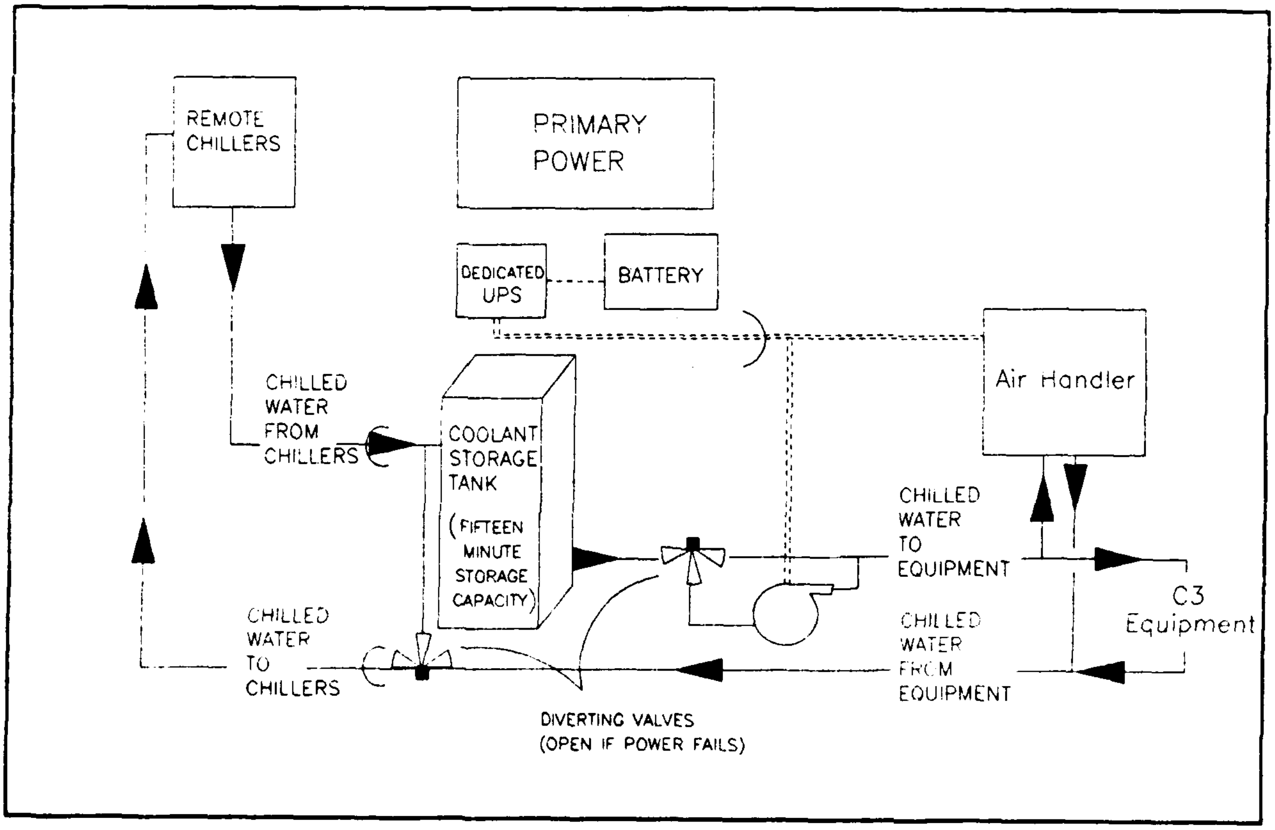

Figure 3-7. System for 15-min backup cooling capacity.

(4) By oversizing the storage tank, the rise in chilled water temperature can be minimized. Another strategy involves installing a second tank on the return side of the chiller. With this arrangement, when the chiller comes back on-line, the warm return water passes through the chiller before being supplied to the equipment and air handlers. The storage strategy used depends on the criticality of the mission, cost constraints, and the chilled water temperature requirements of the electronic equipment.

(5) The uninterrupted power supply (UPS) is dedicated to the backup system. The UPS detects a chiller power outage and sends power to the circulating pump and controls as soon as the primary power fails and the diverting valves are closed. If the chiller serving the C^3I room is in a different building and/or on a different circuit than the C^3I room, special wiring is required, or the coolant temperature and/or flow rate should be monitored instead.

(6) If the chiller is located near the C^3I room, the storage system controls consist of the motorized diverting valves and the UPS serving the pump, valves, and air handlers. The controls must be EMP-protected, and all penetrations caused by chilled water piping or control wires must be treated. The additional sensors and controls required when the chiller is located remotely add to the cost of the backup cooling system, as well as the EMP protection. The extra controls also add to the complexity of the backup system and therefore may have an impact on backup reliability.

(7) The circulating pump is sized for the backup circulating loop pressure drops. It is placed near the bottom of the coolant storage tank for net positive suction head (NPSH).

(8) Packaged electronic equipment chillers are available with "built-in" coolant storage systems that deliver backup cooling to equipment should the primary power fail. These storage modules are physically attached to the liquid chillers, which are located within the C^3I room. The storage modules are available only for small liquid chillers in capacities ranging from 2.5 to 15 tons. The storage modules range in cost from $250 to $700/chiller ton, depending on the size of the chiller. It should be noted that while the storage tanks are generally reliable, some designers avoid locating them inside the C^3I room due to the potential damage to electronic equipment should the tank leak.

b. *Principal Applications.* The main application for this backup system is to provide a minimum of 15 min backup cooling should the power fail and normal chiller operations lose power. Coolant must be maintained to the critical C^3I equipment.

c. *Advantages.* The clearcut advantage is the 15 min backup cooling should a power failure occur, which allows mission-critical operations to continue.

d. *Disadvantages.* Since chilled water flows through the coolant storage tank, higher pressure drops result, which require larger pump sizes. Also, after a power failure, the chiller must provide cooling while simultaneously recharging the storage tank. Thus, the chiller must be sized to handle the additional load imposed by storage recharging.

3-6. Piping System Components

Many system components can be used in a piping system, ranging from temperature gauges to automatic air vents, and check valves to double-suction centrifugal pumps. This paragraph concentrates on the three main components used in typical chilled-water piping applications: piping, valves, and pumps.

a. Piping. Water piping can be divided into three main materials: steel (black and galvanized); wrought iron (black and galvanized); and copper (soft and hard). Less common materials are iron and steel alloys, copper alloys, nickel and nickel alloys, and nonmetallic pipe. The properties of these materials are either not suitable or not cost-effective for water piping.

(7) Steel.

(a) Description. This material is usually black or galvanized. Galvanized pipe is hot-dipped zinc-coated steel pipe and reduces the potential for corrosion.

(b) Principal Application. Steel pipe is one of the most commonly used materials in piping systems. Schedule 40 black steel is usually used for chilled-water, cooling tower, or refrigerant applications. In C^3I applications where underground piping is necessary, a galvanized coated steel pipe can be used effectively to prevent external corrosion. Internal corrosion is not a factor in a closed-loop piping system as long as the water has been treated properly.

(c) Sizes. While normal pipe sizes range from 1/8 to 12 in. diameter, standard pipe sizes of up to 26 in. (od) are available.

(d) Cost. Based on 4-in.-diameter, schedule 40 pipe threaded with couplings and hangers, the cost for black steel is $9/lin ft (material only) or $23/lin ft (installed); the cost for galvanized steel is $10/lin ft (material only) or $24/lin ft (installed).

(e) Advantages.

(i) The chief advantage of steel pipe is its superior strength. Steel pipe has a higher tensile strength than any other standard construction material. In long vertical and horizontal runs, the structural strength and rigidity of steel pipe permit installation with minimum supports.

(ii) Steel also has very little sensitivity to fluctuations in temperature. Copper, for instance, expands about 50 percent more than steel across the temperature spectrum of 20 to 200 °F.

(iii) Steel pipe has proven reliable for up to 70 years. In many old buildings demolished in recent years, steel pipe was found to be in excellent shape and functioning well.

(f) Disadvantages. Steel is not very flexible for use in piping runs that require slight bending around obstacles, thus preventing a straight run. Ungalvanized steel pipe used in corrosive conditions will corrode over time.

(2) Wrought Iron.

(a) Description. Wrought iron is either black or galvanized.

(b) Principal Applications. Wrought iron can be used for chilled water, cooling towers, and refrigerants. Galvanized wrought iron can be used for underground water piping.

(c) Sizes. Normal wrought iron pipe is 1/4 to 12 in. diameter.

(d) Cost. Based on a 4-in. pipe, the cost of wrought iron pipe ranges from $18 to $28/lin ft installed.

(e) Advantages. The cost of wrought iron is relatively inexpensive compared with steel and copper.

(f) Disadvantages. Wrought iron is very hard. This property makes it susceptible to metal fatigue if constant vibration of water hammer occurs. Wrought iron is also heavier than steel and copper and, as such, has a slightly higher installation cost.

(3) Copper.

(a) Description. There are four main types of copper pipe: K, L, M, and DWV. Type K has heavy walls and is available in hard and soft temper. Type L has medium walls and also is available in hard and soft temper. Type M has light walls and is available in hard temper only. Type DWV has light walls and also is available in hard temper only.

(b) Principal Applications. Types K and L are commonly used in chilled-water applications. Copper pipe is usually installed when smaller diameter pipe is necessary. Large diameter copper pipe (above 4 in.) is extremely expensive and is not normally used except in special cases or when other types are not available.

(c) Size. For each type listed above, typical diameters range from 1/4 to 8 in.

(d) Cost. These costs are based on 4-in.-diameter copper pipe. Type K, L, M, and DWV copper pipe range from $38/lin ft to $28/lin ft installed, and from $14/lin ft to $7/lin ft for material only.

(e) Advantages. Copper pipe is very resistant to corrosion. Also, in smaller diameters, copper is very flexible. For example, it can be used when installing a ceiling-hung fan coil in a crowded ceiling where there are many obstacles. Steel pipe would require many elbows to achieve a fan coil.

(f) Disadvantages. Copper is very expensive compared with steel and wrought iron pipe.

b. Valves. The four main types of valves used in HVAC applications are: (1) gate, (2) globe, angle, or "Y," (3) check, and (4) butterfly. Although other types of valves are used in piping systems, these four are the most common. Water valves are usually made of brass, bronze, iron, or steel, and usually have working pressures of 125 psig or 150 percent of the system operating pressure, whichever is greater.

(1) Gate Valves.

(a) Operation. Figure 3-8 shows a typical rising stem gate valve. A gate valve is intended for use as a stop valve. It gives the best service when used in the fully open or fully closed position. With the valve wide open, the wedge or disk is lifted entirely out of the fluid stream, thus providing a straight flow area through the valve.

(b) Principal Applications. A typical location for gate valves is close to the chiller or air handler equipment to permit isolation. This arrangement enables the equipment to be serviced without having to drain or shut down the system.

(c) Sizes. Typical gate valves are 1 to 12 in. diameter; however, much larger valves can be obtained if needed. The 1-1/2-in. and smaller gate valves are usually bronze, whereas 1-1/2 to 2-1/2-in. gate valves can be bronze or cast iron. The 2-1/2-in. and larger gate valves are usually cast iron.

(d) Cost. Based on a 4-in. outside screw and yoke (OS&Y), threaded, iron body, class 125 lb, an installed gate valve is approximately $335, and material only is $260.

(e) Advantages. When gate valves are used in the fully open condition, there is less turbulence and obstruction to flow than with other valves. This results in a lower pressure drop than with other valves.

(f) Disadvantages. Gate valves cannot be used for throttling flow except in an emergency. Vibration, chattering, seat erosion, very high pressure drops, and wide variations in pressure drop would result if these valves were used for throttling.

(2) Globe, Angle, and "Y" Valves.

(a) Operation. Figures 3-9 through 3-11 show typical globe, angle, and "Y" valves, respectively. These three valves are grouped together since they have the same basic design, use, and construction.

(b) Principal Applications. These valves are used primarily for throttling service and allow close regulation of flow. They can be used for shutoff and balancing but are not ideal for this purpose. An angle valve can be used in a piping elbow, thus eliminating the need for an elbow.

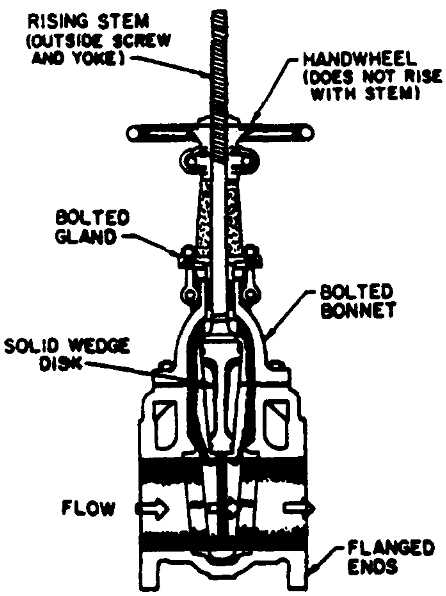

Figure 3-8. Gate valve.

(c) Sizes. Typical valves range from 1/8 to 12 in.; however, larger sizes may be obtained. The 1-1/2-in. and smaller valves are usually bronze. When 2 to 3 in., the valves can be bronze or cast iron. Four-inch and larger valves are usually cast iron.

(d) Cost. Based on a 4-in. diameter, OS&Y, class 125, threaded iron body, a typical globe valve costs approximately $600 installed and $500 for material only.

(e) Advantages. Unlike gate valves, globe, angle, and "Y" valves can be used for regulating flow as well as for shutoff.

(f) Disadvantages. In the full open position, globe valves create more turbulence and friction drop than gate, angle, and "Y" valves. Gate valves are more suitable for operation in the fully open position.

(3) Check Valves.

(a) Operation. Figures 3-12 and 3-13 show two basic types of check valves--swing and lift.

(i) The swing check valve can be used in a horizontal line. It can also be used in a vertical line for upward flow. The flow through the swing check is in a straight line and without restriction at the seat. Swing checks are generally used in combination with gate or butterfly valves.

(ii) The lift check operates much like a globe valve. The disk is seated by back-flow, or by gravity when there is no flow, and is free to rise and fall, depending on the pressure under it. The lift check is used only in horizontal piping, usually in combination with globe, angle, or "Y" valves.

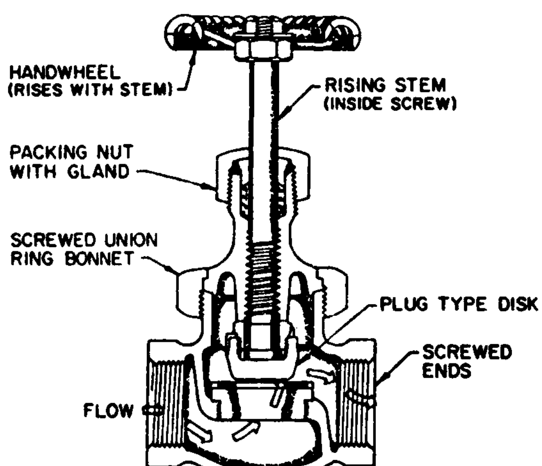

Figure 3-9. Globe valve.

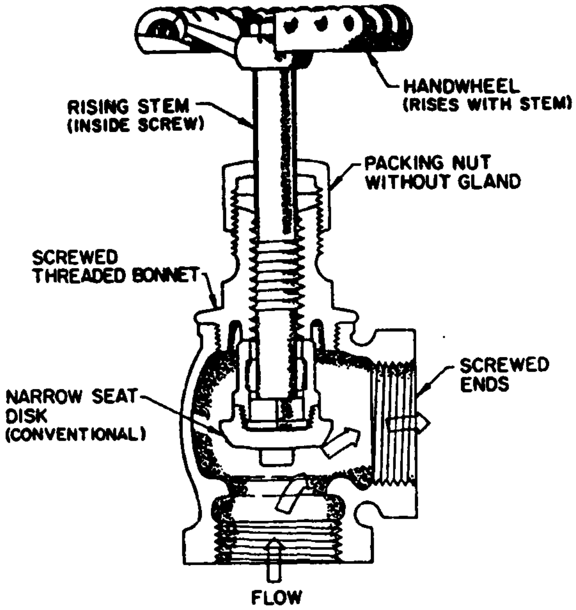

Figure 3-10. Angle valve.

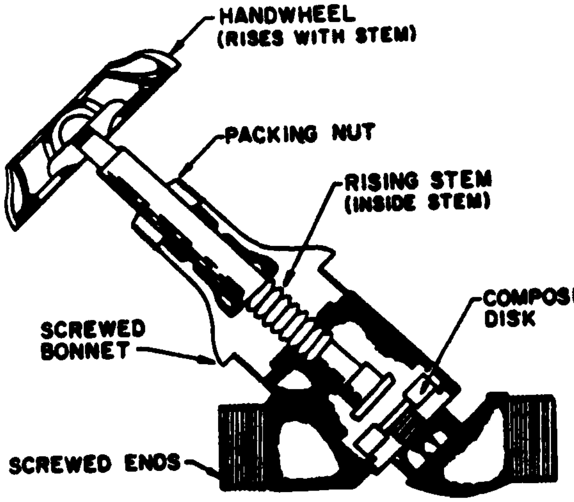

Figure 3-11. "Y" valve.

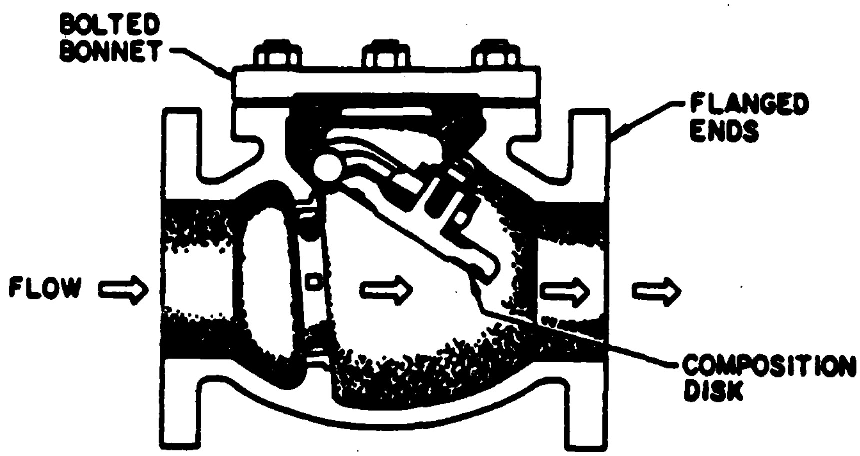

Figure 3-12. Swing check valve.

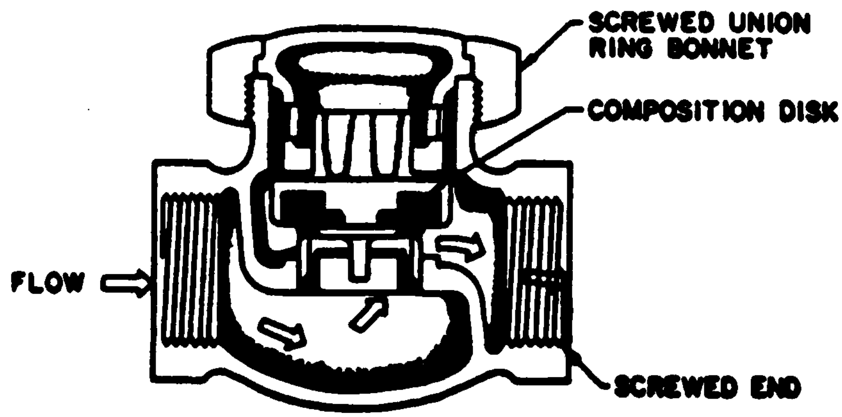

Figure 3-13. Lift check valve.

(b) Principal Applications. Lift (also known as nonslamming or silent) check valves are used in discharge connections for pumps, except for recirculating pumps, and are installed where required or recommended by the manufacturer of the connected equipment.

(c) Sizes. Typical check valves range in size from 1/8 to 12 in. diameter. The 1-1/2 in. and smaller check valves are usually bronze, whereas 2- to 3-in. valves can be bronze or cast iron. Check valves 4 in. and larger are usually cast iron.

(e) Cost. A 4-in., 125-lb, threaded, iron body swing check valve is $270 installed and $200 for material only. Silent check valves are $210 installed and $105 for material only.

(e) Advantages. The clear advantage is backflow prevention.

(f) Disadvantages. If a check valve is installed in a piping system where it is not necessary, pressure drop results.

(4) Butterfly Valves.

(a) Operation. A butterfly valve can operate as a stop valve or a throttling valve. Figure 3-14 shows a typical butterfly valve.

(b) Principal Applications. The use of butterfly valves is rapidly increasing for shutoff, throttling, and balancing of chilled-water flow applications. They may also be used in "tight" applications where space is limited.

(c) Sizes. In overall size, butterfly valves are generally smaller than gate or globe valves. Typical butterfly valve diameters range from 1/8 to 12 in.

(d) Cost. Based on a 4-in. iron body, a butterfly valve costs about $150 installed and $75 for material only.

(e) Advantages. Butterfly valves can be easily turned by hand. Also, they are very versatile since they can regulate, throttle, or balance flow. They are less costly

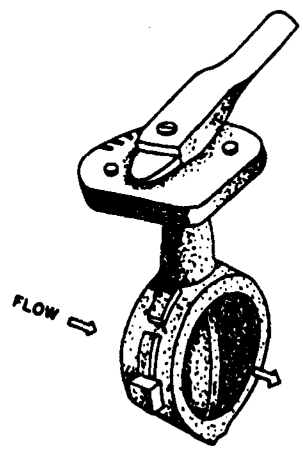

Figure 3-14. Butterfly valve.

than comparable valves, and better for use in tight areas where piping space is a premium since they take up very little space.

(f) Disadvantages. Butterfly valves have higher pressure drops in the fully open position than gate valves and occasionally they do not seat as well as gate valves in the shutoff mode.

c. Pumps. Centrifugal pumps are used in most heating and air-conditioning applications. Other types of pumps such as reciprocating, rotary, and jet are not as well suited for these applications. Therefore, this discussion is limited to centrifugal pumps. Centrifugal pumps have fewer moving parts than other pumps, making them more reliable. Also, they are usually less expensive and require less space. In addition, centrifugal pumps cover a larger range of operating speeds than other types.

(1) Several factors are involved in selecting a pump, including:

- The liquid type
- System head curve
- Pertinent local conditions such as continuous or intermittent service
- Indoor or outdoor operation
- Range of capacity requirement
- Range of pressure requirement
- Temperature
- Type of drive
- Number of pumps desired and percent of standby required for emergency operation
- Electrical service characteristics and controls.

Each pump has an electric motor with a nameplate horsepower rating not less than the actual break horsepower required to drive the pump at any head from zero to shutoff. Continuous duty pumps are used for 24-hr facilities like C^3I rooms and should be rated at 100 percent capacity.

(2) In C^3I applications, multiple parallel pumps are generally installed to provide the required redundancy. Figure 3-15 shows a typical parallel pumping arrangement when pumps are connected to the same header.

(3) The pumps selected to run in parallel must have shutoff heads in excess of the operating head when operating in parallel. If the pumps are supplying underfloor chilled water to water-cooled equipment, three parallel pumps powered by a UPS are commonly used.

(4) The use of multiple pumps operating in parallel is the most common method of eliminating system overpressure, which occurs in controlled flow systems where coils are equipped with two-way control valves.

(5) When multiple pumps are interconnected to the same header, each pump handles the same volume of water. Under partial load conditions, or when one pump is out of the line, the pumps still handle equal water volumes.

(6) Another method of eliminating system overpressure is to arrange the pumps in series. Series pumping can greatly reduce the overpressure in a controlled flow system. However, series pumping should not be used on hydronic systems with flat system head curves. In one-pump operation, a series connection would result in the pump running at shutoff head and producing no flow in the system.

(7) A pumping option that may prove beneficial to C[3]I applications is the use of variable-speed drives. With the proper controls, variable-speed drives allow the pump to follow the system head curve with no overpressure. Several variable-speed drives are available for operating centrifugal pumps in hydronic applications. These include fluid couplings, direct current, wound rotor, and eddy-current drives. A variable-speed drive is typically used when loads fluctuate. In many C[3]I applications, the load remains relatively constant.

(8) The five main types of centrifugal pumps used in cooling applications are:

(a) Circulator.

(b) End-suction (close-coupled and frame-mounted).

(c) Base-mounted, horizontal split case (single-stage double suction and multistage).

(d) Vertical inline.

(e) Vertical turbine.

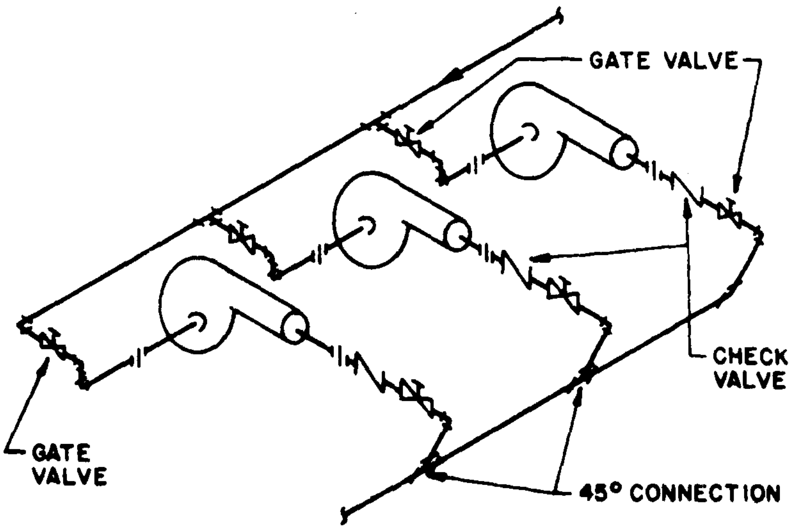

Figure 3-15. Multiple pump piping.

(9) Figure 3-16 shows each of these pumps. Centrifugal pumps are designated according to the:

(a) Type of impellers. The single-suction impeller has one suction or intake, whereas the double-suction impeller has two suctions or intakes.

(b) Number of impellers. Single-stage types have one impeller, whereas multistage models have several impellers.

(c) Type of casing. The volute types include all pumps that collect water from the impeller and discharge it perpendicular to the pump shaft. Diffuser-type casings collect water from the impeller and discharge it parallel to the pump shaft.

(d) Method of connection to the electric motor. The close-coupled pump has the impeller mounted directly on a motor shaft extension, whereas the flexible-coupled pump has an impeller shaft supported by a frame or bracket which is connected to the electric motor through a flexible coupling.

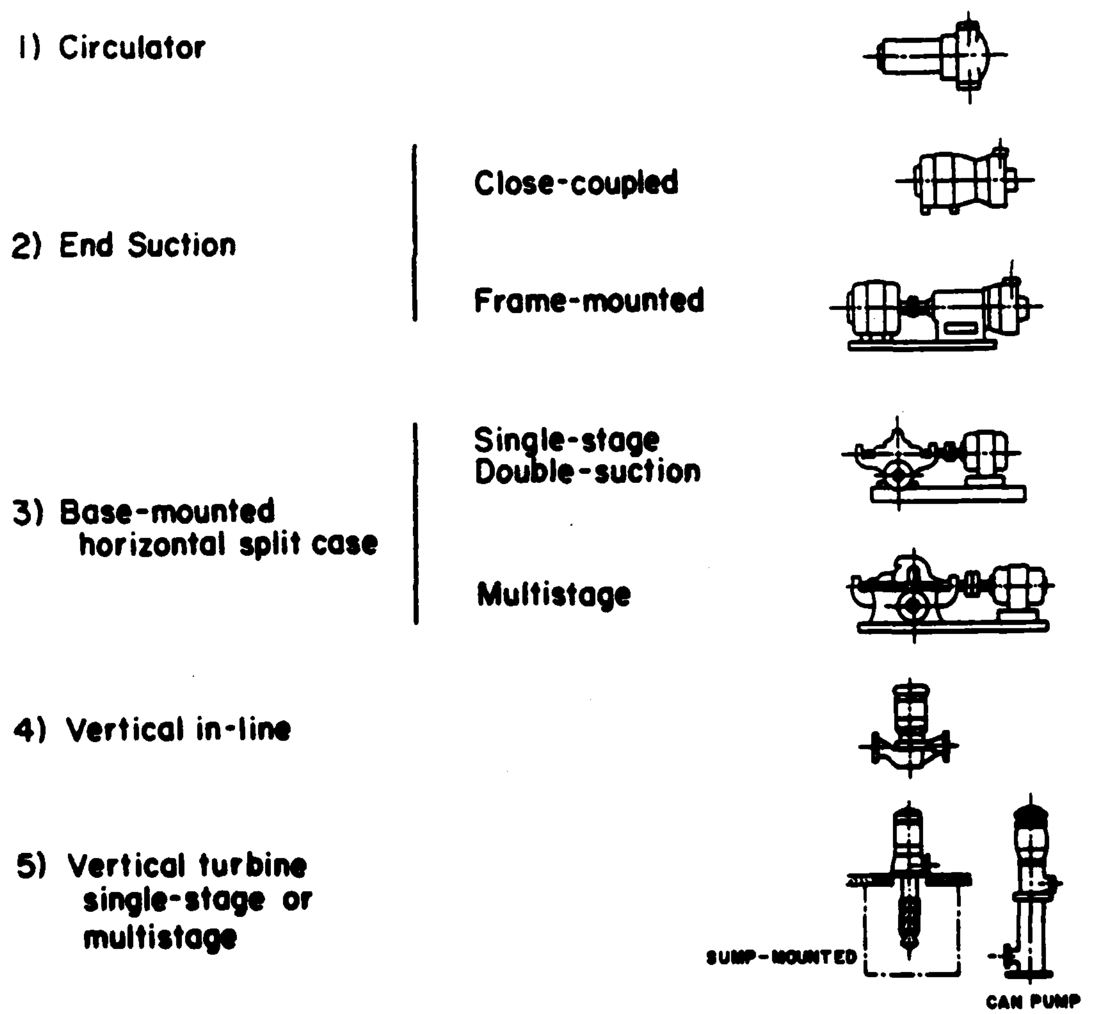

Figure 3-16. Centrifugal pumps used in hydronic systems.

(e) Mounting position. Pump motors can be mounted either horizontally or vertically.

d. Summary of Pump Types.

(1) <u>Circulator Pumps</u>.

(a) Operation. Circulator pumps are usually single-suction, with one impeller, a volute type casing, and a flexible-coupled horizontal motor.

(b) Principal Applications. Some typical applications are residential hydronic systems, domestic hot-water recirculation, and multizone circulation. These pumps are also used in small chilled-water applications.

(c) Sizes. Circulator pumps range in size from 1/4 to 3 HP.

(d) Cost. A 1/2-HP circulating pump costs approximately $700 for material and $900 installed.

(e) Advantages. Circulator pumps are ideal for small pumping applications where low capacities and heads are needed.

(f) Disadvantages. Circular pumps cannot be used for medium to large capacities. The range of pumping capacities and heads is relatively small.

(2) <u>End-Suction Pumps (Close-Coupled, Frame-Mounted)</u>.

(a) Operation. The two basic types of end-suction pumps are close-coupled and frame-mounted. Both have single suction, one or two impellers, a volute-type casing, and a horizontal motor mounting position.

(b) Principal Applications. Some typical end-suction pump applications include cooling towers, condenser water, chilled water, hot water, boiler feed water, condensate return, booster, and irrigation.

(c) Sizes. Capacities are to at least 3500 gallons per minute (GPM) and heads to 500 ft. Typical discharge is 4 to 8 in. in diameter.

(d) Cost. A typical 10-HP end-suction, frame-mounted, single-stage pump costs approximately $2700 for material only. The installed cost is $4200.

(e) Advantages. End-suction pumps are ideal for chilled-water applications that are too large for a circulator pump to handle.

(f) Disadvantages. Like the circulator pump, end-suction pump capacities and heads are relatively low compared with other centrifugal pumps.

(3) <u>Base-Mounted, Horizontal Split-Case Pumps (Single-Stage Double-Suction and Multistage Double-Suction)</u>.

(a) Operation. Base-mounted horizontal split-case pumps are either single-stage, double-suction, or multistage double-suction. They have a volute-type casing and a flexible coupled, horizontal motor.

(b) Principal Applications. Typical applications include cooling towers, condenser water, chilled water, hot water, boiler feed, condensate return, irrigation, and fire protection. These pumps are commonly used in large HVAC applications.

(c) Sizes. Discharge sizes are typically 1 to 14 in., and can be up to 35 in. in diameter. Capacities are to 70,000 GPM, and heads to 500 ft.

(d) Cost. The typical cost for a 10-HP single-stage double-suction pump is about $3400 for material only and $5400 installed.

(e) Advantages. Base-mounted horizontal split-case pumps are ideal for large applications. They are more reliable and easier to maintain than end-suction pumps. The pump can be maintained or repaired without disturbing the piping.

(f) Disadvantages. Horizontal split-case pumps are not very efficient for small capacities.

(4) Vertical Inline Pumps.

(a) Operation. Vertical inline pumps are usually single-suction with one impeller. They have a vertical motor position and can be flexible or close-coupled.

(b) Principal Application. Typical applications are for cooling towers, condenser water, chilled water, hot water, boiler feed, and condenser return.

(c) Sizes. Typical discharge sizes are 1 to 5 in. diameter.

(d) Cost. A typical inline 1-HP vertical pump costs $700 for material and $900 installed.

(e) Advantage. Vertical inline pumps are ideal for small applications where water circulation is continuous.

(f) Disadvantage. A disadvantage of the vertical inline pump is the relatively small capacities compared with end- and double-suction pumps.

(5) Vertical Turbine Pumps.

(a) Operation. Vertical turbine pumps are usually single-suction with 1 to 20 impellers. They have a diffuser-type casing and a flexible-coupled vertical motor.

(b) Principal Applications. These pumps are used in chilled water, hot water, boiler feed, condensate return, and booster applications. They are also commonly used in cooling towers and fire protection services, where high capacities and heads are needed.

(c) Sizes. Typical sizes range from 6 to 57 in. discharge diameter with capacities to 40,000 GPM and heads to 1000 ft.

(d) Cost. A 10-HP vertical turbine pump costs about $4000 for material only and $6000 installed.

(e) Advantage. Vertical turbine pumps are very effective in applications where large heads and capacities are necessary.

(f) Disadvantage. They are not very efficient for smaller applications.

3-7. Other Considerations

a. Piping penetrations should be protected properly to avoid degrading the shielding effectiveness below minimum requirements. Three types of piping penetrations need to be considered:

(1) A metallic pipe that carries a conducting fluid (e.g., steel chilled-water pipe).

(2) A metallic pipe carrying a dielectric fluid (e.g., steel air pipe).

(3) A dielectric pipe carrying a dielectric fluid (e.g., plastic air hose).

b. A fourth possibility, a conducting fluid in a dielectric pipe (or dielectrically lined metal pipe such as glass-lined steel), cannot be EMP-protected properly and must be avoided.

c. The waveguide-beyond-cutoff principle is used for piping penetrations. For a metal pipe, a circumferential weld to the primary shield is required so that current flowing in/on the pipe can be discharged onto the outer surface of the shield. Figure 3-17 shows a metallic pipe penetration. The interior pipe wall serves as the waveguide. The inside diameter (d) should be less than or equal to 4 in. to provide a cutoff frequency of 1.73 GHz. The continuous length (ℓ) adjacent to the penetration must be a minimum of five diameters to attenuate by at least 100 dB at the required frequencies. If a pipe greater than 4 in. id is required, protection options include:

(1) Subdivide into two (or more) pipes that satisfy the dimensional requirements.

(2) Insert a honeycomb filter inside the pipe and treat the penetration as described for air distribution penetrations in Chapter 2. If this method is used, the higher pressure drops incurred must be taken into account when sizing the chilled-water pump.

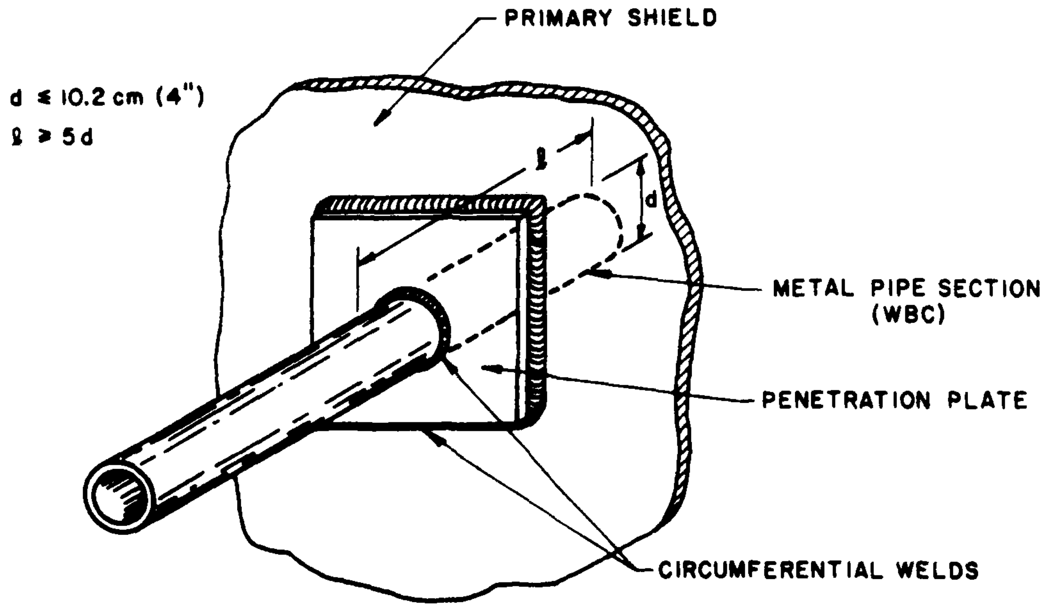

Figure 3-17. Metallic pipe penetration.

d. If a dielectric pipe with dielectric fluid penetrates through a metal waveguide sleeve, the sleeve has the same dimensional requirements that apply to metallic pipe. Figure 3-18 shows a dielectric pipe penetration. It is common practice to fill the waveguide sleeve with epoxy or similar material to prevent the insertion of conductors (e.g., conducting fluids, wires, pull wires). Dielectric pipes greater than 4 in. id follow the same guidelines as metal pipes greater than 4 in. as discussed earlier.

e. The cost to provide EMP shielding for a penetration of a 4-in. steel chilled-water pipe is estimated at approximately $45. If chilled-water flow requires a 5-in. pipe, one of two methods must be followed to provide EMP protection: (1) Subdivide into two 3.5-in. pipes to carry the same flow and satisfy the waveguide dimension restrictions, or (2) attach a honeycomb section (described in para 2-3) within the pipe. Subdividing the 5-in. pipe into two 3.5-in. pipes doubles the cost of protecting the shield's effectiveness. Inserting a honeycomb section into the 5-in. pipe increases the cost about 40 percent over that associated with the circumferential weld method alone. Again, the pressure drop caused by the honeycomb insert must be taken into account when sizing the chilled water pumps.

f. Other considerations in designing piping penetrations include: (1) provisions to accommodate condensation when the piped fluid is cold and (2) pipe supports installed in a such a way as to not create shield penetrations.

g. Chilled liquid distribution controls include motorized valves, pump motor control centers, and pressure and flow sensors. These controls must be protected to ensure their operation during an EMP event. The basic protection method is to surround the controls with a separately grounded conducting barrier. Nonelectrical penetrations must be treated with the methods described earlier. Electrical penetrations can be treated with the various filter and surge limiting strategies described in the EMP literature cited in the **Bibliography.**

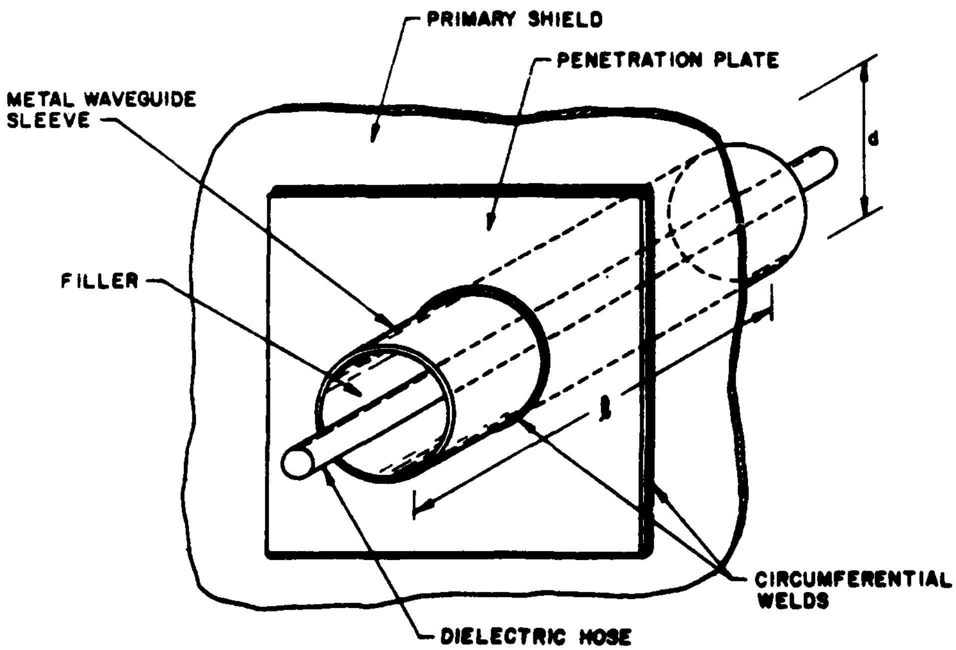

Figure 3-18. Dielectric pipe penetration.

Section III: SUMMARY EVALUATION

3-8. How To Use the Evaluation Table

Table 3-1 ranks each piping configuration for all criteria except the 15-min backup cooling configuration. Although other methods can provide backup cooling, the chilled-water storage system is preferred due to its cost, reliability, and timeliness of switch-over.

a. Table 3-1 is divided into two sections: piping configurations outside the C^3I room and those inside the C^3I room. The three piping configurations outside the C^3I room are:

(1) A central chiller serving C^3I room, C^3I building, and other buildings,

(2) A chiller serving C^3I room and C^3I building,

(3) A dedicated chiller serving the C^3I room and another serving the rest of the building.

b. The piping configurations inside the C^3I room are divided into the following three subsets:

(1) Subset 1--

(a) C^3I room inside loop piping.

(b) C^3I *room perimeter loop piping.*

(2) Subset 2--

(a) Oversized piping.

(b) Redundant piping.

(3) Subset 3--

(a) Dedicated chiller for direct equipment cooling.

(b) Chiller tempering loop for direct equipment cooling.

c. Each of the three subsets are discussed and ranked separately. A comparison between subsets would not be applicable since the three configurations serve different purposes.

d. Each configuration is ranked using the criteria listed in **Section I**. These rankings are qualitative, with a black circle representing a high ranking in the table, a gray circle denoting a medium ranking, and a white circle indicating a low ranking. A low ranking simply means that other configurations are more suitable for a specific criterion, and not that the configuration is a poor choice.

Table 3-1

Summary Evaluation—Chilled Liquid Distribution Systems

PIPING CONFIGURATIONS	Load redistribution	Load expansion	Backup cooling	Reliability	* EMP shielding cost	* Life-cycle cost
Outside the C3 Room						
1) Central Chiller	◍	◍	○	○	◍	◍
2) Building Chiller	◍	◍	◍	○	◍	◍
3) Dedicated Chiller	●	●	●	●	◍	●
Inside the C3 Room						
1) C3 Room Piping	○	○	◍	○	●	●
2) C3 Room Perimeter Piping	●	●	●	●	○	○
1) Oversized Piping	○	●	○	○	●	●
2) Redundant Piping	○	○	●	●	○	○
1) Dedicated Equipment Chiller	●	●	●	●	◍	◍
2) Chiller Tempering Loop	○	○	○	○	◍	●

● = High ◍ = Medium ○ = Low

* A high ranking for cost-related criteria indicates a low relative cost. A low ranking indicates a high relative cost.

3-9. Evaluation Results

a. Load Redistribution.

(1) Outside the C^3I Room. When flexibility in meeting load redistribution is the primary consideration, the dedicated chiller serving the C^3I room, with another serving the building, has the highest ranking. With a multiple chiller arrangement, the dedicated chiller should easily accommodate varying loads. A chiller that is farther away from the C^3I room requires longer piping and extra connections. This design would therefore diminish the response time to varying loads. The central chiller must not only allow for the normal temperature rise between chiller and processor, but also must take into account outside influences that may change throughout the year.

(a) If the load does change at the equipment, it may take some time for the chiller to react and temperature fluctuations may result as the chiller seeks the correct level. This situation could cause internal condensation from too low a water temperature or shutdown if temperature raises too much.

(b) If the total C^3I room load remains relatively constant while the locations of the loads within the room change, all three cases would be about equal. This problem would be handled by the air distribution system.

(2) Inside the C^3I Room. When flexibility in meeting load redistribution is the primary consideration, piping around the perimeter has the highest ranking. Perimeter piping offers high flexibility since there are fewer underfloor obstructions than if the piping loops were run under the C^3I floor. For example, if a load redistribution causes the need to relocate a room air handler, the existing piping can be capped (for future use) inside the room near the wall penetration point. The new piping can penetrate the C^3I wall near the new air handler location. There will be relatively little piping obstruction under the floor. If the load redistribution does not require repiping, all cases will be about equal since the redistribution will be done by the air distribution system.

(a) Oversized piping or redundant piping rank relatively equal when considering load redistribution. Either piping configuration could be used with the same results. Thus, both oversized piping and redundant piping configurations are given a medium ranking.

(b) For dedicated chillers or tempering loops, when there is a constant total C^3I room load, both cases are relatively equal. However, if the total load fluctuates, the dedicated equipment chiller would have the best ranking since it would be more responsive to temperature fluctuations.

b. Load Expansion.

(1) Outside the C^3I Room. When flexibility in meeting load expansion is the primary consideration, the dedicated chiller serving the C^3I room, with another serving the building, has the highest ranking. For the same reason as stated for load redistribution, a dedicated equipment room chiller is much more responsive to changing room requirements. Not only is the central building chiller more difficult to connect initially, but future alterations for additional hardware are costly in terms of both time and money. Rigid piping connections, as well as intricate adjustments in flow rates and pressure, must all be changed when new equipment is added.

(2) Inside the C^3I Room. As in the case for load redistribution, perimeter piping has the highest ranking where flexibility in load expansion is concerned. If an additional air handler needs to be connected, the connections can easily be piped from the perimeter loop closest to the new location.

(a) Oversized piping has the highest ranking for load flexibility since only a throttling valve adjustment would be required to increase chilled water capacities.

(b) A dedicated equipment chiller has a higher ranking over a tempering loop configuration. The dedicated chiller requires fewer control adjustments and, therefore, accommodates load expansion better than a tempering loop.

c. 15-Min Backup Cooling Capacity.

(1) Outside the C^3I Room. When the capacity to provide 15 min of backup cooling is the primary consideration, the dedicated chiller serving the C^3I room, with another serving the building, has the highest ranking. If the chiller serving the C^3I room is in a different building or a considerable distance away from the C^3I room, special wiring is required to achieve the necessary feedback to the backup control system.

(2) Inside the C^3I Room. A perimeter piping loop has a higher ranking than an underfloor piping loop inside the C^3I room. The backup cooling system will occupy valuable space if it is located under the floor. If it is located around the perimeter, not only will less underfloor space be taken, but the backup system can also be inspected easily.

(a) A redundant piping arrangement has a higher ranking than an oversized piping configuration. Since chilled water flows continuously through the backup loop, it will be more costly to pump water through a larger pipe than a smaller one. With redundant piping, one of the two pipes can be used, each smaller than the oversized pipe.

(b) A dedicated equipment chiller has a higher ranking than a tempering loop. Since the power to the chiller needs to be monitored in the 15-min backup case, a dedicated chiller would require less control wiring since it would be closer to the backup configuration.

d. Reliability.

(1) Outside the C^3I Room. When reliability is the primary consideration, a dedicated chiller serving the C^3I room, with another serving the building, has the highest ranking. This case provides for additional chillers and, thus, increases the reliability of maintaining the supply of chilled water to the C^3I room. An ideal setup would have the C^3I room chillers backed up by the building chillers.

(2) Inside the C^3I Room. A perimeter piping chilled-water loop has a higher rating than inside room loop. Since perimeter piping can be inspected easily, a higher degree of reliability is offered.

(a) The redundant pipe configuration has a higher ranking than the oversized pipe. Since the redundant pipe configuration has an "extra" pipe, the reliability is better. If some component were to malfunction in oversized piping, there would be no redundancy.

(b) A dedicated equipment chiller is ranked higher than a tempering loop. Since the dedicated equipment chiller requires fewer control setups than a tempering loop, it will have a slightly higher degree of reliability.

e. Least Cost for EMP Shielding.

(1) Outside the C^3I Room.

(a) When least cost for EMP shielding is the primary consideration, all three configurations are basically equal. In each configuration, chilled water and ducting will penetrate the C^3I room shield. If there are no plans for personnel to occupy the C^3I room, then the piping would be the only HVAC penetrations since ducts to supply air would not be required.

(b) If the entire C^3I building is EMP-protected and the chiller is remotely located, then the chiller will need to have EMP shielding. In this case, the building, or dedicated, chiller will have a higher ranking than a central chiller.

(2) Inside the C^3I Room. Piping inside the computer room has a higher ranking than the perimeter piping configuration. Perimeter piping requires either a larger shield or many penetration points. If the shield is exterior to the C^3I room and the perimeter loop, it will need to be much larger and, of course, more expensive. If the C^3I room is shielded, the perimeter pipe penetrations will require shielding. This will also raise the price considerably.

(a) Oversized piping is ranked higher than redundant piping. The cost to penetrate an EMP shield with one large pipe having a honeycomb insert is some 30 percent less than penetrating the shield with two smaller pipes without honeycomb inserts. Taking into account the larger pump needed to accommodate the additional pressure drops created by the honeycomb insert, and the added maintenance for the honeycomb insert, the two methods are fairly equivalent.

(b) A dedicated chiller has basically the same rating as a tempering loop. In both cases, piping penetrates the shield and is approximately the same size. Therefore, the EMP cost will be the same.

(c) If the entire C^3I building is shielded, the EMP shielding cost will not be a consideration for these cases inside the C^3I room.

f. Lowest Life-Cycle Cost.

(1) Outside the C^3I Room.

(a) When the lowest life-cycle cost is the primary consideration, a dedicated chiller serving the C^3I room, with another serving the C^3I building, has the highest rating. In this case, a high rating means a low life-cycle cost. Conversely, a low rating indicates a high life-cycle cost.

(b) A remote chiller would have to be run 24 hr/day to meet the requirements of a C^3I room. Running a large chiller for long periods of time to supply the relatively small cooling needs of the C^3I room is inefficient and expensive. With a building chiller, the energy consumption of the pumps alone may exceed the total energy use of a dedicated chiller.

(2) Inside the C^3I Room. The inside room loop is rated higher than the perimeter piping loop. Since there is relatively no operating cost difference between the two cases, the initial cost is the basis for comparison. More piping will be required in the perimeter case (since the loop is much larger) than if the piping loop were inside the C^3I room.

(a) Oversized piping has a higher rating than the redundant piping configuration. A redundant piping arrangement will cost more than an oversized piping layout.

(b) A chiller tempering loop has a higher rating than a dedicated chiller. The cost of a dedicated chiller is much greater than that of a tempering loop.

4 AIR-HANDLING UNITS

Section I: GENERAL

This chapter evaluates design approaches for air-handling units capable of supplying air to the air distribution systems described in Chapter 2. The following criteria are used to evaluate each type of air handler:

1. Flexibility in meeting load redistribution.

2. Flexibility in meeting load expansion.

3. Precise humidity control.

4. Filtering of particulate matter.

5. Static electricity control.

6. Capacity to provide 15 min of backup cooling.

7. Reliability.

8. Least cost for EMP shielding.

9. Lowest life-cycle cost.

Some criteria do not apply to air-handling units. In these cases, "N/A" appears in the evaluation table listing the results of the ranking procedure.

4-1. Types of Air Handlers

a. Air-handling units are sold in a wide variety of sizes and capacities. Factory-assembled central station air handlers are available with supply air flow rates up to 65,000 CFM, while field-erected units can supply air at more than 100,000 CFM. At the other extreme, individual fan coil units have air flows as low as 100 CFM. This report concentrates on the four main types of air-handling units used in C^3I rooms:

(1) Central station air-handling units.

(2) Chilled-water packaged units.

(3) Self-contained packaged units.

(4) Fan coil units.

b. In most practical applications, the C^3I room will be served by a combination of the air handler types evaluated. For instance, a central station air handler will provide outside air and dehumidification/humidification functions, and floor-mounted packaged units will provide sensible cooling. The dew point control system mentioned in Chapter 3, and further described in this chapter, is an example of such a system.

Section II: SYSTEM DESCRIPTION

4-2. Air Handlers for C³I Rooms

a. Electronic equipment rooms often must have more than one type of air-handling system to satisfy the redundancy, temperature, and humidity requirements of C^3I rooms. However, to be consistent with the subsystems approach used in these guidelines, each air-handling unit is described and evaluated separately.

b. The self-contained packaged unit differs from the other air handlers evaluated in that it incorporates refrigeration equipment within the unit. C^3I room air is cooled by blowing it over a DX refrigerant coil. Due to the refrigeration equipment, the self-contained packaged units have a higher life-cycle cost than any of the other air handlers evaluated. Since none of the other air handlers have refrigeration equipment, self-contained packaged units cannot be compared equally on the basis of life-cycle cost. However, the units are ranked under the other criteria.

4-3. Central Station Air-Handling Units

a. Operation.

(1) A general equipment arrangement for components of a typical central air-handling unit (AHU) is shown in Figure 4-1. Although the diagram indicates a built-up system, most of the components are available completely assembled by the manufacturer or in subassembled sections that can be bolted together in the field. Figure 4-2 shows an actual central AHU.

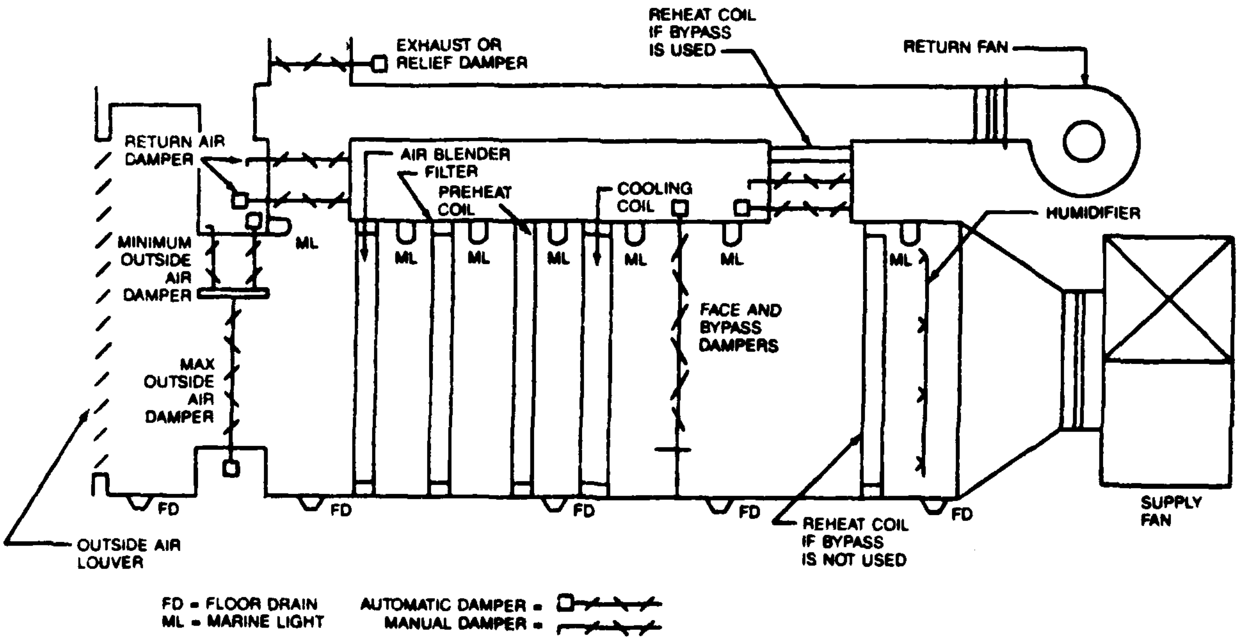

Figure 4-1. Equipment arrangement of a central air-handling unit.

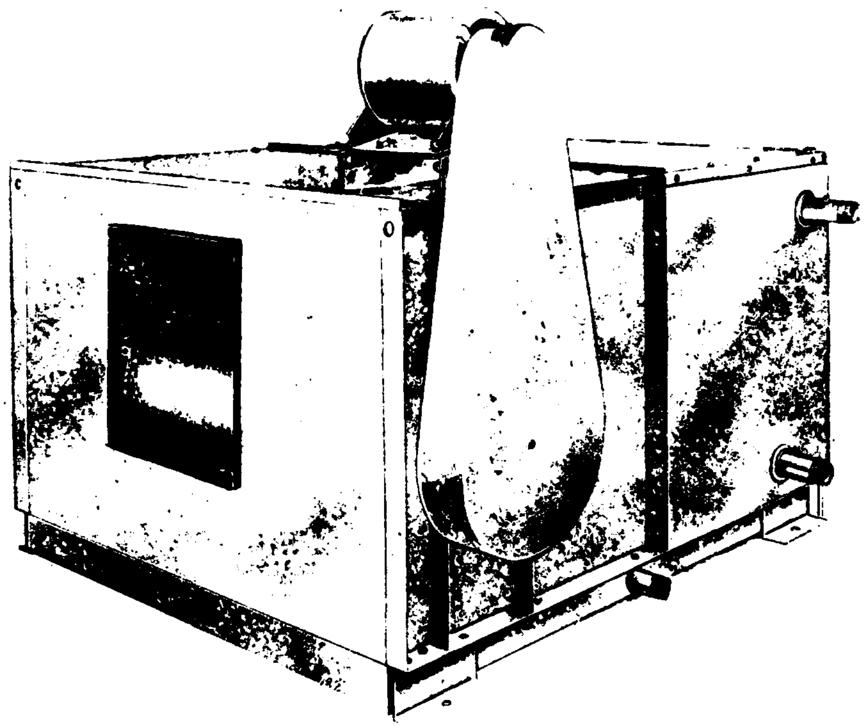

Figure 4-2. Central air-handling unit.

(2) A central station AHU typically consists of:

(a) Return and supply fans.

(b) Automatic dampers.

(c) Relief openings.

(d) Return air dampers.

(e) Outdoor air intakes.

(f) Mixing plenum.

(g) Air filter section.

(h) Preheat coil.

(i) Cooling coil.

(j) Bypass section.

(k) Reheat coil section.

(l) Humidifier.

(3) The return air fan ensures that the proper volume of air returns from the space. It also reduces resistance to the supply fan. The return air fan should handle a slightly smaller quantity of air than the supply air fan to ensure a positive pressure in the C^3I room.

(4) The supply air fan can be axial flow or centrifugal. Large systems usually are centrifugal. The supply fans must be matched carefully with the return fans in VAV systems to ensure proper distribution pressures. The basic central AHU can be designed to supply a variable or a constant air volume. It can supply the air for low-, medium-, and high-pressure air distribution.

(5) The automatic dampers can be arranged with opposing blade dampers for the outdoor, return, and relief air streams. This design allows a very high degree of control.

(6) Relief openings for large facilities must be constructed similar to outdoor air intakes. The main difference is that they should have backdraft dampers to prevent high wind pressures or stack action from causing the air flow to reverse when the automatic air dampers are open.

(7) Return air dampers must accommodate the pressure drop across the damper plus the difference between the positive pressure-relief air plenum and the negative pressure in the outdoor air intake plenum. The positive pressure in the relief air plenum is a function of the static pressure loss through the exhaust or relief damper, the exhaust duct between the plenum and outside, and the relief louver. The negative pressure in the outdoor air intake plenum is a function of the static pressure loss through the outside air louvers, dampers, and ducts.

(8) Outdoor air intakes should have a minimum pressure loss of less than 0.10 in. of water. High-efficiency, low-pressure loss louvers that effectively limit carryover of rain should also be used.

(9) Mixing plenums are provided to mix the outside and return air. Return air dampers should be set so that any deflection of air is toward the outside to maximize turbulence and mixing. While opposed blade dampers offer better control, properly proportioned parallel blade dampers are more effective than opposed blade dampers for mixing air streams of different temperatures. Dampers must be arranged to achieve proper mixing, regardless of the final type and configuration. Otherwise, the preheat coil will waste heat or the cooling coil may freeze.

(10) Air filter sections contribute much to system performance. A dirty filter increases system resistance, diminishes air flow, and lowers the life of the fan equipment. Primary considerations with filters are selection and location. The filter must be easily accessible for cleaning or replacement. A roller-type filter can be used that automatically rolls new filter media into place when the existing filter section becomes dirty.

(11) Preheat coils should be protected by filters, have wide fin spacing, and be accessible for easy cleaning. The coils are usually heated with steam or hot water. If steam is used, an inner distribution tube or integral face and bypass coils are preferable. If hot water is used, a constant flow recirculating pump should be used with parallel piping so that the coldest air will contact the warmest coil surface first.

(12) Cooling coils remove sensible and latent heat from the air. For close control of room conditioning, as in C^3I rooms, a deep coil with spray or an air washer might be

required. A copper fin coil is required for sprays. Coil freezing can be a serious problem with chilled water coils. Full-flow circulation of chilled water during freezing weather diminishes the chances of freezing. Antifreeze solutions or complete coil draining also can prevent coil freezing.

(13) The bypass sections mix air for a third time. At reduced internal loads, the room thermostat opens the bypass damper, which permits return air to enter the bypass section. At the same time, the face damper on the cooling section closes and reduces the flow of cooled and dehumidified air. The temperature of the mixed air rises, and overcooling is prevented. Since there is a large pressure drop across the bypass damper, it has heavy leakage when closed. The apparatus dew point can be lowered to compensate.

(14) Reheat coil sections are used where it is essential to control temperature and relative humidity accurately. Heating coils located in the reheat position, as shown in Figure 4-1, are often used for warmup, although a coil in the preheat position is preferable. The reheat coils can be heated by hot water, steam, or electricity, with hot water providing the highest degree of control.

(15) Steam grid humidifiers or recirculating sprays with dew point control usually are used for accurate humidity control. In this application, the evaporation heat should be replaced by heating the recirculated water rather than increasing the size of the preheat coil.

b. *Principal Applications.*

(1) Central systems are typically used in the following cases:

(a) Uniform loads--generally large areas with small external loads.

(b) Multiple systems in large areas--very large facilities such as factories, hangars, and hospitals may require multiple central stations.

(c) As a primary source of conditioned air for other subsystems such as VAVs and mixing boxes.

(2) The central station equipment is located outside the computer room, usually in a basement, penthouse, or service area. It can be adjacent to, or a considerable distance away from, the primary heating and refrigeration equipment.

(3) In the dew point control scheme described in the section on chilled-water packaged units, the central station supplies cool air at the room dew point temperature, while the room AHUs supply sensible cooling only. In this case, a sprayed coil dehumidifier is used in the central unit to make maximum use of recovered heat to warm the air so that it will accept moisture. Also, the air is saturated or nearly saturated upon leaving the unit and allows accurate control of dew point temperature when dehumidifying.

c. *Sizes.* A central AHU typically ranges from about 3000 CFM to 65,000 CFM.

d. *Cost.* The equipment costs of a central chilled water AHU range from $400/ton for a 7-ton unit to $260/ton for a 115-ton unit. Installed costs range from $550/ton for a 7-ton unit to $350/ton for a 115-ton unit.

e. Advantages.

(1) In central station supply systems, components with larger capacities can be used than in self-contained room air-conditioning equipment. Also, since the equipment is not located within the computer room, a central station AHU can be used with a greater variety of choices in air-conditioning design and arrangement.

(2) No C^3I room floor space is required, and virtually all servicing and maintenance operations can be performed in areas specifically devoted to air-conditioning equipment.

(3) Roller-type filter mechanisms can be used that automatically roll new filter media into place when the existing filter material becomes dirty. This feature eliminates downtime for changing the filters and minimizes the possibility of operating the AHU with inadequate filtration.

f. Disadvantages. Since the AHU is outside the C^3I room, physical security is lessened. In addition, large-diameter ducts penetrate the C^3I shield, thus increasing the EMP shielding cost. If the system is used for other operations, the setpoints must be altered for them. Also, a central station AHU usually serves overhead ducted supply systems. As discussed in Chapter 2, overhead ducted supplies may not be as flexible in accommodating load redistribution or expansion as room AHUs would be.

4-4. Chilled-Water Packaged Units

a. Operation.

(1) Figure 4-3 shows a typical arrangement for a chilled-water packaged unit. Components common to most chilled-water packaged units include: a filter rack, chilled water coil, humidifier, reheat coil, fan, and fan motor. Chilled-water packaged units designed for electronic equipment room environments are built with a higher level of performance and reliability than conventional packaged units designed for comfort applications. These units are usually mounted on the raised floor of the C^3I room. However, in smaller cooling capacity ranges, they may be hung from the ceiling. Floor-mounted units are available with both upward and downward air flow patterns.

(2) Downward-flow, floor-mounted, chilled-water packaged units typically use horizontal filter racks located at the top. Upward-flow units are arranged with vertical filters located on the front face. Ceiling-hung chilled-water packaged units have either vertical filters in the air intake section or horizontal filters located inside the air intake ceiling panels. Filters generally can be replaced without interrupting unit operation.

(3) Most units are furnished with disposable filters that have ASHRAE Standard 52-76 dust spot efficiencies ranging between 35 and 60 percent. ASHRAE recommends a filtration efficiency of 45 percent for electronic equipment room applications.

(4) It should be noted that floor-mounted, chilled-water packaged units generally use a free return system and, therefore, do not introduce or filter outside air required for minimum ventilation and positive pressurization. Outside air requirements must be satisfied by a separate duct, fan, and filter system or by one air handler that serves the entire C^3I room area, such as a central station air handler. The outside air duct penetrates the EMP shield, and therefore, precautions must be taken to protect the effectiveness of the EMP shield. These precautions were addressed in Chapter 2.

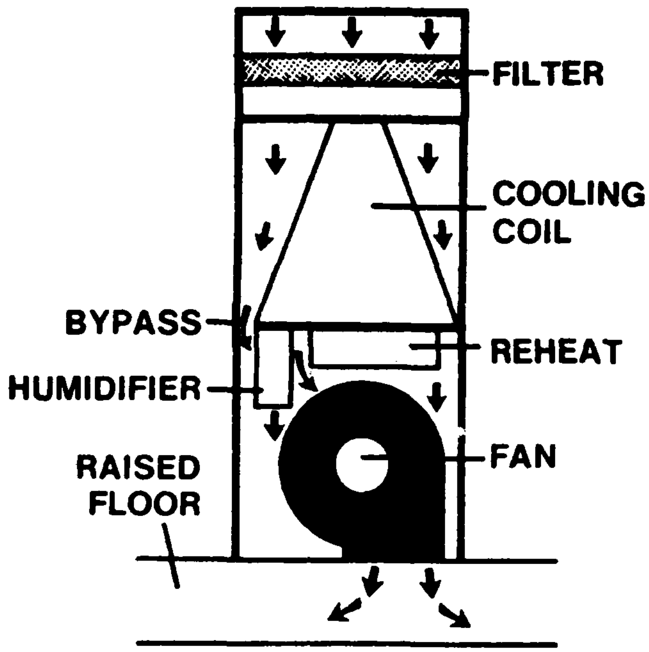

Figure 4-3. Chilled-water packaged unit. (Source: *ASHRAE 1987 HVAC Systems and Applications Handbook*. Used with permission.)

(5) Most chilled-water packaged units are equipped with either a vertical or A-frame chilled-water coil. Since C^3I rooms have mostly sensible cooling loads, the chilled-water coils tend to have large face areas and air velocities sufficient to produce a high sensible heat ratio. This minimizes excess dehumidification and reduces humidification costs. However, sometimes air must be dehumidified even at partial load conditions. To address this situation, some manufacturers provide chilled-water coils with multiple chilled water circuits, each with their own control valve.

(6) The humidifiers in most chilled-water packaged units are of the steam-generating variety. For proper humidity control, steam is typically introduced into the bypass air stream. Electrical resistance is most commonly used for steam generation. Recent steam-generating humidifier designs adjust the water level automatically to balance the output by electronically sensing the water conductivity.

(7) While most chilled-water packaged units are equipped with an electric resistance reheat coil, steam and hot-water coils are available as options. Electric resistance reheat coils are standard equipment because of their simplicity and reduced number of required piping connections. In areas with high electrical demand charges, the high operating cost of electric reheat coils should be considered. Also, the existing electrical service may not be adequate to handle the reheating requirements of multiple units.

(8) Most chilled-water packaged units use forward-curved, double-width housings. Since fan impellers in these units are typically less than 2 ft in diameter, the high speed associated with backward-inclined or airfoil fans would cause excessive belt speeds. The forward-curved fan, because of its low speeds, is particularly suitable to applications having two or more fans on a common shaft. The higher speed of a

backward-inclined fan would require a shaft of prohibitive size. Two fan belts are normally used in chilled-water packaged units, each capable of handling the load alone. This redundancy ensures continued operation should one belt fail.

b. *Principal Applications.*

(1) Chilled-water packaged units are installed almost exclusively within the C^3I room itself. Locating the units inside the C^3I room enhances the flexibility to handle load redistribution and expansion.

(2) Floor-mounted, chilled-water packaged units can supply air to an underfloor plenum. With this configuration, air can be returned to the unit by a free return system or a ceiling plenum return system. The floor-mounted units are also available with an upward air flow arrangement, supplying air to a ceiling plenum or overhead ducted supply system. With an upward-flow packaged unit, a free return system is normally used to return air to the front face of the unit. Supply air from ceiling-hung units is usually transported to the C^3I room through round, flexible ducts. Air is typically returned to ceiling-hung units through perforated ceiling panels attached to the bottom of the unit.

(3) In large applications, where multiple chilled-water packaged units are required, it is a common practice to provide outside air to one unit that serves the entire electronic equipment room area. One such configuration is described in the literature.[3] This configuration, called a "dew point control system," uses floor-mounted chilled-water packaged units to provide sensible cooling only. A separate, central sprayed-coil dehumidification unit mixes return and outside air and controls the dew point of the discharge air instead of attempting to control relative humidity. With this method, the multiple chilled-water packaged units do not "fight" each other (e.g., one unit dehumidifying and reheating, while another is humidifying).

c. *Sizes.*

(1) Floor-mounted chilled water packaged units are available in a size range from 2 to 40 tons. Ceiling-hung units are available from 1 to 5 tons.

(2) The air discharge area in many of the floor-mounted units is designed to match the dimensions of standard raised floor panels (24 by 24 in.). For example, the air discharge area of some floor-mounted units under 15 tons is 24 by 24 in. Over 15 tons, the air discharge area is 24 by 72 in. (the dimensions of three floor panels). This design minimizes cutting of floor panels and reduces the time and cost of installation.

d. *Cost.*

(1) Floor-mounted chilled-water packaged units vary in equipment costs from $200/ton for a 40-ton unit to $1100/ton for a 5-ton unit. Installed costs range from $350/ton for a 40-ton unit to $1700/ton for a 5-ton unit. These costs do not include the separate liquid chilling system.

(2) Ceiling-hung units range in equipment costs from $700/ton for a 5-ton unit to $2300/ton for a 1-ton unit. Installed costs range from $1200/ton for a 5-ton unit to

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[3] M. C. Connor and L. Hannaver, "Computer Center Design," *ASHRAE Journal* (April 1988).

$4000/ton for a 1-ton unit. These costs do not include the separate liquid chilling system.

e. Advantages.

(1) Locating chilled-water packaged units within the C^3I room increases their ability to accommodate load redistribution and expansion. Existing units can be relocated to satisfy load redistribution within the C^3I room. New units can be added to handle increased load without interrupting the operation of existing chilled-water packaged units. The use of multiple air handlers provides the redundancy required by C^3I rooms. Ceiling-hung units free valuable C^3I room floor area.

(2) Load expansion can also be accommodated by oversizing the chilled water packaged units. Initially, the units will operate at less than full load. As the load increases, the units will gradually operate at capacities approaching full load. In this instance, the chilled-water piping would be sized for the future full load. The coil control valves would modulate for the first part load operation. By initially oversizing the cooling coil for future load expansion, the beginning partial load performance is actually improved due to the larger coil face and/or number of rows, but at a significant added cost.

(3) Since chilled-water packaged units do not contain refrigeration equipment, maintenance performed within the C^3I room is less than for self-contained packaged units. However, chilled-water packaged units require more maintenance within the C^3I room than central station air handlers and about the same as fan coil units. Chilled-water packaged units designed for electronic equipment room environments are built with a higher level of performance and reliability than conventional packaged units designed for comfort applications. Locating the units inside the C^3I room increases security compared with central station air handlers located in a mechanical room.

(4) With chilled-water packaged units, piping for chilled water, condensate, and humidifier water supply, and ductwork for outside air penetrate the EMP shield. Central station air handlers, in comparison, have substantially larger penetrations for supply and return air ductwork. Since C^3I rooms usually require chilled-water piping to serve water-cooled electronic equipment, the additional cost for piping to the chilled-water packaged units is reduced over that of the central station air handlers.

f. Disadvantages.

(1) While locating the floor-mounted chilled-water packaged units within the C^3I room provides flexibility in meeting load redistribution and expansion, the units occupy floor space that is frequently quite valuable in C^3I room applications. By locating the chilled-water packaged units within the C^3I room, the chance of damaging water leaks is increased. Also, maintenance of the units must be performed inside the C^3I room, possibly interfering with C^3I operations. Chilled-water, condensate, and humidifier water supply piping associated with chilled-water packaged units typically penetrates the C^3I room EMP shield. The methods of providing shielding effectiveness for piping penetrations, described in Chapter 3, must be used.

(2) If the controls of individual chilled-water packaged units are even slightly out of calibration in multiple unit applications, the units may compete with each other. However, the dew point control system described above can prevent this problem.

(3) Although chilled-water packaged units have alarms to indicate when filters are clogged, the filters have to be changed manually, increasing the likelihood that the units will operate part of the time without adequate filtration. Central station air handlers are available with roller-type filter mechanisms that automatically roll clean, new filter media into place when the existing filter becomes dirty.

4-5. Self-Contained Packaged Units

a. Operation.

(1) Self-contained packaged units are similar to chilled-water packaged units except that the refrigeration equipment is self-contained. Instead of the chilled-water coil used in a chilled-water packaged unit, the self-contained unit uses a DX refrigerant coil to cool the C^3I room air. Figure 4-4 shows a self-contained packaged unit. Components of typical self-contained packaged units include: a filter rack, single or multiple compressors, DX refrigerant coil, humidifier, reheat coil, fan, and fan motor.

(2) Like chilled-water packaged units, the self-contained units are usually mounted on the raised floor of the C^3I room. However, for smaller cooling capacities, ceiling-hung units are available. Floor-mounted models are available with both upward and downward air flow patterns. Self-contained packaged units designed for electronic equipment room environments are built with a higher level of performance and reliability than conventional units designed for comfort applications.

(3) Compressors used in self-contained packaged units are usually the hermetic reciprocating type. Units with a capacity of more than 5 tons commonly use dual compressors, each with their own refrigerant circuit. Some manufacturers claim to use heat pump duty compressors to extend the service life of the unit.

(4) The filter arrangement in self-contained packaged units is the same as that for chilled-water packaged units. Most manufacturers' units are furnished with disposable filters with ASHRAE Standard 52-76 dust spot efficiencies ranging between 35 and 60 percent. ASHRAE recommends a filtration efficiency of 45 percent for electronic equipment room applications.

(5) It should be noted that, like chilled-water packaged units, floor-mounted, self-contained packaged units generally use a free return system and, therefore, do not introduce or filter outside air required for minimum ventilation and positive pressurization. Outside air requirements must be satisfied by a separate duct, fan, and filter system, or by one air handler that serves the entire C^3I room area. The outside air duct penetrates the EMP shield and, therefore, the penetration must be treated to protect the shielding effectiveness. EMP shielding associated with air distribution systems was addressed in Chapter 2.

(6) Most self-contained packaged units are equipped with either a vertical or A-frame DX refrigerant coil. If dual compressors are used, the DX coil typically contains two separate refrigerant circuits. Each circuit supplies refrigerant over the entire face of the coil.

(7) The humidifiers in most self-contained packaged units are of the steam-generating variety. For proper humidity control, steam is typically introduced into the bypass air stream. Electrical resistance is most commonly used for steam generation. Newly developed steam-generating humidifiers automatically adjust the water level to balance the output by electronically sensing the water conductivity.

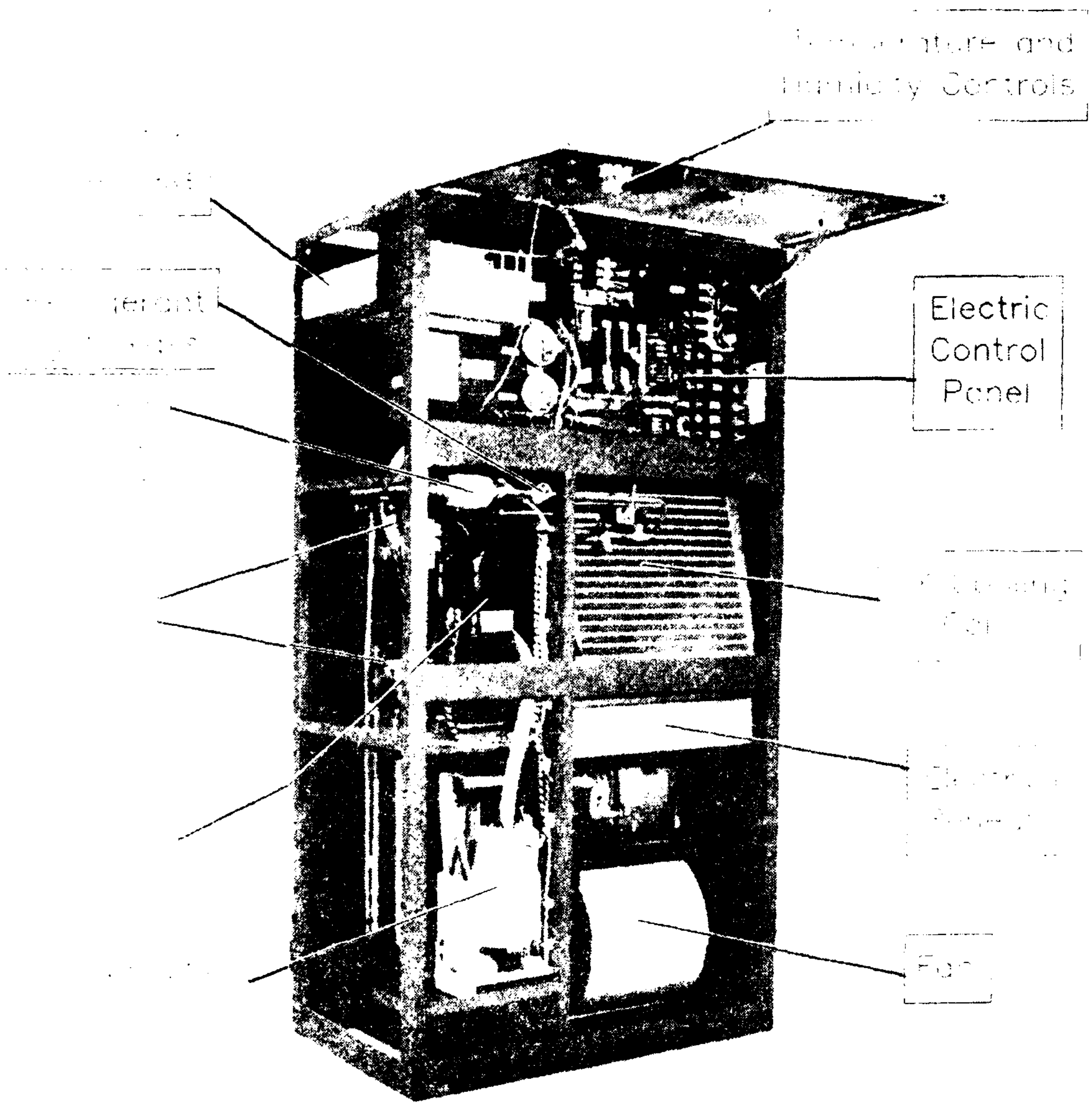


Figure 4-4. Self-contained packaged unit. (Source: *Airfow Product Catalog.* Used with permission.)

(8) While most self-contained packaged units are equipped with an electric resistance reheat coil, steam, hot water, and hot refrigerant gas reheat coils are available as options. Electric resistance reheat coils are standard equipment because of their simplicity and low number of piping connections required. In areas with high electrical demand charges, the high operating cost of electric reheat coils should be considered. Also, the existing electrical service may not be adequate to handle the reheating requirements of multiple units.

(9) Like chilled-water packaged units, the self-contained units primarily use forward-curved, double width, double inlet centrifugal fan impellers within scroll-type housings. Two fan belts are normally used in chilled-water packaged units, each capable of handling the load alone. This redundancy ensures continued operation should one belt fail.

b. Principal Applications.

(1) Self-contained packaged units are typically installed inside the C^3I room. Locating the units inside the C^3I room improves flexibility in handling load redistribution and expansion. The upward and downward air flow properties of the supply and return air distribution systems used are the same as those discussed previously for chilled-water packaged units. The dew point control system described for chilled-water packaged units can also be used with floor-mounted, self-contained packaged units. Self-contained packaged units can be water-cooled by a remote cooling tower, cooled by a remote dry cooler using water or glycol as the condensing fluid, or air-cooled by a remote air-cooled condenser.

(2) One configuration for cooling both the space and equipment of C^3I rooms combines floor-mounted, self-contained packaged units with self-contained packaged liquid chillers. The floor-mounted, self-contained packaged units cool air supplied to an underfloor plenum and self-contained packaged liquid chillers located within the C^3I room provide water-cooled electronic equipment with chilled water. An interesting feature of the water-chilling portion of this system is that it incorporates a built-in water storage tank and separate circulating pump that provides 20 min of backup cooling to the water-cooled electronic equipment in the event of a power failure.

c. Sizes. Floor-mounted self-contained packaged units are available in size ranges from approximately 2 to 20 tons. Ceiling-hung units are available from 1 to 7 tons. As with the chilled-water packaged units, the air discharge area in many of the floor-mounted, self-contained packaged units is designed to match the dimensions of standard raised floor panels.

d. Cost.

(1) Floor-mounted, water-cooled, self-contained packaged units vary in equipment costs from $770/ton for a 23-ton unit to $1470/ton for a 5-ton unit. Installed costs range from $950/ton for a 23-ton unit to $1800/ton for a 5-ton unit. These costs do not include the condenser, water supply, or cooling tower.

(2) Equipment and installed costs for glycol-cooled, floor-mounted units are about 30 percent higher than for water-cooled, floor-mounted units. This cost premium is due to the dry cooler, glycol pump, and controls associated with the glycol-cooled units.

(3) Water-cooled, ceiling-hung units range in equipment costs from $1000/ton for a 7-ton unit to $2300/ton for a 1-ton unit. Installed costs range from $1600/ton for a 7-ton unit to $4000/ton for a 1-ton unit. These costs do not include the condenser, water supply, or cooling tower. Glycol-cooled, ceiling-hung units have an equipment cost of between $2500 and $3500 for the dry cooler, glycol pump, and associated controls.

e. Advantages. Similar to the chilled-water packaged units, locating the self-contained units within the C^3I room increases their ability to accommodate load redistribution and expansion. Existing units can be relocated to satisfy load redistribution within the C^3I room. New units can be added to handle increased load without interrupting the operation of existing self-contained packaged units. The use of multiple air handlers provides the redundancy required by C^3I rooms. Locating the units inside the C^3I room increases security compared with central station air handlers located in a mechanical room. With self-contained packaged units, only condenser water or glycol piping and small ductwork for outside air penetrate the EMP shield.

f. Disadvantages.

(1) Although locating the self-contained packaged units within the C^3I room provides flexibility in meeting load redistribution and expansion, the units occupy space that is frequently quite valuable in C^3I room applications. In addition to the actual space occupied by the unit, clearance around the unit is required for maintenance. Also, placing the self-contained packaged units within the C^3I room increases the chance of damage due to refrigerant, water, or glycol leaks. Since self-contained packaged units contain refrigeration equipment, more maintenance must be performed on the units inside the C^3I room than with chilled-water packaged units, possibly interfering with C^3I operations.

(2) Similar to chilled-water packaged units, if the controls of individual self-contained packaged units are even slightly out of calibration in multiple unit applications, the units can compete. However, the dew point control system described for the chilled-water packaged units can alleviate this problem.

(3) Although self-contained packaged units have alarms to indicate when filters are clogged, the filters have to be changed manually, increasing the likelihood that the units will operate part of the time without adequate filtration. As noted earlier, central station air handlers are available with roller-type filter mechanisms that automatically roll clean, new filter media into place when the existing filter material becomes dirty.

4-6. Fan Coil Units

a. Operation.

(1) A typical fan coil unit, as shown in Figure 4-5, consists of: a fan and motor, chilled-water coil, air filter, casing, and controls. The fan blows across the coils at various speeds, depending on room requirements. A thermostat controls the fan speed.

(2) The fan motors usually have a high, medium, and low speed. A dual fan arrangement can be used for increased capacity or the second unit can be used as backup if the primary one fails.

(3) The chilled water coils can come in various diameters and numbers of rows, depending on the design cooling load. The filters can be disposable or cleanable and reusable. Supply and return air flow can be ducted or open.

b. Principal Applications. Fan coils are used in areas where chilled water is readily available and a concentrated heat load is present. The flexibility to handle load redistribution and expansion is increased.

c. Sizes. Fan coils typically range from 0.5 to 15 tons.

d. Cost. Cabinet-mounted fan coil units vary in equipment costs from $1000/ton for a 0.5-ton unit to $200/ton for a 15-ton unit. Installed costs range from $1400/ton for a 0.5-ton unit to $220/ton for a 15-ton unit.

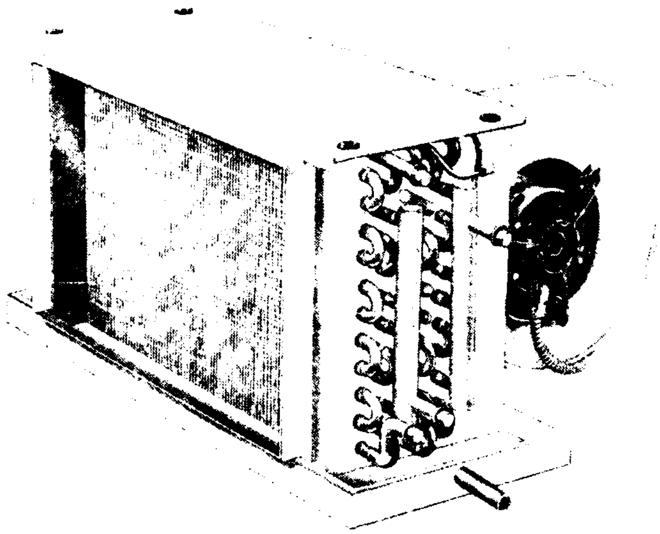


Figure 4-5. Fan coil unit.

e. Advantages. If multiple fan coils are used, temperature control can be achieved for varying concentrated loads. In addition, fan coil units have a lower first cost than central air handlers and bring additional savings due to reduced ductwork requirements.

f. Disadvantages. There is a possibility of leaks in the chilled-water piping system within the C^3I room. Also, the fan coils do not introduce fresh air or humidity into the room so that additional equipment is required. Maintenance on the units must be done in the room, possibly interfering with C^3I operations.

4-7. Other Considerations

Air handler operation must be maintained during an EMP event. All components and controls susceptible to damage from an EMP burst must be protected. Since chilled-water and self-contained packaged units and the fan coil unit are typically located inside the C^3I room, their components and controls are protected by the C^3I room shield. However, a central station AHU requires additional protection. The fan motor and electric valves and controls must be shielded. The design engineer must determine if it is more cost-effective to shield the individual components or the entire air handler. Factors in this assessment include: (1) the fact that shielding cost per unit volume decreases as the total volume increases, (2) the total number of penetrations necessary, which varies between strategies, and (3) the need for honeycomb air vent panels when shielding the entire AHU.

Section III: SUMMARY EVALUATION

4-8. How To Use the Evaluation Table

a. Table 4-1 ranks each air-handling unit for the criteria presented in **Section I.** These rankings are qualitative, with a black circle on the table representing a high ranking, a gray circle denoting a medium ranking, and a white circle indicating a low rating. It should be noted that a low rating simply means that other air handlers are more suitable for a specific criterion, and not that the unit is a poor choice in general.

b. When ranking each air-handling unit, certain assumptions were made regarding the system configuration. These assumptions were based on the most typical system configuration in which each air handler is used. It was assumed that the chilled-water and self-contained packaged units are located inside the C^3I room and supply an underfloor plenum. The central station AHU was assumed to be located outside the C^3I room and to supply air to an overhead ducted supply system. Fan coil units were assumed to be installed around the perimeter of the C^3I room. In ranking the lowest cost for EMP shielding, it was assumed that the C^3I room itself is shielded.

4-9. Evaluation Results

a. Load Redistribution.

(1) Chilled-water packaged units, self-contained packaged units, and fan coil units can all be relocated to accommodate load redistribution. The only way a central station AHU can adapt to load redistribution is to modify the air distribution system by adjusting duct volume dampers or VAV boxes. If the air distribution system is not designed to handle load redistribution, the ductwork may have to be demolished, redesigned, and replaced. To more easily accommodate load redistribution, flexible ducting can be used. However, flexible ducting run lengths are generally limited to a maximum of 6 ft,

Table 4-1. Summary Evaluation—Air-Handling Systems

AIR-HANDLING UNITS	Load redistribution	Load expansion	Humidity control	Filtering	Static electricity	Backup cooling	Reliability	EMP shielding cost *	Life-cycle cost *
1) Central Station Air-Handling Unit	Low	Low	High	High	High	N/A	Low	Low	Medium
2) Chilled Water Packaged Unit	High	High	Medium	Medium	Medium	N/A	High	Medium	Medium
3) Self-Contained Packaged Unit	High	High	Medium	Medium	Medium	N/A	Medium	Medium	N/A
4) Fan Coil Unit	High	High	Low	Low	Low	N/A	High	High	High

● = High   ◎ = Medium   ○ = Low

* A high ranking for cost-related criteria indicates a low relative cost. A low ranking indicates a high relative cost.

due to the relatively high-pressure loss associated with it. Furthermore, flexible ducting is not generally used in high-pressure applications.

(2) If multiple units are used, the chilled-water and self-contained packaged units and the fan coil units can be oversized to accommodate load redistribution. When the load concentration is relocated, the unit serving the area of concentrated load will run at nearly full load. The unit serving the area from which the load has been shifted will then run at partial load. Oversizing a central station AHU will increase its ability to handle load expansion, but will not improve its ability to handle load redistribution.

(3) For these reasons, a high ranking was given to chilled-water packaged units, self-contained packaged units, and fan coil units. The central station AHU was given a medium ranking due to its lower flexibility.

b. *Load Expansion.*

(1) All of the air-handling units evaluated can be oversized to accommodate load expansion. A well designed chilled-water distribution system within the C^3I room is built with extra piping connections and valves to plan for the installation of additional chilled-water packaged units or fan coil units. The additional units will handle load expansion and can be installed without interrupting operation of the existing units. Similarly, the condenser piping system can be designed and built to accommodate additional self-contained packaged units.

(2) With a central station AHU, the fan speed can be increased up to a certain point to increase the supply air flow rate, thus accommodating load expansion. Unless a variable- or multiple-speed fan is used, the variable-pitch fan pulleys must be adjusted manually to increase the fan speed and supply air flow rate. This would require shutting down the unit while adjustments are made. However, the fan speed can be increased only up to a certain maximum limit. If the maximum speed results in a supply flow rate that is still inadequate to handle the increased load, the cooling coil must be replaced with one of larger capacity. Again, this modification requires that the unit be shut down.

(3) Due to the factors just discussed, a high ranking was given to the chilled-water packaged units, self-contained packaged units, and fan coil units. The central station AHU received a medium ranking because of the complexities involved with increasing its cooling capacity.

c. *Precise Control of Humidity.*

(1) Fan coil units can dehumidify, but not humidify. Therefore, fan coil units were given a low ranking under this criterion.

(2) As mentioned in the system description, if multiple packaged units (chilled-water or self-contained) are used, and the controls of the individual units drift out of calibration, the units can "fight" each other. Although the dew point control system described previously could solve this problem, the system uses a central AHU to provide humidity control. Thus, the chilled-water and self-contained packaged units received a medium ranking and the central station AHU was given a high ranking. It should be noted that, in applications requiring only one chilled-water or self-contained packaged unit, unit "fighting" is not an issue. In these applications, the chilled-water and self-contained packaged units would receive high rankings.

d. Filtering of Particulates.

(1) Central station AHUs are available with automatic, roller-type filter mechanisms. When a differential pressure controller senses that the filter is dirty, new filter material is rolled into place. Continuous, adequate filtration is achieved without requiring the time and effort of a maintenance engineer. Furthermore, central station AHUs typically filter both return air and outside air. For this reason, the central station AHU received a high rating.

(2) While both chilled-water and self-contained packaged units are available with alarms that signal when a filter is dirty, the filters must be changed manually. This arrangement increases the chance that the units will operate at least some of the time with clogged filters. Also, packaged units usually filter only return air. Outside air must be filtered by a separate system, such as a central station AHU. Thus, both the chilled-water and self-contained packaged units received a medium ranking.

(3) Fan coil units do not have alarms that indicate when a filter is dirty. Without a dedicated, comprehensive maintenance program, fan coil units could operate for a long time with dirty filters, which decreases efficiency. Also, fan coil units filter only return air. Thus, the fan coil unit was given a low ranking.

e. Static Electricity Control. For AHUs, the control of static electricity is mainly a function of regulating the relative humidity of the air supplied to the C^3I room. Thus, the rankings for static electricity control are the same as those for the humidity control criterion. The central station AHU received a high ranking, the chilled-water self-contained packaged unit was ranked medium, and the fan coil unit was given a low ranking.

f. Backup Cooling Capacity. In the event of a power failure, chilled-water air handlers receive chilled water from the backup cooling system instead of the chiller. Self-contained packaged units are connected to a UPS to provide cooling during a power outage. Thus, backup cooling capacity is a function of the chilled water distribution and UPS, and does not apply to air handlers. Therefore, "N/A" appears in the evaluation table for all AHUs under this criterion.

g. Reliability.

(1) While all of the AHUs are generally reliable, rankings for this criterion were based on the relative number of components within the air handlers that are subject to failure. The level of redundancy offered by systems using each type of AHU was also considered in the ranking process.

(2) Since fan coil units have the fewest components of any of the units evaluated and since the use of multiple units provides a certain amount of redundancy, these units received a high ranking. Chilled-water and self-contained packaged units designed for C^3I environments are built with a higher level of reliability than conventional packaged units for comfort applications. Since multiple packaged units are typically used in C^3I rooms, redundancy and reliability are enhanced. Thus, chilled-water packaged units received a high ranking. Since self-contained packaged units have more components (e.g., compressors) than chilled-water packaged units, they were given a medium ranking.

(3) With all of its return, relief, and outside air dampers and actuators, the central station AHU has more points of potential failure than the other units. Typically, only one central station AHU serves the C^3I room. The number of components and lack of redundancy resulted in a low ranking for central station AHUs.

h. Least Cost for EMP Shielding.

(1) The ranking of least cost for EMP shielding was based, in part, on the relative size and number of the shielding penetrations associated with the typical configuration of the AHU. This ranking was also based on the air handlers' requirements for additional EMP shielding to ensure operation during an EMP event. The supply and return air ducts of a central station AHU must be sized to handle the total conditioned air flow of the C^3I room, in addition to outside air, for minimum ventilation and positive pressurization. Assuming the C^3I room itself is EMP-shielded, the central station AHU has the largest area of EMP penetrations of any other air handler evaluated. Also, central station AHUs are normally located remotely from the C^3I room, and therefore outside the EMP shield. The central station AHU must be provided with separate shielding, greatly increasing the cost of EMP protection. Thus, the central station AHU received a low ranking.

(2) Chilled-water packaged units have penetrations for chilled water, condensate, and humidifier water supply piping. Self-contained packaged units have condenser, condensate, and humidifier water supply piping penetrating the EMP shield. The size and number of EMP shield penetrations are about the same for both types of packaged unit, but less than that for central station AHUs. Since packaged units are typically located inside the C^3I room, the units do not require additional shielding. Both chilled-water and self-contained packaged units received medium rankings.

(3) Fan coil units have only chilled-water and condensate piping to penetrate the EMP shield. Furthermore, fan coil units are located inside the C^3I room and do not require additional shielding to ensure their operation during an EMP event. Fan coil units were given a high ranking.

g. Lowest Life-Cycle Cost.

(1) Life-cycle cost for air-handling units is primarily a function of first cost, energy operating cost, and maintenance cost. Since the fan coil unit is the most basic of the air handlers evaluated and has the fewest number of components, it has the lowest first cost and maintenance cost of any of the air handlers. Also, the fan coil unit has no humidifier or reheat coil. Therefore, it has the lowest energy operating cost. Thus, the fan coil unit received a high ranking for lowest life-cycle cost.

(2) Per ton, the central station AHU has a lower first cost than the chilled-water packaged unit. Due to the fact that larger motors generally have higher efficiencies than smaller ones, the fan energy per ton is less for the central station air handler than for the chilled-water packaged unit. Furthermore, if the control system is not designed properly, multiple chilled-water packaged units can compete, contributing to a higher energy operating cost than with central station AHUs. However, the labor to keep all of the return, relief, and outside air dampers in a central AHU adjusted properly produces higher maintenance costs compared with chilled-water packaged units. All of these factors result in relatively equal life-cycle costs. Thus, both the central station AHU and the chilled-water packaged unit received medium rankings.

(3) Since self-contained packaged units incorporate refrigeration equipment, their first cost, energy operating cost, and maintenance cost are higher than any of the other air handlers evaluated and would be very difficult to evaluate on an equal basis with the other AHUs. Thus, the self-contained packaged units are not ranked in the table.

(4) It should be noted that using self-contained packaged units will reduce the required capacity of the liquid chilling system compared with chilled-water AHUs. However, the COP of a large liquid chilling system is much higher than that of the multiple small compressors used in self-contained packaged units. Furthermore, it is difficult to compare fan coil units directly to the other air handlers evaluated since they do not have humidifiers or reheat coils.

5 LIQUID CHILLING SYSTEMS

Section I: GENERAL

This chapter evaluates design approaches for liquid chilling systems serving electronic equipment rooms. The following criteria are used to evaluate each system component:

1. Flexibility in meeting load expansion.
2. Capacity to provide 15 min of backup cooling.
3. Reliability.
4. Least cost for EMP shielding.
5. Lowest life-cycle cost.

5-1. Liquid Chiller Types

Only the liquid chilling systems used most often to serve C^3I facilities are evaluated. Less common chillers, such as steam turbine units, are not addressed. However, the steam turbine power plant usually serves the compressors and associated components covered in this chapter. The following liquid chilling systems are described:

a. Reciprocating chillers.

b. Rotary screw chillers.

c. Centrifugal chillers.

d. Absorption chillers.

e. Gas engine-driven chillers.

Section II: SYSTEM DESCRIPTION

5-2. Classes and Components

a. Liquid chilling systems can be divided into two main categories: (1) vapor compression and (2) absorption. Vapor compression liquid chilling systems consist of four basic components: (1) compressor, (2) condenser, (3) evaporator, and (4) expansion device. Vapor compression liquid chilling systems are usually specified by the type of compressor used to achieve a refrigeration effect (e.g., centrifugal or reciprocating). Compressors are categorized as either positive displacement or turbo-compressors (centrifugal).

b. Vapor compression chillers can use a variety of condensing methods. A water-cooled unit uses an evaporative cooling tower. A remote air-cooled condenser unit uses a remote water coil/fan condenser combination to cool water that has picked up heat from the refrigerant vapor. An air-cooled packaged unit uses air blown directly over refrigerant piping to cool the refrigerant vapor. A water-cooled condenser allows the highest efficiency operation. It enables heat rejection to a lower temperature (near the wet bulb temperature) than an air-cooled condenser. Also, water is a more efficient heat transfer medium than air.

c. Absorption chillers have the following basic components: (1) generator, (2) absorber, (3) evaporator, (4) condenser, (5) solution heat exchanger, (6) solution and evaporator pumps, and (7) expansion device. Absorption chillers are usually specified in terms of both the heat source and generator type (e.g., steam single-stage, gas-fired two-stage). In general, absorption chillers are water-cooled, although they are available as air-cooled in small size ranges.

5-3. Reciprocating Chillers

a. Operation.

Reciprocating chillers use a positive displacement compressor in which refrigerant gas is compressed by a piston that reduces the volume of the refrigerant in a closed cylinder. Packaged reciprocating chillers with cooling capacities over 100 tons typically have several compressors in a single unit. The cooling capacity is generally controlled by a combination of compressor cycling and cylinder unloading in which the suction gas to some cylinders is shut off. Partial load efficiencies within a few percent of design values down to 20 percent load can be obtained with this control strategy.

(2) A water-cooled reciprocating chiller (Figure 5-1) of 120 tons operating at American Refrigeration Institute (ARI) conditions (85 °F entering condenser water and 45 °F exiting chilled water temperature) can achieve COPs of up to 4.07 (0.86 kW/ton). When the exiting chilled water temperature is increased to 50 °F, the COP can reach 4.28 (0.82 kW/ton).

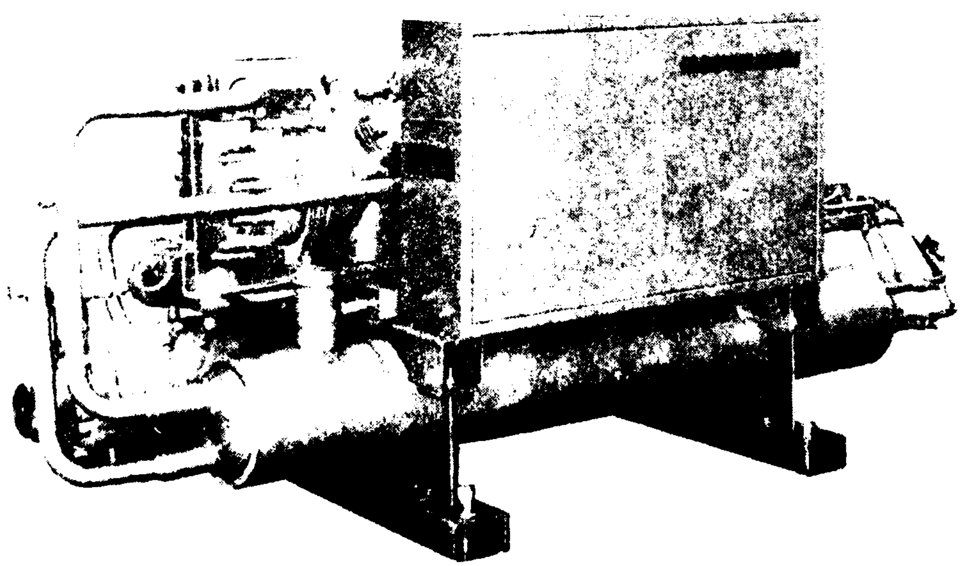


Figure 5-1. Water-cooled reciprocating chiller.

(3) When the same 120-ton reciprocating chiller uses a remote air-cooled condenser at ARI conditions of 95 °F air entering the remote condenser and 45 °F exiting chilled water temperature, the COP is 3.14 (1.12 kW/ton). When the exiting chilled water temperature is raised from 45 °F to 50 °F, the COP increases to 3.31 (1.06 kW/ton).

(4) When a 120-ton reciprocating chiller is directly air-cooled (Figure 5-2), the COP at ARI conditions is 2.90 (1.21 kW/ton). If the exiting chilled water temperature is raised to 50 °F, the COP rises to 2.99 (1.18 kW/ton).

(5) Since reciprocating chillers use stepwise capacity reduction instead of continuous modulation, the control scheme must combine precise temperature regulation with operation that avoids excessive compressor cycling and cylinder loading/unloading. This control is typically accomplished by locating the chilled water temperature sensor in the return line. The result is a thermal flywheel effect that dampens excessive cycling. While more precise, responsive temperature control can be achieved by sensing the supply chilled water temperature, unstable operation may result if the load changes rapidly. Since electronic equipment rooms usually have fairly constant cooling loads, either control scheme is appropriate.

b. Principal Applications.

(1) Reciprocating chillers are well suited for air-cooled condenser applications since they can retain almost full cooling capacity even at the elevated condensing temperatures associated with using air as the heat sink. This condition occurs because the volume flow rate of the reciprocating compressor is impacted very little by the increase in pressure resulting from higher temperatures. Ethylene glycol is often substituted for water as the condenser fluid for freeze protection in remote air-cooled systems. When ethylene glycol is used instead of water, the cooling capacity is somewhat reduced and the condenser loop pumping power requirements are slightly increased since glycol has a lower heat capacity and higher viscosity than water.

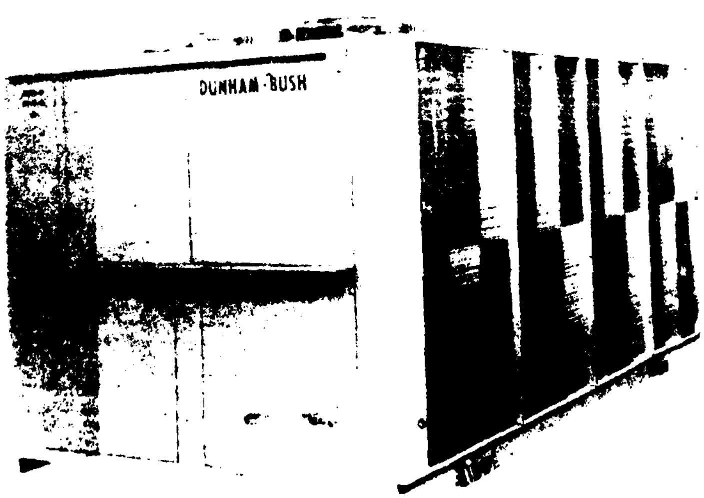


Figure 5-2. Air-cooled reciprocating chiller.

(2) Reciprocating chillers can be used in C^3I room applications as the main chiller when cooling loads are less than 250 tons, which is the upper limit of available capacity. They can handle small steps of load expansion since they can be installed with various incremental capacities. When a large central or building chiller serves the C^3I room and other loads during normal office hours, it often needs to operate only at a very light load at night (other building loads are not present). Reciprocating chillers can be used efficiently at night to closely match the C^3I room cooling load.

(3) Air-cooled packaged chillers are well suited for retrofit applications because they can be located on the roof near the C^3I room. In this case, only the chilled-water piping needs to be run to the C^3I room and air handler.

c. Sizes. Reciprocating chillers are available with capacities ranging from 2 to 250 tons.

d. Cost.

(1) Air-cooled reciprocating chillers range in equipment cost from around $400/ton for capacities over 100 tons to $850/ton for capacities under 10 tons. Chilled-water piping and pumps, controls, and installation result in installed costs of $800 to $1450/ton.

(2) Water-cooled reciprocating chillers range in equipment cost from about $300/ton for capacities over 100 tons to $900/ton for capacities under 10 tons. Cooling towers, condenser water piping and pumps, controls, and installation result in installed costs ranging from $725 to $1700/ton.

(3) These costs do not include EMP shielding. Estimated costs for shielding chiller equipment are addressed in paragraph 5-7.

e. Advantages.

(1) Since reciprocating chillers are well suited for air-cooled condensers, they are quite versatile for retrofit applications. In air-cooled packaged configurations, reciprocating chillers can be easily installed on a rooftop near the location to be served. Even when a remote air-cooled condenser is used, retrofit installations are straightforward, requiring minimal demolition.

(2) Reciprocating chillers are the most compact in volume, on a per-ton basis, of all chillers evaluated. Besides volume, another important size consideration is the occupied floor area, often called the "footprint" of the chiller. Below 450 tons, the reciprocating chiller has the smallest footprint of all chillers evaluated. At a capacity of 300 tons, the footprint of two 150-ton reciprocating chillers is smaller than the other single chiller rated at 300 tons. However, above 450 tons, the footprint of the centrifugal chiller is smaller than that of the reciprocating chiller.

(3) Reciprocating compressors are also versatile because they can be used to cool air as well as water. By using a DX coil, air can be cooled directly without the need for chilled water and associated piping. One configuration for cooling both the space and equipment of C^3I rooms uses self-contained DX air handlers to cool air supplied to an underfloor plenum, and reciprocating chillers to provide water-cooled electronic equipment with chilled water. An interesting feature of the water-chilling portion of this system is that it incorporates a water storage tank module that provides 20 min of back-up cooling to the water-cooled electronic equipment in the event of a power failure.

f. Disadvantages. The suction and discharge valves associated with the reciprocating compressor can cause excessive pressure losses due to either high gas velocities or poor mechanical action. These conditions can reduce the COP and also the service life of the unit. Additional pressure losses can be caused by the suction shutoff valves often used for capacity control. Air-cooled reciprocating chillers have the lowest COP of all chillers evaluated, except for air-cooled centrifugal chillers. Because of the reciprocating motion and large number of contacting parts, reciprocating chillers tend to be noisier than other chillers of equal capacity.

5-4. Rotary Screw Chillers

a. Operations.

(1) A helical rotary screw compressor forces the refrigerant vapor through a continuously decreasing volume between two counterrotating, multilobed helical rotors until it is finally discharged under high pressure. Uniform, nonpulsating gas flow can be achieved by using multiple rotors. Figure 5-3 shows a rotary screw chiller and Figure 5-4 demonstrates the compression process.

(2) The capacity is contolled by a slide valve that varies the opening of the rotor housing, effectively varying the compressor displacement. Since the slide valve does not introduce additional losses, the partial load performance of rotary screw chillers remains high down to 10 percent full load. Water-cooled rotary screw chillers can achieve COPs up to 4.83 (0.73 kW/ton) at 85 °F entering condenser water and 45 °F exiting chilled water. If the leaving chilled-water temperature is raised to 50 °F, the COP increases to 5.14 (0.68 kW/ton). Rotary screw chillers with remote air-cooled condensers can achieve COPs up to 3.1 (1.14 kW/ton).

(3) Rotary screw compressors are oil-injected to provide lubrication and cooling, and to seal potential leaks through the clearance between moving parts. In addition to cooling and improving the pumping efficiency, oil injection reduces noise levels. Since rotary screw chillers are designed for oil injection, they can also handle occasional slugging of liquid refrigerant without damage to the compressor.

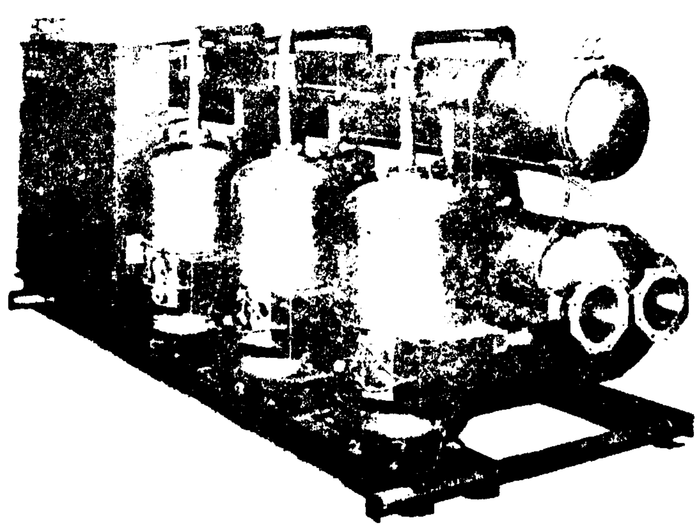


Figure 5-3. Rotary screw chiller.

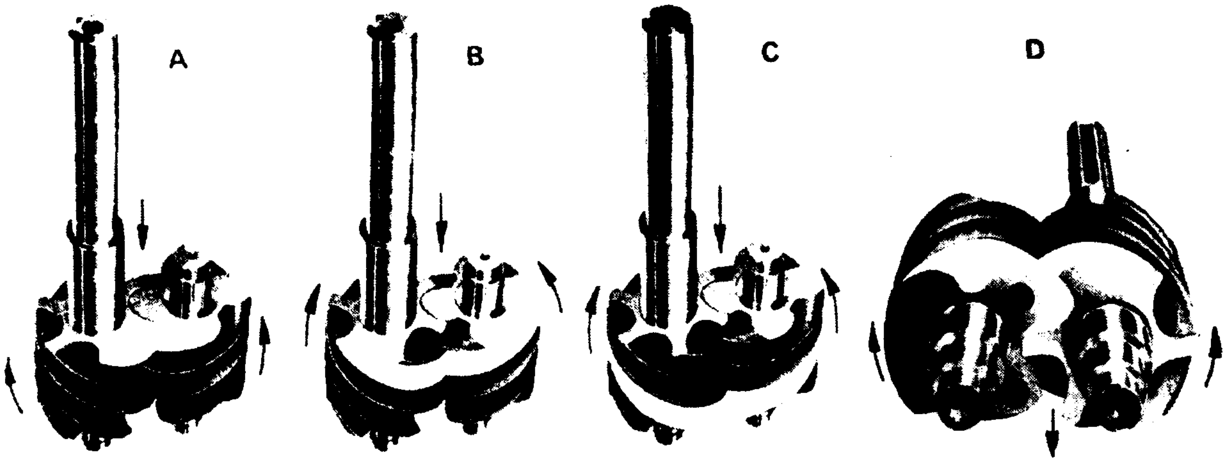


Figure 5-4. **Compression process in a rotary screw chiller.** Suction phase: a void is created and is drawn in through the inlet port (A) and gas flows continuously into the compressor (B). The entire length of the interlobe space is completely filled with gas (C). Compression phase: the occupied volume of trapped gas decreases and the gas pressure increases. Discharge phase: the discharge port is uncovered and the compressed gas is discharged (D).

(4) An economizer that increases the cooling capacity and improves the COP is an enhancement in more recent rotary screw compressors. The economizer is an additional suction port located between the normal suction port and the discharge port. After gas from the normal suction port has begun compression, additional vapor, at a higher pressure, is introduced through the economizer port. The net effect is to increase the refrigeration capacity by a greater amount than the power input is increased.

b. *Principal Applications.*

(1) In the past, rotary screw chillers were most often used in low-temperature refrigeration plants. However, they are recently seeing more use for air-conditioning applications because of their reasonable cost, durability, slug tolerance, and nonsurging operation.

(2) Rotary screw chillers can be used in C^3I rooms as the main chiller up to a maximum cooling load of 850 tons, which is the upper limit of available capacity. Like reciprocating chillers, they can handle small steps of load expansion since they can be installed with various incremental capacities. In addition, they can handle gradual load expansion since the slide valve capacity control provides continuous modulation. When a large central or building chiller serves the C^3I room and other spaces during normal office hours, it often operates at a very light load at night (other building loads are not present). Rotary screw chillers can closely match the C^3I room cooling load (even more efficiently than reciprocating chillers).

c. *Sizes.* Rotary screw chillers are available in sizes ranging from 40 to 850 tons.

d. Cost. Equipment costs for rotary screw chillers range from $220 to $360/ton. Cooling towers, condenser water piping and pumps, chilled-water piping and pumps, controls, and installation result in installed costs ranging from $550 to $800/ton. These costs do not include EMP shielding; see paragraph 5-7.

e. Advantages.

(1) Since the slide valve used for capacity control displaces the intake charge before compression begins, little or no losses occur. For this reason, there is high partial load performance down to 10 percent of full load. Thus, a rotary screw chiller can be installed when gradual future load expansion is anticipated. Before the load is added, the rotary screw chiller can act as a backup unit to, say, a centrifugal base chiller. As increments of load are added, the rotary screw chiller operates efficiently at low load to satisfy the additional cooling requirements.

(2) Because rotary screw chillers are oil-injected, they have lower discharge temperatures and run quieter than comparable reciprocating chillers. Rotary screw chillers are generally more compact in volume (per-ton) than centrifugal and absorption chillers.

f. Disadvantages. Oil injection requires the use of an oil separator to remove oil from the high-pressure refrigerant. Also, additional equipment is needed to cool the oil. This equipment requires proper maintenance to ensure trouble-free operation of the rotary screw chiller. Although more compact in volume than the centrifugal and absorption chillers, the footprint of a rotary screw chiller is larger than that of a reciprocating or centrifugal chiller of equal capacity.

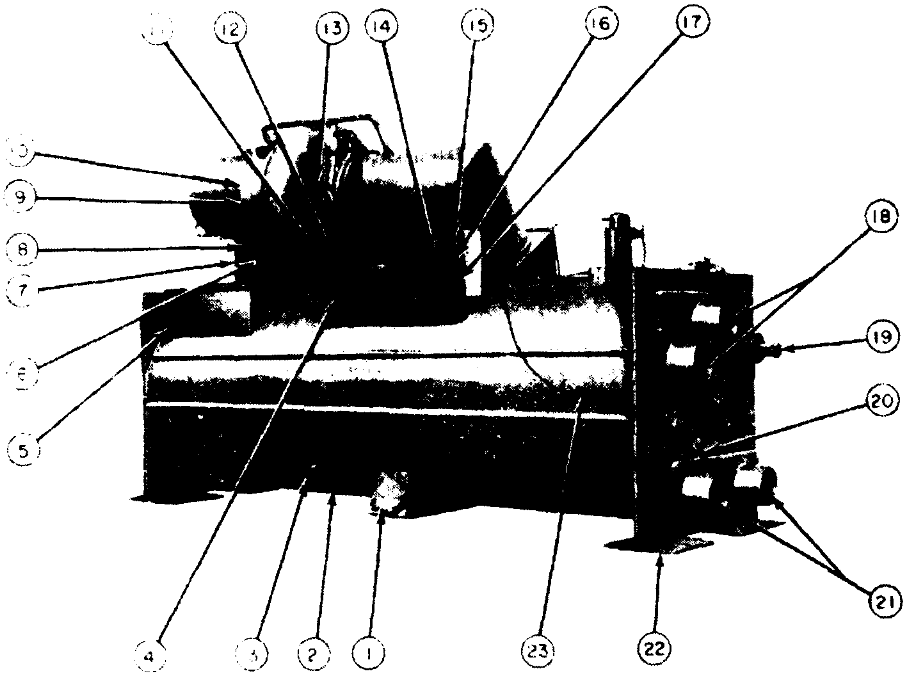
5-5. Centrifugal Chillers

a. Operation.

(1) Centrifugal chillers (Figure 5-5) use a compressor that increases the pressure of the refrigerant vapor by raising its speed in a high-speed, rotating impeller and then passing the vapor through a diffuser. Single- or multistage compressors can be used, with multistage compressors providing improved performance and smoother operation at light loads. The capacity of centrifugal chillers is generally controlled by inlet guide vanes that prerotate the refrigerant vapor before it reaches the impeller. This action lowers the amount of momentum that can be transferred to the vapor by the rotating impeller, and thus reduces the increase in pressure.

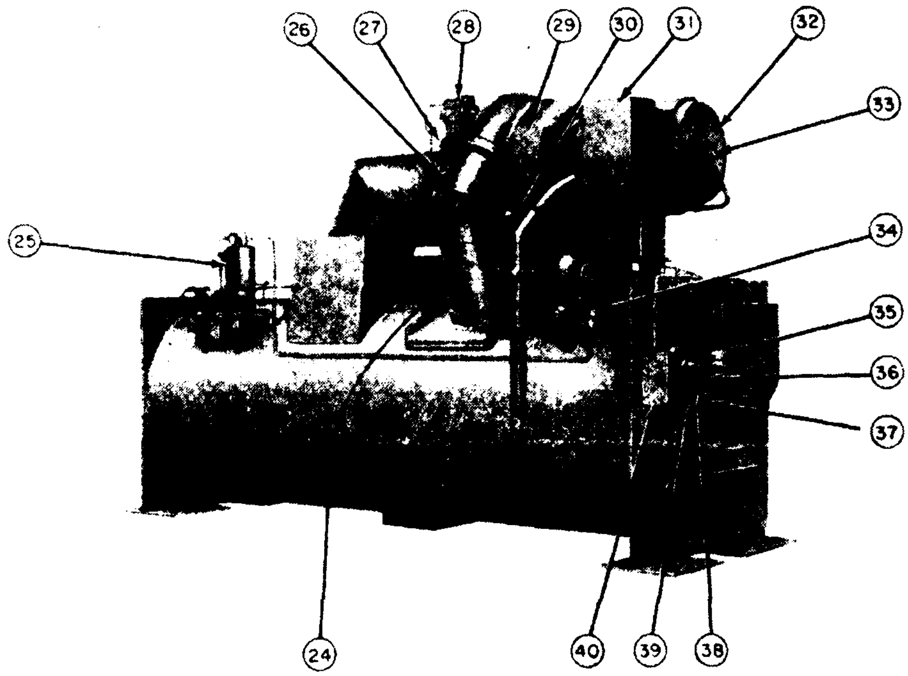
(2) The partial load performance of centrifugal chillers is very good from 40 to 100 percent load. However, below 40 percent load, the performance declines dramatically. The ability to continuously modulate capacity with nearly proportionate changes in input power enables the centrifugal chiller to provide precise temperature control with energy efficiency. COPs of up to 5.74 (0.61 kW/ton) are obtainable from water-cooled centrifugal chillers. With 85 °F entering condenser water and 44 °F exiting chilled water, the COP of a 450-ton water-cooled centrifugal chiller is 5.57 (0.63 kW/ton). If the exiting chilled water temperature is raised to 46 °F, the COP increases to 5.62.

FRONT VIEW



1 — Flow Valve Chamber
2 — Cooler Charging Valve
3 — Refrigerant Level Sight Glasses
4 — Field Wiring Knockouts
5 — Machine Informative Plate Location
6 — Oil Reservoir Temperature Gage
7 — Oil Heater and Thermostat Terminal Box
8 — Oil Level Sight Glass
9 — Return-Oil Temperature Gage (Hidden)
10 — Compressor Nameplate (Hidden)
11 — Bellows Seal (Hidden)
12 — Diffuser Feedback Potentiometer (Hidden)
13 — Vane Seal Oiler
14 — Microprocessor Control Panel
15 — Condenser Pressure Gage
16 — Cooler Pressure Gage
17 — Oil Pump Differential Pressure Gage
18 — Condenser Water Nozzles
19 — Safety Relief Device (Rupture Disc)
20 — Chilled Water Temperature Sensor
21 — Cooler Water Nozzles
22 — Support Plates
23 — Condenser Refrigerant Temperature Sensor

REAR VIEW



24 — Cooler Refrigerant Temperature Sensor
25 — Purge Assembly
26 — Discharge Temperature Sensor (Hidden)
27 — Diffuser Solenoid Valves (Hidden)
28 — Guide Vane Actuator
29 — Compressor Access Plate
30 — Inlet Volute Drain Strainer
31 — Compressor Terminal Box
32 — Motor End Cover
33 — Motor Rotation Sight Glass
34 — Refrigerant Filter
35 — Oil Cooler Solenoid Valve and Plug Valve
36 — Oil Pump, Cooler and Filter Assembly
37 — Oil Cooler Drain Plug
38 — Oil Charging Valve
39 — Oil Pressure Regulating Valve (Factory Set)
40 — Oil Pump Starter, Factory Installed

Figure 5-5. Centrifugal chiller. (Source: *Carrier Product Catalog, 1988.* Used with permission.)

b. *Principal Applications.*

(1) Since centrifugal chillers are typically used in the larger capacities (over 250 tons), they are usually water-cooled by a flooded, shell-and-tube condenser connected to an evaporative cooling tower. Air-cooled condensers can be used with chillers that use higher pressure refrigerants, but with a substantial energy penalty. Since, ton-for-ton, centrifugal compressors have a higher volumetric flow than positive displacement compressors, air cooling is not as effective for centrifugal chillers as it is for positive displacement chillers. The performance penalty associated with air-cooling a centrifugal chiller is as follows:

	Water-Cooled Evaporative Cooling Tower COP	Remote Air-Cooled Condenser COP
150-Ton reciprocating chiller	4.13	3.58
150-Ton centrifugal chiller	4.43	2.52

(2) Since centrifugal chillers have the highest COP of all chillers evaluated, they can be used effectively as the main chiller in C^3I room applications. Above 40 percent load, their partial load performance is excellent. Thus, a centrifugal chiller can be installed when gradual future load expansion is anticipated. Before the load is added, the chiller will act as a backup unit to the base chiller. As load is gradually added, the centrifugal chiller will operate efficiently at partial load to satisfy the additional cooling requirements.

c. *Sizes.* Packaged centrifugal chillers are available in sizes ranging from about 80 to 2400 tons. Within the ranges of 200 to 400 tons and 400 to 600 tons, the required cooling capacity can be matched very closely by mixing and matching different compressors, motors, and condensers. Field-erected units can achieve capacities of up to 10,000 tons.

d. *Cost.*

(1) Depending on cooling capacity, centrifugal chillers range in equipment cost from $130 to $280/ton. Cooling towers, condenser water piping and pumps, chilled-water piping and pumps, controls, and installation result in installed costs ranging from $450 to $700/ton.

(2) These costs do not include EMP shielding; see paragraph 5-7.

e. *Advantages.*

(1) Water-cooled centrifugal chillers larger than 400 tons can achieve a COP of up to 5.74 (0.61 kW/ton)--the highest COP of any other chiller evaluated. The partial load performance of centrifugal chillers is very good from 40 to 100 percent load. The ability to continuously modulate capacity with nearly proportionate changes in input power allows the centrifugal chiller to combine precise temperature control and high-energy efficiency.

(2) For a given capacity, centrifugal compressors have a greater volumetric flow of refrigerant vapor than positive displacement compressors. For adequate compression, a centrifugal compressor must spin faster than positive displacement compressors. However, minimal vibration or wear results from the higher speed because of the steadiness of motion and the absence of contacting parts.

(3) For chillers of equal capacity, the volume of centrifugal chillers is larger than reciprocating and rotary screw chillers and smaller than absorption chillers. While ranking third among the evaluated chillers in volume, centrifugal chillers rank much better on the basis of footprint size. Below 450 tons, the footprint of the centrifugal chiller is smaller than that of the rotary screw and absorption chillers. Above 450 tons, the footprint of the centrifugal chiller is smaller in area than that of any other chiller evaluated.

f. Disadvantages. Partial load performance is limited to around 40 percent of full load. At partial loads below 40 percent, unstable operation called "surging" occurs. When a centrifugal chiller surges, refrigerant travels back and forth through the unit, causing considerable vibration, noise, and heat. If allowed to operate in a surging condition for an extended duration, the centrifugal compressor will become damaged.

5-6. Absorption Chillers

a. Operation.

(1) Absorption chillers are similar to vapor compression chillers in that both evaporate and condense refrigerant at two different pressures. However, instead of a compressor, absorption chillers use steam- or gas-fired generators to produce a pressure differential. While vapor compression chillers rely on an expansion valve alone to create low pressure, absorption chillers use a combination of an expansion valve and the strong affinity of an absorbent for the refrigerant to achieve this result.

(2) The most common absorbent/refrigerant combination is lithium bromide/water, with water acting as refrigerant. However, chillers in the 3- to 5-ton capacity range sometimes use an ammonia/water combination, with ammonia as refrigerant.

(3) Figure 5-6 shows a simple lithium bromide/water absorption chiller. Since the refrigerant vapor in this system is generated in one step, the machine is referred to as a single-stage, or single-effect, absorption chiller. Operation of the single-stage absorption chiller consists of the following steps:

(a) Refrigerant vapor from the evaporator is absorbed in the absorber by a concentrated solution with a strong affinity for the refrigerant.

(b) As the solution absorbs more refrigerant, it becomes weaker in its affinity for the refrigerant. The weak solution is pumped to the generator where heat is added to boil off the refrigerant and, thus, regenerate the weak solution into strong solution. The strong solution then flows back to the absorber.

(c) A heat exchanger is used to recover heat from the strong solution returning to the absorber, preheating the weak solution before it enters the generator.

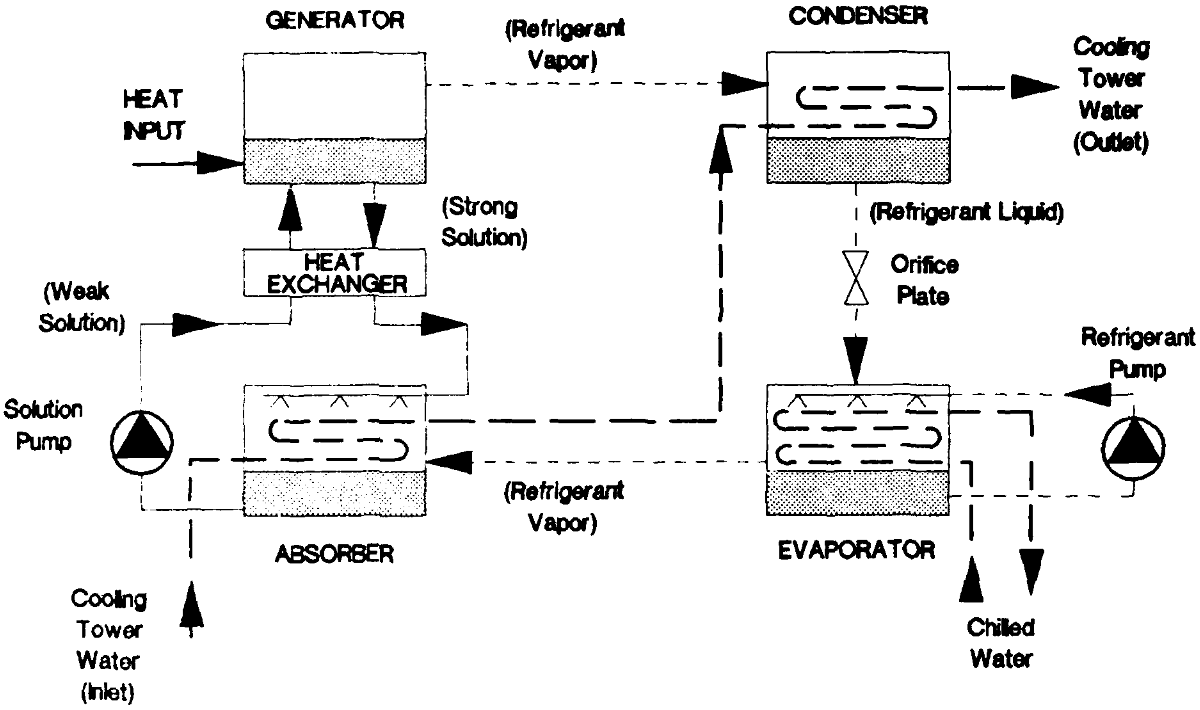


Figure 5-6. Single-stage absorption chiller.

(d) The refrigerant vapor that is released from the solution in the generator is passed through a condenser, expansion valve, and evaporator in a way similar to that in a vapor compression chiller.

(4) A common variation of the single-stage unit is called the "double effect," or "two-stage," absorption chiller. It uses a two-stage generator, which is actually two separate generators: the first stage, or high-temperature generator and the second stage, or low-temperature generator. Heat supplied to the first-stage generator boils off refrigerant from the weak absorbent. The hot refrigerant vapor is passed to the second-stage generator where it is used to drive off additional refrigerant from absorbent of medium strength coming from the first-stage generator. Approximately 30 to 40 percent more refrigerant vapor is generated in two-stage units than in single-stage units. A two-stage unit uses about 35 percent less energy per ton of cooling than a single-stage machine.

(5) In a single-stage unit, the optimal generator temperature is about 200 °F. The most common heat source for single-stage machines is low-pressure steam. Two-stage absorption chillers require generator temperatures of about 300 °F. Consequently, they require high-pressure steam or must be direct-fired with natural gas or oil.

(6) The cooling capacity of absorption chillers is controlled by regulating the amount of heat supplied to the generator. This mechanism modulates the concentration of the absorbent and, in turn, the flow of refrigerant to the evaporator.

(7) The thermal COP used to rate the performance of an absorption chiller cannot be directly compared to the mechanical COP of vapor compression chillers. To compare the performance of an absorption chiller with that of a vapor compression

chiller, the COP of the vapor compression chiller must be modified to include the efficiency of the electricity-generating plant and distribution system. For example, if the efficiency of the generating plant and distribution system is 33 percent, then the comparable COP of a vapor compression chiller with a mechanical COP of 5.0 is 1.65 (5.0 x 33 percent). Single-effect absorption chillers have thermal COPs averaging between 0.5 and 0.7, whereas two-stage absorption chillers can have thermal COPs of up to 1.06.

b. *Principal Applications.*

(1) Lithium bromide/water units are used more for large commercial cooling loads and ammonia/water units tend to be used more for residential or small commercial applications. However, ammonia/water absorption chillers have been used for large industrial process cooling applications requiring low temperatures.

(2) Lithium bromide/water absorption chillers are typically water-cooled by evaporative cooling towers whereas ammonia/water units are usually air-cooled. Water-cooled, lithium bromide/water absorption chillers may be either direct-fired, using natural gas or fuel oil, or indirectly fired, using steam, hot water, or waste heat. Small air-cooled, ammonia/water absorption chillers are typically available only with direct firing.

(3) Direct-fired, lithium bromide/water absorption chillers are two-stage units. Direct-fired systems can provide both heating and cooling by passing hot refrigerant vapor from the generator to a hot-water heat exchanger. Indirect-fired units can be either single- or double-effect and are particularly advantageous when adequate quantities of waste heat are available.

(4) Other factors that must be addressed when comparing an absorption chiller with an electrical chiller include the ability to install a chimney, gas service piping requirements (direct-fired only), and available electrical service (new chillers). Furthermore, the absorption chiller can also be used for heating.

c. *Sizes.* Direct-fired, lithium bromide/water, double-effect units are available in sizes ranging from 7.5 to 1500 tons. Indirect-fired, lithium bromide/water units are available in the single-effect configuration at sizes from 1.3 to 1660 tons and in the two-stage design at sizes from 100 to 1500 tons. Gas-fired, ammonia/water, air-cooled units are sized from 3 to 5 tons, or as combination packages from 8 to 25 tons.

d. *Cost.*

(1) Steam single-stage absorption chillers in the 100-ton range cost from $300 to $500/ton for equipment only. Installed costs in this range are between $730 and $1100/ton. Over 500 tons, the material costs fall to between $230 and $300/ton and installed costs run from $480 to $620.

(2) For steam double-effect absorption chillers, the equipment costs in the 100-ton range are between $540 and $900/ton. Over 500 tons, double-effect chillers have equipment costs ranging from $380 to $440/ton. Double-effect chillers in this range have installed costs that vary from $780 to $840/ton. Installed costs include condenser and chilled water-piping and pumps, cooling towers, controls, and installation. The costs of EMP shielding are not included; these costs are addressed in paragraph 5-7 below.

e. Advantages.

(1) Gas-fired absorption chillers have low electrical power requirements. For applications where electricity demand charges are high and gas prices are low, the operating cost of absorption chillers can be lower than that of electric vapor compression chillers.

(2) In cogeneration configurations, hot exhaust gases can be used as the heat source for indirect-fired systems. Other forms of waste heat, such as incinerator heat recovery, can also be used in indirect-fired systems.

(3) Gas-fired absorption chillers can also supply heat, eliminating the need for a separate boiler. In general, absorption chillers provide quiet, vibration-free, reliable operation.

f. Disadvantages.

(1) The major disadvantage of absorption chillers is their high first cost compared with electric chillers. In addition, both single- and double-effect absorption chillers require cooling towers with about 60 percent more cooling capacity than do vapor compression units of equal size due to the extra cooling requirement of the absorber. The condenser water piping and pumps must be correspondingly larger than those for an electric chiller of equal capacity. In facilities not using a gas boiler for heating, the chimney and flue piping required by an absorption unit is an additional cost compared with electric units.

(2) In general, absorption chillers have larger space requirements than electric chillers of equal capacity. Response times are also longer than for electric chillers.

5-7. Other Considerations

a. Gas Engine-Driven Chillers. In addition to the chillers evaluated, other types of chillers used in the past and currently being redeveloped are used in certain applications. One of these is the gas engine-driven chiller. In large applications, the chillers are field-assembled using commercially available components. Large diesel engines are typically used to drive the compressor. Packaged units in the 150- to 450-ton range that use automotive gasoline engines are currently under development. For a complete evaluation in this report, the gas engine-driven chiller is described. However, since the packaged chillers in the capacities applicable to this report (i.e., 100 to 600 ton) are still under development, their reliability has not been proven adequately. Therefore, the gas engine-driven chiller was not ranked in the evaluation process.

(1) Operation.

(a) Figure 5-7 is a simplified schematic of a typical gas engine-driven chiller system. Gas engine chiller systems couple the shaft output from a gas-powered engine to the compressor of an open-drive mechanical chiller. Coupling is usually done through an intermediate gearbox in order to match the optimal speeds of the engine and compressor.

(b) Many different options are possible, depending on the size and application. In small sizes, the engine could be a conventional reciprocating internal combustion engine (IC), rotary engine, or a Stirling cycle engine. The compressor driven by these engines could be the reciprocating or rotary design. For larger units, the IC engine is usually the most effective. Reciprocating, screw, or centrifugal compressors could, in principle, all be coupled to a gas engine drive. However, centrifugal compressors may not be a good application because of potential problems with excessive shaft harmonics caused by the engine drive. Reciprocating units are generally limited in size to less than about 120 tons, and do not provide the operating speed range necessary to take full advantage of the engine's inherent variable speed capability. Screw compressors appear to offer the best match for larger gas engine-driven chiller systems.

(c) Because a typical engine converts less than one-third of its fuel input to shaft power, opportunities exist for recovery of engine waste heat. This waste heat can be recovered from three sources: (1) jacket cooling water, (2) exhaust gases, and (3) lubrication system heat. The recovered heat can be used directly for space heating or hot-water heating, or to fire an absorption chiller to obtain additional cooling capacity from the unit.

(d) Engine-driven chillers are usually field-assembled units. However, several manufacturers are developing packaged applications. Industrial-type natural IC

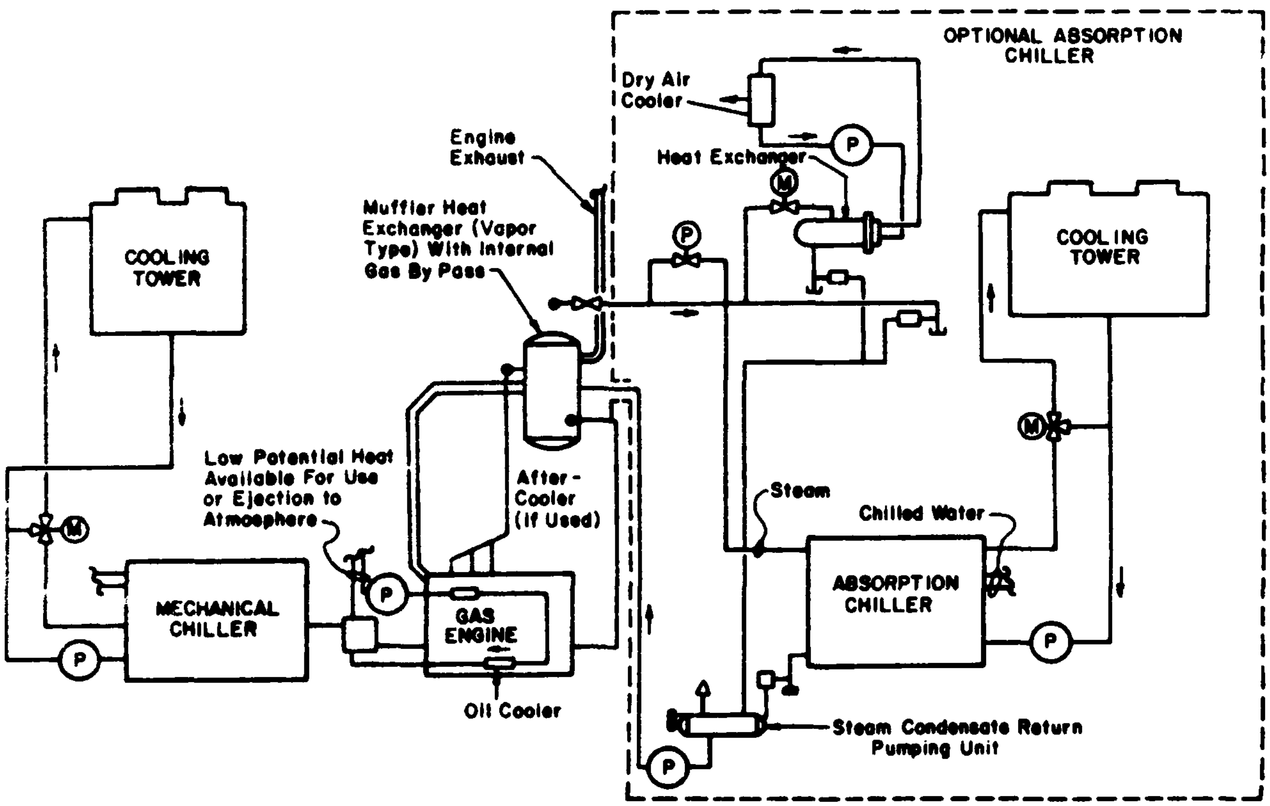


Figure 5-7. Main components of a gas engine-driven chiller. (Source: *Natural Gas Cooling Manual* [American Gas Association, 1982]. Used with permission.)

engines are available from several manufacturers with efficiencies of 20 to 32 percent (shaft power to fuel input), based on the higher heating value of natural gas.

(e) Gas engines can be categorized as naturally aspirated or turbocharged. Turbocharging increases the power production and, to a lesser extent, the efficiency of gas IC engines, by compressing combustion air and therefore allowing a greater amount of fuel to be burned in each cylinder. For example, a 100-kW naturally aspirated engine with an efficiency of 29.9 percent will produce 150 kW at 30.8 percent efficiency when turbocharged. Current open-drive screw chillers have COPs of 4.7 to 5.2 (0.9 to 1.0 HP/ton). Field-assembled chillers, using standard compressors and heat exchangers, can be constructed with COPs as high as 5.2 (0.9 HP/ton).

(f) With these engine and chiller efficiencies, systems can be assembled with overall COPs of up to 1.7. When coupled with a single-stage absorption chiller utilizing the engine waste heat at a COP of 0.7, the COP can increase up to 2.0. A current package under development at the Gas Research Institute (GRI) has achieved COPs of 1.50 without absorption and 1.9 with absorption. Partial load performance of engine-driven chillers is generally very good because of the ability to operate at variable speeds.

(2) Principal Applications.

(a) Any facility with a central chilled-water plant is a candidate for a gas engine-driven chiller. However, these units are typically found in applications where the waste heat can be used for hot water or space heating.

(b) It is feasible for a gas engine-driven chiller to serve as standby to an electric-driven chiller, providing backup cooling in the event of an electrical power failure. However, the liquid chilling needs of a C^3I facility require an uninterrupted flow of chilled water. The gas engine-driven chiller would take several minutes to bring online and provide useful cooling. The backup cooling requirements of a C^3I facility would be better served by the storage system described in Chapter 3.

(3) Sizes. Individual off-the-shelf components can be combined in a wide range of sizes, from 20 tons up to 700 tons or possibly higher. Reciprocating compressors are the most appropriate for the lower end of this range (less than 100 tons), whereas screw compressors are better suited for the larger sizes.

(4) Cost. Industrial-type gas engines cost between $100 and $150/HP for the engine alone. With accessories, unit costs run from $180 to $250/HP. Open-drive chiller costs range from about $200 to $300/ton. Installed costs of engine/chiller combinations range from $500 to $650/ton. Cooling towers, condenser water piping and pumps, chilled-water piping and pumps, controls, and installation result in total installed costs ranging from $830 to $1100/ton.

(5) Advantages/Disadvantages. The main advantages of engine-driven systems are the potential for high thermal overall COPs (relative to direct-fired absorption systems) and subsequently lower operating costs, and the variable speed capability that will improve partial load performance. The disadvantages of these systems are potentially poor engine reliability and maintenance problems, noise, the lack of currently available packaged systems, and the high first cost of current systems, particularly those with an absorption chiller heat recovery feature.

b. EMP Shielding.

(1) EMP shielding of liquid chilling systems is typically achieved by surrounding the equipment and controls with a conductive barrier that is grounded separately to dissipate EMP voltages and currents. Piping penetrations in the shield are treated by welding the pipe circumferentially to the shield (see Chapter 4). Electrical penetrations are treated with various methods using filters and surge-limiting devices. There are many sources describing EMP hardening techniques and design, as listed earlier. MIL-HDBK-419 provides standardized information on EMP hardening of electronic equipment for DOD.

(2) It is very difficult to estimate EMP shielding costs. No single design is suitable to the majority of applications. In addition, EMP shielding costs per unit volume vary widely, depending on the shield volume. Hardening an entire mechanical room could cost between $5 and $10/cu ft. However, protecting one small electric motor could cost between $100 and $200/cu ft. Providing an EMP shield around a 200-ton chiller could cost between $15 and $25/cu ft. Because of the ventilation requirements, shielding costs for an air-cooled condenser of similar size could increase to a range between $30 and $50/cu ft. It should be noted that these cost estimates are based on rough generic designs and that actual costs may vary significantly.

Section III: SUMMARY EVALUATION

5-8. How To Use the Evaluation Table

a. Table 5-1 ranks each liquid chilling system for the criteria presented in **Section I.** These rankings are qualitative, with a black circle in the table representing a high ranking, a gray circle denoting a medium ranking, and a white circle indicating a low rating. It should be noted again that a low rating simply means other systems are more suitable for a specific criterion, and not necessarily that the system is a poor choice in general.

b. When ranking each system, certain assumptions were made regarding the system configuration. For example, when considering life-cycle cost, it was assumed that all systems are served by an evaporative cooling tower in order to evaluate the different liquid chilling systems on an equal basis.

5-9. Evaluation Results

a. Load Expansion.

(1) When load expansion is the primary consideration, the rotary screw and centrifugal chillers have the best ranking. These liquid chilling systems can continuously modulate capacity in response to cooling load without excessive cycling or on/off operation. Therefore, both the rotary screw and centrifugal chillers can adapt smoothly to gradual load expansion.

Table 5-1

Summary Evaluation: Liquid Chilling Systems

	Load expansion	Backup cooling	Reliability	* EMP shielding cost	* Life-cycle cost
1) Reciprocating Chillers	Medium	High	Medium	High	Medium
2) Rotary Screw Chillers	High	High	High	High	High
3) Centrifugal Chillers	High	High	High	High	High
4) Absorption Chillers	Low	High	Medium	Low	Low

● = High ◎ = Medium ○ = Low

* A high ranking for cost-related criteria indicates a low relative cost. A low ranking indicates a high relative cost.

(2) When significant steps of load are added to the C^3I room, the availability of chillers with a variety of incremental steps of capacity becomes a consideration. Both reciprocating and rotary screw chillers are available with small incremental steps of capacity. However, since a more gradual load expansion would cause the reciprocating chiller to experience a certain amount of cylinder loading and unloading, it was given a medium ranking. Absorption chillers received a low rating primarily because their larger size per ton of cooling limits the amount of capacity that can be added within the mechanical room.

(3) The partial load performance of a chiller at very low load can have an impact on the ability to handle load expansion. For example, to plan for future load expansion, more capacity than is needed at present can be installed. When the load increases, a chiller with good partial load performance can operate satisfactorily until more load is added so that the extra capacity is used at full load. The reciprocating and rotary screw chillers have good partial load performance at low loads.

b. *Backup Cooling Capacity*. The capacity of cooling systems to provide 15 min of backup cooling in the event of a power failure was discussed in detail in Chapter 2. All chillers evaluated are capable of providing chilled water at temperatures adequate to satisfy the cooling requirements of C^3I room space and equipment. All are capable of supplying the backup cooling system described in Chapter 3. For this reason, all chillers evaluated received a high ranking for this criterion.

c. *Reliability*.

(1) All chillers evaluated are generally reliable. For vapor compression chillers, relative reliability is primarily a function of the compressor. The use of an air-cooled condenser will result in increased gas discharge temperatures, which, in turn, will reduce the equipment service life. On the other hand, air-cooled condensers generally require less maintenance than cooling towers. However, to compare the chillers on an equal basis under this criterion, it was assumed that all units are served by an evaporative cooling tower.

(2) The steadiness of motion and minimum number of contacting parts in a centrifugal compressor result in little wear and vibration. The *1987 ASHRAE HVAC Systems and Applications Handbook* lists an expected service life of 23 years for centrifugal chillers. Thus, centrifugal chillers received a high rating for reliability.

(3) Rotary screw chillers have been typically used in low-temperature industrial process cooling applications that require rugged, durable equipment. In air-conditioning applications, they show the same high reliability. Rotary screw compressors use an oil injection system to lubricate, cool, and seal. Since the rotary screw compressor is designed to handle oil, it can also tolerate liquid refrigerant slugging without damage. Therefore, the rotary screw chiller received a high rating.

(4) The reciprocating chiller, though generally reliable, has the most contacting components of any of the chillers evaluated. The suction and discharge valves are the components with the greatest potential for failure. Even though the *1987 ASHRAE HVAC Systems and Applications Handbook* shows an expected service life of 20 years, this is less than the 23 years listed for centrifugal and absorption chillers. Therefore, the reciprocating chiller was given a medium rating.

(5) ASHRAE estimates the service life of an absorption chiller at 23 years. Absorption chillers have strict requirements for keeping all internal surfaces clean and all vacuum vessels airtight for the life of the equipment. Failure to prevent the accumulation of air and other noncondensable gases can allow salt to crystallize inside the unit. Crystallization renders the chiller inoperable and can corrode internal components over time. While crystallization is not a problem in new, well maintained absorption chillers, the service life and trouble-free operation depend on the success of efforts to keep the unit clean and airtight. Thus, while the estimated service life of 23 years equals that of the centrifugal chiller, more exacting maintenance procedures are required to keep the absorption chiller operational. For this reason, the absorption chiller was given a medium ranking.

d. Least Cost for EMP Shielding.

(1) If a chiller serves an area that is EMP-protected, then the chiller and related equipment must also be provided with EMP protection. If the entire mechanical room is shielded, the overall size of the cooling equipment (and, therefore, the size of the mechanical room) will impact the cost of EMP protection. If only the electric motors and controls are protected, the size and number of the motors and controls dictate the cost of EMP shielding. For a particular piping configuration, the cost to protect the integrity of the C^3I room EMP shield from piping penetrations is the same for each of the chillers evaluated, since the same flow of chilled water and, therefore, the pipe size are the same for each chiller.

(2) In cases for which the entire mechanical room is EMP-shielded, the chillers are ranked according to increasing total volume since larger chillers would require correspondingly larger mechanical rooms and, therefore, would carry higher costs for EMP shielding. It should be noted that chiller package footprints and volumes for some types of chillers vary among manufacturers. This evaluation is based on a review of specific chiller packages and is a generalization.

(3) In cases for which only the electric motors and controls are EMP-shielded, the vapor compression chillers are ranked equally since their compressor, chilled-water pump, and condenser water pump motors and controls are approximately the same size for a given capacity. While electric motor requirements of an absorption chiller are

small, two pump motors (the solution and evaporator pumps) must be EMP-shielded compared with the one large compressor motor for the vapor compression chillers. The difference in cost between EMP shielding two small motors and that to shield one large motor is not great. The labor cost of protecting two motors offsets the reduced material cost of the smaller shields.

(4) Another factor is that absorption chillers require a cooling tower that is approximately 60 percent larger in capacity than a vapor compression chiller of equal size. Therefore, the related condenser water pumps, motors, and piping are larger for an absorption chiller. In conclusion, when only the electric motors are EMP-shielded, it costs more to protect an absorption chiller than to protect a vapor compression chiller.

(5) Since it is more typical to EMP-shield the entire mechanical room than to protect just the electric motors and controls, the rankings are based on chiller volume and footprint area. The reciprocating, rotary screw, and centrifugal chillers received a high ranking, and the absorption chiller was ranked low. A ranking of high in the least cost for EMP shielding criterion indicates a relatively low cost. Conversely, a low rating denotes a relatively high cost for EMP shielding.

e. Lowest Life-Cycle Cost.

(1) Components of the total life-cycle cost of a liquid chilling system include the present value of: the energy operating costs over the economic life of the equipment; the investment costs of the equipment design, purchase, and installation; the annually recurring (nonfuel) operating and maintenance (O&M) costs; the nonannually recurring (nonfuel) O&M costs; the replacement costs; and the salvage value. A life-cycle cost analysis was conducted for chiller capacities of 100, 200, 400, and 600 tons. Based on this analysis, the centrifugal chiller has the lowest life-cycle cost for all capacity sizes except 100 tons and, therefore, was given a high ranking. In the 100-ton capacity, the centrifugal chiller was less than 1 percent higher in total life-cycle cost than the rotary chiller, which had the lowest cost in this size category. Though first cost is lower for the centrifugal chiller in the higher capacity ranges, the primary reason for the high ranking of the centrifugal chiller was its high COP and associated low energy operating costs.

(2) The rotary screw chiller had the next lowest life-cycle cost and received a high ranking. As mentioned, in the 100-ton size, the rotary screw chiller had the lowest life-cycle cost. In the other sizes, the rotary screw chiller was second to the centrifugal chiller in lowest life-cycle cost. Again, energy operating costs had the most impact on the life-cycle cost and the COP of the rotary screw chiller was second only to the centrifugal chiller.

(3) The reciprocating chiller had relatively high energy operating costs and, in turn, high life-cycle costs. In the 400- and 600-ton analyses, multiple reciprocating chillers, each rated at 200 tons, had to be used because reciprocating chillers are not available in sizes above 250 tons. Thus, the reciprocating chiller received a medium ranking. The absorption chiller had the highest energy operating costs and, therefore, the highest life-cycle cost of any of the chillers evaluated. The absorption chiller was given a low rating for this reason.

(4) The energy prices and electrical demand charges used in the life-cycle cost analysis are national averages. Local fuel prices and electrical demand charges can have dramatic impacts on the results of the analysis. For example, if the local electrical demand charges are high and the natural gas prices are low, the absorption chiller may have lower energy operating costs than some of the vapor compression chillers. Also, if

low-cost waste heat is available, the energy operating cost of the absorption chiller would be reduced greatly.

(5) The chiller and chilled-water distribution system configuration can impact the life-cycle cost of liquid chilling systems. For example, certain electronic equipment requires direct water cooling. The temperature of chilled water required varies between equipment and between models. Typical ranges are from 50 to 70 °F. If a chiller is dedicated to serving water-cooled electronic equipment, the chilled water temperature setpoint can be raised above the standard setpoint of 45 °F. Raising the chilled water temperature setpoint improves the COP, reduces the present value of the energy costs, and reduces the life-cycle cost.

(6) Another example in which the chilled water temperature setpoint can be raised is a case in which air handlers within the C^3I room provide sensible cooling only. Humidification and dehumidification are controlled by a central system served by the building chiller. Occupant areas in the C^3I room are also served by an air handler that receives chilled water from the building chiller. A dedicated chiller serves water-cooled electronic equipment and the air handlers that provide sensible cooling only. The electronic equipment and air handler loads can be satisfied with 50 °F chilled water. The COP of the dedicated chiller is improved by raising the chilled water setpoint, and the total life-cycle cost is reduced.

(7) The installation of multiple chillers in C^3I rooms for redundancy and backup has significant impacts on partial load performance and life-cycle costs. Frequently, three chillers, each capable of handling more than the total cooling load, are installed in a C^3I facility. This is sometimes referred to as the "N+2 rule" as described earlier. Normally, two of the chillers are run at partial load to satisfy the cooling requirements. The third chiller is used as a backup. If one of the two operating chillers fails, the other can handle the whole load, while the third chiller still provides the required redundancy. Since two chillers are normally running at less than full load, the partial load performance of the selected chillers dictates the energy operating costs and, therefore, the life-cycle costs.

6 CONCLUSION

6-1. Four types of cooling systems have been evaluated in terms of different performance criteria to present guidance for C^3I facility designers. The four types of systems were: air distribution, chilled-liquid distribution, air-handling, and liquid chilling systems. Performance criteria included flexibility in meeting load redistribution and load expansion requirements, humidity control, particulate matter filtering ability, static electricity control, backup cooling, reliability, lowest cost for EMP shielding, and lowest life-cycle cost. Advantages and disadvantages of each type of unit have been presented. Relative costs have been estimated to allow comparisons.

6-2. The information in this report is not intended to replace the design analysis. The goal was to provide designers with a comprehensive assessment of the systems and components available and their predicted performance under a variety of conditions. Cooling systems must be designed on a case-by-case basis, with the designer considering factors such as cost, location, and mission criticality of the C^3I facility.

METRIC CONVERSION TABLE

1 in.	=	2.54 cm
1 ft	=	0.305 m
1 sq ft	=	0.092 m^2
1 cu ft	=	0.028 m^3
1 lb	=	0.453 kg
1 ton	=	907.2 kg
1 gal	=	3.785 L
1 psig	=	6.895 kPa
1 HP	=	0.75 kW
°F	=	(°C x 1.8) + 32

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ABBREVIATIONS

AGA	American Gas Association
AHU	air-handling unit
ARI	American Refrigeration Institute
ASHRAE	American Society of Heating, Refrigerating, and Air-Conditioning Engineers
C^3I	command, control, communications, and intelligence
CFM	cubic feet per minute
COP	coefficient of performance
CV	constant volume
DOD	Department of Defense
DX	direct expansion
EMP	electromagnetic pulse
FPM	feet per minute
GPM	gallons per minute
GRI	Gas Research Institute
HP	horsepower
IC	internal combustion
id	inside diameter
HVAC	heating, ventilating, and air-conditioning
LF	linear foot
N/A	not applicable
NBC	nuclear, biological, and chemical [contamination]
NPSH	net positive suction head
od	outside diameter
O&M	operation and maintenance
OS&Y	outside screw and yoke
PSIG	pounds per square inch (gauge)
RFI	radiofrequency interference
UPS	uninterruptible power supply
UPW	uniform present worth
VAV	variable-air-volume

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