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INTERIM DESIGN CRITERIA. TECHNICAL GUIDELINES FOR ENERGY CONSER--ETC(U)
JAN 75 K N PATEL, H C SHANER, P J SARACENI N00025-74-C-0022

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INTERIM DESIGN CRITERIA

TECHNICAL GUIDELINES FOR ENGERGY CONSERVATION IN NEW BUILDINGS

JANUARY 1975

DEPARTMENT OF THE NAVY
NAVAL FACILITIES ENGINEERING COMMAND
200 STOVALL STREET
ALEXANDRIA, VA. 22332

SECURITY CLASSIFICATION OF THIS PAGE (*When Data Entered*)

REPORT DOCUMENTATION PAGE		READ INSTRUCTIONS BEFORE COMPLETING FORM
1. REPORT NUMBER	2. GOVT ACCESSION NO.	3. RECIPIENT'S CATALOG NUMBER
4. TITLE (*and Subtitle*) Interim Design Criteria Technical Guidelines for Energy conservation in New Buildings		5. TYPE OF REPORT & PERIOD COVERED Interim Design Criteria
		6. PERFORMING ORG. REPORT NUMBER
7. AUTHOR(*s*)		8. CONTRACT OR GRANT NUMBER(*s*) N00025-74-C-0022
9. PERFORMING ORGANIZATION NAME AND ADDRESS Kling-Lindquist, Inc., Engineers Philadelphia, Pa. 19107		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS
11. CONTROLLING OFFICE NAME AND ADDRESS Naval Facilities Engineering Command 200 Stovall Street Code 043 Alexandria, Va. 22332		12. REPORT DATE January 1975
		13. NUMBER OF PAGES 207
14. MONITORING AGENCY NAME & ADDRESS(*if different from Controlling Office*)		15. SECURITY CLASS. (*of this report*)
		15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (*of this Report*)

Unclassified/ unlimited

17. DISTRIBUTION STATEMENT (*of the abstract entered in Block 20, if different from Report*)

18. SUPPLEMENTARY NOTES

19. KEY WORDS (*Continue on reverse side if necessary and identify by block number*)

design criteria new
energy conservation

20. ABSTRACT (*Continue on reverse side if necessary and identify by block number*)

This manual contains description, illustration and design criteria to be applied for energy conservation in buildings. Also included in the manual is the description and our recommendation of commercially available computer programs together with the sample printouts of a computer analysis on an example building.

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PREPARED FOR:
THE NAVAL FACILITIES ENGINEERING COMMAND
NAVY CONTRACT N00025-74-C-0022

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ABSTRACT

This submittal comprises final draft of design manual entitled "Technical Guidelines for Energy Conservation in New Buildings". The manual contains description, illustration and design criteria to be applied for energy conservation in buildings. Also included in the manual is the description and our recommendation of commercially available computer programs together with the sample printouts of a computer analysis on an example building.

INTERIM DESIGN CRITERIA

"TECHNICAL GUIDELINES

FOR

ENERGY CONSERVATION

IN

NEW BUILDINGS"

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INTRODUCTION TO ENERGY MANAGEMENT

MANAGEMENT FOR CONSERVATION. The management of energy conservation for a given facility, like the management of any other function, requires planning, organizing, implementing and controlling the energy expenditure. For a facility not yet built, the planning involves the specification of certain guidelines, such as, the building envelope and ventilation criteria given in Chapter 1. It requires definition and evaluation of alternatives as outlined in the computer analysis discussion in Chapters 1 and 5 and the economic decisions discussed in Chapter 6. For every building project an energy balance diagram (Figure 2-2) should be prepared to graphically display the amount of internal energy available for recycling and to show the minimum amount of new energy required to maintain environmental conditions. This chart will guide the selection of alternative energy conservative systems. Once the alternative systems have been analyzed and one system selected, the computer model printout for that system should be preserved for later use by the responsible building operating personnel. In organizing or specifying, the building systems, the performance characteristics desired (especially coefficients of performance) must be clearly spelled out, and in implementing the design sufficient tests should be made of systems and their components to assure that, as installed, they meet the design requirements. On every project the building operator should be furnished with a complete set of as-built drawings and specifications and, also, with systems diagrams, operating and maintenance instructions, instructions for recording usage and demand of all energy sources. Actual usage of energy versus that estimated by the designer should be reported on a regular basis to assure that necessary corrections are made to controls for energy efficient operation. For each installation the officer-in-charge should designate an energy conservation officer to coordinate conservation efforts.

CHAPTER 1. DESIGN PARAMETERS

The United States Government, one of the largest users of energy, can contribute considerably in achieving the President's energy conservation goal. About 85% of the total U. S. Government expenditure for electric energy is attributable by the Department of Defense (DOD). Energy Conservation programs may require changing DOD construction standards, and creation of energy management programs. To be effective, these programs must begin in the design stages of the new construction when establishing exterior envelope and thermal criteria requirements. Energy conservation design criteria will include consideration of siting or orientation, outdoor design conditions, indoor design conditions, ventilation and infiltration rates, solar screening; building envelope, insulation factors - and methods for calculating heating and cooling loads.

The purpose of this manual is to present criteria for the design of energy efficient buildings with full consideration of life cycle costs.

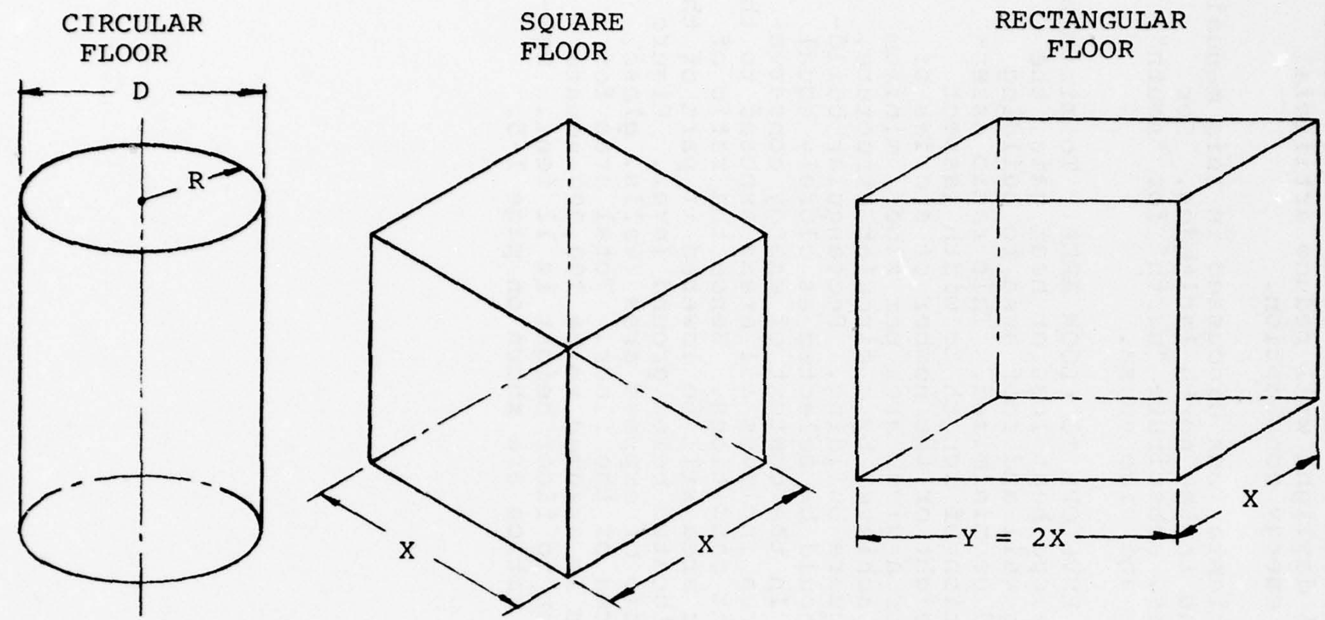
Section 1. SITING AND GEOMETRY

1.1.1 BUILDING SURROUNDING. A preferred location for a completely air-conditioned building is the one that will take maximum advantage of any available natural shade such as from trees, hills or adjacent buildings. This type location would not be desirable for a building with little or no air conditioning load but with a large heating load. A location close to a marsh or swampy area, will increase the ambient design wet bulb and increase cooling load due to higher moisture content in the outside air. Tall trees, hills or other natural barriers will function to break the penetrating wind. Also avoid when possible a location that is too close to industrial plants that are discharging pollutants such as flue, gases, exhaust from processes, steam and other harmful pollutants which will increase the maintenance, replacement and energy requirements of filters.

1.1.2 COMPASS ORIENTATION. Energy required for heating and cooling can be significantly affected by building orientation. For air conditioned buildings consider the requirements of both heating and cooling seasons when establishing the optimum building orientation. Factors that require evaluation include prevailing wind direction, solar-altitude, intensity and azimuth. In regions where heating is required, facade in the direction of the prevailing wind should be designed to reduce infiltration leakages and hence the corresponding heating and cooling loads. In tropical regions, the building should be oriented to gain advantage of natural ventilation. The degree of exposure to solar radiation has a major impact on the energy requirements of all heated and cooled buildings. Wall and glass exposure to the west, south and southwest should be minimized for mechanically cooled buildings. Locating the elevator shafts and perimeter corridors along the west exposure will reduce the peak afternoon solar heat gain. Where there is a lobby requiring large glass area, locate this exposed to south and put overhang above the glass. This will minimize heat gain from higher sun elevation in summer and take maximum advantage of the winter sun. In heated but not cooled buildings east, west and south exposure should be designed to take maximum advantage of winter solar heat. Utilization of natural daylight will reduce artificial lighting use and energy consumption.

NOTE: Compass orientations discussed in this manual refer to locations in the north latitudes. For southern latitudes, substitute "north" for "south" solar orientation and vice versa.

1.1.3 RATIOS OF ENVELOPE TO FLOOR AREA. To minimize the building envelope heat loss or heat gain, the ratio of building wall and roof area to building floor area should be minimized. This ratio essentially is a function of length to width (aspect ratio) and the height or the number of stories of the building. For a given area per floor, minimum envelope area is achieved by a circular structure. Next best is a square building. Rectangular building dimensions should be selected as closely equal as is feasible. In the context of energy conservation, envelope area includes all areas exposed to the outside or ambient conditions. Hence the ratio of envelope to floor area will be lowered if part of the building can be located below ground level. Figure 1-1 shows the ratio of exposed area (walls, glass, roof) per unit area of the floor. Total area for this illustration is assumed to be 100,000 square feet and the floor to floor height is 12 feet. Calculation of area ratios are shown on page 1-5.



OPTIMUM E/F = 0.496 E/F = 0.538 E/F = 0.56

THE OPTIMUM VALUE OF EXPOSED AREA PER UNIT AREA OF THE FLOOR E/F, (FOR 10^5 SQUARE FEET BUILDING) SHOWS THAT THE VARIATION IS FROM 0.496 to 0.612, I.E. +24%.

FIGURE 1-1

EXPOSED AREA FOR VARIOUS CONFIGURATIONS

SEE COMPUTED RATIOS ON PAGE 1-5

TABLE 1-1

RATIO OF EXPOSED AREA TO FLOOR AREA E/F (10^5 SQ.FT.)

	3 FLOORS Height 36'	5 FLOORS Height 60'	6 FLOORS Height 72'	8 FLOORS Height 96'	10 FLOORS Height 120'
Circular Floor	D = 206'-1" E/F = 0.566	D = 159'-6" E/F = 0.5	D = 145'-8" $\boxed{\text{E/F = 0.496}}$	D = 126'-2" E/F = 0.505	D = 112'-10" E/F = 0.525
Square Floor Aspect Ratio = 1	X = 182'-7" E/F = 0.596	X = 141'-5" E/F = 0.539	X = 129'-0" $\boxed{\text{E/F = 0.538}}$	X = 111'-10" E/F = 0.554	X = 100'-0" E/F = 0.580
Rectangular Floor Aspect Ration = 2	Y = 258'-2" X = 129'-1" E/F = 0.612	Y = 200'-0" X = 100'-0" $\boxed{\text{E/F = 0.56}}$	Y = 182'-6" X = 91'-3" E/F = 0.561	Y = 158'-0" X = 79'-0" E/F = 0.58	Y = 141'-4" X = 70'-8" E/F = 0.609

Section 2. OUTDOOR DESIGN CONDITIONS

1.2.1 OUTDOOR DESIGN. Ambient outdoor design conditions include dry bulb temperature °F, wet bulb temperature °F, dew-point temperature °F, percent relative humidity, air velocity; solar intensity-direct, diffuse; cloud cover, and variation of temperature (diurnal) for each latitude, season and time of day. Design conditions are given in the current edition of the Joint Services Manual NAVFAC P-89, TM 5-785, AFM 88-8, Chapter 6, "Engineering Weather Data." (For any missing or additional data, the latest copy of ASHRAE Handbook of Fundamentals should be used.) Outdoor design condition when used in conjunction with other parameters such as indoor design, ventilation, infiltration, envelope requirements, etc. determine the building envelope heat gain and loss.

1.2.2 SUMMER OUTDOOR DESIGN. Summer outdoor conditions for environmental cooling systems shall be based on the 2-1/2 percent frequency dry bulb temperature and the 5 percent frequency wet bulb temperature as specified in the Joint Service Manual P-89.* This does not apply to those priorities (1 through 4, see DOD 4270.1-M, paragraph 8-5.2) where specialized technical requirements demand using one percent frequency for both dry bulb and wet bulb temperatures. Summer outdoor frequency for United States locations (including Alaska and Hawaii) relates to the four warmest months, June through September (2,928 hours). This means that the outdoor dry bulb design temperature (2-1/2 percent frequency) will be exceeded for approximately 75 hours and the wet bulb temperature (5 percent frequency) will be exceeded for approximately 150 hours. When calculating the solar heat gain, and for energy analysis, haze cover or cloud cover must be accounted for to avoid oversizing of the system capacity and to give realistic energy consumption. Also allow for shading coefficient due to screening, blinds, barrier, etc.

1.2.3 WINTER OUTDOOR DESIGN CONDITIONS. The outdoor design temperature shall be based on 97-1/2 percent frequency as specified in the Joint Service Manual P-89.* Winter outdoor frequency for United States locations (including Alaska and Hawaii) represents the three coldest months, December, January and February (2160 hours). This means that the outdoor temperature can be expected to fall below the design value for approximately 54 hours per typical year.

*NOTE: Joint Service Manual P-89 of 15 June 1967, in effect at the time of preparation of this manual, is being revised. The revised P-89 will publish the coincident dry bulb and wet bulb temperatures to be used for summer conditions. The wet bulb temperature coincident with the 2-1/2 percent dry bulb temperature will be a more realistic design criterion.

Section 3. INDOOR DESIGN CONDITIONS

1.3.1 INDOOR DESIGN CONDITIONS. The indoor environmental design parameters suggested herein are to be used as criteria for human comfort in general applications. For special usage such as hospitals, laboratories, etc., indoor design conditions for temperature and humidity shall be those suggested in the latest copy of ASHRAE Applications Handbook. Thermal comfort is established by controlling the dry bulb temperature, mean radiant temperature (MRT), relative humidity (water pressure) and air cleanliness and velocity (distribution). Exterior envelope design influences the mean radiant temperature. Lower coefficient of heat transfer values for all exterior envelope components (glass, wall, roof, floor) improves the comfort conditions by raising the MRT in winter and reducing the MRT in summer.

1.3.2 SUMMER INDOOR DESIGN CONDITION. For general personnel comfort applications the indoor summer design condition shall be 78°F dry bulb and 60 percent maximum relative humidity. In reheat systems, reheat capacity shall be limited to that required to provide a dry bulb temperature of 73°F.

1.3.3 WINTER INDOOR DESIGN CONDITION. Shall be 70°F and 25 percent relative humidity for conventional facilities. Special areas such as hospitals and laboratories shall be in accordance with ASHRAE Standard 55-74. For utilitarian type facilities such as shops, warehouses, etc. there is no humidification required (except for processing plants), and indoor design temperature shall be 65°F, except for storage areas which shall be 40°F for prevention of freezing. Winter humidification enhances human comfort by reducing evaporation from the membranes of the nose and throat. Humidification also reduces drying of the skin and hair; unpleasant static sparks to people walking over carpets; difficulties in handling sheets of paper, fibers, and fabrics; and hazardous situations when volatile gases are present.

Section 4. VENTILATION AND INFILTRATION

1.4.1 VENTILATION AIR. Ventilation air may be outside air provided by forced ventilation or by infiltration or recirculated air purified by charcoal filters. During the heating season and during the cooling season if mechanical cooling is used, the ventilation air shall be limited to the design values contained in this section. Where power ventilation is provided, the total ventilation air quantity (including pressurization and infiltration) used for calculating heating and cooling loads shall be established as the greater of the total exhaust requirement or 0.125 CFM per square foot of net floor area, provided that ventilation air exclusive of infiltration is furnished at 5 CFM per person. Exhaust requirement values for toilets, kitchens, etc. shall be set at the minimum values recommended by ASHRAE Standard 62-73. Ventilation air quantity of 0.125 CFM per square foot will be adequate to meet general exhaust requirements and to pressurize the building to minimize infiltration. Five CFM/person will take care of heavily populated areas such as auditoriums, churches, theaters, arenas, convention halls, classrooms, cafeterias, conference rooms, meeting places, etc. Smoking should be prohibited in all assembly type areas.

1.4.1.1 Family Housing. Normally an outside air supply is not required in air conditioning family housing. Any exception is quarters for officers of flag rank for which an outside air supply shall be provided to meet the above requirements. The above ventilation rates are applicable for general use. ASHRAE's recommendation for ventilation shall be considered for special areas such as hospitals, laboratories, bakeries, restaurants, laundries, swimming pools, explosive manufacturing, contaminated areas such as smoking, toxic gases, etc., but in no case shall less than 5 CFM per person be allowed. For kitchen and dishwashing areas, the ventilation requirement for cooling shall be based on the internal load and air temperature rise of 15°F.

1.4.1.2 Variable Ventilation. For energy conservation provision for variable ventilation rates corresponding to occupancy should be incorporated in the design of assembly type areas. All mechanical ventilation systems shall be equipped with readily accessible means for volume reduction and/or shut-off when ventilation is not required. Outside ventilation air intake in unoccupied buildings should be closed during early morning pick up cycle to achieve rapid acceleration to proper temperature (air cooling or heating), e.g., to bring an air-conditioned building down to temperature in the morning (0600 to 0730) before workers arrive.

1.4.2 INFILTRATION. Specifications should require that building windows and doors be designed and installed so that infiltration will be limited to 0.5 and 0.75 CFM per foot of crack for windows and doors respectively when tested at 1.567 pounds/square foot. Tests to confirm these rates of infiltration shall meet ASTM Standard E283-73 requirements. Infiltration must be considered as a room heat load and not as a system load, since it bypasses the air handling units and comes directly into the room. Caulking and weatherstripping reduces infiltration. Knowledge of prevailing wind will aid judgment in considering the crack length to be assumed in computing infiltration air quantity, but in no case will less than one-half the total crack be used.

1.4.3 VENTILATION FOR SPECIAL AREAS. Ventilation is required to provide adequate oxygen per person and to avoid build up of concentration of carbon dioxide in the space. The other purpose of ventilation is to remove body odors and where there is large volume per occupant less outside air is needed. Ventilation air requirements should be minimized to a value that will not unduly penalize the heating/cooling system energy requirements while providing a healthful environment.

1.4.4 THERMAL LOADS. The thermal load of ventilation air is directly proportional to the total quantity of the outside air and its magnitude is different for summer and winter conditions. The summer ventilation air contributes to sensible and latent heat loads, whereas winter ventilation load has a sensible component only. Any induced humidification provides the latent load in winter.

Summer ventilation load:

$$H_V \text{ (summer)} = 4.5 \times Q_V \times (h_o - h_i)$$

Winter ventilation load:

$$H_V \text{ (winter)} = 1.08 \times Q_V \times (t_i - t_o)$$

Winter humidification load:

$$W \text{ \#/HR} = \frac{Q_V \text{ (CFM)} \times 60 \text{ Min./Hr.} \times (G_i - G_o) \text{ (grains)}}{13.33 \text{ cu.ft./lb.} \times 7000 \text{ grains/lb.}}$$

$$= 0.00064 \times Q_V \times (G_i - G_o) \text{ \#/HR (Moisture)}$$

Where H_V = Ventilation air thermal load at outdoor design conditions (summer and winter) in BTUH

Q_V = Outside ventilation air, CFM at standard conditions (70°F, 0% R.H. and 29.92" Hg pressure)

Note: All air quantities mentioned in this manual correspond to Standard air conditions, and corresponding corrections must be applied for other conditions.

h_o = Enthalpy of outside air in BTU/lb. of air

h_i = Enthalpy of inside or room air in BTU/lb. of air

t_i = Room dry bulb temperature, °F

t_o = Outside winter design temperature, °F

W = Pounds per hour of moisture or steam required for humidification

G_i = Grains per pound of air at inside winter design conditions (at 70°F and 25% R.H. = 27 grains/lb.)

G_o = Grains per pound of air at outdoor winter design conditions (from "Engineering Weather Data")

1.4.5 EXAMPLE. Proper selection of ventilation air quantity reduces the first and the operating cost during both the heating and the cooling seasons. A 1000 CFM reduction in outdoor air (in the Philadelphia area) will represent a saving of about 2.7 tons in refrigeration equipment, 58,320 BTUH in heating equipment, and 14.1 pounds per hour of steam for humidification.

$$\text{Summer tons} = \frac{4.5 \times 1000 \times (39.57 - 32.4)}{12{,}000} = 2.70$$

Outside - 91 DB, 76 WB
Inside - 78 DB, 60% RH

Winter BTUH = 1.08 x 1000 x (70-16) = 58,320

Winter Humidification = 0.00064 x 1000 x (27-5) = 14.1

Reduction in air requirements for kitchen and industrial applications to achieve these savings may be obtained by the following considerations:

(1) Use of vented hoods directly over the heat producing equipment.

(2) Directly bringing the outside air into the area of hot spots or equipment to pick up the heat without causing draft conditions.

(3) Heat the outside air by recovering heat from hot waste gases through heat exchangers.

(4) Tighten up all unnecessary openings by installing well-fitted dampers for fireplaces. Fireplaces increase the heating requirements in a space so their use should be minimized.

(5) Use of double glazing and weatherstripping.

Section 5. SOLAR SCREENING

1.5.1 SOLAR SCREENING. The use of glass must be carefully controlled, since glass permits the greatest loss of energy per unit surface area of all the building components. All glass facing south, southwest, and west should be protected from summer time solar exposure by architectural shading, tinted glass or solar screening. Solar screening reduces the direct component of the solar heat load. The other portion of solar heat gain is caused by diffused radiation which affects all exposures. Proper design of solar screening includes consideration of latitude, elevation, orientation, percent of glass, heating and cooling loads, cost, obstruction and inconveniences to such activities as window washing. Consider roof overhangs, horizontal and vertical building projections, louvers, reflective glass coating and roof covering, internal shades, venetian blinds, draperies, awnings, eyebrow reveals, or vertical/horizontal fins.

1.5.2 OVERHANGS AND PROJECTIONS. Roof overhangs and horizontal projections of reasonable length provide shading to east and west exposure for the entire year and on the south, southeast and southwest exposures during the late spring, summer and early fall. When properly designed they will no longer shade south exposures during the winter heating season when the sun is too low to be intercepted without excessively long projections. See Figure 1-2. For projection length, see ASHRAE Handbook of Fundamentals 1972, Chapter 22, Table 25.

1.5.3 VERTICAL PROJECTIONS. Vertical projections, fins or louvers, (horizontal or vertical) are more effective when they are movable and on the outside of the building. They are effective on the east and the west walls. Outside motor operated louvers have been applied successfully for solar control. They also reduce the impact of higher wind velocity and increases the thermal resistance of the outside air film on the wall and the glass. Awnings will be beneficial for some applications such as residential. See Figure 1-2.

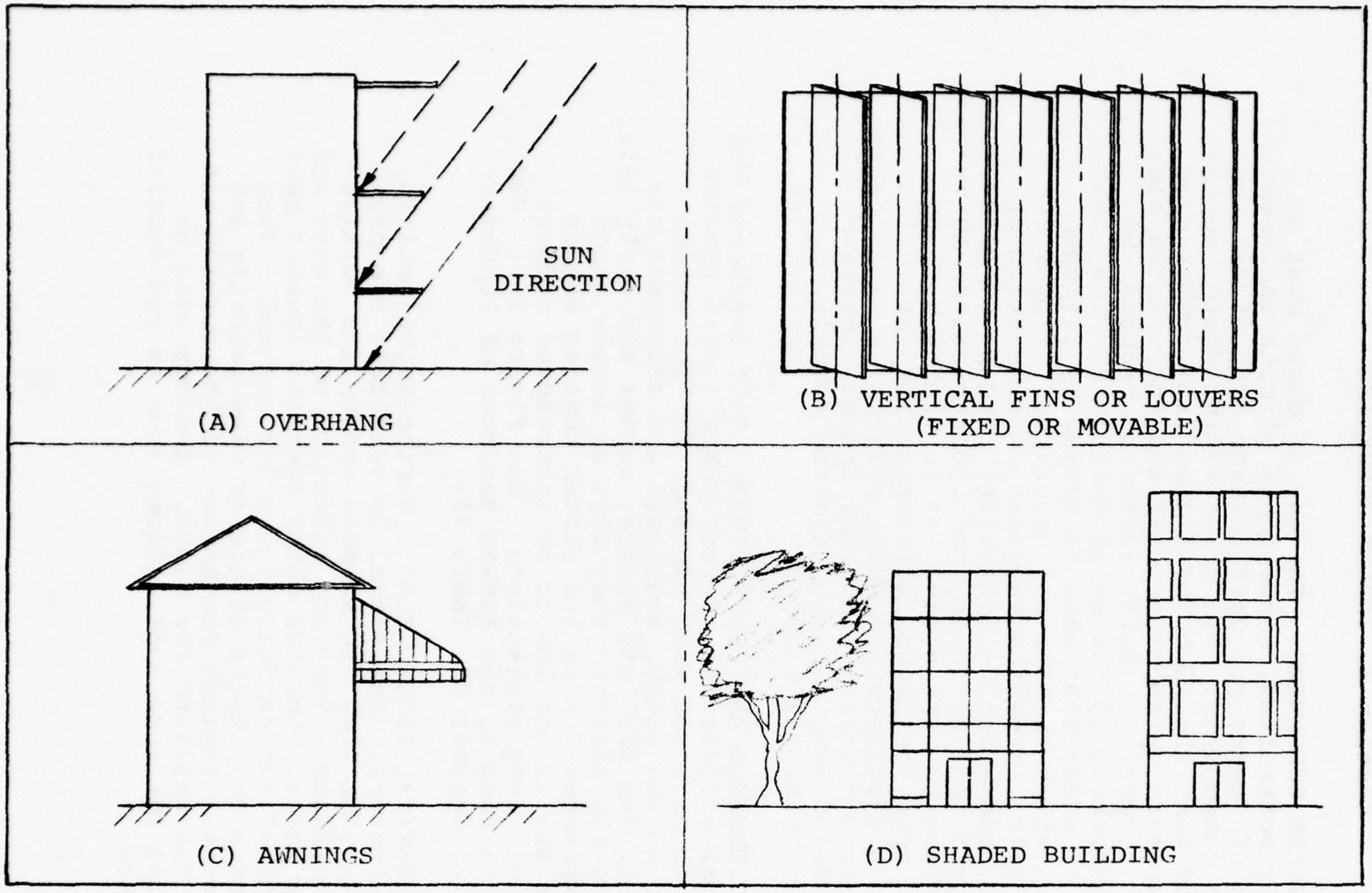


FIGURE 1-2

SOLAR SCREENING

1.5.4 INTERNAL SHADING. Internal shading devices cost less but they are less effective.

1.5.5 SHADOW EFFECT. Adequate credit when selecting an air conditioning system must be taken for the "shadow" effect of adjacent trees, hills and buildings. For calculating this effect, follow ASHRAE method as outlined in "Procedure for Determining Heating and Cooling Loads for Computerized Energy Calculation."

Section 6. EXTERIOR ENVELOPE -

WALLS, FLOOR, GLASS AND ROOF

1.6.1 EXTERIOR ENVELOPE REQUIREMENTS - HEATING AND COOLING. The exterior envelope requirement shall apply to all buildings including special applications such as hospitals and laboratories. The intent of this requirement is to thermally optimize the exterior shell of a building, thus minimizing winter heat loss and summer heat gain. The selection of Heat Transmission Factor "U" (Btu/Hour/Square Foot/°F) is made by comparing heating and cooling criteria requirements and selecting the most restrictive value, i.e., the lower value of the two.

1.6.1.1 Heating Design Criteria. The heat transmission factors for walls, roof and floors shall not exceed the values shown in Table 1-2. (No interpolation for intermediate Degree Days values is permitted.) Glass selection for all buildings shall be based on economics, but in no case shall the overall heat transfer coefficient value (U_O) shown in Table 1-1 be exceeded when used in conjunction with the following equation:

Equation: $U_O A_O = U_W \times A_W + U_G \times A_G + U_D \times A_D$

Where U_O = the average thermal transmittance of the gross wall area and A_O = a unit area of gross wall.

U_W = thermal transmittance of opaque (net) wall area.

A_W = ratio of opaque (net) wall area to gross wall area.

U_G = thermal transmittance of window or glass.

A_G = ratio of window area to gross wall area.

U_D = thermal transmittance of door.

A_D = ratio of door area to gross wall area.

TABLE 1-2

Maximum[1] Walls, Roof, Floor and Overall Transmission Factor (U and U_O)

Degree-Days	Walls U_W	Roof U_R	Floor U_F	Family Quarters Gross U_O For Wall	All Other Buildings Gross U_O For Wall
0-2200	0.14	0.05	0.15	0.32	0.38
2201-4400	0.10	0.05	0.13	0.27	0.36
4401-6600	0.08	0.05	0.11	0.23	0.31
6601 and above	0.07	0.05	0.10	0.19	0.28

Degree-Days values from NAVFAC P-89 shall be used when available. Until the revised P-89 is available, use 1973 ASHRAE Systems Handbook or its latest issue. Value of U for wall, roof and floor shown in Table 1-2 shall not be greater than the following values corresponding to 97-1/2% winter ambient design temperatures: i.e., use the lowest of the two values obtained, one based on Degree-Day criteria and the other on winter ambient design criteria.

Temperature	Walls	Floor
-40°F to -10°F	.07	.05
- 9°F to +10°F	.10	.07
+11°F to +50°F	.15	.10

[1] Maximum U_O value will put a limitation on the allowable percentage of glass to gross wall area in a building. Insulation glass on the building will allow higher percentage of glass in comparison with single glass.

1.6.1.2 Perimeter Insulation. When heated spaces are adjacent to exterior walls in slab-on-grade construction, perimeter insulation shall be installed on the interior of foundation walls as follows: 1-inch thick when annual heating degree days of aggregate from 3,500 to 4,500, and 2 inches thick when the annual heating degree days are 4,500 and over. Installation of the insulation shall be in accordance with the ASHRAE Guide.

1.6.1.3 Condensation Control - Heating. The design of the building envelope shall provide protection against cold weather water-vapor condensation on or in roofs, attics, walls, windows, doors, and floors. For opaque areas of ceilings, roofs, floors, and walls containing dry thermal insulation, a continuous vapor barrier having a water vapor permeance not exceeding 0.5 perm (grains/hr ft^2 (in.-Hg) is required on the winter-time warm side of the insulation. Slab-on-grade floors shall have a vapor barrier with lapped joints under the slab not exceeding 0.1 perm. A vapor barrier not exceeding 0.1 perm shall be required to cover the ground area of a crawl space beneath floors. Ceiling,roof, floor, and wall constructions shall contain thermal breaks to prevent excessive heat transmission through framing members.

1.6.1.4 Cooling Design Criteria. Materials shall be specified so that wall (net area) and roof heat gain shall not exceed 2.0 Btuh per square foot at design conditions. All glass, except north glass, shall have a shading device (e.g., shades, venetian blinds, draperies, awnings, eyebrow reveals, or vertical/horizontal fins), and maximum instantaneous transmission and solar gain through glass shall not exceed 70 Btuh per square foot as an average for the entire building (i.e., block load figure). This average of maximum instantaneous solar and transmission factors includes shading factor. Thermal storage effect due to mass of building must be accounted for to produce properly sized system, capable of balancing the actual loads. For buildings that are cooled only the overall thermal transmittance U_O for the gross wall shall not exceed 0.32 for family quarters and 0.38 for all other buildings.

Section 7. HEATING, COOLING LOAD AND ENERGY CALCULATIONS

1.7.1 HEATING AND COOLING LOADS. These loads shall be calculated in accordance with one of the procedures specified in the 1972 or latest issue of the American Society of Heating, Refrigerating and Air Conditioning Engineers (ASHRAE) Handbook of Fundamentals.

1.7.1.1 Simplified Method for Small Buildings. For family quarters and other buildings with a gross floor area less than or equal to 10,000 square feet, if both heated and cooled or 40,000 square feet if heated only, the simplified and manual method as discussed in Part 1, General Cooling Load Calculations, Chapter 22, in the 1972 ASHRAE Handbook of Fundamentals, shall be used. Show design calculations, method, results, and heat balance chart (see Chapter 2). Annual energy requirements for residential type buildings,such as single family dwellings and family quarters, using Degree Day method for heating and Equivalent Full-load Hour method for cooling as discussed in Chapter 43 of 1973 ASHRAE Systems Handbook or its latest edition. For other smaller buildings annual energy requirements should be obtained by the "bin" or temperature frequency occurence method as discussed in Chapter 43, Page 43.13 of 1973 ASHRAE Systems Handbook or its latest edition. Bin system data at 5° interval must be obtained from NAVFAC P-89 "Engineering Weather Data" manual. Adequate credit must be taken for the heat reclaiming systems. Additional data on system incorporating heat pumps is given in Chapter 11 of 1973 ASHRAE Systems Handbook.

1.7.2 COMPUTER ANALYSIS. For buildings larger than 10,000 square feet, an extensive hourly dynamic analysis for load as well as energy calculating procedure shall be made using advanced computer techniques and hourly weather data prepared from the sources of NAVFAC P-89 or as obtained from The National Climatic Center of National Oceanic and Atmospheric Administration, U.S. Department of Commerce, Environmental Data Service, Federal Building, Ashville, North Carolina 28801. Load and energy calculations shall be made using the design parameters as discussed in the earlier section on site, outdoor design conditions, indoor design conditions, ventilation, infiltration, solar screening; and exterior envelope-walls, floor, glass and roof. The annual hourly energy analysis (8,760 hours) shall incorporate the effectiveness of energy conservation features and systems as discussed in Chapter 3. A computer program for dynamic analysis to simulate the operation of all the buildings through a full-year operating period shall be of sufficient detail to permit the evaluation of the effect of system design and operational characteristics (such as space temperature and humidity control, supply air flow and temperature, nature of fuel or energy source, outside air quantities, lighting and occupancy schedules) and mechanical plant characteristics (such as part-load profiles, sequencing and accessories) on annual energy usage. See Chapter 5 for more details. Manufacturer's data or comparable field test data shall be used, when available, in the simulation of all systems and equipment. The calculation procedure shall utilize simulation techniques similar to the current recommendations in accordance with ASHRAE Task Group publication entitled "Proposed Procedure for Simulating the Performance of Components and Systems for Energy Calculations." The energy consumption for heating and cooling are directly related to the actual weather conditions and also the systems selected. The purpose of this calculation procedure is to get overall savings in owning and operating costs due to more precise sizing of the equipment selected and careful control of heating and cooling system operation. The computer programs need minimum repetition and maximum utilization of the input data. Alternate designs for the same building can be obtained with very little effort by changing few design constants or type of systems and submitting to the program for a rerun. To allow the equipment selected to operate close to design capacity, no additional safety factors, above what are inherent in the ASHRAE method, shall be allowed.

CHAPTER 2. HEAT BALANCE

2.1 HEAT BALANCE ANALYSIS. In order to establish the maximum potential benefit of energy conservation, a heat balance analysis should be made of all major heat load contributors at the beginning of design. The analysis involves first, a calculation of building envelope heat gain or loss and the probable indoor heat contributors for all conditions of outside temperature. Secondly, the net heating/cooling requirement for the building as a whole is plotted graphically to permit visual evaluation of the heat balance components. The analysis should always include the six following internal heat contributing parameters:

(1) Exterior envelope conduction load of walls, glass, floor, and roof, i.e., heat loss in winter and heat gain in summer. This is a sensible load due to different ambient and space temperatures.

(2) Ventilation load is the outside air load brought through air handling systems and infiltration air. The infiltration air is air leakage in the building through walls, doors, glass, and roof due to wind pressure differential and due to the chimney or stack effect created by temperature differences. The ventilation load consists of both sensible and latent load components, the former being predominant in winter. The intent of using this load is to get the breakeven temperature which most of the time is in the vicinity of winter design temperature. For this reason,and for simplicity,only sensible load figures will be used for heat balance.

(3) Light and power load is due to luminaires and electrical machinery (typewriters, copying machines, calculators, etc.). For heat balance, light load (H_L) may be taken as 90 percent of installed capacity. This is because all lights may not be on at the same time and some fixtures may be in need of repair or replacement. For power load (H_p) use no less than 50 percent diversity on the installed capacity.

(4) Occupancy or people load is obtained by multiplying the number of people (occupancy for which the building is designed, less visitors) times sensible load/person. Only sensible load is to be used for the same reason mentioned earlier. Typical value of this depends upon the degree of activity and is, for standing, light work or walking slowly = 250 BTU/hour. Values for other activities may be obtained from the 1972 ASHRAE Handbook of Fundamentals, Table 29, page 416. For constructing a heat balance chart use 80 percent of the calculated occupancy load assuming all persons will not be present in the building at the same time.

(5) Equipment load includes energy of fans, pumps, computers, heat of compression and any other heat producing equipment. Heat of compression to be used only in the system incorporating double bundle or heat pumps. See Chapter 3.

(6) Solar load can be obtained from heat gain data (cooling load).

All the above values can be easily obtained from computer printouts where they are used for load calculations and energy analysis.

For small projects on which computer analysis is not used the values may be simply derived as follows:

A. $H_T = \Sigma UAdt$

B. $H_{VS} = 1.08 \times Q_V \times \Delta t$

C. Total light and power load: $H_{LP} = 0.90 \times H_L + 0.50 \times H_P$ BTUH, where H_L and H_P = kilowatts x 3413.

D. H_O = No. of people x sensible load/person x 0.80

E. $H_E = \dfrac{\text{Brakehorsepower (BHP)} \times \text{\% load} \times 2545}{\text{Efficiency (at corresponding load)}}$

Notations:

H_T - Conduction load at various ambient temperatures, BTUH

H_{TPW} - Peak winter conduction load, BTUH

T_W - Ambient winter design dry bulb temperature, degrees F.

T_I - Inside or room design dry bulb temperature, degrees F.

H_{VS} - Sensible ventilation load, BTUH

Q_V - Outside or ventilation air, CFM

H_{LP} - Diversified light and power load, BTUH

H_O - People sensible load, BTUH

H_E - Equipment load, BTUH

H_S - Solar heat gain, BTUH

The heat balance chart as well as monthly energy usage chart must be prepared for all buildings.

2.2 HEAT BALANCE DIAGRAM. A graphic representation of the calculated heat loads is plotted to give a simplified graphic overview of the heat contribution from all energy components and to establish the relative merits of balanced heat recovery at various outdoor temperatures. Figures 2-1A and 1B show individual major load components which are summarized in the composite heat balance illustrated in Figure 2-2. Breakeven temperature (T_{BE}) the temperature corresponding to the outdoor temperature at which the heat from the internal energy components balances with the heat losses in a building, with and without the sun, can be obtained from the graph. Breakeven temperature for most buildings falls in the vicinity of winter design temperatures. There is surplus heat in the interior spaces and heat is required along the exterior envelope to offset winter heat losses. It is possible to eliminate any external heat source for energy demands during occupied hours for a system designed to take full benefit of transfer of heat from surplus zones to where it is needed. Consideration of heat recovery techniques can

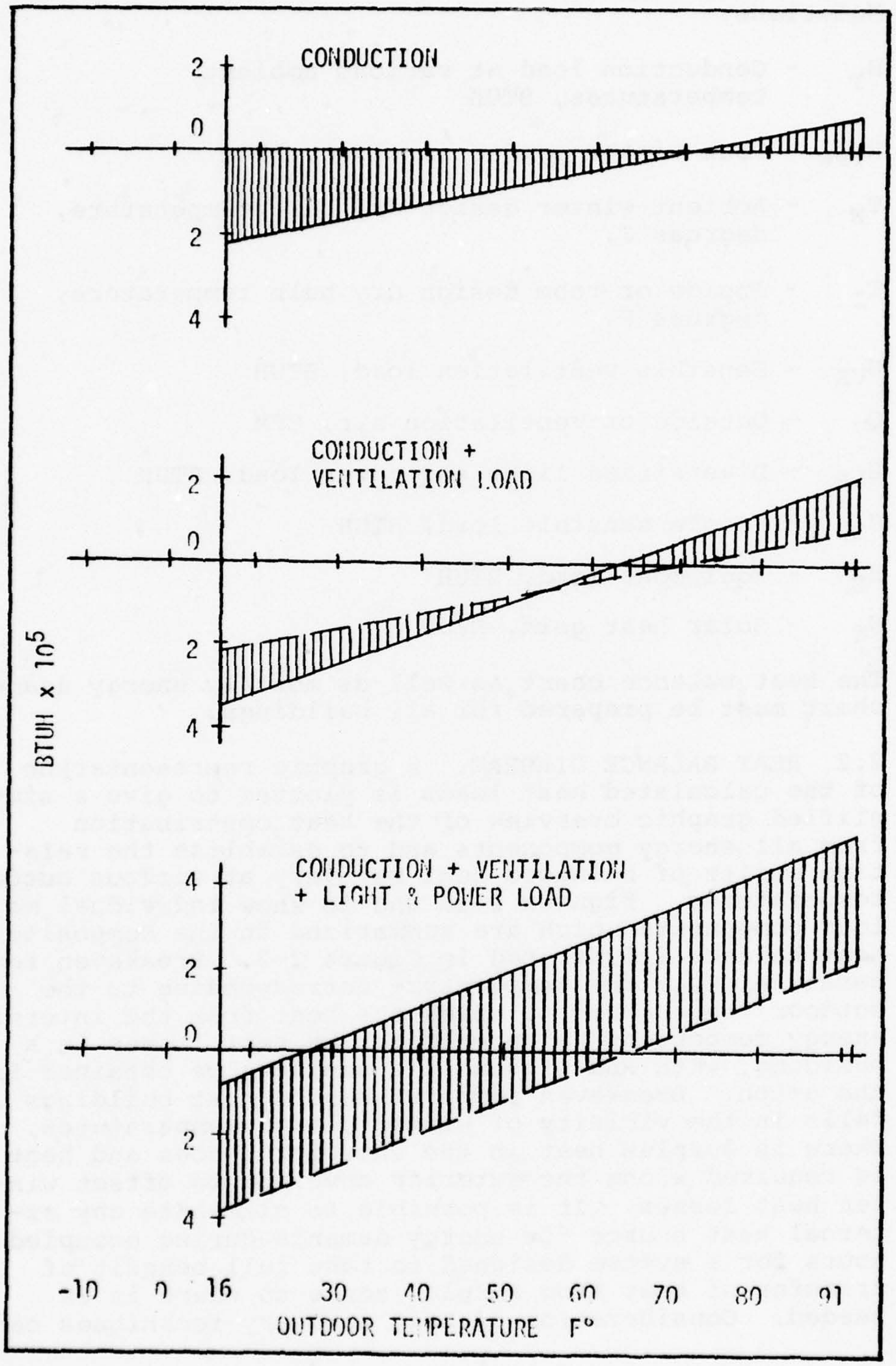


FIGURE 2-1A
MAJOR LOAD COMPONENTS

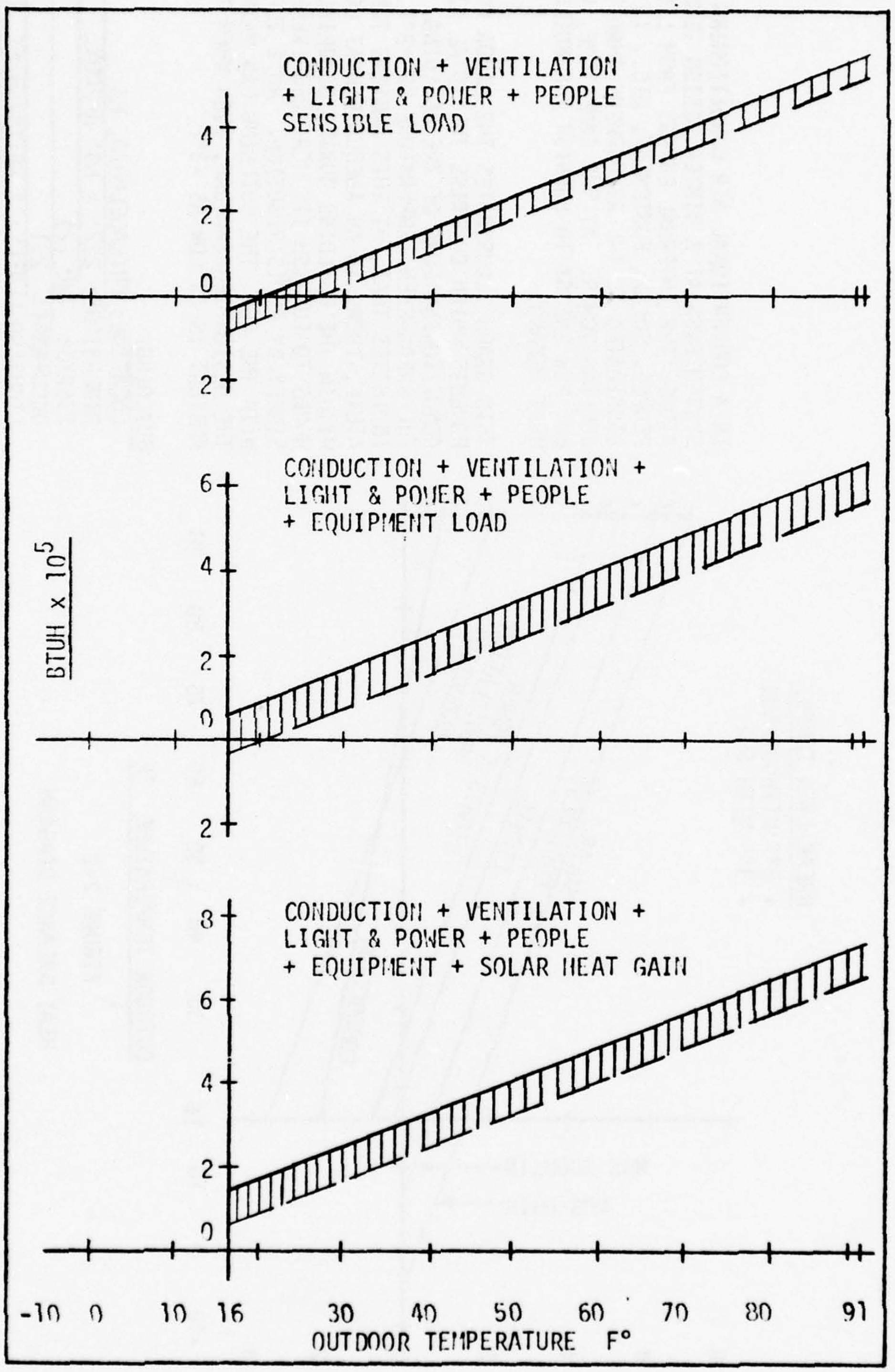


FIGURE 2-1B

MAJOR LOAD COMPONENTS

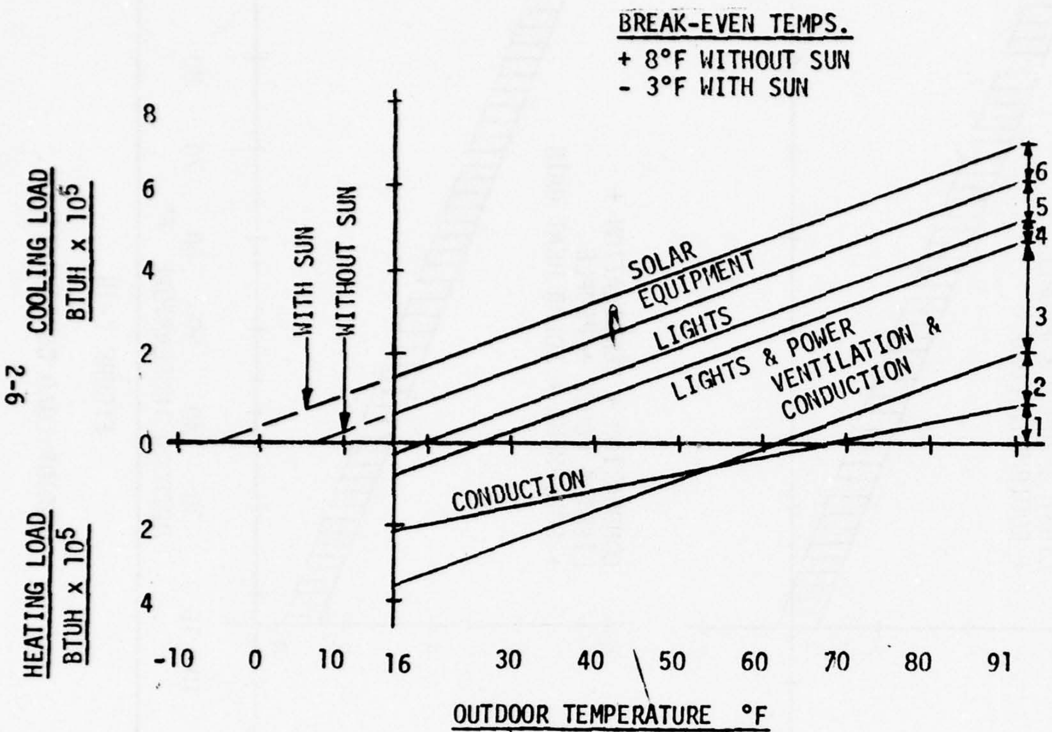


FIGURE 2-2

HEAT BALANCE DIAGRAM

IN A CONVENTIONAL AIR CONDITIONING SYSTEM, EVEN AT A WINTER DESIGN TEMPERATURE, THE INTERNAL ENERGY FROM LIGHTS, PEOPLE, SOLAR, EQUIPMENT, ETC., IS DISSIPATED TO THE ATMOSPHERE THRU THE COOLING TOWER. AT THE SAME TIME, WE BUY NEW ENERGY TO FURNISH THE BUILDING HEAT DEMAND.

THIS GRAPH ILLUSTRATES THE MAJOR COMPONENTS WHICH COMPRISE THE TOTAL AIR CONDITIONING LOAD OF THE BUILDING. THE BREAK-EVEN TEMPERATURE OF +8°F. INDICATES THAT, AT THIS AMBIENT TEMPERATURE, THERE IS AN ADEQUATE HEAT LOAD WITHIN THE BUILDING DURING OCCUPIED HOURS TO FURNISH ITS HEAT REQUIREMENTS SIMPLY BY REDISTRIBUTION. NOTE THAT WITH THE SUN, THE BUILDING CAN MAINTAIN THE DESIGN TEMPERATURE, EVEN WHEN THE AMBIENT IS AS LOW AS -3°F.

BUILDING
LOCATION: PHILADELPHIA, PA.
DIMENSIONS: 222' x 60' APPROX.
STORYS: TWO (2)
OCCUPANCY: 231
LIGHTING LEVELS: 3 WATTS/SQ.FT. (70F.C.) + 1/2 WATTS/SQ.FT. (FOR POWER)

eliminate the undesirable wastage which was inherent in the past conventional air conditioning systems, where the boiler was supplying the heat in the building and simultaneously heat was dissipated to the outside through cooling towers. Recognizing that the heat balance diagram is drawn for a building during occupied hours, supplementary heat is required for the following reasons:

(1) At winter design temperature when the outdoor temperature is so low that internal heat is not adequate to meet the project requirement.

(2) To furnish the heating requirements during nights, weekends and holidays when the building systems may be shut down.

(3) As a standby equipment for special buildings and in case of repairs to the heat recovery component.

A heat balance analysis displays the penalties that result if heat recovery techniques are omitted. The graphic portrayal of energy requirements aids in the analysis of system operation and in the selection of the fuel source, optimum equipment, and system.

2.3 BREAKEVEN TEMPERATURE (T_{BE}) BY CALCULATION. Breakeven temperature can be obtained either graphically as illustrated in Figure 2-2, or can be calculated if peak values of various energy components are known at winter design temperature. We suggest doing it both ways for checking purposes.

Derivation Heat loss = Heat gain

$$H_{TPW} \times \frac{(T_I - T_{BE})}{(T_I - T_W)} + H_{VS} \times \frac{(T_I - T_{BE})}{(T_I - T_W)}$$

$$= H_{LP} + H_O + H_E + H_S$$

Simplified Equation:

$$T_{BE} = T_I - \frac{(H_{LP} + H_O + H_S + H_E) \quad (T_I - T_W)}{(H_{TPW} + H_{VS})}$$

Above T_{BE} is with sun; T_{BE} without sun is obtained by substituting $H_S = 0$ in the above equation.

Example: The purpose of the example to calculate the breakeven temperature is to get the idea of its general value as applicable to an office building.

Location: Philadelphia, Pa.

Dimensions: 222' x 60' approx.

Storys: Two (2)

Areas: Gross total 26,230 ⏀ , Net 23,136 ⏀

Occupancy: 231 (100 ⏀/person of net area)

Lighting Levels: 70 foot candles (3 watts/⏀),
misc. power (additional 1/2 w/⏀)

Most of the values for this example are taken from Run 1 of computer analysis. Only exception being that value of 100 ⏀/person is used instead of 50 ⏀/person and no electronic equipment load is considered since this applies to the laboratory area of the building. The following values were used for breakeven temperature calculations:

H_{TPW} - Peak winter conduction load = 207,150 BTUH

T_W - Winter design dry-bulb 16°F (97-1/2% frequency)

T_I - Inside winter design 70°F

H_{VS} - Sensible ventilation load 168,487 BTUH

H_{LP} - Light and power load 276,371 BTUH

H_O - People sensible load 57,750 BTUH

H_E - Equipment load 101,314 BTUH

H_S - Solar heat gain 96,074 BTUH

Calculation for T_{BE}:

(1) T_{BE} with sun

$$= 70 - \frac{(276,371 + 57,750 + 101,314 + 96,074)(70-16)}{207,150 + 168,487}$$

$$70 - \frac{531,509 \times 54}{375,637} = -6.4°F.$$

(2) T_{BE} without sun ($H_S = 0$)

$= +7.4°F.$

Summary:	From Graph	By Calculations
Breakeven temperature T_{BE} with sun	-3°F.	-6.4°F.
Breakeven temperature T_{BE} without sun	+8°F.	+7.4°F.

CHAPTER 3. SYSTEM TYPES

Section 1. TYPES OF ENVIRONMENTAL SYSTEMS - ADVANTAGES AND DISADVANTAGES

3.1.1 DIRECTION OF DESIGN. The engineering profession, by its works, must display its recognition that our resources are limited and must show its extreme concern for their conservation and judicious use. It must become our goal in the design of building services to reduce the amount of prime energy and to extract all possible benefits from that expended in each building. Efforts to date in this direction insofar as system types are concerned have taken three general courses:

(1) To discontinue or limit to the least possible the use of new energy for reheat control.

(2) To recover heat from all energy used in the interior or specific process areas and transfer this to the perimeter for winter heating or where essential for reheating.

(3) To exchange or transfer heat between outdoor air coming into the building and air being exhausted. Through these efforts significant savings are being made in the use of energy in buildings with estimates running to 30 percent and more. There is not yet sufficient tabulated data to completely evaluate the many methods being designed. Each building presents a different character to the engineer and the problem becomes which system is suitable and which method of energy conservation is adaptable to that system. The practice of control by reheat has been a standard of design for environmental systems for many years. The advantages of constant flow, building pressurization, and accuracy of control of dry bulb temperature and relative humidity in reheat systems are still unequaled in simplicity and flexibility in buildings of diverse usage and variable occupancy. However, the use of prime or new energy for this purpose is extremely wasteful and must be

discontinued. All of the principal designs of environmental control systems which have evolved over the years were developed because of some particular adaptation to building type or service. A less expensive or more convenient method was presented to solve some difficult design problem. Some to simplify treatment of high rise, some for H or L shaped buildings with extensive exposed walls, others for large interior compartmented spaces. All of these systems will continue to be used. Most types incorporate in one way or another basic control by reheat. And in the past this was provided directly by new energy. Fortunately, new methods for the recovery and reuse of energy in the form of heat have been devised. Much of the heat previously rejected from our buildings can now be returned to further service. Section 2 of this chapter describes the many devices available in modern systems to improve efficiency in our use of energy. Our problem is to decide which or how many can be incorporated in each of our designs. The recycling of heat as it is being rejected from condensers in the range of 105° to 130° is directly suitable for circulation through perimeter radiation or air heaters. In discussing the many types of systems in general use and their possible employment of secondary energy let us mention first the two most recent developments. These are variable air volume and reversed cycle, closed loop heat pumps. Both are highly efficient in energy requirements and both have come into favor with engineers and owners within the past few years.

3.1.1.1 Variable Air Volume. The development of this system in the past eight years has provided the industry with the best method yet devised for energy conservation. In addition to minimizing or eliminating the need for reheat, savings may also be made in weight of air circulated and therefore fan horsepower. The optimum at this moment appears to be wall-to-wall variable air volume with perimeter radiation by means of heated condenser water. Limitations are the perimeter loads. Loads requiring more than 3 CFM/sq. ft. for cooling can become difficult to introduce to perimeter spaces without drafts using present air diffusion techniques. Ductwork is sized at medium to high pressure and because of the usually uncomplicated layout, results in an appreciable reduction in installation costs compared to low velocity systems. The size of trunk ducts should be based on diversity since maximum load demand follows the solar incidence for each exposure. Since little or no reheat is involved, all of the hot condenser water is available to the radiation circuits. Both centrifugal fans with inlet vane control and controllable pitch axial flow fans are used for supply and return, see Figure 3-1 and Figure 3-1A. During periods of light load, static pressure regulators reduce the total volume of air flowing through the system by gradually closing the inlet vanes or reducing the pitch angle on the axial fans. A proportional saving in fan horsepower is realized with each fan type, the controllable pitch axial being the more efficient. This system can be designed with no new energy consumed in reheating.

3.1.1.2 Closed Loop Heat Pump. This design has been a number of years in development and at one time was termed the "California Heat Pump System." Early installations were confined to apartment and motels. It consisted of a series of self contained air conditioning units capable, with the use of reversing valve, of inverting the function of evaporator and condenser. The space served could then be either cooled or heated without regard to any adjacent space. The closed loop is maintained between, roughly 70° and 95°, and in this range suitable for condensing or reheating. The closed loop includes a dry cooler to reject heat when the majority of units are cooling, and a boiler or electric water heater for supplementary heat in winter when most of the units need heating. This proved to be an inexpensive and adequate heating and cooling system for multiroom, variably occupied buildings such as motels, and received immediate acceptance. The idea of a closed loop lies now enlarged to a complete heat recovery system and its use extended to all types of buildings. In addition to the early concept of perimeter units, larger systems up to 20 tons are used to add interior heat to the loop. In this manner, the need for supplementary heat is reduced and the interior energy is made available to heat the building exposures. Where interior zones are large, storage tanks may be installed to extend the period of use of surplus day time heat. If the closed loop system is to be

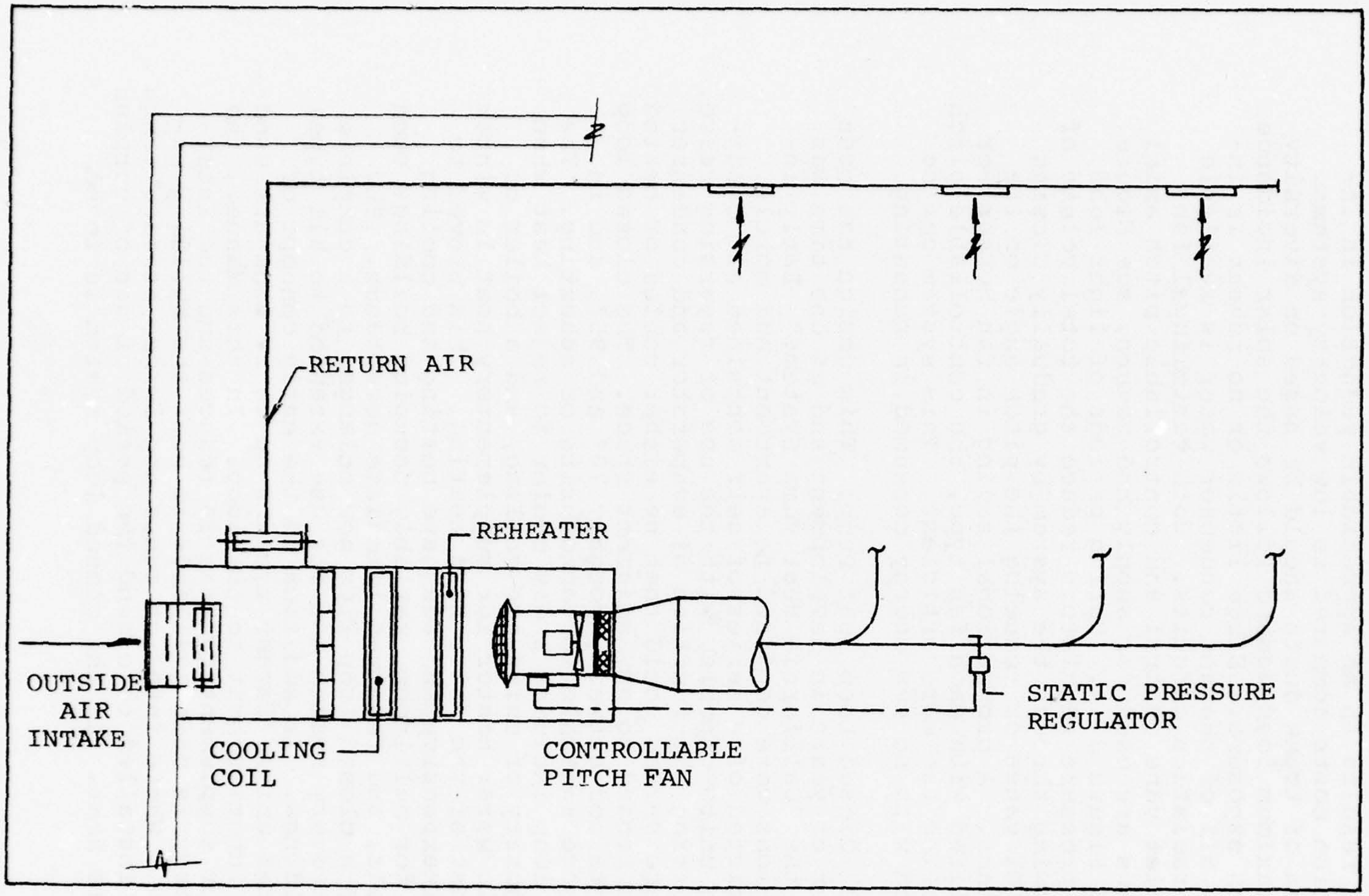


FIGURE 3-1

VARIABLE AIR VOLUME SYSTEM

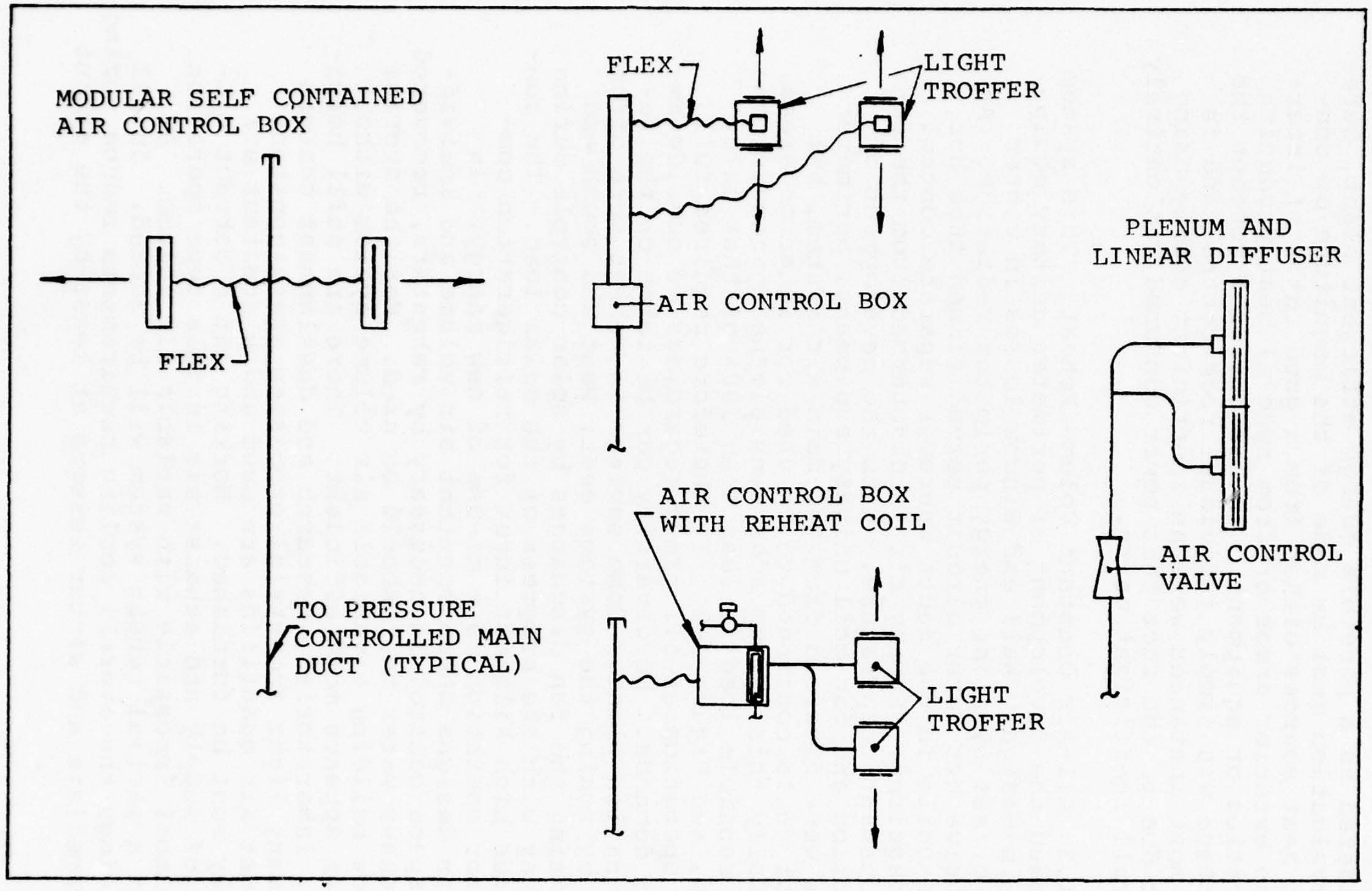


FIGURE 3-1A

TYPES OF AIR DISTRIBUTION FOR VARIABLE VOLUME SYSTEM considered as a possible energy efficient plant, careful evaluation must be made of the magnitude of constant heat sources either from a good ratio of interior to exterior areas or from special heat producing activities or equipment. In its original concept the advantage was simply individual room control and in fact most instances were an inefficient energy using plant due to the fact that power consumed was entirely by small inefficient units.

3.1.1.3 All-Air Constant Volume Reheat. This system preceded the development of perimeter unitary equipment in design. Wall and window losses in winter were offset by prime energy perimeter radiation. An extensive hot water circuit served finned tube hot water coils in the ducts wherever separate control was required. Supply air is discharged from the fan at constant temperature. With the development of induction and fan coil unitary equipment, perimeter loads were divorced from the main air system, but reheat coils continued to be used for interior areas. Basically this system adds sensible heat whenever the area sensible load is less than 100% of that calculated, see Figure 3-2. It therefore requires full load operation at all times regardless of outside ambient demands. No diversity can be taken on the refrigeration plant. Some savings in reheat were often made by zoning the systems east, west and south and resetting the fan discharges by solar controls during the day with the progress of the solar load. The continuous high kilowatt input for refrigeration compressor operation is a misuse of new energy. In modern designs where constant air volume and individual space control is necessary by reheaters, recovered condenser water heat should be used. For the average office building a variable air volume system without reheat appears most efficient. There are still hospitals, laboratories, research and development centers and many light industrial processes where constant exhaust air quantities are used and a constant air supply must be furnished. Working out a correct balance of supply and exhaust air in this type operation is almost impossible with variable air volume. At least a partial reheat system will be needed. In all buildings the overall cooling requirements reduce during intermediate and winter seasons at least by the extent

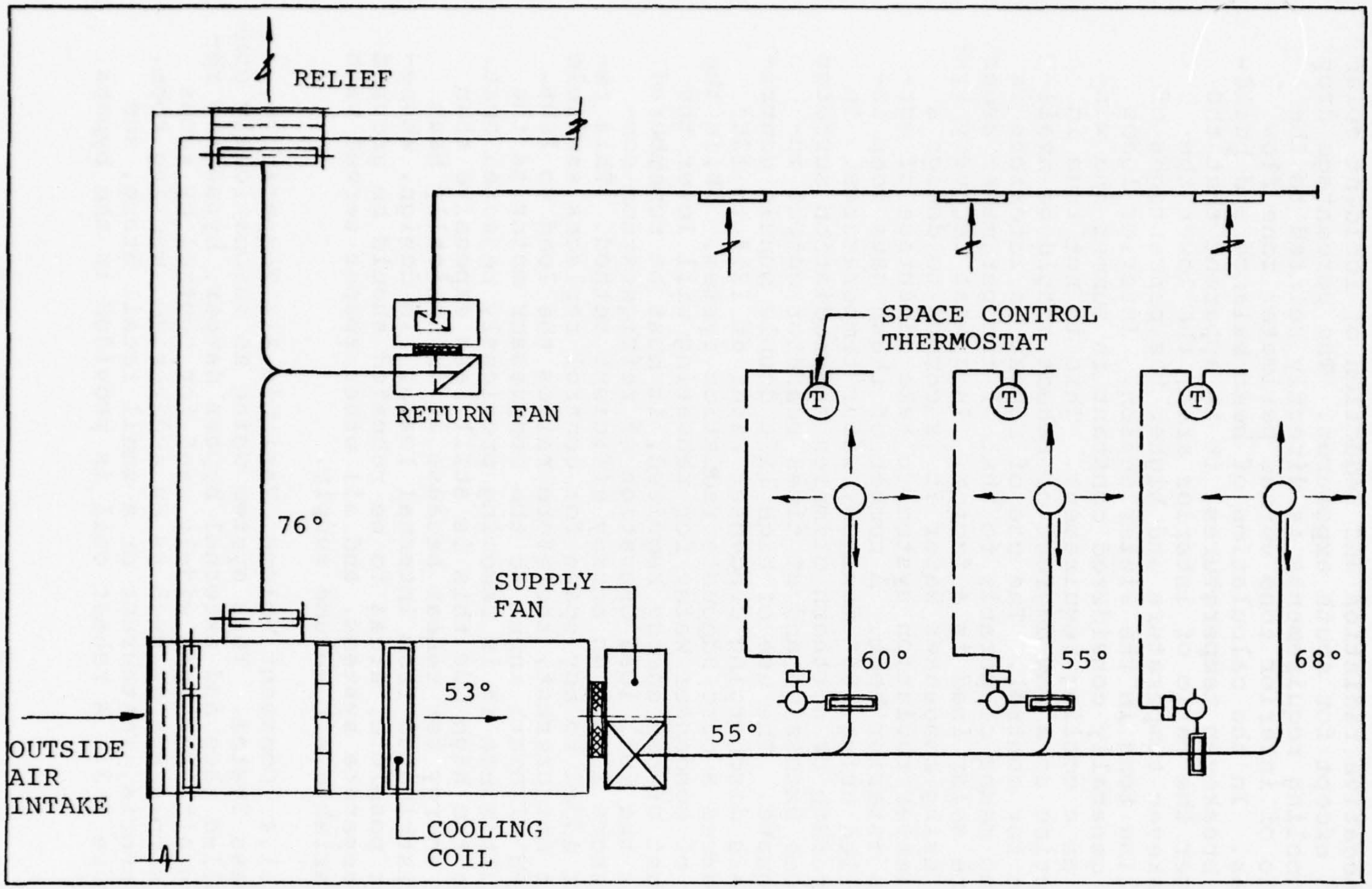


FIGURE 3-2

CONSTANT VOLUME - TERMINAL REHEAT SYSTEM of negative radiation and reduction of incident solar load except for south exposures. The percentage drop in cooling requirements is directly related to the ratio of interior zone versus perimeter zone flow areas. In the calculation of heat balance and building breakeven temperatures, it is apparent that the larger the ratio of interior area, the lower the breakeven temperature and higher the percentages of cooling tons in the winter season. Interior loads are generally considered constant in summer and winter as a cooling requirement. This is not true in practice so a few degrees of reheat should be available for control. The use of reheat in interiors is by no means comparable to that of the perimeter areas where solar load is a factor. In a heat recovery system using condenser water it is common to design a perimeter radiation system to take advantage of surplus interior heat. A number of these have been installed using 105° maximum, water temperatures. In temperate or northern climates the radiation surfaces become excessive and at times radiator output inadequate. The use of high lift double bundle centrifugals discharging condenser water at 125° to 130° produces a more adequate radiation system. While the use of condenser water for reheating will lower the amount of new energy required, it must be remembered that the full load operation of refrigeration compressors is not an energy efficient method. This reheat added to any space for control replaces sensible heat not present, therefore raises the load to 100%. Added kilowatt input to the compressor motor is the result since it is removing previously rejected heat. With the high COP this is still less expensive than new energy for reheat because it is basically heat redistributed from internal loads. In design, whereever possible, areas to be reheated should be grouped on separate systems, and all other spaces served with a variable air volume supply.

3.1.1.4 Constant Volume, Variable Air Temperature, Bypass System. This system using an automatically controlled face and external bypass damper, bypassing return air, has been widely used for control by zones or simple spaces such as an auditorium, meeting room, cafeteria, restaurant or a small retail store, see Figure 3-3. A reheat coil is provided in the bypass.

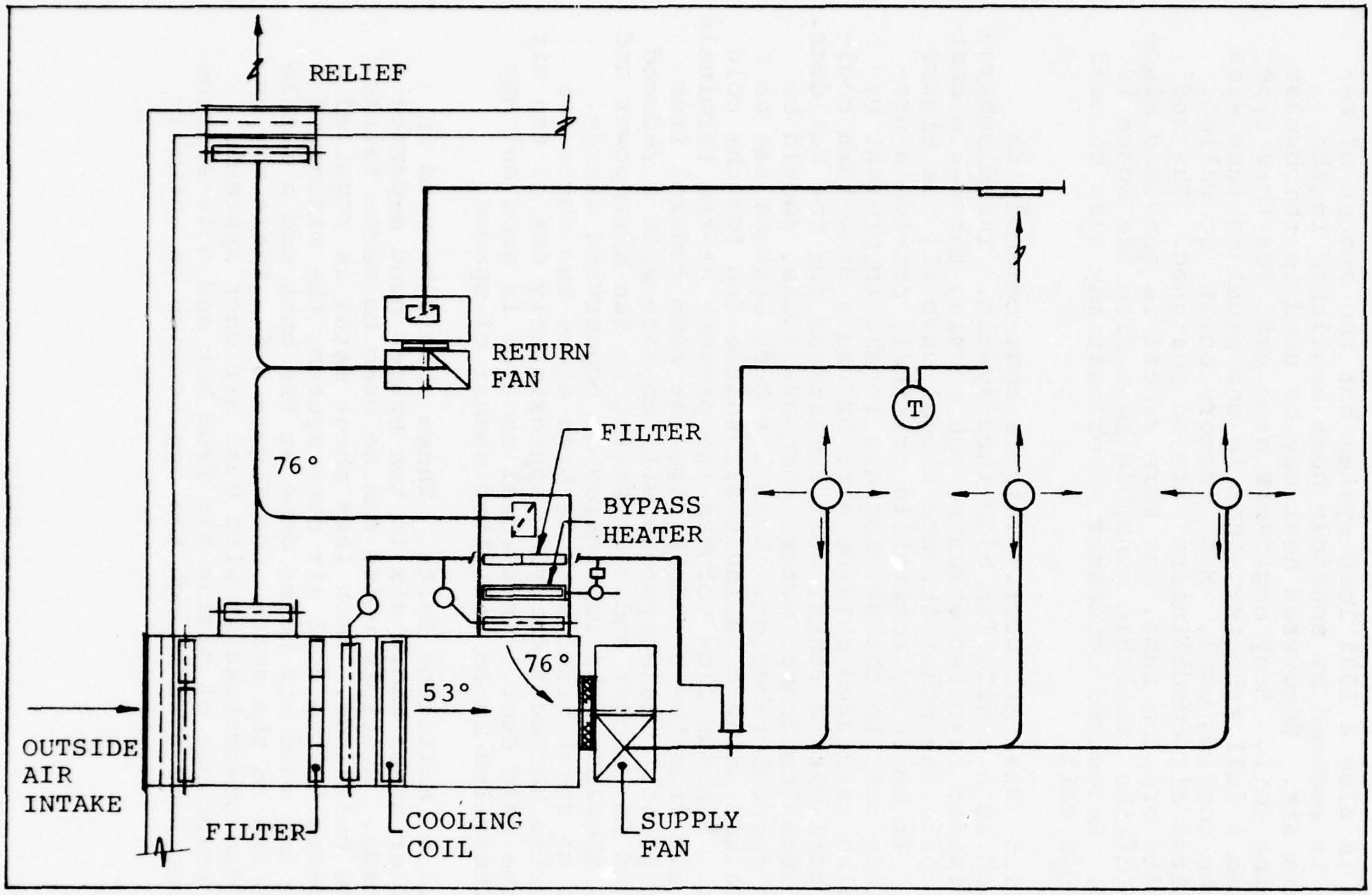


FIGURE 3-3

CONSTANT VOLUME-VARIABLE TEMPERATURE BYPASS SYSTEM

This is also a 100% load system but the amount of reheat is reduced by sensible heat available in the return air. Recovered heat may be used in the bypass heating coil. Many engineers have used for this type system a full reheater, that is one equal in face area to the cooling coil. Most factory built, so called, "central air conditioners" are so designed. The end results are the same, but more reheat is consumed since none of the available sensible heat from the space is used. Recovered condenser water heat may also be used in this coil.

3.1.1.5 Dual Air Duct. This is most commonly designed as a single fan blow-thru system. The discharge is divided into two streams, one passing through a heating coil, the other through a cooling coil, see Figure 3-4. If humidity control is critical, provide a precooling coil in the outside air intake to prevent bypassing untreated outside air; or use a draw-thru cooling coil with a reheat coil downstream for the hot deck. Recovered condenser water, when available, should be used for the reheater. Dual air duct systems can be installed using a variable air volume fan for the cold deck. Minimum flow volumes are present at the terminals and air from the hot deck used for room control from this point. These systems will operate with a reduced demand for reheat and some saving in fan horsepower and may, therefore, be considered as conserving energy. Cost of sheet metal work is high since two ducts are run, the hot duct carrying approximately 60% of the air in the cold duct. Individual control is good and may be installed in an unlimited number of spaces.

3.1.1.6 Multizone Units. These should be used in smaller areas where six to ten spaces need separate control. Multiple units can be used to serve fairly large buildings. Much less sheet metal is required compared to the dual air duct system, the mixing of warm and cold air being done at the unit and a single duct run to the space, see Figure 3-5. These may be classed operationally with dual air duct systems. The principle of mixing air from hot and cold sources is the same even though the application is entirely

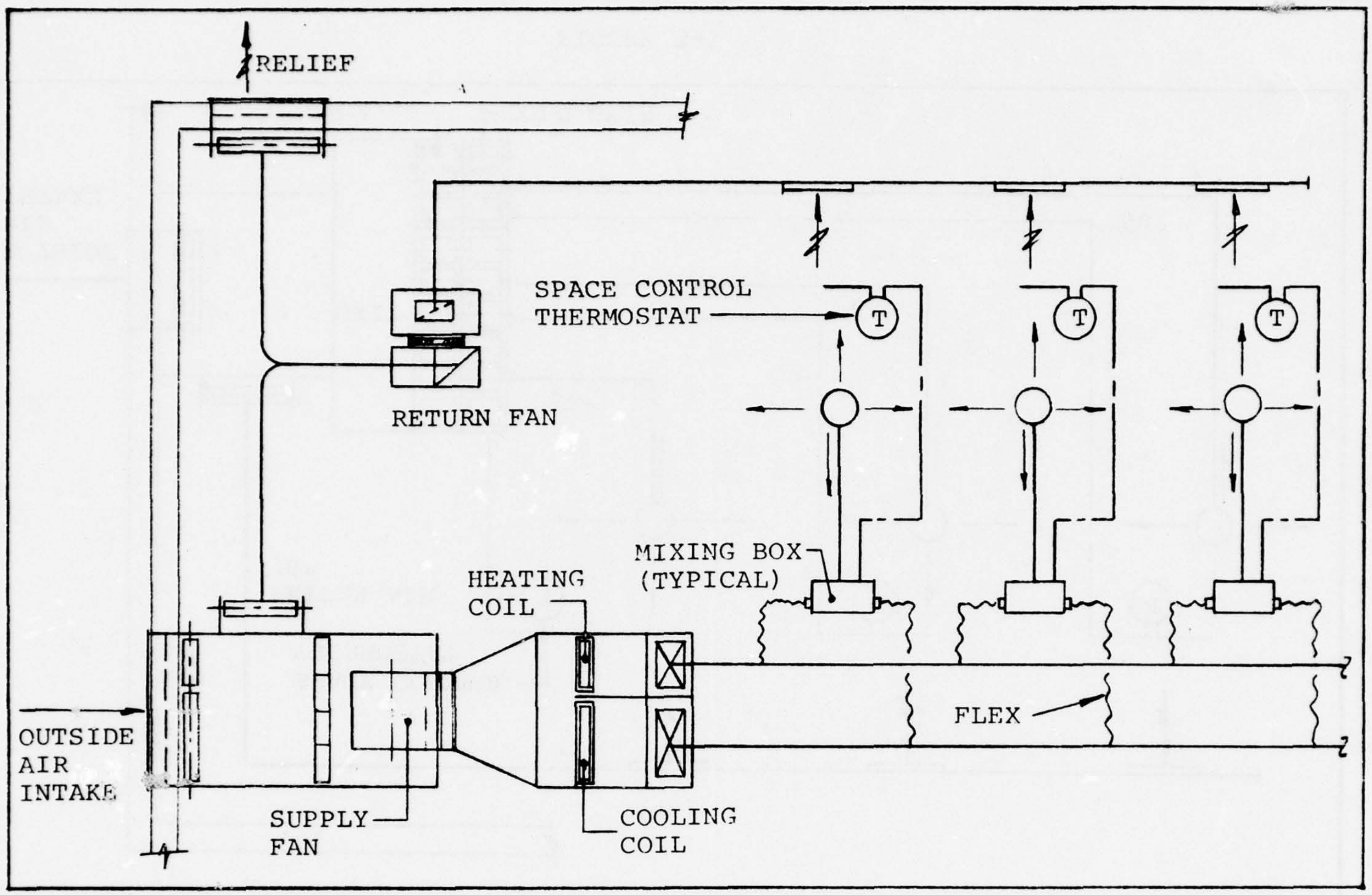


FIGURE 3-4

DUAL AIR DUCT SYSTEM

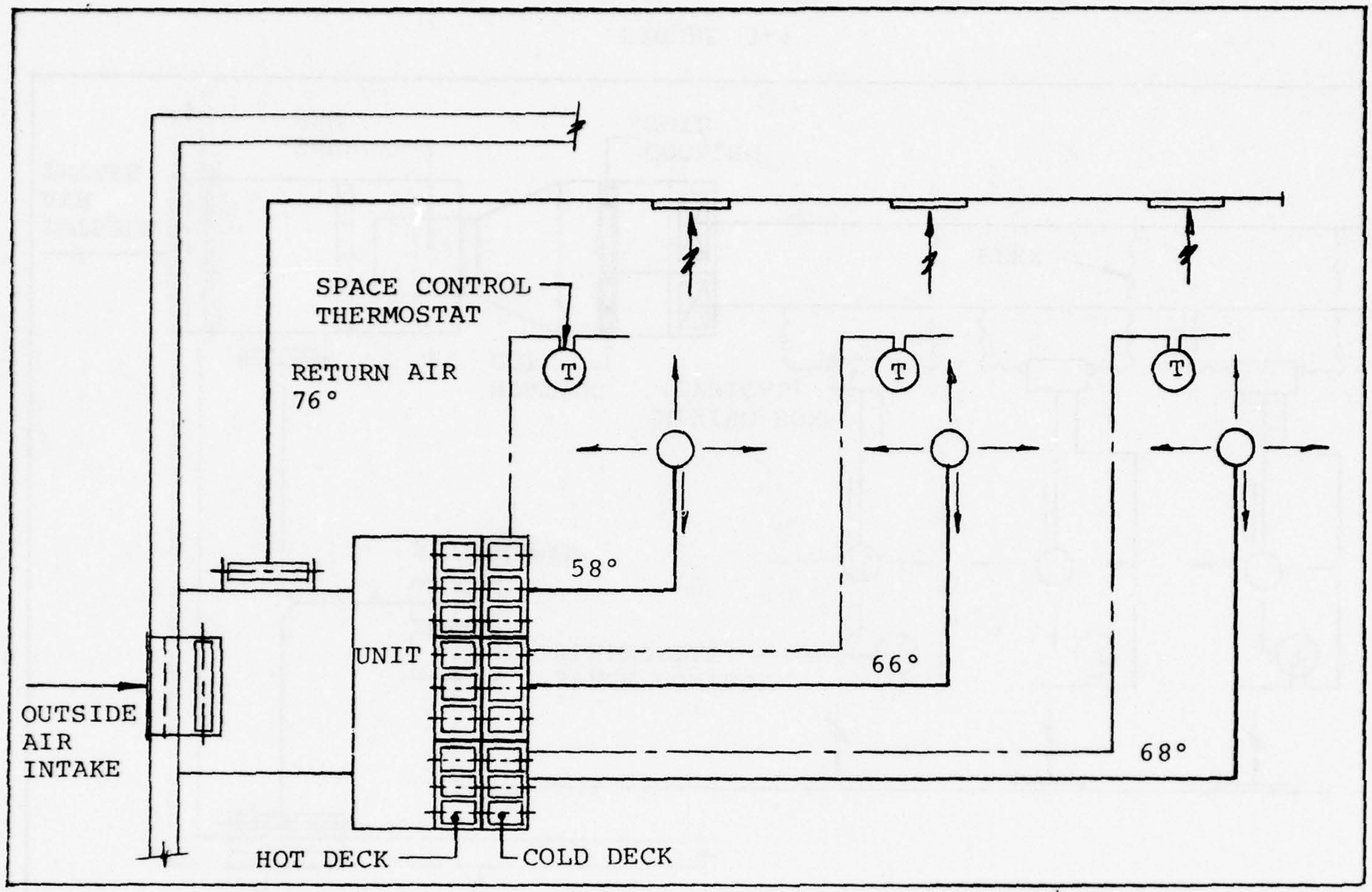


FIGURE 3-5

MULTIZONE SYSTEM

different. Condenser water, when available, should be used in the heating coil.

3.1.1.7 Dual Air Conduit System. In this design a separate duct system provides for the transmission load of wall and glass either in summer or winter. Solar load is assigned to the main air system. The perimeter system is usually only recirculated air and consists of fan, cooling coil and heating coil. See Figure 3-6. Recovered heat should be used for the heating coil. Air may be distributed upward along the window sills or down the wall from linear ceiling diffusers. In operation an outdoor thermostat resets the supply fan discharge temperature on a preset schedule to exactly balance heat gains or losses at the skin of the building. With recovered heat at a maximum of 125°F, a definite maximum discharge temperature is established for heating. The overhead supply of warmed air should be limited to moderate climates (outside design temperature 26° and higher). Where winter ambient temperatures are 25°F or below, design for upward distribution at sill height to avoid cold down drafts. During unoccupied hours, this system requires continuous or cycling fan operation during the heating season. For a complete economic study, operation and maintenance of the fans must be equated with water pump costs for a radiation system.

3.1.1.8 Split Systems (Air and Hydronic). When perimeter loads exceed 3 or 4 CFM per sq.ft., either through the use of large glass areas or in the case of high internal gains (art, photographic, business machines) some combination of interior air supply and perimeter unitary system is required. The units will then cool or heat to compensate for the transmission and solar variations and in many cases part or all of the internal heat. The two most frequently installed unitary systems are the high pressure induction system or simple fan and coil units. Either of these systems can be designed to control the complete perimeter zone, usually considered to be from 12 to 16 feet in from the outside wall. Assuming either of these systems to be two pipe, the principal difference is that the induction system provides the necessary ventilation and

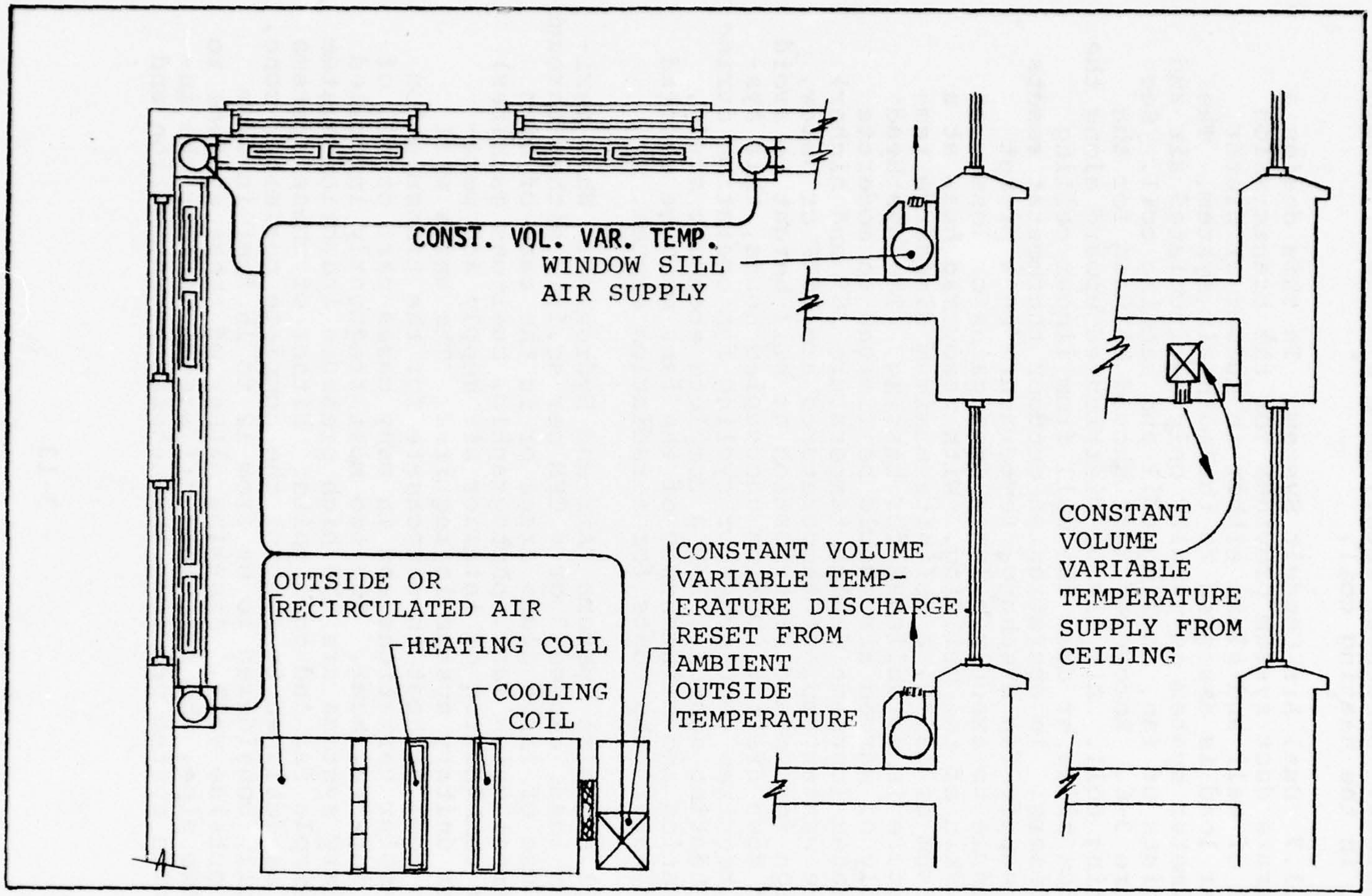


FIGURE 3-6

DUAL CONDUIT SYSTEM (USED WITH V.A.V. INTERIOR) dehumidified air for the perimeter zone, while the fan coil is simply a recirculating unit and ventilation air must be provided from another source. In both systems the water in the cooling coil is normally at or above 53°F and no latent cooling is expected from the window units. Heat recovered from the interior should be used for winter heating in either of these systems. The fan coil system is always a change-over system; that is, the water coil receives cold water in the cooling season and heated water in the heating season. The high pressure induction system may be designed as either a change-over or non-change-over system. This means the coil always received cold water when designed for nonchange-over and the air stream must be heated to compensate for the wall and window losses. Since primary air quantities are small in the induction system, this limits the use of nonchange-over systems to regions of relatively mild winters that have a degree day range from 0-2200. Primary air discharge temperatures rise to 130°F and above in severe climates which would be beyond normal recovered heat ranges. Where heating requirements are minimal, a nonchange-over induction system could be used. Analysis of winter solar loads and correct zoning for control is necessary for either induction or fan coil systems. When the induction system is designed for change-over (hot water coil and cold primary air stream) it becomes basically a reheat system because in winter the hot water must bring the primary air to room temperature plus the additional ΔT to offset wall and window losses. The fan coil unit will supply only the heat required for the wall and window losses. Supplemental air is heated by interior gains. In combination with a variable air volume interior supply, the fan coil system will be more economical in energy usage than the induction system. Opportunities for the use of hot condenser water are therefore present in each of the above systems for the conservation of energy. The variable air volume system may be used to the best advantages since it virtually eliminates reheating.

3.1.1.9 Heat of Light. Information on heat of light recovery is discussed in Section 2 of this Chapter.

Section 2. ENERGY RECLAMATION COMPONENTS

3.2.1 COMPONENTS. Heat recovery components reclaim energy that might otherwise be wasted. For optimum energy use in HVAC systems the techniques to be considered should include:

(1) Exhaust Air Heat Recovery - Rotary air wheels, static heat exchanger, heat pipe, run-around system.

(2) Heat of Light Recovery.

(3) Refrigeration-Type Heat Recovery - Refrigeration coil, heat pumps, single and double condenser circuits.

3.2.1.1 Techniques. Heat recovery techniques should be considered for all systems greater than 25 tons. Economic evaluations should include run-around system, heat pipe, thermal wheel, cooling coils in exhaust ducts (chilled water/direct expansion type with double bundle or heat pump application), double bundle refrigeration machines, heat pumps, solid waste recovery boilers (Chapter 4), and heat of light.

3.2.2 EXHAUST AIR HEAT RECOVERY.

3.2.2.1 Rotary Air Wheels. These are rotary wheels that transfer heat between exhuast and makeup air streams, i.e., air-to-air heat recovery. They are available in two types, one transfers sensible heat and the other transfers both sensible and latent heat simultaneously. The latter is also known as a total transfer or enthalpy wheel. Economic justification for installing either of the two must be made in relation to the systems. In a system that has double bundle heat recovery machines, application of the wheel is only beneficial below the breakeven temperature (T_{BE} - see Chapter 2). The number of occupied hours below this temperature should be obtained for the location from NAVFAC P-89, "Engineering Weather Data." For further information on wheel design, efficiency construction and economics, see 1972 ASHRAE Guide and Data Book, Chapter 34. The application of wheels also must consider the relative duct location of two air streams, effect on fan static pressure, space required by the wheel, and limitation on the contamination criteria for the building design. Heat recovery wheels are available in single units ranging from 300 to 50,000 CFM. Multiple units may be installed for larger capacities. Typical value of efficiency at equal flow ranges from 60 to 80 percent for both types of wheel. Air flow should be designed for counterflow for maximum efficiency and to keep the wheels clean, see Figure 3-7.

(1) Sensible Wheel: This transfers sensible heat only, in summer as well as winter. This is done by using heat absorbing corrugated metal mesh such as stainless steel or aluminum.

3.2.2.1.1 Temperature Calculations. Supply air temperature T_S °F. at the wheel outlet for equal supply and exhaust CFM is given by:

$$T_S = T_O + (T_E - T_O) \times \eta \quad \text{(wheel sensible efficiency)}$$

$$T_S = T_O + \frac{Q_E}{Q_S} \times (T_E - T_O) \times \eta \quad \text{(wheel sensible efficiency)}$$

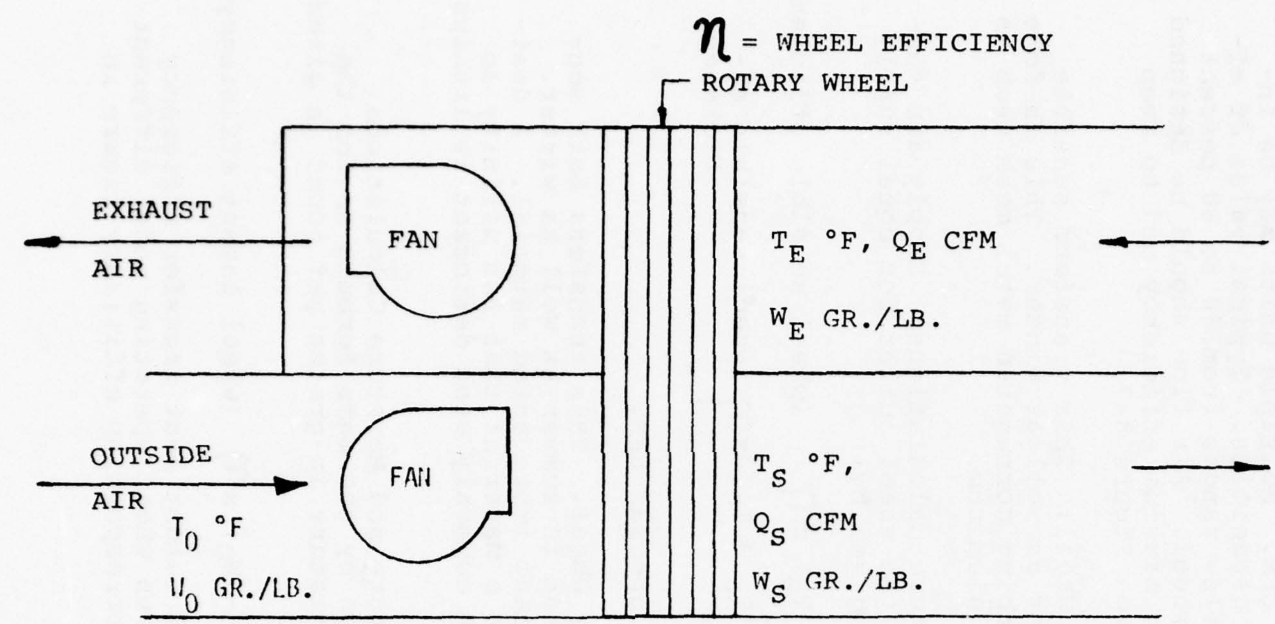
See Figure 3-7 for unequal CFM.

3.2.2.1.2 Enthalpy Wheel. This transfers both sensible and latent heat in summer as well as winter. It employs a desiccant impreganted material. A desiccant is defined as a material that has affinity to absorb moisture. A commonly used desiccant is lithium chloride (LiCl).

3.2.2.1.3 Temperature and Moisture Calculations. Temperature is given by the same formula as for the sensible wheel. Moisture in grains per pound is given by:

$$W_S = W_O + (W_E - W_O) \times \eta \quad \text{(wheel latent efficiency)}$$

Note: Sensible and latent heat transfer efficiency will vary for a given wheel operating under different conditions. Use corresponding efficiency figure in calculations.



T_O, T_S, T_E = DRY BULB TEMPERATURE, °F OF OUTSIDE AIR, SUPPLY AND EXHAUST AIR RESPECTIVELY.

W_O, W_S, W_E = MOISTURE CONTENT IN GRAINS PER POUND OF AIR IN OUTSIDE AIR, SUPPLY AND EXHAUST AIR RESPECTIVELY.

Q_S, Q_E = SUPPLY AND EXHAUST AIR QUANTITY, CUBIC FEET PER MINUTE.

FIGURE 3-7

HEAT RECOVERY WHEEL

3.2.2.2 Static Heat Exchanger. This device has no moving parts and sensible heat exchange takes place in alternate passages that carry exhaust and outside air through it in a counterflow fashion. This eliminates contamination and the mode of heat transfer is conduction. See Figure 3-8.

3.2.2.3 Heat Pipe. The heat pipe is a self-contained closed system capable of transporting large quantities of heat between exhaust and outside air streams. It consists of a bundle or finned array of 5/8" copper tubes similar to a dehumidifying or a cooling coil. Each tube is sealed at both ends and it is filled with a wick and a charge of working fluid. Common working fluid used for comfort air conditioning system includes refrigerant, water, and methanol; but for high temperature application liquid metals are the preferred working fluid. This device transfers only sensible heat and there is no contamination since the heat pipes are installed with opposite ends projecting into each air stream. See Figure 3-9 which shows a typical tube cross-section. The working fluid in liquid state is transferred towards the warm air stream by capillary action through the wick, where it evaporates absorbing heat from the hot exhaust air. The evaporated fluid then flows towards the cold end where it condenses with the release of heat to the outside air. This happens in winter; the function of each side reverses in summer. Counterflow air stream design gives optimum efficiency.

3.2.2.4 Run-Around System (Closed Loop). The exhaust air heat recovery components discussed so far necessitate that the exhaust and outside air intake ducts must be close to one another. The run-around system does not require this since it consists of two coil banks with a pump and closed pipe loop. A run-around system is a hydronic system that transfers sensible heat and in some cases latent heat also from air to fluid medium (glycol, water, lithium chloride, etc.) and back from medium to air. The coils when employed are finned and may be sprayed to acquire better recovery for summer operation. The pump may be located at any convenient location in the loop but an expansion tank must be installed on the suction side to allow

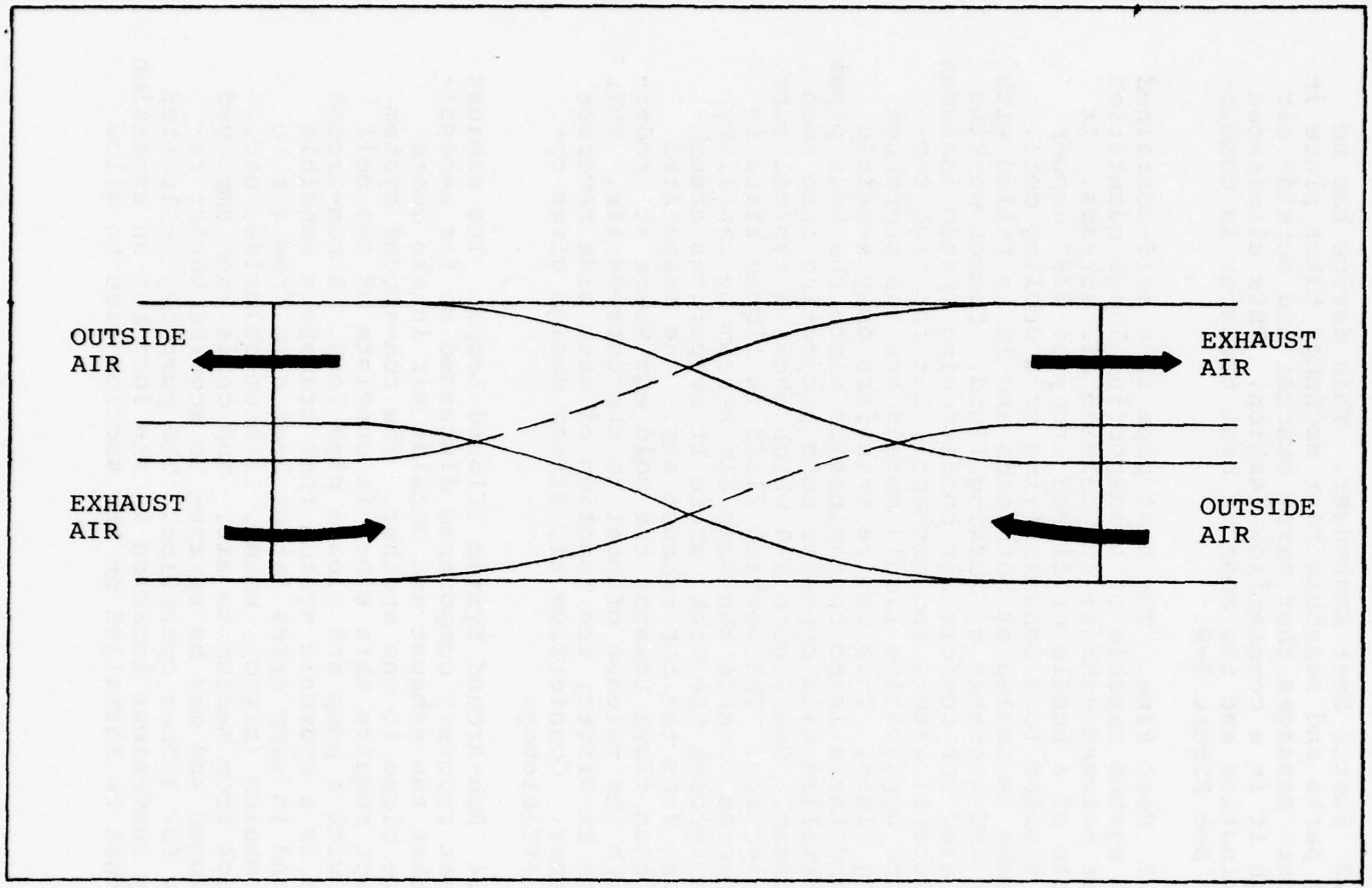
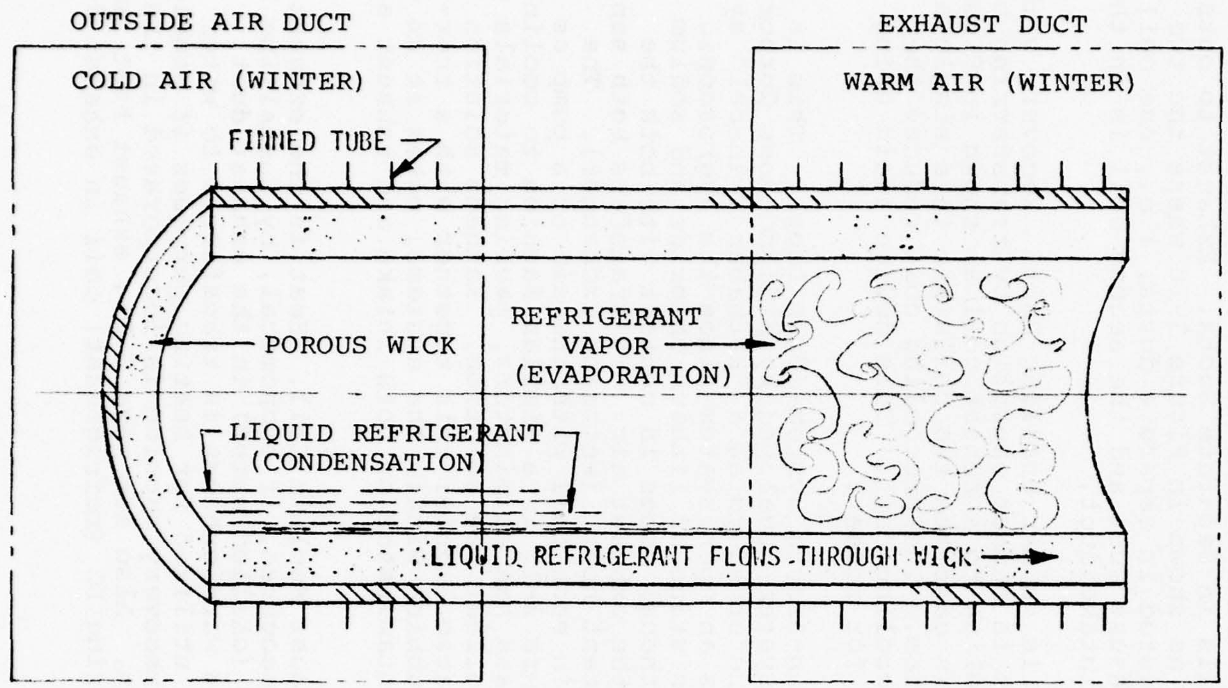


FIGURE 3-8

STATIC AIR-TO-AIR HEAT EXCHANGER



NOTE: ABOVE INDICATES WINTER FLOW SEQUENCE. IN SUMMER THE ABOVE FUNCTION OF EVAPORATOR AND CONDENSER REVERSES.

FIGURE 3-9

HEAT PIPE

for volumetric variation in the water and also to insure a net positive suction head. The run-around system may be used for the following applications:

(1) Coils in Separate Ducts. Exhaust to outside air exchange as shown in Figure 3-10 where the two coils are located in separate ducts, i.e., one coil is in the exhaust duct and the second coil is in the outside air intake duct.

(2) Coils in Air Hnadling Unit. Recovery within a single unit in summer operation by transferring heat from a precooling to a reheat coil as shown in Figure 3-11. In this case the two coils are in a single air handling system. The precooling coil reduces the load on the cooling coil. This device avoid using prime energy for reheat.

3.2.2.4.1 Run-Around System (Open Loop). This is a proprietary system developed by Midland-Ross Corporation and is referred to as a "Kathabar Twin-cel" system. This is an open system since the hygroscopic solution consisting of lithium chloride and sodium chromate (Kathene) comes in contact with both the outside and the exhaust air. It transfers both sensible and latent heat, (enthalpy exchanger). The fluid flows in each cell with the aid of a pump as shown in Figure 3-12 in a similar fashion to cooling tower flow, and has eliminators, packing materials and a basin to collect the solution. Kathene solution is a bacteriostatic liquid; this together with a thorough air scrubbing within the systems, makes it an efficient decontaminator of both intake and exhaust air streams.

3.2.2.5 Exhaust Reclaim Coil. Heat in the exhaust air may be recovered, if economical, by installing a cooling coil (chilled water) in the exhaust duct. Heat from the warm exhaust is transferred to water and could be utilized for heating purposes if double bundle heat recovery machine is incorporated in the central plant. Also see page 3-26, exhaust heat recovery by utilizing DX (refrigerant) coil in exhaust duct.

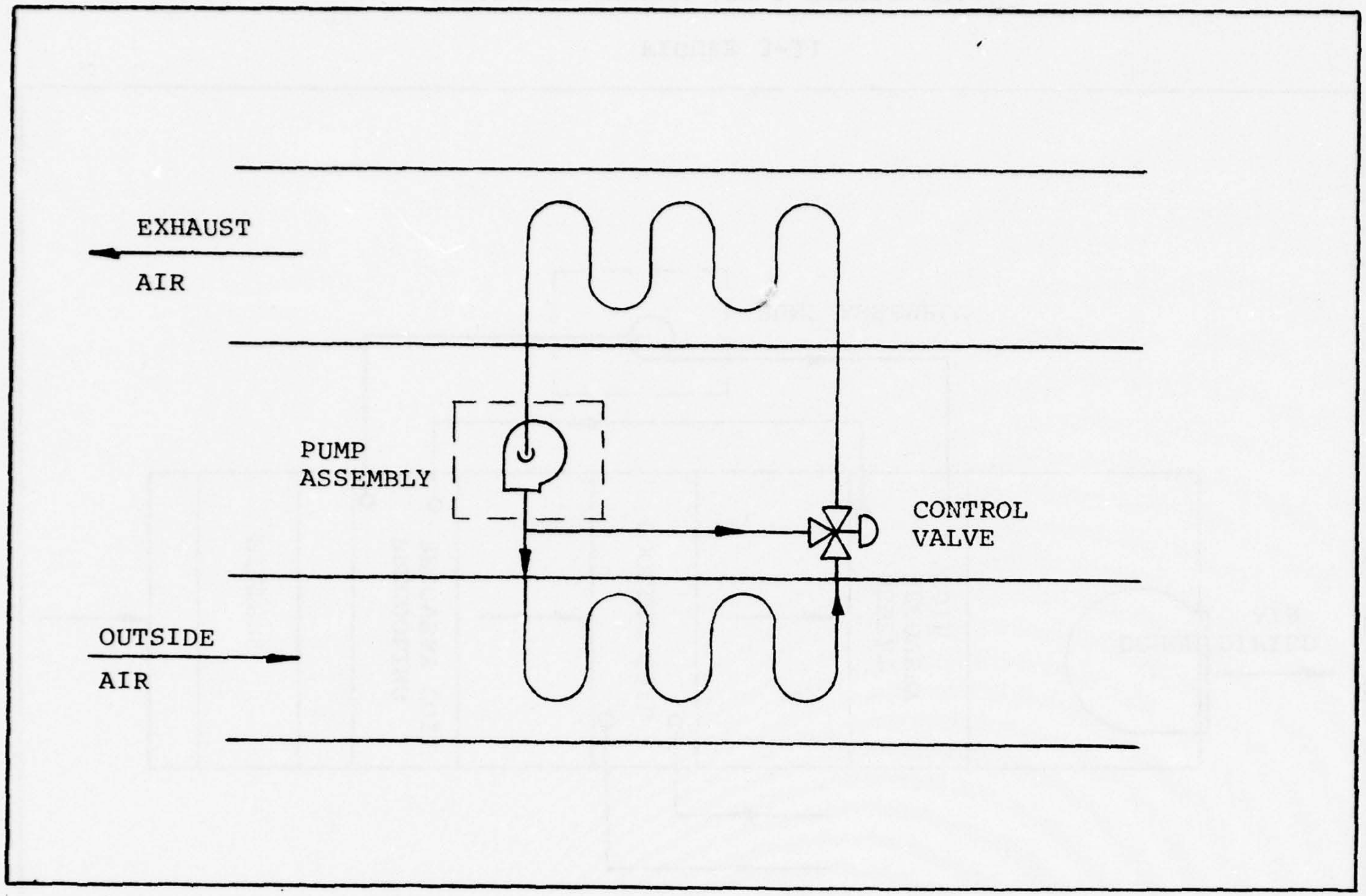


FIGURE 3-10

RUNAROUND SYSTEM (COIL IN SEPARATE DUCT)

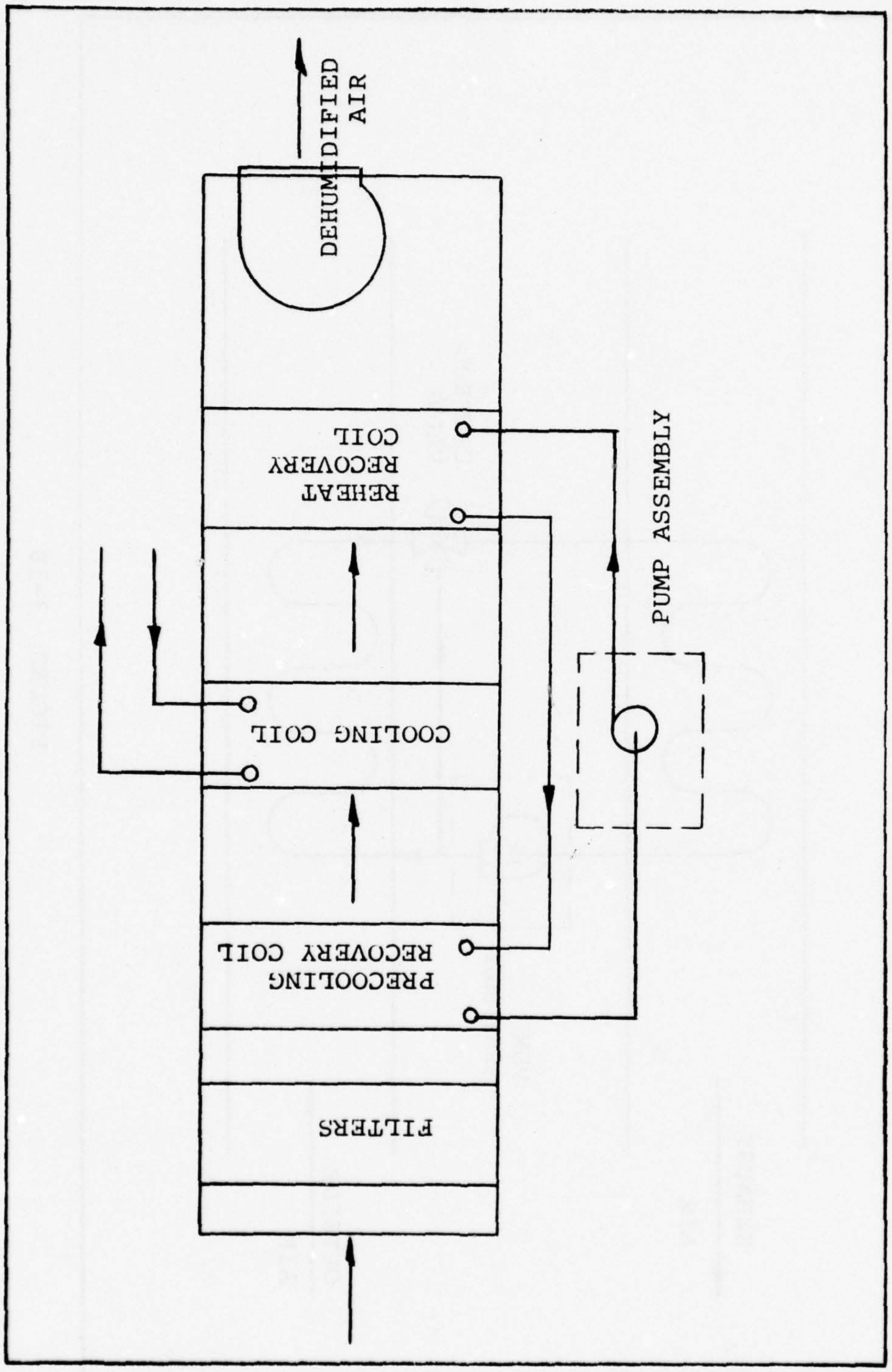


FIGURE 3-11

RUNAROUND SYSTEM (IN A SINGLE UNIT)

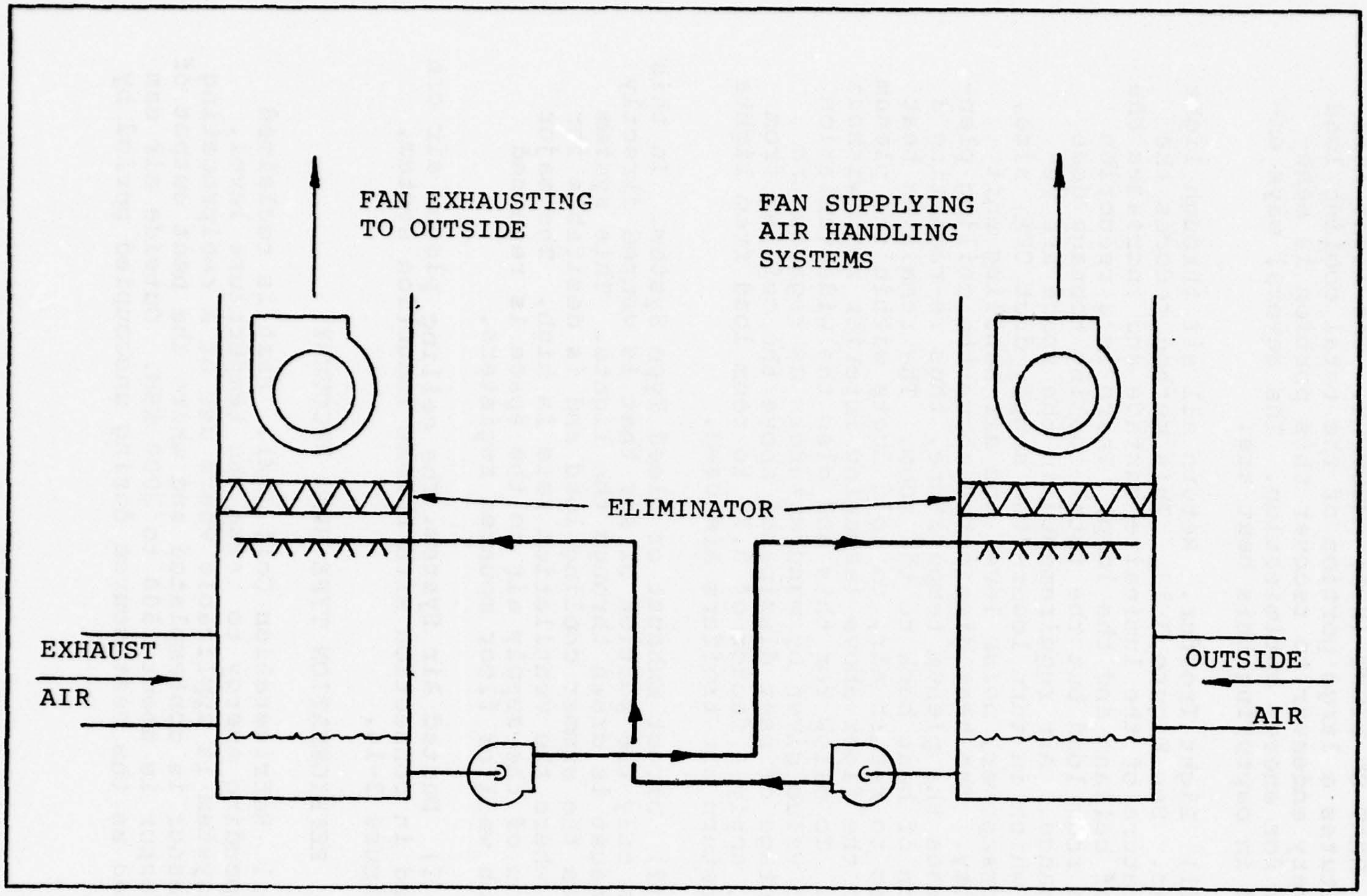


FIGURE 3-12

OPEN RUNAROUND SYSTEM (ENTHALPY TRANSFER)
(KATHABAR TWIN-CEL)

3.2.3 HEAT-OF-LIGHT (HOL) RECOVERY. The light load constitutes a large portion of the total cooling load and every endeavor to recover this portion is beneficial for energy optimization. The several ways employed in capturing this heat are:

(1) Light Troffer. Return all air through light troffer. See Figure 3-13. This method reduces the temperature of the luminaire surface and increases the life of ballast and the lamp. There is a reduction in the room load but the total cooling tonnage does not change. Air requirements in the space are reduced which in turn lowers the supply duct CFM, size, fan horsepower, noise level and air handling unit capacity. The heat dissipated above the ceiling plenum raises the plenum temperature, thus re-radiating a portion of heat back to the room. The remaining heat is lost to return air, to cold ducts within the plenum and to the floor above (assuming interior room without roof). To allow for this and also the wide variation in the value given by manufacturers as regards the percentage of heat dissipated above the ceiling from light, apply a factor of 0.75 to room load from lights when return air troffers are used.

(2) Direct Exhaust or Bleed Type System. In this system only the portion of air that is vented directly to exhaust is drawn through the lights. This system reduces the summer cooling load and is desirable for areas where the ventilation rate is high. The major portion of the supply air to the space is returned through wall or floor mounted registers.

(3) Ducted Air System. The ceiling plenum air can be used in connection with reheat induction system, see Figure 3-14.

3.2.4 REFRIGERATION TYPE HEAT RECOVERY.

3.2.4.1 Refrigeration Coil (DX). Heat is reclaimed by expending energy to raise the temperature level. This system is applicable where use of a reciprocating compressor is contemplated and where the heat output of compressor is about 1500 to 2000 KBH. Outside air can be used as the heat source during unoccupied period by

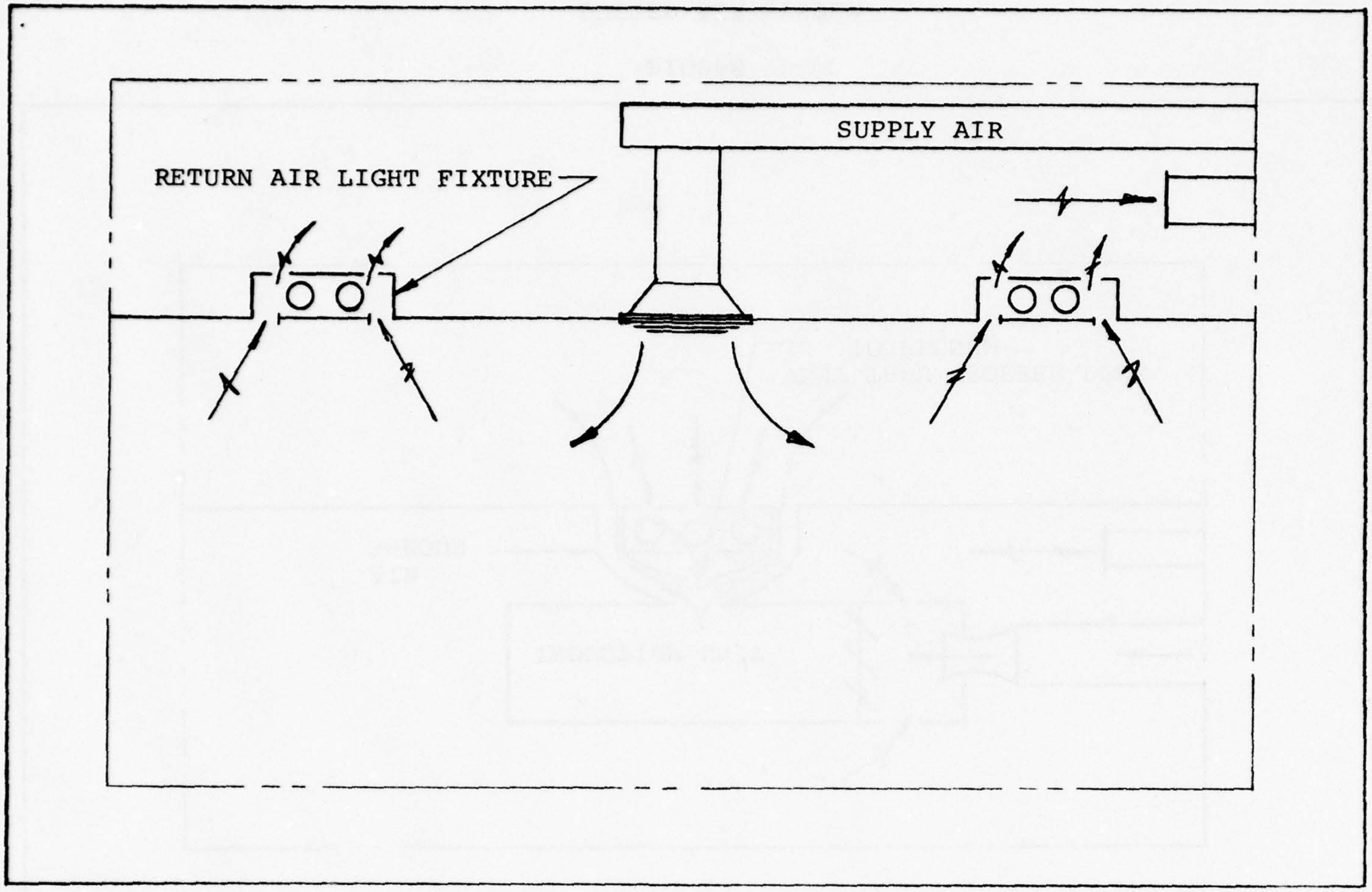


FIGURE 3-13

HEAT-OF-LIGHT RETURN

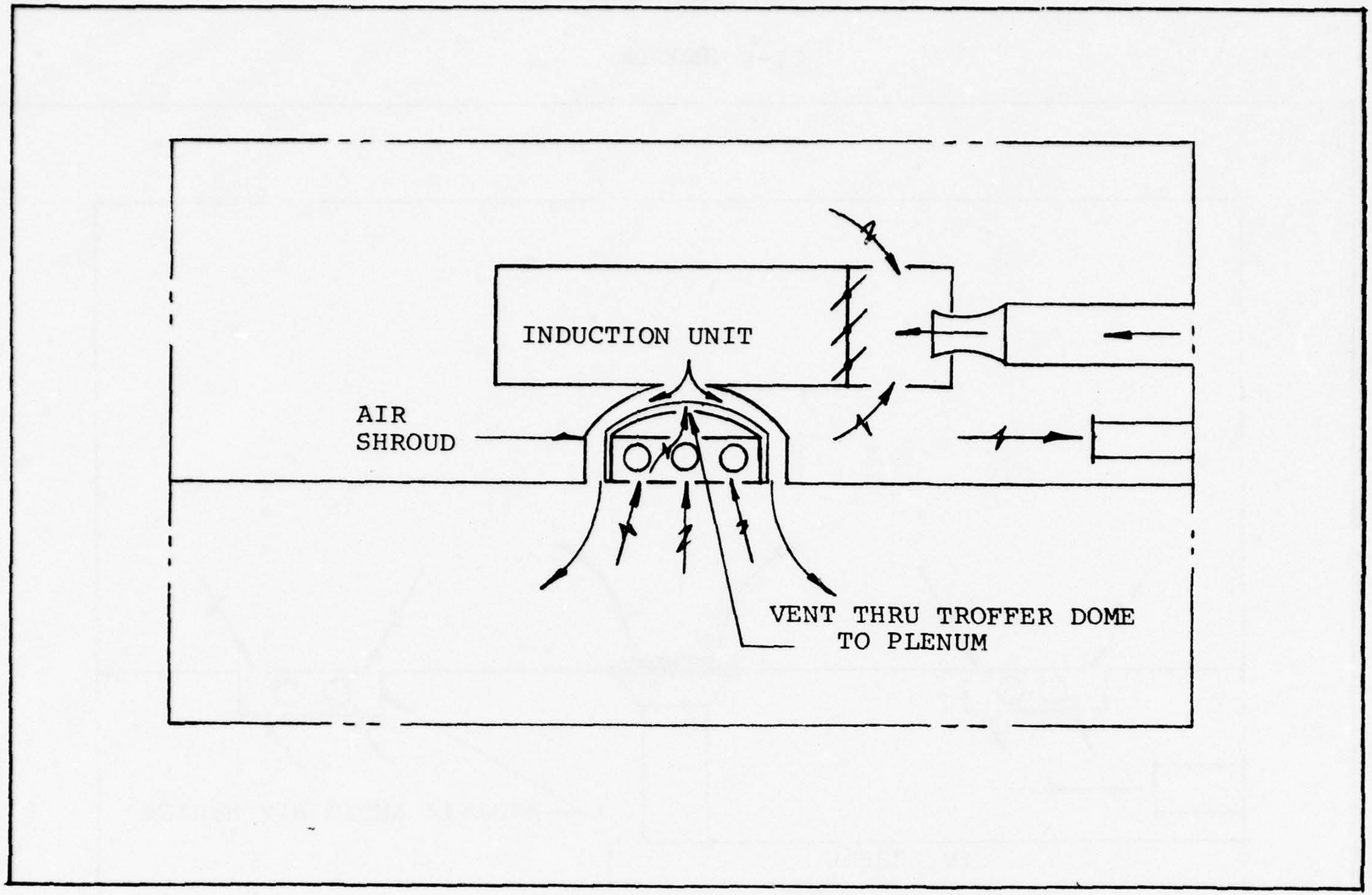


FIGURE 3-14

DUCTED AIR SYSTEM
(HEAT-OF-LIGHT)

reversing the cycle. Application of a reciprocating machine eliminates the surge problems that are associated with centrifugal type machines. Figure 3-15 shows cooling coil (DX) located in the all year-around interior system and the heating coil (condenser coil) located in the perimeter supply air system.

3.2.4.2 Heat Pumps. These are constructed of the basic refrigeration cycle components to utilize as much as possible both heating and cooling effects. Unitary heat pumps are available in sizes ranging from 1-1/2 to 20 tons. A heat pump is used where year-round air conditioning for commercial and large residential applications is required. Coefficient of Performance is a very important term widely used not only for heat pumps but also for all refrigeration type heat recovery systems. Its definition and purpose is explained by the Basic Refrigeration Cycle, Figure 3-16. Heat pumps may be classified as external source heat pumps and internal source heat pumps. Coefficient of performance (C.O.P.) is defined as the ratio of desired effect to energy input in consistent units. The C.O.P. value based on heating cycle is about 2 to 3, depending upon the heat source, unit selected and the operating conditions. An air to air heat pump has a slightly lower value and is about 1.4 to 2.2

Cooling $(C.O.P._C)$

$$= \frac{H_E \text{ (BTUH)}}{\text{Horsepower (HP)} \times 2545}$$

Heating $(C.O.P._H)$

$$= \frac{H_C \text{ (BTUH)}}{\text{Horsepower (HP)} \times 2545}$$

3.2.4.3 External Source Heat Pumps. Heating is the desired effect at the condenser (H_C) and the heat source at the evaporator (H_E) is obtained from outside the building from air, well water, solar or earth (ground). In Figure 3-17, the flow sequence in winter is 1,2,3,4,5,6,7,8,1 and in summer is 1,6,7,8,5,2,3,4,1.

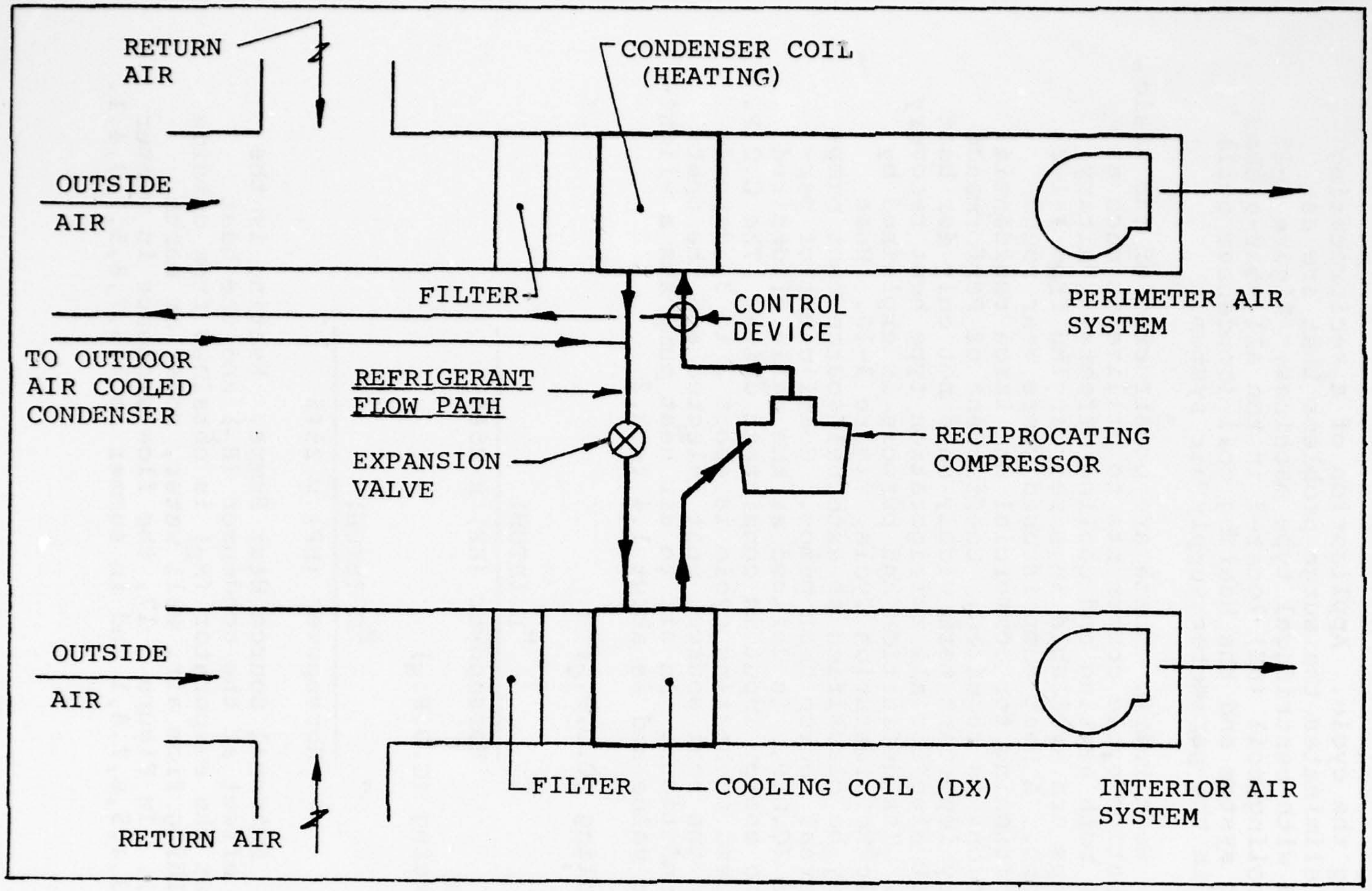


FIGURE 3-15

REFRIGERATION COIL (DX) HEAT RECOVERY SYSTEM

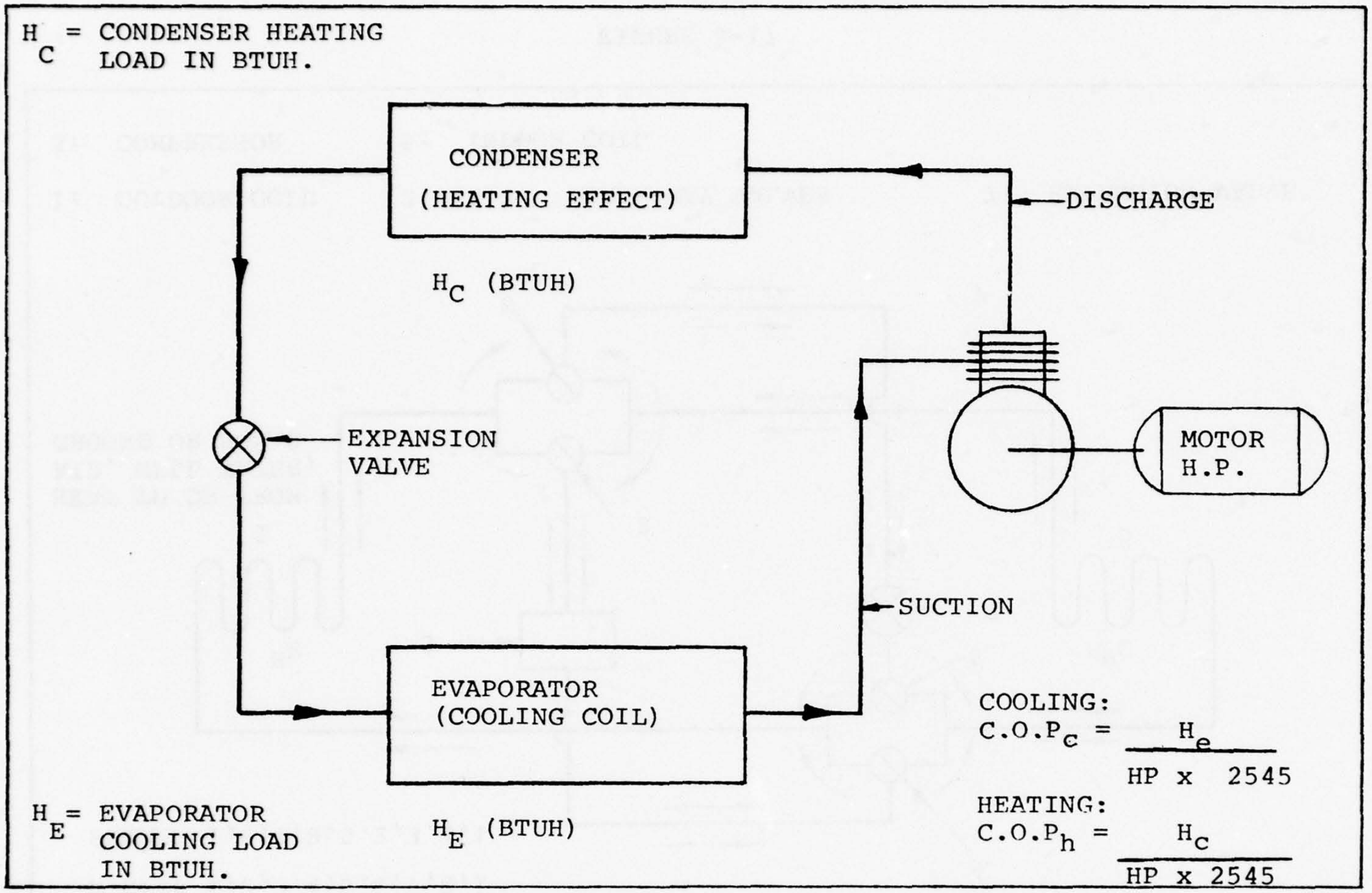


FIGURE 3-16

BASIC REFRIGERATION CYCLE

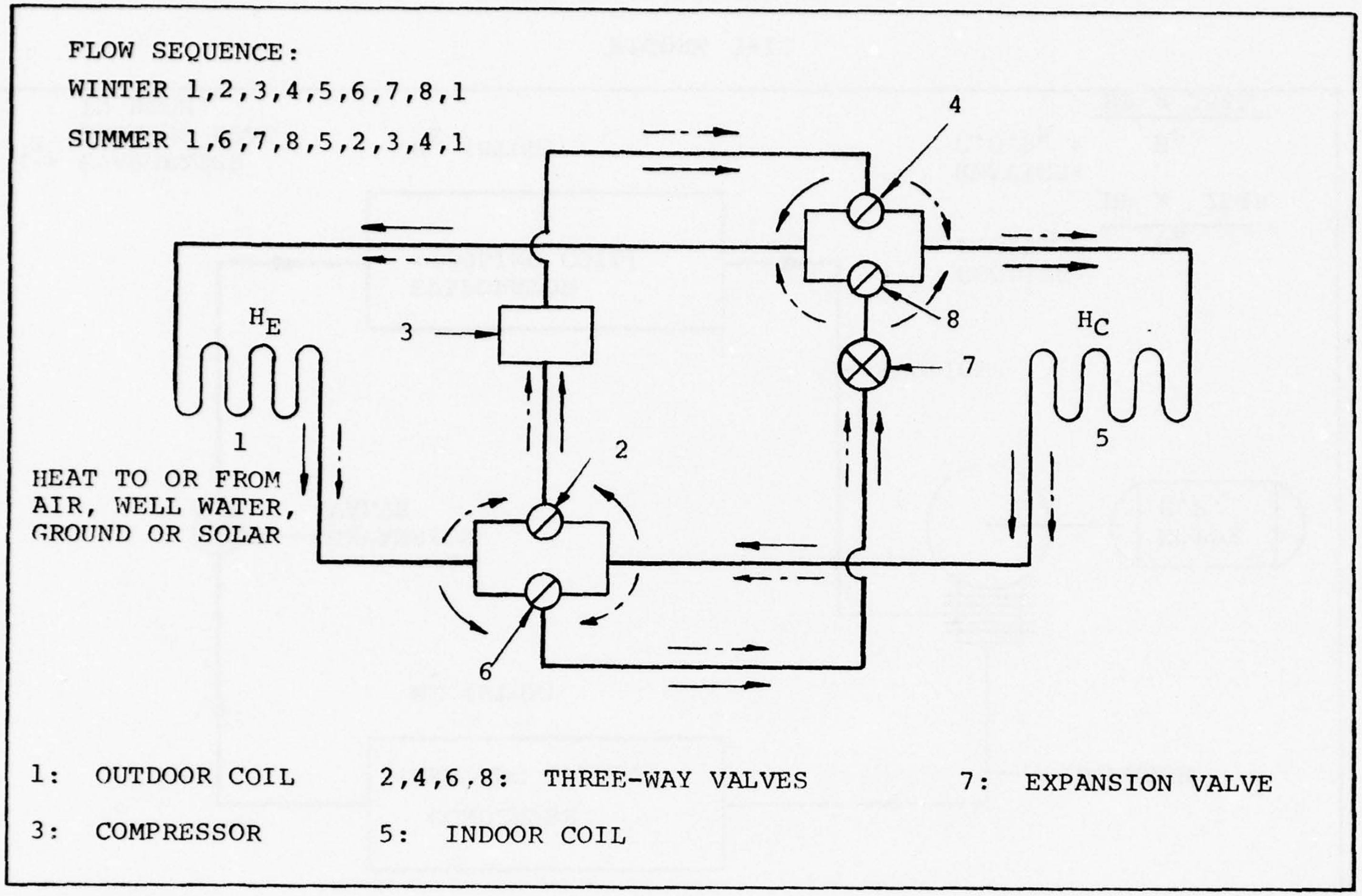


FIGURE 3-17

EXTERNAL SOURCE HEAT PUMP

Supplementary heating may be required for severe winter conditions. A provision must be made for the defrost cycle below 32°F and for drainage of water resulting from the defrost.

3.2.4.4 Internal Source Heat Pumps. With this type heat pump system separate areas of a building can be heated and cooled simultaneously. Heat removed from the space requiring cooling is transferred to the space requiring heating through a closed loop water circuit. This is used in a building that needs year-round cooling which could result due to high percentage of interior areas and high internal loads from lights, people, equipment and processes. The heat is extracted from and rejected to a common closed water loop system. Changing from heating to a cooling cycle reverses the function of condenser and evaporator as shown in Figures 3-18 and 3-19. Use heat pump principle in winter to transfer excess heat from windowless interior space or high heat gain areas such as EDP space to perimeter space. Figure 3-20 shows a typical heat pump system utilizing boiler and evaporative cooler. Boiler will furnish heat to maintain approximately 60°F in the main loop, and the evaporative cooler will dissipate heat to maintain approximately 90°F in the main loop.

3.2.4.5 Recovery with Single Condenser Circuit. A single condenser conventional machine with closed loop water cooled/evaporative cooling tower system: application of this limits the water temperature to a maximum of 110°F. This may be used with any type of compressor-reciprocating, centrifugal or screw type. See Figure 3-21. A modification to this system sometimes used is the substitution of a water to water heat exchanger and open cooling tower system for the closed loop evaporative cooling system. See Figure 3-22.

3.2.4.5.1 Conventional Machine with Water or Air Condensing. An air cooled system eliminates the problems associated with water evaporation such as make-up, bleed-off, and chemical treatment. Because of large space requirements this system is usually for lower tonnage. In this range mostly reciprocating compres-

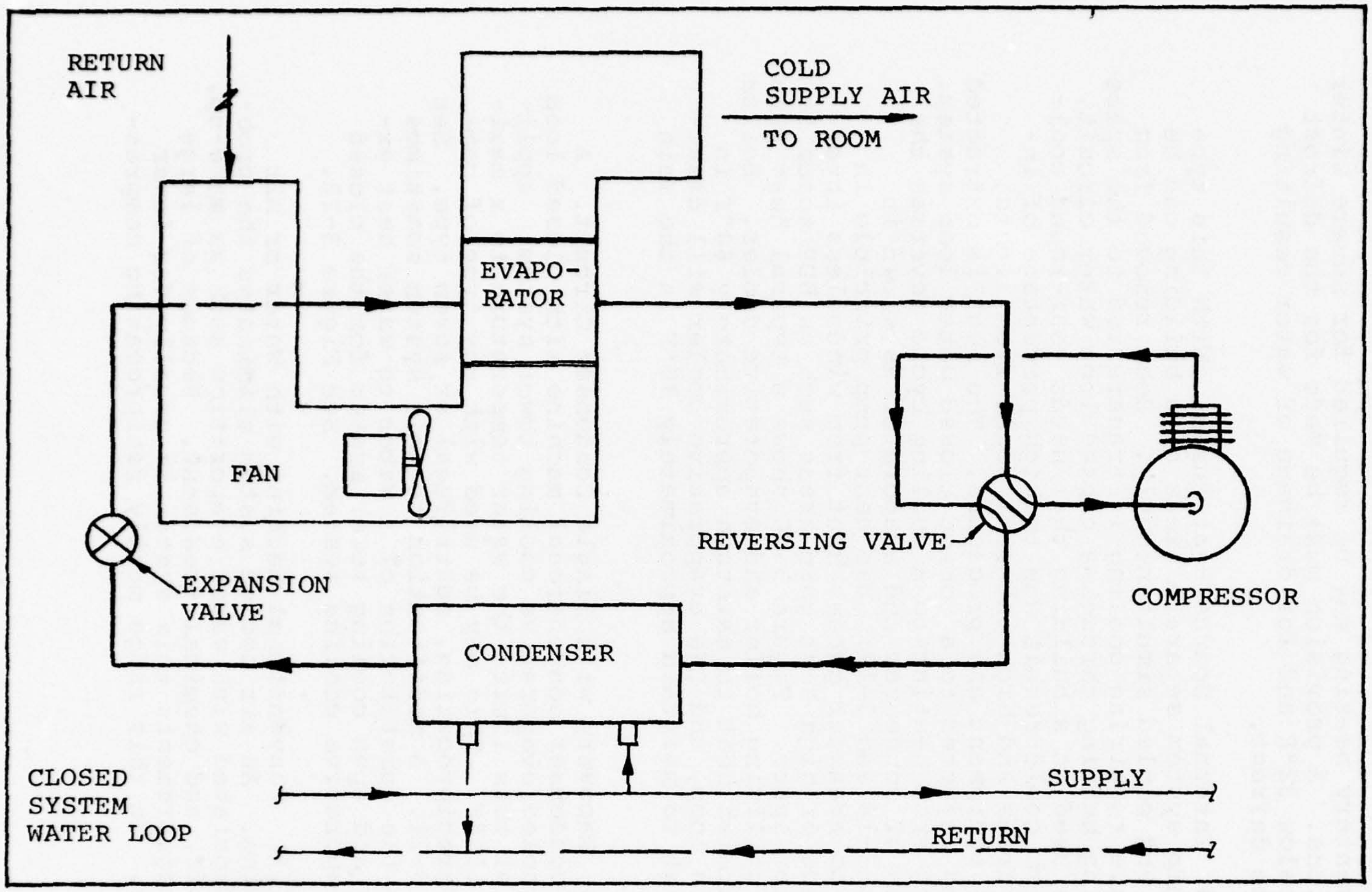


FIGURE 3-18

HEAT PUMP - COOLING MODE (INTERNAL SOURCE, CLOSED LOOP)

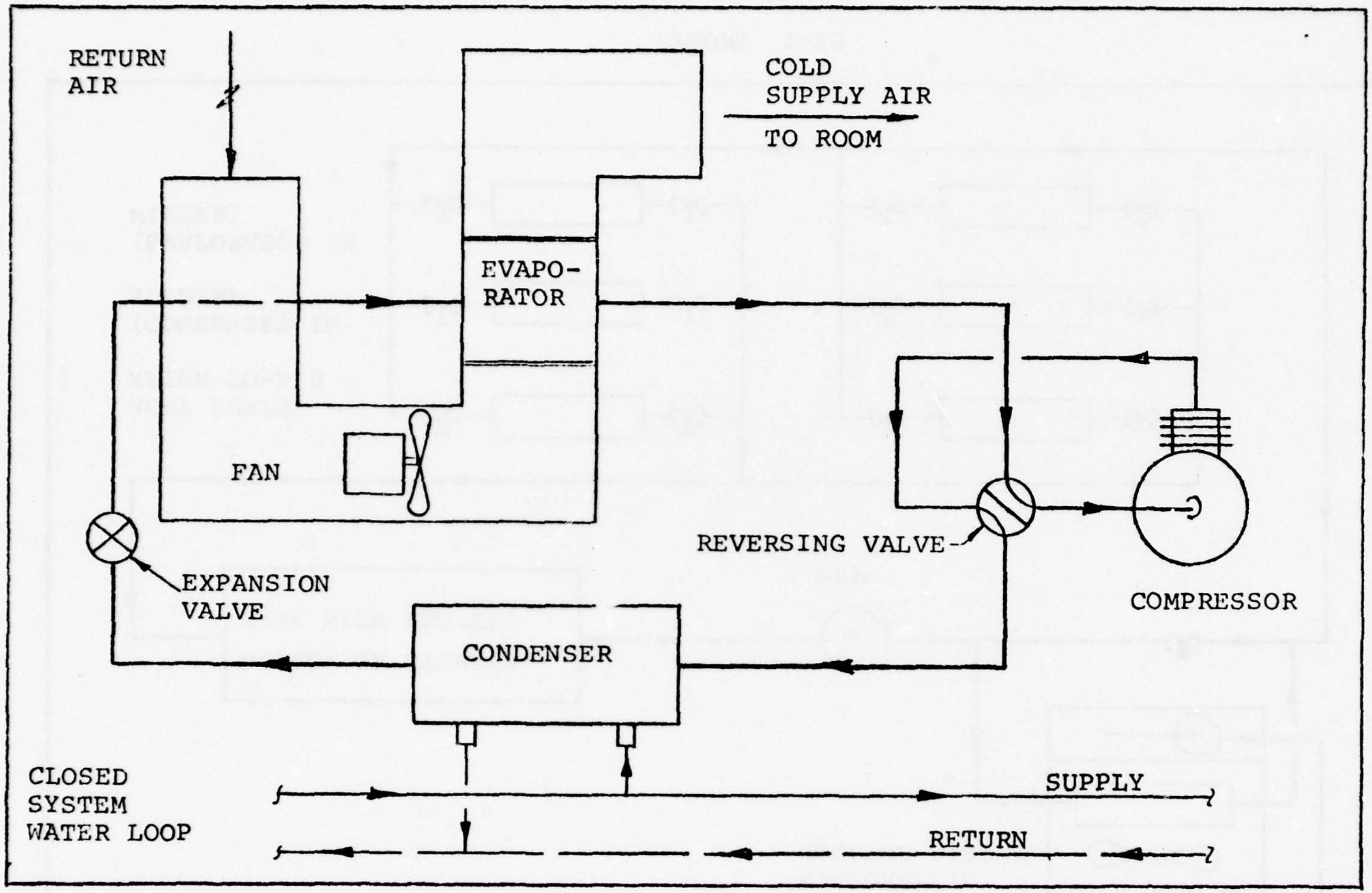


FIGURE 3-19

HEAT PUMP - COOLING MODE (INTERNAL SOURCE, CLOSED LOOP)

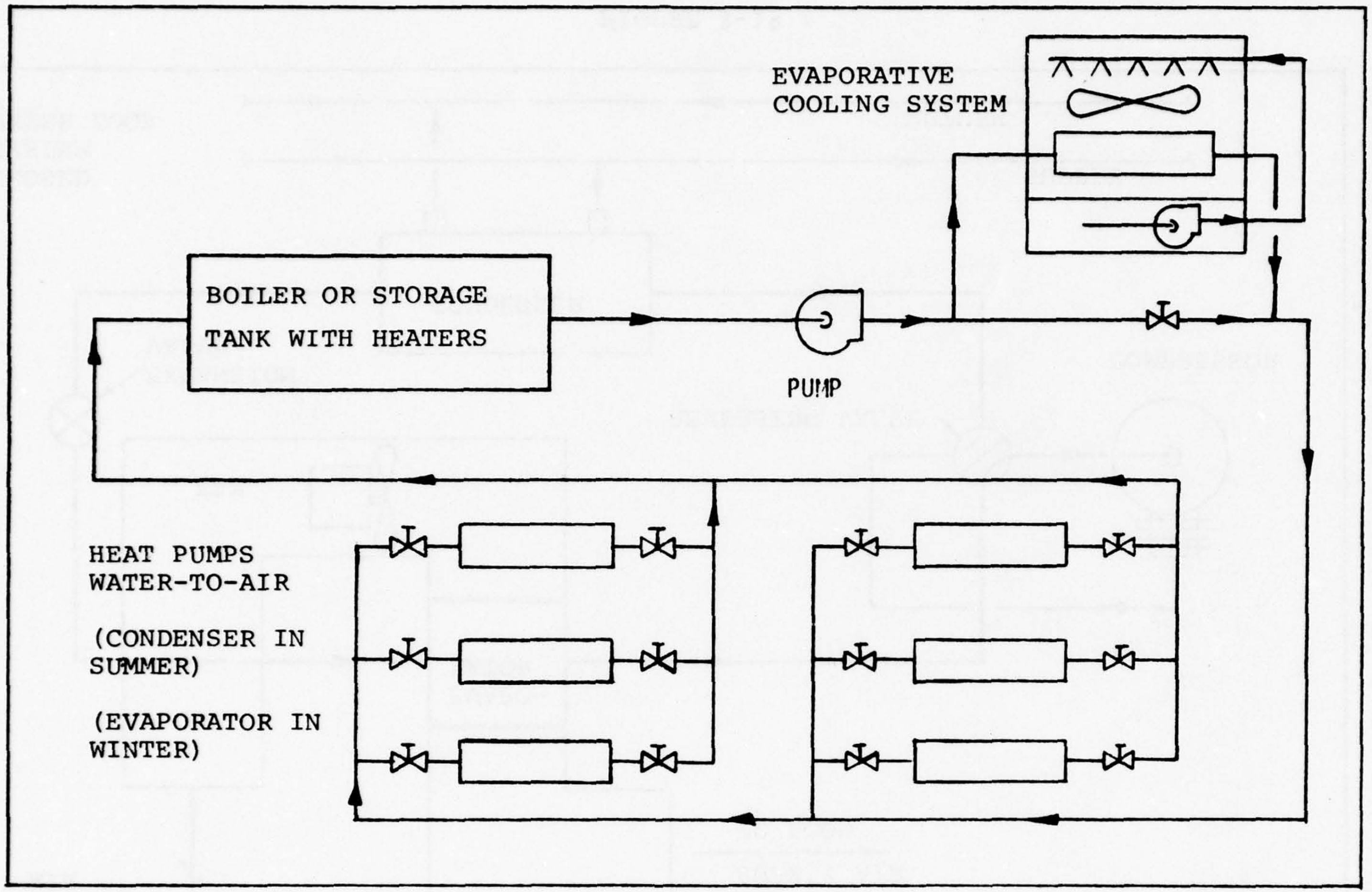


FIGURE 3-20

TYPICAL HEAT PUMP SYSTEM WITH BOILER AND EVAPORATIVE COOLER

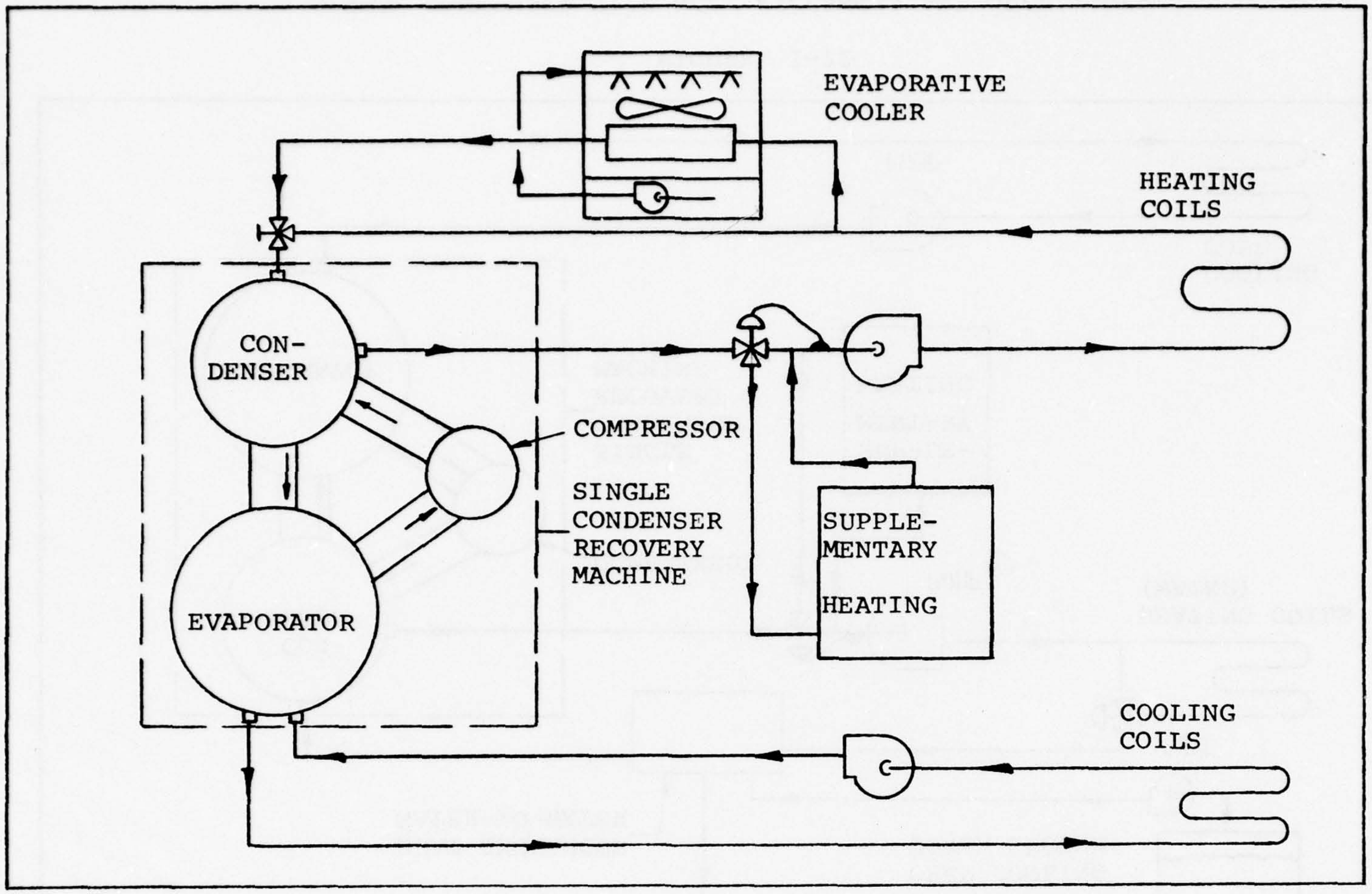


FIGURE 3-21

SINGLE CONDENSER WITH EVAPORATIVE COOLING SYSTEM

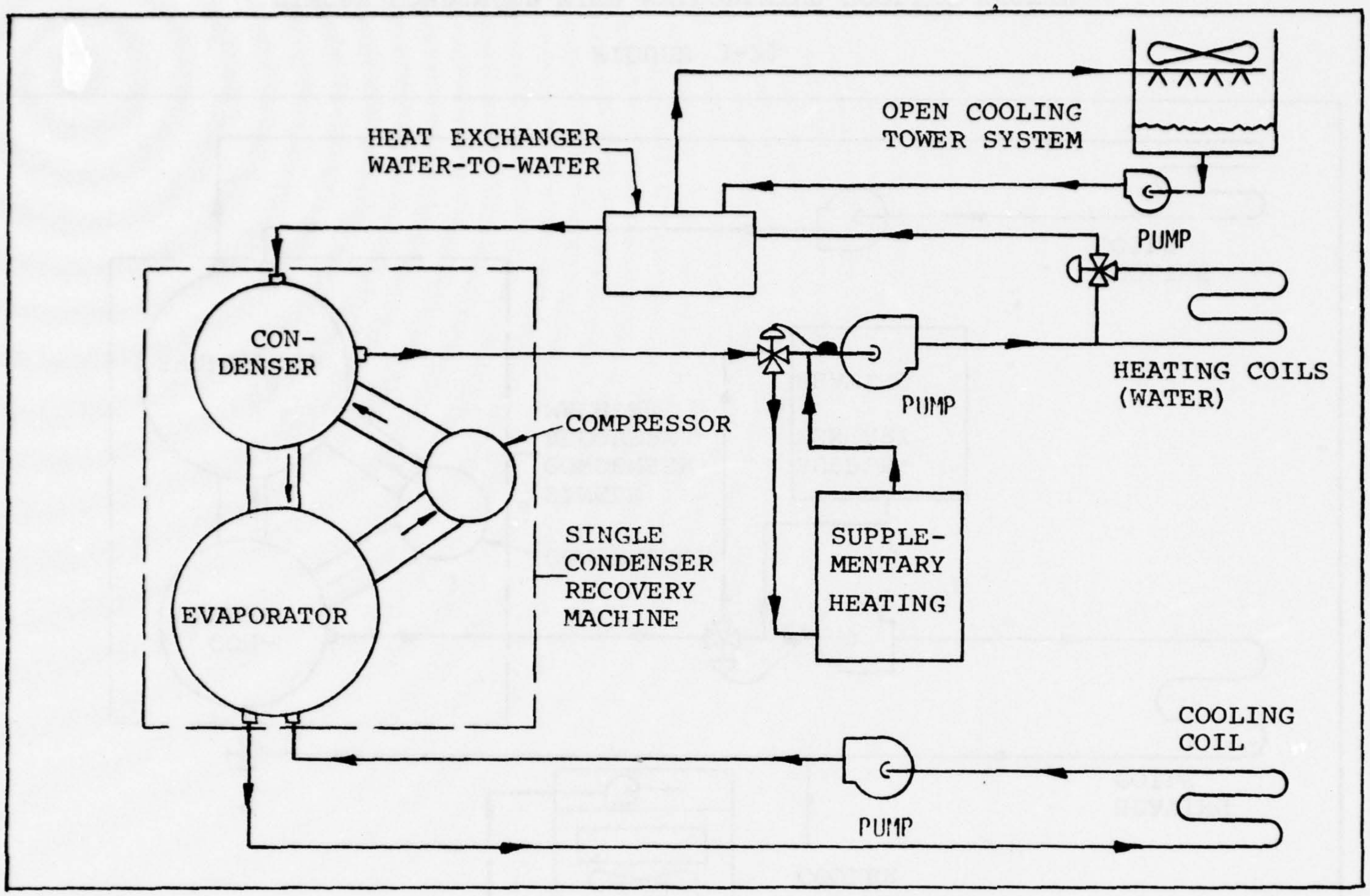


FIGURE 3-22

SINGLE CONDENSER WITH HEAT EXCHANGER AND OPEN TOWER sors are used although centrifugal compressors may be used if their use is justified economically. The hot water system heat exchanger is connected in parallel with the air cooled condenser as shown in Figure 3-23. A series connection also is feasible. This system is suitable where the condensing unit and the heating system are remotely located from each other. In some designs a condensing refrigerant coil may be substituted for the interchanger and water pump for direct air heating, see Figure 3-24.

3.2.4.6 Heat Recovery with Double Condenser or Circuits. Double-bundle or split condenser heat recovery with an open cooling tower system: This requires a modified refrigeration machine with the condenser and the evaporator isolated from one another. The condenser shell incorporates two water circuits, one for the building heating system, and the other for the open cooling tower system. See Figure 3-25. This is done to isolate the contamination and corrosion problems inherent with the open cooling tower system. This construction also gives efficient heat transfer and lower heating system maintenance cost. Water temperature in the range of 125° is obtained by using higher compressor speed, larger impeller, or 2 or more stage compressors. Storage tanks may be incorporated at the location shown in the figure. Coefficient of performance for heating ($C.O.P._H$) for this system at peak heating load is about 4 to 4.5 depending upon the machine selection. Economic evaluation shall be made for use of heat recovery machines in all central plants. If economically justified such a central plant may be designed for multiple machine installation (using conventional or regular machines in conjunction with heat recovery machines, also known as double-bundle or split condenser machines), with as much tonnage as possible sized for efficient summer operation. The coefficient of performance ($C.O.P._H$)of a heat recovery machine in a heating mode shall not be below 4.2.

3.2.4.6.1 Satellite Condenser System. This involves addition of an external or auxiliary condenser to a conventional, universal type of machine, where the cooler and the condensers are housed in a common shell. This may be designed to provide 100°F. or 105°F. water

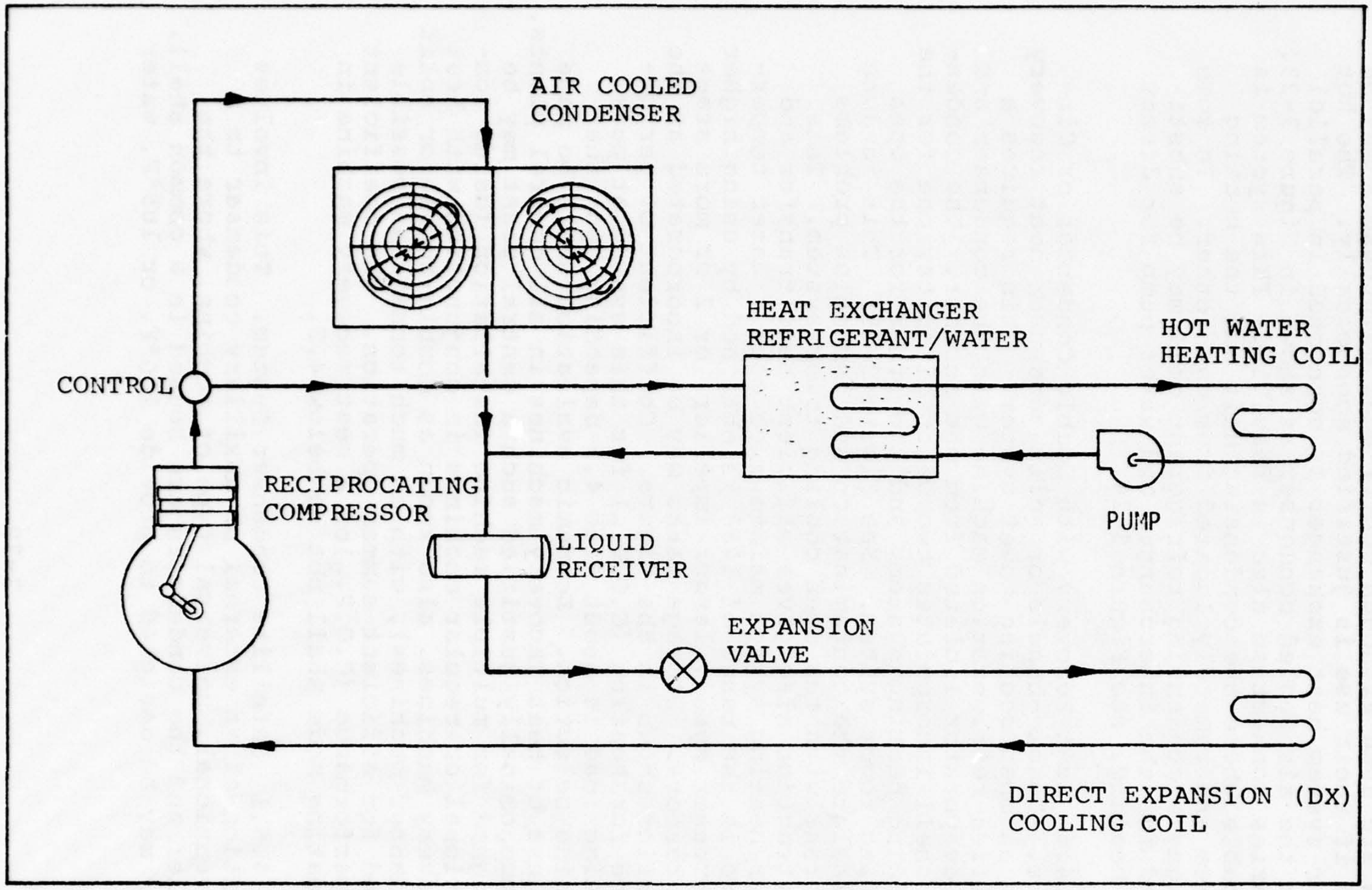


FIGURE 3-23

AIR COOLED CONDENSER WITH REFRIGERANT TO WATER HEAT EXCHANGER

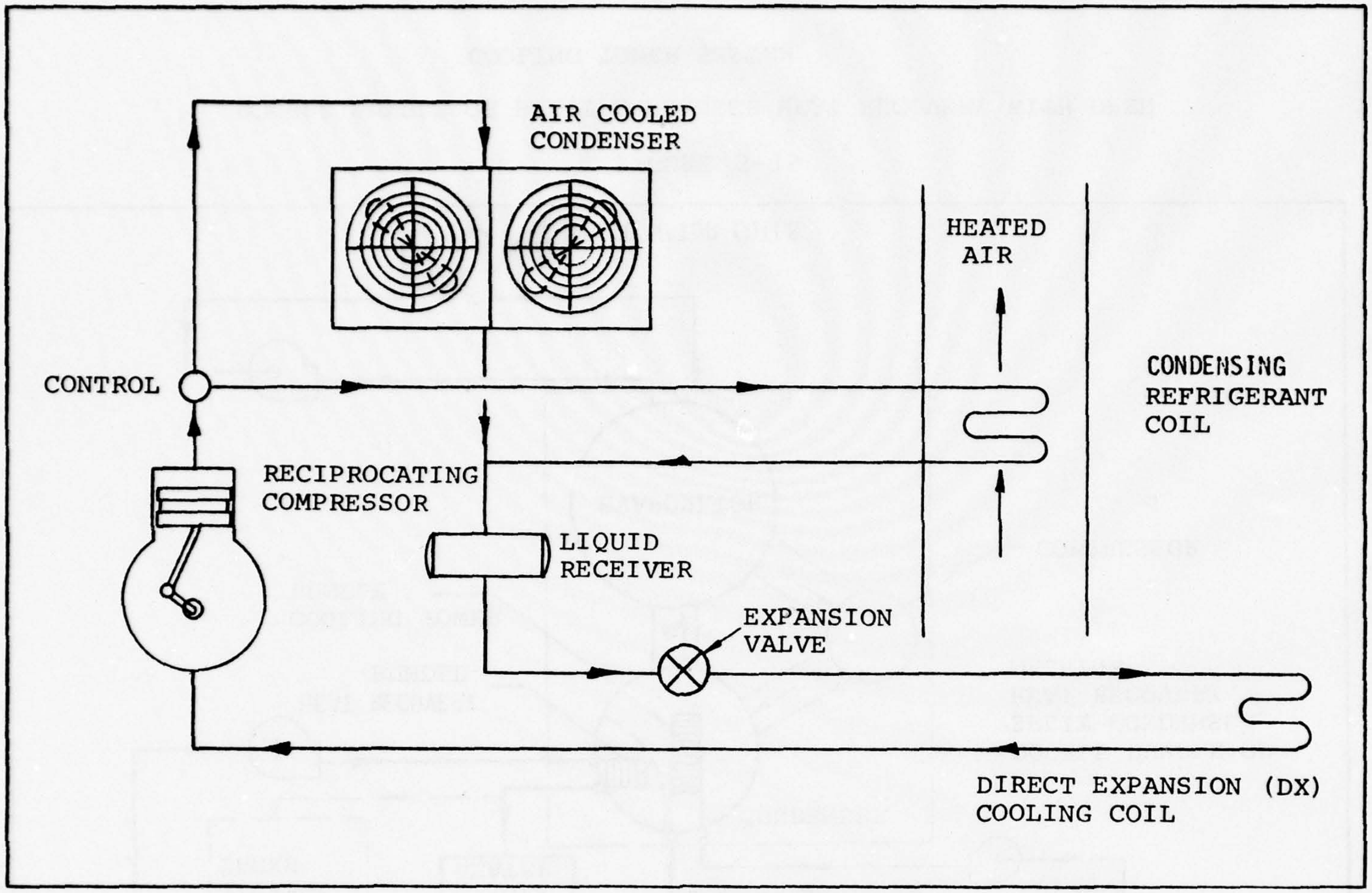


FIGURE 3-24

AIR COOLED CONDENSER WITH HEATING REFRIGERANT COIL
(MODIFIED SYSTEM)

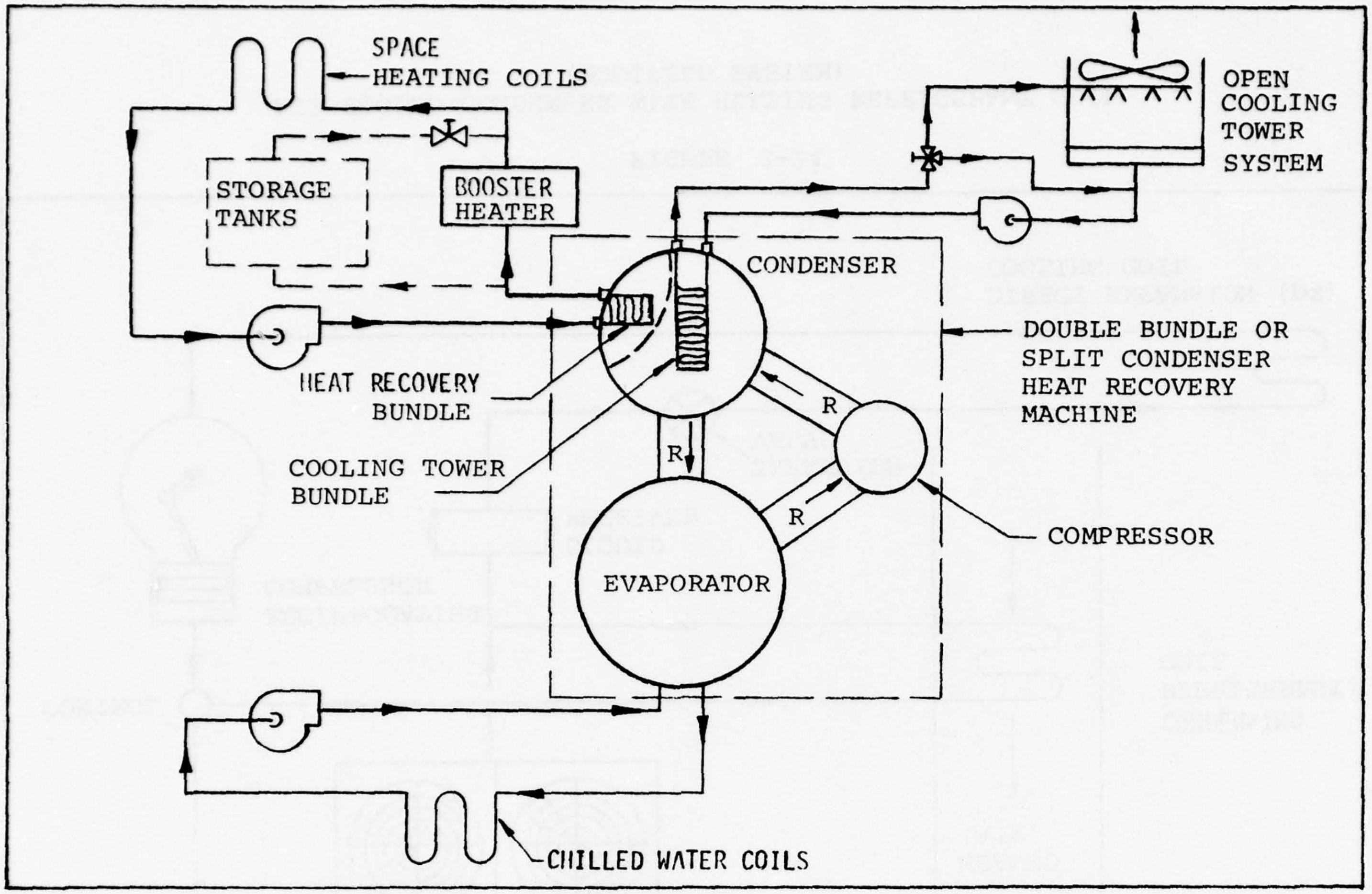


FIGURE 3-25

DOUBLE BUNDLE OR SPLIT CONDENSER HEAT RECOVERY WITH OPEN COOLING TOWER SYSTEM temperature. The double bundle recovery system should be considered for all projects that have a year-round cooling load and where the recovered heat can be used beneficially. The recovered heat may be used to furnish heat to system utilizing one or more of the following: perimeter radiation, unit heaters, domestic hot water, snow melting, heating coils in air handling, fan coil and induction units, reheat system (where required) or storage tanks. When the total recovered heat is less than the heating demand, the deficit is made up by the supplementary heating source. This additional heat is rarely used and is required only below the building breakeven temperature. When the recovered heat exceeds the heating demand, the surplus heat is dissipated to the outside by means of a cooling tower or evaporative cooler.

3.2.5 STORAGE TANKS. The objective of any heat recovery system is to salvage heat which would otherwise be wasted. It can be used immediately or stored for use at another time. The heat salvaged usually comes from the large lighting, equipment, and people loads generated during daytime building usage. The stored heat can then be used during the night to supply the building heating demand which consists mainly of the transmission loss through the walls, glass, etc. Heat storage is accomplished by circulating water from storage tanks to heat reclaim or double-bundle heat recovery machines. The storage tanks may be utilized in several different ways, such as: (1) to store chilled water to minimize peak demand, (2) installing an electric resistance heater to provide supplementary heat after the tanks are depleted or, (3) charging the tanks at night with low cost electrical energy, and (4) with sufficiently high storage temperatures, the tanks can supply building heat directly, thereby reducing the operating time of the booster heater. To optimize tank size for capacity and size, the tank should be located so that it receives the hottest water from the heating circuit and the coldest water from the chilled water circuit. This same location will also be desirable from the hydraulic standpoint, since it will minimize pressure in the storage system. In multiple tank installations, series piping of the tanks will decrease balance problems.

Section 3. OTHER TECHNICAL GUIDELINES AND MANDATORY REQUIREMENTS FOR ENERGY CONSERVATION

3.3.1 ECONOMY/ENTHALPY CYCLE. Figure 3-26 shows the regions for enthalpy control. Each HVAC system with cooling capacity greater than 120,000 Btu/hour in buildings not equipped with internal/external zone heat recovery equipment shall be designed to use maximum outdoor air for cooling whenever cooling is needed at an outdoor condition such that:

(1) The enthalpy of the outdoor air is lower than that of the indoor air, and/or

(2) The outdoor dry bulb temperature is lower than that of the indoor air, and

(3) The humidification requirements shall not exceed 2.5 times that required when sized for minimum outside air.

3.3.2 MECHANICAL DRIVE SYSTEMS. The use of a mechanical drive, such as steam turbine, diesel engine, or combustion turbine should also be considered as the drive for refrigeration compressors, especially for larger cold storage warehouses or where there is a year-round requirement for air conditioning. For very large installations, full consideration shall be given to a "piggy-back" configuration of a steam turbine driving a centrifugal or helical rotary-screw compressor and exhausting into an absorption refrigeration machine. Mechanical drive of air conditioning equipment may prove economical, even without heat reclamation, in areas of high electric power cost.

3.3.3 CONTROL VALVES. Use two-way control valves for chilled water and hot water coils in lieu of three-way valves to acquire variable flow and reduction in energy requirements. Valve selected shall be equal percentage contoured type to maintain minimum 25 percent of the flow rate. Also consider variable speed pumping if economical.

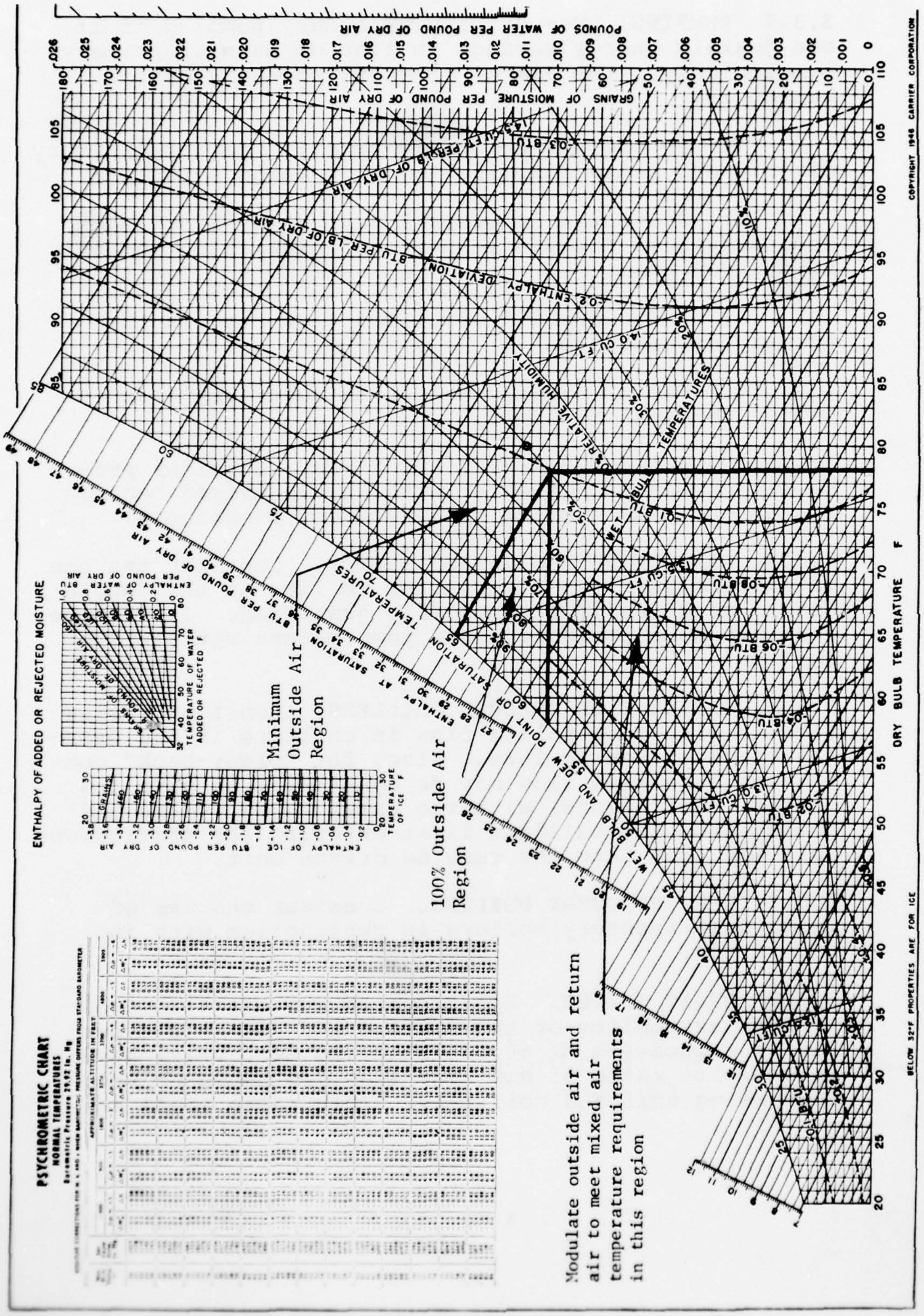


Plate I Psychrometric chart for normal temperatures

FIGURE - 3-26. REGIONS FOR ENTHALPY CONTROL

3.3.4 PUMPING. Use primary-secondary pumping if it can justify energy savings in lieu of main or primary pumping only. The aim of primary-secondary distribution is:

(1) To isolate secondary circuit from the primary circuit and hence minimize the flow balance problem.

(2) Reduction in overall pump power since each secondary pump circuit is pumped in isolation and only to its own need.

(3) Reduced and diversified flow rate in the primary distribution.

(4) Use of two-way valves in secondary distribution terminal control.

3.3.5 PREHEATING DOMESTIC HOT WATER. Consider preheating domestic hot water with the recovered heat in system that incorporate heat recovery techniques.

3.3.6 CASCADE REFRIGERATION SYSTEM. Consider the use of cascade refrigeration system in lieu of the double-bundle machines if economically justified. It will be possible to get higher water temperature with this system.

3.3.7 SERIES FLOW THROUGH CHILLERS. Consider series flow in lieu of parallel flow in chillers if the overall kilowatt/ton is less. Also, for "piggy-back" configuration series flow may be beneficial to give low energy requirements, with the water to be cooled and coming from the building first entering the absorption unit and then into the turbine driven unit.

3.3.8 HEAT RECOVERY BOILERS. Consider the use of waste heat recovery boilers in conjunction with incinerators.

3.3.9 SPLIT SYSTEM. Split system unitary air conditioning assemblies of the RCU-A-C and RCU-A-CB types having capacities of 60,000 Btuh and less shall have a Btuh/watt ratio of not less than 7.5 based on the condensing unit and coil only. This ratio shall be established for both types of assemblies from the capacity and power ratings listed for RCU-A-C assemblies in the ARI publication "Directory of Certified Unitary Air conditioners" (latest edition). In determining the ratio for a RCU-A-CB assembly, when the condensing unit is listed under RCU-A-C assemblies with different coils, the condenser coil assembly with the highest Btuh/watt ratio shall be used to determine the acceptability of the RCU-A-CB assembly. In cases where the condensing unit used with a RCU-A-CB assembly is not listed as part of RCU-A-C assembly, the Btuh/watt ratio, based on the information listed for the RCU-A-CB assembly shall not be less than 6.5.

3.3.10 WINDOW UNITS. When room (window) air conditioning units are used, they shall produce not less than 8.5 Btuh per watt input for 120 volts and not less than 8.0 Btuh per watt input for 230 volt units. In order to establish these ratings, the Association of Home Appliance Manufacturers' publication "Directory of Certified Room Air Conditioners" (latest edition) shall be the sole determination. Energy rates for through-the-wall units shall be as specified in Fed. Spec. 00-A-372. All future replacements of room units shall conform to these requirements.

3.3.11 AIR CONDITIONING SYSTEMS. Design air conditioning systems so that all return air passes through louvers in lighting fixtures to prevent lighting and ballast heat from entering the occupied space. Because this method eliminates the lighting heat at the source, it is possible to use smaller air handlers, coils, and ducts since the room load has been lowered. Use this warm air as reheat in air conditioning systems. See paragraph 3.2.3.

3.3.12 SENSING UNIT. Use an outside temperature sensing unit to modulate temperature of hot water heating systems by increasing water temperature as outside air temperature drops and decreasing water temperature as outside air temperature rises. When fan coil units are used to provide both heating and air conditioning, the hot water should be modulated down to a maximum temperature of 75°F. when the outside temperature is 60°F.

3.3.13 SHUT-OFF. Provide a positive shut-off of heating systems when outside air temperature reaches 65°F. In well insulated buildings, cut-off can occur at 60°F.

3.3.14 PROGRAMMED CONTROLS. Use programmed controls through clocks or other systems for night, weekend, and holiday temperature set back (or cut-off) to reduce air conditioning and heating loads. Normally, air conditioning for personnel comfort will be cut off during unoccupied hours and heating reduced by 10°F. to 15°F. depending upon equipment capacity, pick-up limitations, and if no additional energy is consumed.

3.3.15 WINTER AIR CONDITIONING. Use air cooled condensers or cooling towers with indoor sumps to eliminate the need to heat cooling tower basins where air conditioning is required in below freezing weather.

3.3.16 CONDENSER WATER TEMPERATURE. Condenser water temperature for the refrigeration machine (chiller) shall be as low as the ambient temperature will permit but not below the minimum temperature recommended by the chiller manufacturer. This will reduce the kilowatt/ton of refrigeration. Also, if possible, consider varying the condenser water temperature with respect to ambient to lower the energy requirement. Also check with the manufacturer the possibility of obtaining free cooling, i.e., chilled water without running the compressor (thermocycle or free cooling principle). Thermocycle or free cooling provides up to 30 to 40 percent of total chiller capacity by utilizing the cold from outdoor air to get chilled water. Cold water from cooling tower flows through the condenser lowering the refrigerant pressure. This causes the refrigerant from the evaporator to flow to the condenser, where it is condensed. The liquid refrigerant drains by gravity. This process is accentuated by a gas by-pass from condenser to evaporator and liquid refrigerant spray pump that sprays the refrigerant from the bottom of the evaporator and over the chilled water tubes. The system's compressor remains idle during the entire process.

3.3.17 INSULATION. Piping and ductwork insulation shall be optimized to minimize the cost of heat gain or loss when compared with cost of insulation, but in no case should the thickness be less than that specified in NAVFAC specifications.

3.3.18 BUILDING ORIENTATION. Building orientation and configuration shall be selected to minimize infiltration and solar loads.

3.3.19 INDICATING DEVICES. Install meters and gauges on energy sources to each building and separately submeter energy used for space heating, hot water heating, space cooling, lighting and special power. All new buildings shall incorporate indicating devices such as integrated Btu meters, kilowatt-hours meters, flow meters, temperature gauges, pressure gauges, velometers, fuel meters for boilers, smoke density recorder, orsat, CO_2 and O_2 analyzers; gas meters in each system to monitor and evaluate energy performance.

3.3.20 HIGH Δt. High temperature rise or drop is desirable to minimize the flow rate and to conserve fluid transport energy. The following should be used as a minimum: supply air summer =20° Δt, winter =40° Δt, chilled water =20° Δt, and hot water =40° t.

$$\text{Frictional Head} \propto (\text{Flow})^2$$

$$\text{Horsepower} \propto (\text{Flow})^3$$

Theoretically, for 25 percent reduction in flow, the frictional head is 56.3 percent and horsepower is 42.2 percent of its original value. Similarly, for 50 percent reduction in flow, the frictional head is 25 percent and horsepower is 12.5 percent of its original value. This means 25 percent and 50 percent reduction in the flow rate saves 57.8 percent and 87.5 percent of energy respectively.

3.3.21 ENERGY UTILIZATION. Specifications should require that efficiency of all major equipment such as boilers, chillers, central air conditioning equipment, fans, water pumps, heat pumps, etc. be certified by manufacturers. Efficiencies should be based on full load, 75 percent and 50 percent operation. The criteria for selection and sizing of equipment shall meet the requirement of ASHRAE Standard 90-P. Certification from nationally recognized programs such as those of AGA, IBR, ABMA, AMCA, ASME, ANSI, ASHRAE, ARI, etc. is required.

3.3.22 CREDIT. A credit for internal heat of light and people shall be taken when sizing the heating equipment for the occupied period.

3.3.23 FOIL LOCATION. To provide the greatest resistance to heat flow downward, use aluminum foil backed insulation, or aluminum foil backed gypsum board facing upward or outward and facing a closed air space.

3.3.24 EVAPORATIVE COOLERS. Use single stage evaporative coolers as precooler for outside air make-up in air conditioning systems in arid zones.

3.3.25 CONDENSERS. Use air cooled condensers in series with cooling towers to minimize equipment sizes and reduce electrical consumption. Use a small cooling tower in series with a large air cooled condenser for peak shaving particularly in arid zones.

3.3.26 REFRIGERATION MACHINES. Consider use of the double effect absorption refrigeration machines instead of single effect. This reduces the steam requirements from 18 lb./hr./ton to about 12 lb./hr./ton.

3.3.26.1 Note: Double effect machine is a proprietary item by the Trane Company, LaCrosse, Wisconsin.

3.3.27 LOW RESISTANCE FILTERS. Use low resistance filters, registers and grilles.

3.3.28 EXHAUST HOODS. Where possible the air supply on exhaust hoods shall be direct to preclude heating and cooling large quantities of fresh make-up air, then exhausting it outside.

3.3.29 VEHICLE TRAFFIC. Revise in-plant vehicle traffic to cut down on door openings.

3.3.30 HEATERS. Locate all heaters clear of obstructions.

3.3.31 STEAM CONDENSATE. Return steam condensate to conserve both energy and water.

3.3.32 VENTILATION IN ATTIC. Provide ventilation in attic spaces between ceiling and roof for family quarters and other similar buildings with attic space.

3.3.33 FUEL SOURCES. Log all fuel sources, their consumption at least on a monthly basis for each department.

3.3.34 AUTOMATION SYSTEM. A centralized automation system shall be considered for HVAC, load shedding, control of electric equipment, etc., where economically justifiable.

3.3.35 CONTROL DEVICES. The use of control devices such as staging controls, industrial grade controllers, static pressure sensors, time based programmers, etc. shall also be considered where applicable.

3.3.36 CONTROLS. Controls system shall be energy efficient with each zone provided with adjustable automatic control device to maintain the design set point. The set point shall be adjustable within 10° deadband to avoid usage of heating or cooling equipment. A 10°F. to 15°F. night setback for winter unoccupied periods shall be employed, provided additional energy is not required to attain the setback temperature.

3.3.37 REHEAT. Recovered energy is desirable for reheat for control of temperature. The reheat use of energy sources other than recovered shall be minimized by limiting the air temperature rise across the reheat coil to about 7-1/2°F. Proper zoning selection must be considered.

3.3.38 EXHAUST AIR. Minimize exhaust air quantities and provide variable speed or inlet vane control on exhaust fans. This is done to minimize exhaust air since all areas such as toilets need no exhaust unless they are occupied. Exhaust fans should be controlled with static pressure controller. Install individual switch for toilet exhaust instead of tying it with lights.

3.3.39 INTEGRATED AIR CONDITIONING AND LIGHTING SYSTEM. With the exception of clean rooms, animal laboratories, and laboratories with toxic, explosive or bacteriological exhaust requirements, all air conditioned spaces where the general lighting level is 2.5 watts per square foot or greater shall have an integrated air conditioning-lighting system. Use heat of light return to reduce air requirements for the space. See paragraph 3.2.3.

3.3.40 LUMINAIRES. Additionally, the use of air cooled luminaires shall be considered. See paragraph 3.2.3.(1).

CHAPTER 4. ENERGY SOURCES, SOLID WASTE HEAT RECOVERY AND TOTAL ENERGY SYSTEMS

4.1 FUELS. The conventional energy sources for building energy requirements are the fossil fuels: coal, fuel oil and fuel gas. Since the end of World War II, coal has declined steadily as a heating fuel leaving this market almost entirely to oil and gas. By 1985, unless the oil situation worsens or gas continues in short supply, coal should be completely out of the heating picture for heating plants serving individual buildings. However, increased use of coal in large central boiler plants is expected.

4.1.1 Coal. Coal lends itself best to a large central heating plant or to a remotely located total energy type of installation. This is because of the large building volumes and adjacent land areas needed for the storage, processing and handling of coal and to house the larger boilers required for its burning. Coal handling operations require conveyors, pulverizers, and additional fans, all of which are noisy and dusty even though some of the operations can be partially enclosed. Also, equipment is required for ash quenching and removal, which consumes large quantities of water, and additional space is required for ash storage as well as remote sites for final ash dumping.

4.1.2 Oil. Fuel oil is more convenient to handle than coal. Its storage is simpler and there is no ash disposal problem. The lighter No. 2 fuel oil should be considered for small steam and hot water heating plants and for diesel engine and gas turbine total energy plants. For steam plants having water tube boilers, No. 6 (Bunker C) fuel oil should be considered based on the economics of providing fuel oil storage tank heaters, pump and heater sets and traced and insulated fuel oil lines required for No. 6 oil.

4.1.3 Natural Gas. Natural gas is by far the least cumbersome of the fossil fuels. Storage is the problem of the supplier, not the user. Handling is clean and simple. Combustion safeguards are more numerous than in a coal or oil installation, but they have been standardized and are extremely reliable. One of the limitations of natural gas has been its distribution.

The development of various purification and drying processes for natural gas (which is largely methane) has led to its liquification, at low temperatures (-259°F). A mass of gas equivalent to 600 cu. ft. at low pressure can be liquified to 1 cu. ft. of liquid (LNG), and this reduced volume can be shipped by tank trucks to any point within reach of a highway. Natural gases rich in propane and butane, are usually separated and the liquified propane and butane (LPG) can also be shipped by tank trucks. Some LPG and LNG is being imported into the United States via specially modified and refrigerated tanker ships. In accordance with current DoD policy, the use of natural gas is prohibited in boilers of greater than 20,000 lbs/hr capacity.

4.1.4 Other Fuels. The feasibility of burning fuels other than fossil fuels should be considered when available, and when proven to be economical. Examples of other fuels that should be considered are wood, paper, sawdust and municipal waste. In a few areas in the world, peat and lignite, which are the early stages in the formation of coal are readily available, but, as can be seen from Table 4-1 their heat (BTU) content is low relative to coal. Wood, of course, had been used as a fuel for centuries before the advent of coal and it is still used in many industries where bark, scrap wood, sawdust, etc. are waste or by-products. Paper, also, is available as waste in many offices, merchandizing and manufacturing operations. As a matter of fact, in many governmental installations tons of paper containing classified information must be incinerated as a routine. Plastics, too, are available as a waste and many can be incinerated. Care must be taken with plastics, however, as many do give off toxic gases. See paragraph 4.2 for a discussion of municipal waste as a fuel.

4.1.5 Pollution. All combustible fuels are faced with the problem of getting rid of exhaust gases, including products of combustion, some of which may be toxic. Anti-pollution regulations in most states and large cities prohibit the release of such gases without prior treatment to render them harmless. In the case of the fossil fuels, sulphur is the prime polluter causing the release of sulphur dioxide, a particularly noxious pollutant. The corrective measures are to burn fossil fuels with low sulphur contents, or to install sulphur dioxide removal equipment to treat the exhaust gases. Low-sulphur fossil fuel, which is not in plentiful supply, ranges in price from 50% to 100% higher than ordinary fossil fuels which may range from 3.5% sulphur in coal to as high as 18% in some natural gases. Sulphur removal equipment in a power plant ranges in cost from 25% to 35% of the total installed plant cost.

4.1.6 Fuel Selection. The selection of the specific fuel, or fuels, for a specific energy plant is based on economics, availability of fuel, availability of storage space, anti-pollution regulations and safety requirements. The economic factors have been tabulated in Table 4-1 for the Philadelphia area. These may vary for other parts of the country.

4.2 SOLID WASTE HEAT RECOVERY. Municipal refuse and industrial waste by-products also are valuable energy sources. In special industries there are many steam boilers fired by solid waste fuels such as bagasse (squeezed sugar cane), coffee hulls, nut shells, wood bark, shavings, and sawdust. Municipal incinerators, which dispose of refuse (solid wastes) while generating steam, are commonplace in Europe and Japan, and are now being installed in Canada and the United States. Two units in the Norfolk Naval Shipyard, Virginia, have been operating successfully since 1967.

4.2.1 Waterwall Incinerator-Type Waste Heat Recovery Boilers. They differ very little from stoker-type coal-fired boilers. The furnace volume generally is 15-20% larger for the incinerator boiler due to the greater liberation of gases and to allow for complete combustion of the lighter solids which may be lifted off the grate. Flame temperatures of 2500°F are usual with only 30% excess air. A special type of air-cooled reciprocating grate stoker is required with a steep angle downward in the direction of the feed. The waterwalls and drums of the A-type arrangement are best suited for this type of boiler. Auxiliary fuel oil or gas is necessary to stabilize the flame at low (35% and below) outputs. Except for bulky waste the solid wastes are fed directly into the furnace without processing.

TABLE 4-1

Comparative Energy Costs of Fuel[1] (October 1974)

Fuel	Heat Content, BTU /Lb.	Heat Content, BTU /Gal.	Heat Content, BTU /Cu.Ft.	Cost $/Unit	Comb. Eff.	Rel.Cost $/MM BTU
<u>Coal</u>						
Anthracite	13,300					
Semi-anthracite	13,100					
Semi-bituminous	14,100					
Bituminous	13,900			$35[2]	80%	1.57
Sub-Bituminous	9,400					
Lignite	7,100					
Peat	3,600					
<u>Oil</u>						
Crude						
No. 6 (Low Sulfur)		148,000		0.3517	80%	2.97
No. 2		138,500		0.349	80%	3.15
JP Fuel						
<u>Gas</u>						
Natural			1,080	$2.66/ 1000cf	78%	3.16
LNG						
LPG		91,500				
<u>Waste</u>						
Wood	5,800			$4/tn[4]	60%	0.69
Paper	5,500			$4/tn	60%	0.73
Plastics						
Municipal Waste (IIA[3] Type 1-M)	5,000			$5/tn	60%	1.00

1. For the Philadelphia Area.
2. The cost is per ton of coal.
3. Incinerator Institute of America.
4. The cost is for collection and preparation (shredding).

4.2.2 Refractory-Lined Incinerator With Waste-Heat Boiler. The refractory-lined incinerator furnace requires about 300-500% excess air to help keep the furnace walls cool. Flame temperatures of 1500-1600°F are usual. Small incinerators, up to a capacity of 150 tons of refuse per 24 hours, are refractory-lined circular and generally equipped with stationary grates and rotating stoking arms. Larger refractory incinerators (150 tons per 24 hours and above) are generally rectangular in shape and equipped with traveling, reciprocating or rocking grates. Refractory furnaces are susceptible to clinkering and refractory damage as well as excessive fireside tube corrosion and ash deposits on the boiler tubes. It is desirable to make the material homogeneous by shredding to improve combustion.

4.2.3 Solid Waste Boilers With Supplementary Fuel Firing.

(1) Processing. Solid waste boilers are generally designed as multi-fueled boilers capable of burning a wide variety of industrial and municipal wastes. Most of these solid wastes are a heterogeneous mixture of combustibles (paper, wood, plastics) and non-combustibles (tin cans, bottles, metal scrap such as loose-leaf binder rings, spiral springs, paper clips, pins, zippers, buttons, etc.). Separation of the non-combustibles is followed by shredding (cutting) and/or hogging (tearing apart) of the remaining combustibles. Shredding, followed by air classification into a light fraction (fuel) and a heavy fraction (reclaimable metals, etc.), is the usual preparation process prior to incineration.

(2) Existing Boilers. Municipal incinerator/steam plants are commonplace in Europe and Japan. In Germany, especially, where each municipality is responsible for supplying district heating steam as well as electricity, municipal waste is a highly-prized fuel. Incinerator/steam plants are making their appearance in the United States. A municipal unit has been in operation at Oceanside (Long Island), New York since the early 1960's. Two units have been operating in the Norfolk Naval Shipyard, Virginia, since 1967, and two more are presently under construction there.

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4.3 WASTE HEAT RECOVERY FROM BOILERS, ENGINES AND TURBINES. The heart of a steam generating plant is the steam boiler. Figure 4-1 shows a total energy plant of the steam boiler type but if we neglect the main turbine-generator and the auxiliary turbines, we have an ordinary steam generating plant. In this type of plant the only recoverable heat is in the stack gases, and most of this is used to pre-heat the combustion air and feedwater for the boiler. The steam is delivered for end use through a pressure reducing station rather than through a turbine. Auxiliaries may be electric motor-driven, turbine-driven, or both, in which case it may be possible, with proper controls and disconnects to keep a boiler in operation during an electric power outage.

4.3.1 Limitation. There are limitations in recovering heat from boiler stack gases. First, it is important that the stack gases be kept above the dew point (approximately 250°F) to prevent condensation in the stack. Any water contacting the stack gases forms a mild sulphurous or sulphuric acid which is highly corrosive. Often, it is necessary to use expensive, corrosion-resistant materials in the stack gas heat exchangers. Second, stack gases cannot be used to pre-heat fuels. This is a hazardous operation and it is prohibited by all safety codes relating to boilers.

4.3.2 Steam Turbines. They provide five sources of recoverable heat: the inlet steam, interstage bleed steam, exhaust steam, gland seal condenser water and bearing cooling water. Inlet steam is usually used to drive auxiliaries and for fuel oil atomization in the boiler. Interstage bleed and exhaust steam is used for No. 6 fuel oil pre-heating in the storage tank, pump and heater set, and in fuel oil line tracing. It can also be used in steam coils for combustion air pre-heating to the boiler, and in steam radiators for space heating. Steam gland seal condenser cooling water and bearing cooling water can be used in hot water radiators for space heating and in heat exchangers to provide hot water service for building use.

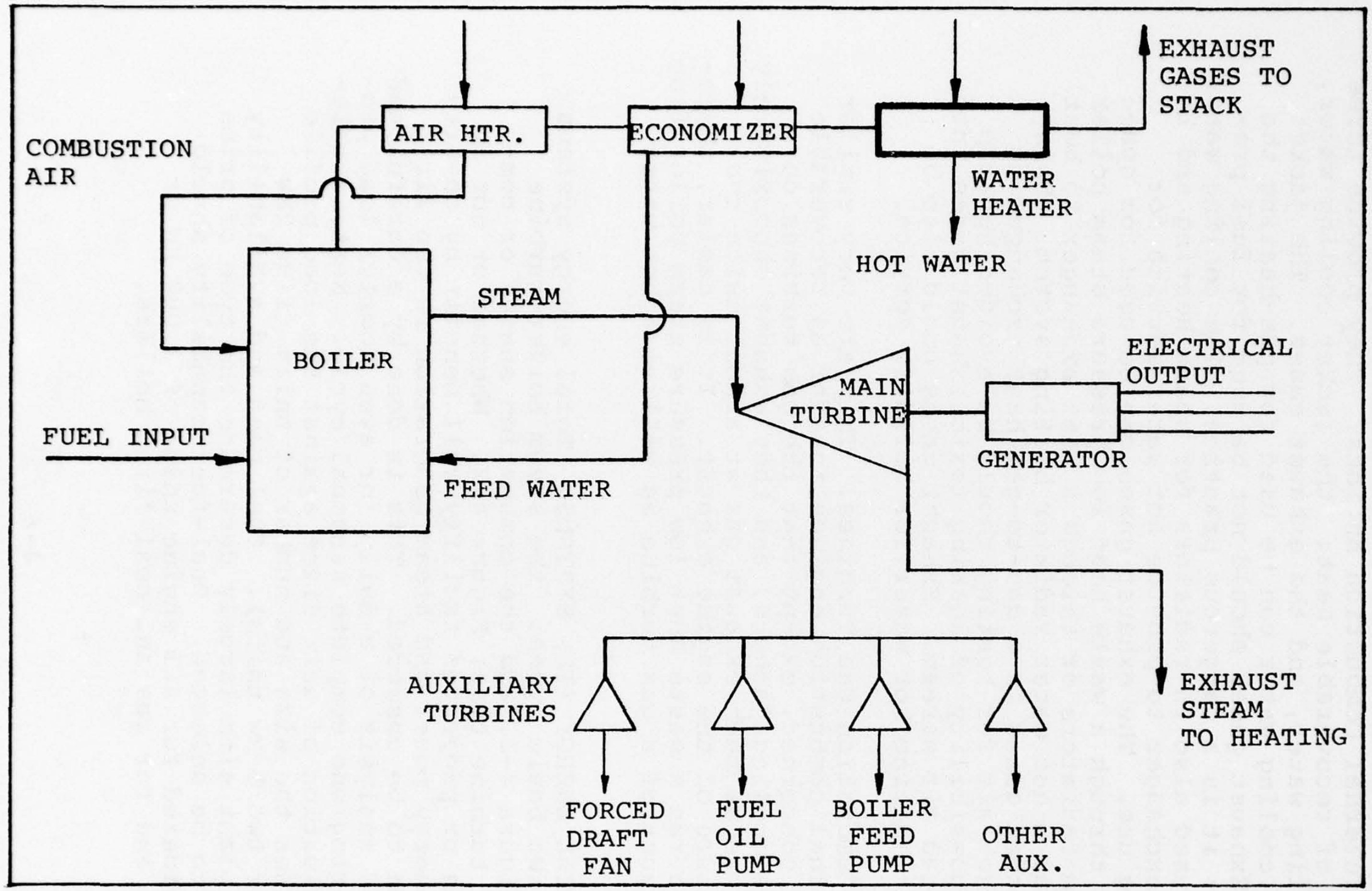

FIGURE 4-1

TOTAL ENERGY SYSTEM USING STEAM BOILER

4.3.3 Internal Combustion Engines. They provide three sources of recoverable heat: the jacket cooling water, oil cooling water, and the exhaust gases. The jacket and oil cooling water can be used for pre-heating the fuel. Exhaust gases should not be used for fuel pre-heating; it is a dangerous practice. The cooling water can be used also in radiators for space heating and in a heat exchanger to provide hot water service for building use. The exhaust gases can be used for space heating through a waste heat low pressure steam boiler to steam radiators or through a heat exchanger to heat water for a hot water radiator heating system. Using the exhaust gases in a gas-to-gas heat exchanger to supply hot air for heating should be avoided because of the possibility of leaking toxic exhaust gases into the heated air stream. Exhaust gases could also be used for heating hot water for building service.

4.3.4 Combustion Gas Turbines. They are very similar to internal combustion engines insofar as recoverable heat is concerned, except that the gas turbines do not have cooling jackets, and they exhaust approximately four times as much exhaust gas at almost twice the temperature of the engine exhaust. It is easier, therefore, to run a waste heat low pressure steam boiler from the exhaust of a gas turbine as well as a hot water heater.

4.4 TOTAL ENERGY (TE) SYSTEMS. Total energy systems are of two basic types: the steam boiler-turbine type, Figure 4-1, and the combustion engine or combustion turbine type, Figure 4-2. Whether or not an existing or projected facility will benefit by on-site total energy power and steam generation is the first question to be answered. This is done by a careful and detailed analysis of a daily (or even hourly) load profile during one complete seasonal cycle. Next, a careful evaluation of unit sizes against the load profile determines the size and number of units (five 2Mw units or two 5 Mw units). Fuel cost and availability at the plant site largely determine the type of prime movers to be selected. Dual-fuel capability should be considered for all engine units of 1,000 KW or larger, and for gas and coal-fired boilers.

4.4.1 Areas for TE Consideration. Total energy systems should be considered and evaluated for all larger buildings or complex of buildings which require both heating and air conditioning. Special consideration should be given to essential facilities such as hospitals, communications centers, and other facilities, which must continue to operate during electric power failures.

4.4.2 Steam Boiler-Turbine Type Total Energy Systems. They are generally used where large amounts of steam are needed for heating and/or process requirements. When the heating load is decreased, in the summer, the steam may be used in absorption machines for air conditioning. This sort of piggy-back system is described in Chapter 3-3. In this type of system the main generator is driven by a steam turbine and most, if not all, of the auxiliaries are driven by steam turbines. Most modern industrial and office facilities need precision-controlled frequency and voltage for computers, clocks, and process control. The steam turbine with its excellent regulation, load-following and control characteristics lends itself perfectly to this application. Steam is available at boiler pressure, interstage bleed pressure and turbine exhaust pressure for auxiliary drive and for heating purposes. Further heat recovery can be obtained from the boiler stack gases, turbine gland seal condenser water, and bearing cooling (lube oil cooler) in water.

4.4.3 Combustion Engine Type Total Energy Plant. This is more desirable where less of a steam load is required. The engines may be connected to serve as peaking power units or standby emergency units. Another possible arrangement is to connect the engines to a heavy constant load, such as air conditioning, and by quick disconnect couplings be able to switch over to emergency electric power generation in case of a power outage. Dual-fuel supply should be considered using Diesel oil (No. 2) and natural gas. Heat can be recovered from the engine jacket cooling water, bearing (lube oil) cooling water and exhaust gases.

4.4.4 Combustion Gas Turbine Plant. They operate with 200% to 400% excess air in order to keep the

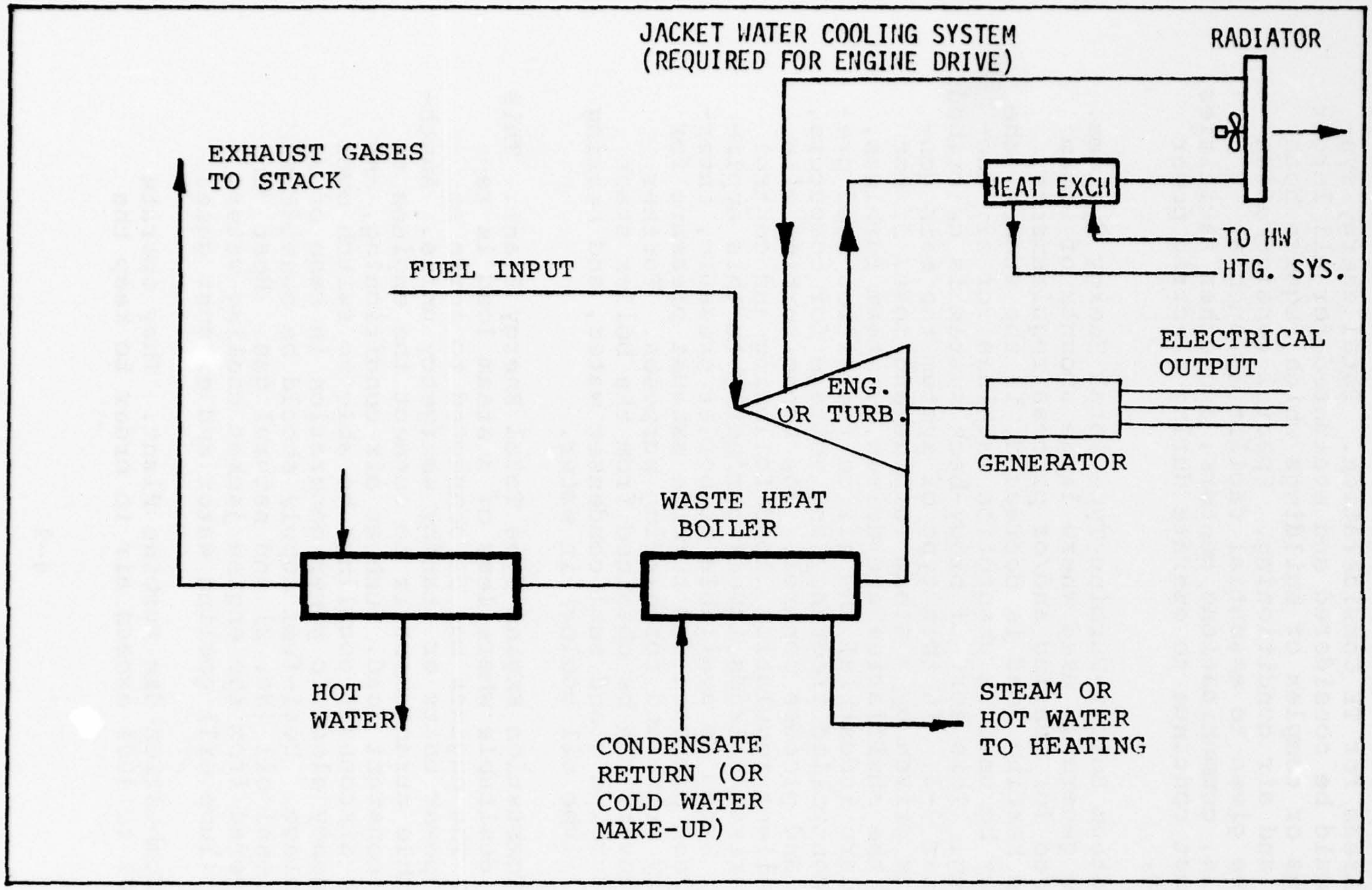

FIGURE 4-2

TOTAL ENERGY SYSTEM USING COMBUSTION ENGINE OR TURBINE mechanical parts of the turbine cool and to insure complete combustion within the turbine. Exhaust temperatures are on the order of 1,000°F. With such volumes of air at such high temperatures, a waste heat recovery boiler can be placed in the system to generate low pressure steam. The pressure and quantity of steam available is dependent upon the size of the gas turbine and the amount of heat available in the exhaust gases. Gas turbines may be arranged and operated in very much the same way combustion engines are arranged and operated. The major difference is that gas turbines are practically instant-starting with no warm-up period required. Heat is recoverable from the bearing (lube oil) cooling water and the exhaust gases. The gas turbine can be made to operate with practically any liquid or gaseous fossil fuel available, which gives it a multi-fuel capability.

CHAPTER 5. COMPUTER PROGRAMS

Section 1. COMPUTER PROGRAMS FOR ENERGY SYSTEM ANALYSIS

5.1.1 Commercially available computer programs which are suitable for performing building energy use analysis and HVAC system selection include the following:

(A) Energy System Analysis Series (ESA)

(A) AXCESS Energy Analysis Computer Program

(A) ECUBE - Energy Conservation Uitlizing Better Engineering

(B) MACE - McDonnell Annual Consumption of Energy

(B) TRACE - Trane Air Conditioning Economics

(B) Westinghouse Energy Study

The computer programs are grouped above in order of capability. The first three are recommended since they provide hour by hour energy use estimates for 8760 hours per year. A supplementary computer program, Heating Cooling Calculations (HCC-III), is also included in the program descriptions. It is a heating and cooling design load program and lends itself to energy analysis only by way of peak load definition which is required input for some of the programs. A description of the available programs is given in the following pages of the section.

5.1.2 THE ENERGY SYSTEM ANALYSIS SERIES. The Energy System Analysis Series is a library of computer programs developed by Ross F. Meriwether and Associates, Inc. for hour-by-hour calculation of the annual energy consumption of various types of air-side systems and mechanical plants, for applying local utility rate schedules to these demands and consumptions, and for combining these costs with other owning and operating costs for year-by-year cashflow projections. Each major step in a complete energy system analysis is handled by a different program, thereby permitting the engineer to evaluate the results of one part before finalizing inputs and proceeding with the next part. The Energy System Analysis Series is designed to calculate monthly and annual energy requirements and costs, not design point heating and cooling loads. These programs begin with design point loads for the overall building or major building sections and distribute them throughout a full year cycle of the building's operation. The six programs in the library are explained as follows: (Fig. 5-1).

5.1.2.1 Energy Requirements Input Data Check (ERCK). This is a preliminary, free-standing program which reads the input data for the various buildings or sections to be run in the building energy requirements estimate program (ERE) and expresses them on a unitized basis to permit a check of the magnitude of the potential loads and system capacities. The input to the ERCK program is the same as the data for the ERE program. The output of the ERCK program begins with a display of the input data. It affords an opportunity to check all the values that will be used in the computer and correct any errors that could cause the ERE program to abort. Percentage variation profiles and operating schedules are shown on an expanded hourly basis for checking and reference. The unitized input data follows and allows the user to compare his input loads to reference "rule-of-the-thumb". It also permits the user to compare his installed heating and cooling system capacities to the design point loads to verify that the equipment is sized adequately.

5.1.2.2 Energy Requirements Estimate (ERE). The ERE program uses the calculated design point heating and cooling loads as a reference and determines the hourly loads using actual values of dry bulb temperature, dew point temperature, cloud cover, solar radiation, and percentage profiles for various building operating schedules. Hourly weather data that is used in the energy requirements calculations is obtained from the National Climatic Center (see Section 5.1.2.8 for address) and consists of 8760 hourly values of dry bulb, dew point, and cloud cover for some typical year. Programs have been developed for selecting and preparing this weather data for use in the Energy System Analysis Series. Solar radiation tables as published in the

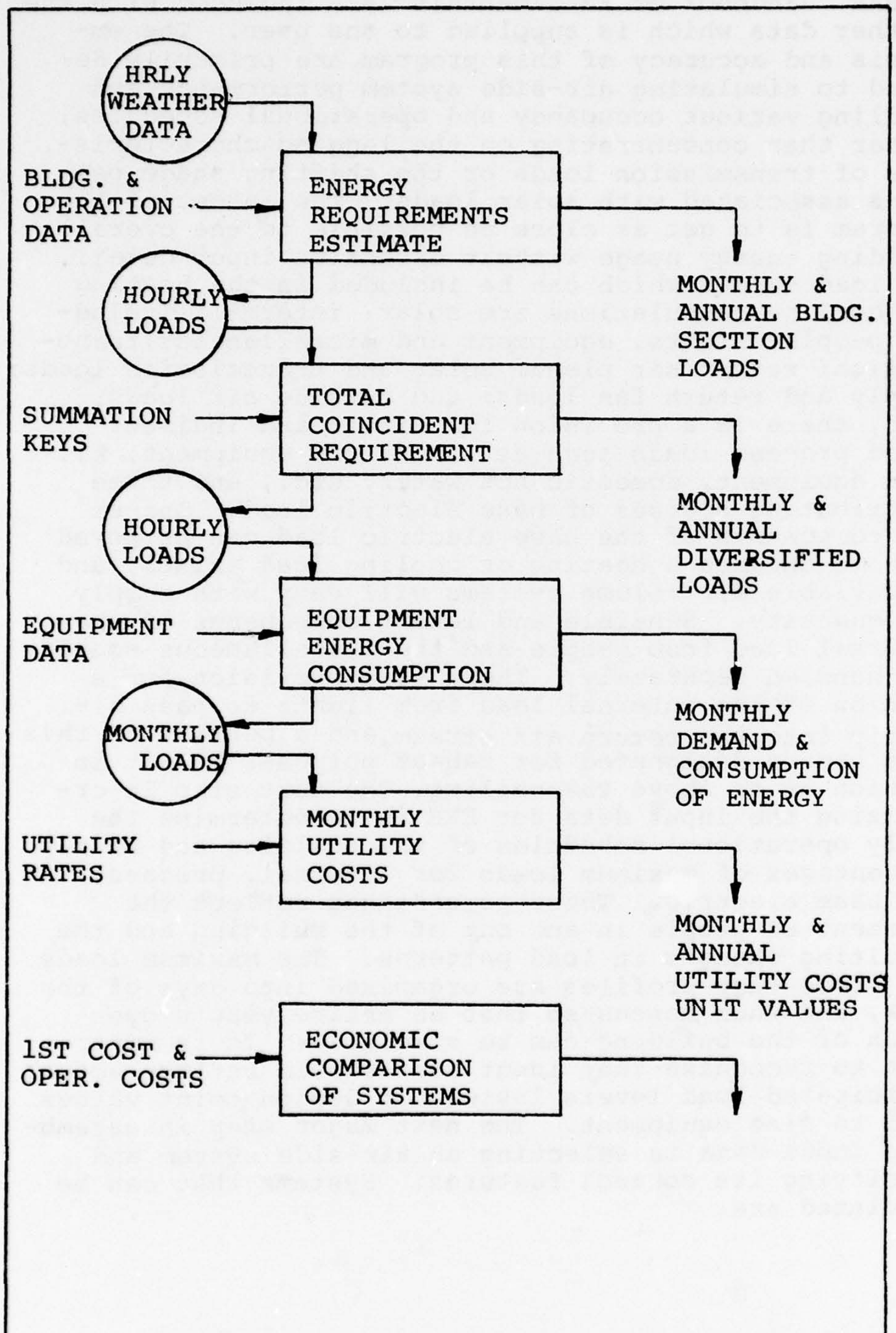

FIGURE 5-1

THE ENERGY SYSTEM ANALYSIS SERIES

ASHRAE "Handbook of Fundamentals" are included with the weather data which is supplied to the user. The emphasis and accuracy of this program are primarily devoted to simulating air-side system performance and handling various occupancy and operational schedules, rather than concentrating on the lagging characteristics of transmission loads or the shifting shade patterns associated with solar loads. The intent of the program is to get as close as possible to the overall building energy usage without excessive input detail. The load values which can be included in the heating and cooling calculations are solar; internal, including people, lights, equipment and miscellaneous;transmission; return air plenum solar and transmission loads; supply and return fan loads; and outside air loads. Also, there is a provision for direct and indirect fired process loads such as laboratory equipment, kitchen equipment, domestic hot water, etc., and three contributing sources of base electric load. One of the components of the base electric load can be keyed to come on when a heating or cooling load exists, and in variable air volume systems will vary with supply air quantity. Sensible and latent components of the internal load from people and the miscellaneous source are handled separately. There is a provision for a portion of the internal load from lights to pass directly into the return air stream,and a portion of this heat can be designated for reheat purposes in air induction units above the ceiling. The next step in organizing the input data for ERE is to determine the daily operational schedules of the building and hourly percentages of maximum loads for internal, process, and base electric. These percentages reflect the movement of people in and out of the building and the resulting changes in load patterns. The maximum loads and percentage profiles are organized into days of the week, and then months so that an entire year's operation of the building can be simulated. It is important to recognize that input data should reflect actual anticipated load levels instead of design point values used to size equipment. The next major step in assembling input data is selecting an air-side system and specifying its control features. Systems that can be simulated are:

(1) Single duct, constant volume, variable discharge temperatures set by demand with no excess cooling or reheating.

(2) Terminal reheat with scheduled cold coil discharge temperature during the cooling cycle.

(3) Terminal reheat with scheduled cold coil discharge temperature during cooling or heating cycles.

(4) Induction or fan-coil type system with scheduled primary air temperature.

(5) Terminal reheat with cold coil discharge temperature set by maximum demand of any section.

(6) Dual-duct with scheduled hot and cold deck temperatures.

(7) Dual-duct with deck temperatures set by greatest demand.

(8) Split conduit system with single duct, constant volume; variable temperature system to offset transmission loads and a variable volume system for solar and internal.

(9) Standard variable volume, scheduled temperature system.

Additional features that can be utilized in the ERE program are outside air economizer cycle, cold deck reset schedules according to ambient temperature or time clock, heat recovery devices operating between return and outside airstreams, supplemental perimeter heating systems that are independent of central system, holiday scheduling for accurate representation of operating schedules, distinction between on and off peak time periods for electrical service, capability to interrupt gas service and switch to auxiliary fuel, calculation of heat storage effects caused by shutoff and setback, and the ability to print selected days during the year to observe the hourly behavior of the system. The output printout includes monthly and annual peaks and consumptions for heating, cooling, process and base electric loads. The hours of heating and cooling system operation are shown for each month and annually.

Data is also given for minimum and maximum room temperature that occurred during the year, number of hours capacities were exceeded and unitized peak and consumption values for heating cooling, process and base electric.

5.1.2.3 Total Coincident Requirements Program (TCR). The input consists of multiple ERE hourly load output tapes plus the data called for on the input forms. A multiplier can be used with each building or section if ERE runs have been made on a unit basis, such as in an apartment complex. The output of the TCR program shows the diversified peaks and time of occurance, as well as the sum of the individual building peaks. The monthly and annual consumptions are shown for the combined plant loads, and unitized values are printed for peaks and consumptions. If desired, a printout is given for the number of hours at ten percent increments of assumed peak loads to aid in equipment sizing. Coincidental peak steam and electrical loads can also be obtained, if desired.

5.1.2.4 Equipment Energy Consumption Program (EEC/B). The input consists of an hourly load tape from ERE or TCR plus the data entered on the input forms. The rated capacities and part load performance are required for each piece of equipment, and this equipment is then organized into the various systems and the appropriate accessory equipment loads. Typical systems that can be simulated are total energy with heat recovery, gas heating and cooling, gas heating and electric cooling, all electric, and purchased chilled water and steam or hot water. Energy sources may be gas, electricity, or any other fuel specified by the user. Chillers in each system can be any combination of direct or indirect-fired absorption machines, steam-turbine driven machines, gas engine chillers, or electric motor chillers. Boilers can be gas or electric or any special fuel. Generator sets can be driven by reciprocating gas engines, gas turbines, or steam turbines. Heat recovery can be utilized from prime movers or from chiller condenser water. Features of this program include: different methods of scheduling machines on the line; various sequencing schedules for accessories; separation of on-peak electrical usage for special rates; flexibility in handling recoverable heat and heating requirements in each system; multiple ERE or TCR hourly load tapes representing different sections in a building or different buildings in a complex, premitting separate systems in a single complex to be grouped on a single meter; provision for a multiplier to be used with any system; capability to adjust chiller and generator performance characteristics and their associated accessories for ambient temperature effects; simulation of cooling tower performance with constant or variable condenser water temperature; provision to limit chiller capacity in selected months; provision to limit electrical demand in selected months; and improve methods of handling accessory equipment scheduling. The output information includes monthly and annual peaks and consumptions for energy input to the equipment in each system. Unitized values are also shown for comparison and checking. The number of operating hours is shown for each machine monthly and annually. Other useful information shown in the printout includes utilization of recoverable heat, loads in excess of equipment capacity, and a summary of the energy usage by source.

5.1.2.5 Monthly Utility Costs (MUC). Utility demand and consumption values from a tape generated in the EEC program are input to the MUC along with input cards containing the specific utility rate steps. Alternatively, the demands and consumptions can be entered on cards and the program run independently from any previous program. The MUC program is capable of calculating demand and consumption charges for gas service, electric service, chilled water, steam or hot water, and any special auxiliary fuel. The output shows monthly and annual costs for each energy form in each system as well as average costs per unit of energy and per square foot. The total yearly energy costs (all forms) are shown for each system.

5.1.2.6 Economic Comparison of Systems (ECS/B). This program uses the energy costs determined in MUC plus other annual operating costs, such as maintenance and operating labor, and combines these costs with initial investment and associated owning cost factors, such as depreciation, to find the annual cash flow each year for the life of the system. Annual utility costs and other operating costs can be independently escalated each year by a percentage supplied by the user. The initial investment may be divided into two segments with different depreciation periods, and a provision exists for four additional reinvestments (for equipment replacement or staged projects) which can be on a recurring basis. In addition to straight cash flow and a discounted cash flow for each system on an independent basis, a comparison can be made of each system to the lowest first cost system to show the net savings of reduced operating costs compared to higher owning costs. The output for each system shows the year-by-year and total period values of debt balance, debt interest, equity balance, equity interest, total owning cost, total operating cost, total owning plus operating cost, and equivalent current dollars. If a comparative analysis is performed, the printout shows annual and total values of owning and operating costs savings, total owning and operating cost savings before principal payment difference, depreciation difference, etc., total owning and operating cost savings. The discounted rate of return on total gross investment difference and the number of years to recover initial gross investment difference are also shown.

5.1.2.7 Availability. For additional information and program availability contact:

Ross F. Meriwetner and Assoc., Inc.
1600 N.E. Loop 410, Suite 241
San Antonio, Texas 78209

5.1.2.8 Weather Data. Hourly weather may be obtained from the National Climatic Center at the following address:

NATIONAL CLIMATIC CENTER OF
NATIONAL OCEANIC and ATMOSPHERIC
ADMINISTRATION
U.S. Department of Commerce
Environmental Data Center
Federal Building
Ashville, N.C. 28801

5.1.3 AXCESS ENERGY ANALYSIS COMPUTER PROGRAM. The AXCESS Energy Analysis Computer Program developed by the Electric Energy Association (EEA), is a means for estimating the difference in energy consumption and demand for up to six different mechanical/electrical designs for a planned building, or complex of buildings. Effectively a comprehensive feasibility study method, AXCESS is divided into 4 sections. The first three sections cover the engineering phases, each of which provides monthly and annual monetary values:

(1) ENERGY ANALYSIS - The cost of supplying all of the energy needs of a building in which alternate mechanical/electrical systems are to be considered.

(2) FIRST COST DIFFERENTIALS - The first cost differentials between such alternate systems.

(3) MAINTENANCE & OPERATING PERSONNEL COSTS - Differentials in cost for operating personnel, maintenance and unscheduled repairs.

(4) FINANCIAL ANALYSIS - The engineering-derived dollar values are utilized in methods of evaluation meaningful to the investor, i.e., rate of return on the investment, yield, discounted cash flow or net present value. These techniques are covered in a separate training manual developed by Price Waterhouse & Co. In addition, a computer program to perform the analyses is available from EEA.

5.1.3.1 Program Method. Using the input design loads, the program calculates hourly zone solar and transmission loads for the year. If desired, these may be input from another program. The program calculates the base energy loads each hour as the profile percentage multiplied by the peak usage, taking proper credit for any waste heat application. From these and zone data, it then calculates the zone internal gains and the net zone space conditioning requirement. Terminal system operation is simulated in one of ten subroutines using input set-points or "standard"designs. Based on the input performance data, and the calculated loads, each primary system is then simulated in one of three general subroutines. All calculated energy usages are summed hourly into "meters". These meters function exactly as electrical usage and demand meters but may be used with any fuel. A unique feature of the program is its ability to receive as input a variety of thermal load types. The user may input hourly, zone-by-zone loads, as derived from other programs. If hourly and zonal loads are not available, the user may input building total loads with or without a breakdown between glass, wall and roof transmission values and solar loads, either as a design total or on an exposure-by-exposure basis. This is one of the program features which make it extremely practicable at various stages of building systems design. Most of the HVAC systems in popular use today are simulated in the computer. These systems range from unitary equipment to the most complex systems (at least from the standpoint of estimating energy consumption), such as heat recovery cycles and ceiling induction units utilizing lighting cavity heat. HVAC simulations are broken down by terminal and primary system types. Not only does this allow extreme flexibility in formulating an energy analysis study but it also provides ideal latitude in keeping the program always up-to-date. When any new terminal or primary system appears in the future, a new subroutine may be written and added in modular form to the existing computer program. On-site systems have been simulated on a modular basis so that any combination of HVAC terminals and primaries may be included with them in a particular scheme. Both thermal balance and electrical balance isolated generation systems may be studied. Up to six separate mechanical/electrical systems may be analyzed on a single computer run. In addition, a unique metering stage allows the user to benefit from supplementary information supplied from within each of the six schemes. The program uses standard engineering principles but bases all its calculations on input; so that as the quality of input increases, so will the quality of the output. It is well, however, to remember that the program's purpose is to provide a comparison of alternate designs, and the results for each scheme have more validity when viewed in relation to each other. The program's basic calculations and reset guidelines conform to standard engineering practice and are consistent with publications of ASHRAE.

5.1.3.2 Program Input. The program input is of two types - that provided by the user for a particular job, and that provided from other sources, usually standard, such as hourly weather data. The non-user input consists of the weather data and control information and is covered in the program description manual. The weather data is obtained from the National Climatic Center (see Section 5.1.2.8 for address) in a format for use with AXCESS. The user merely selects a weather station and year although it is possible to select a composite weather year if that is considered desirable. Weather data used in the program are hourly readings of dry-bulb temperature, percent relative humidity and cloud cover factor. The user input is divided into four groups - physical dimensions and description, HVAC information, other energy uses, and fuel/energy sources. These are further subdivided into nine sections on the field note input forms. The nine subsections of input are as follows:

(1) Section I is project description data. This includes general information for identification purposes as well as basic building operation and dimensions.

(2) Section II covers the base load items of usage. Each of the basic non-HVAC loads may be described via an installed KW or peak BTU and a "profile" describing the percent usage each hour. The input reference section of this manual contains typical profiles for some loads. The magnitude of the interior lighting load may vary among schemes and up to six different interior lighting loads may be described. Only one of each of the other pre-titled base loads is allowed, but up to thirty "other items of usage" may be input. There is also a total limit of thirty base load items of usage. Any base load may be referenced to any profile and there is a limit of thirty profiles. The profiles may be changed during a "special period" of building operation (such as summer session for a school). There is no limit on the number of special periods. Any of these loads may be partially or totally met by waste heat from an HVAC system or from another base load. (For example, domestic hot water preheated by rejected heat from a heat pump.)

(3) Section III is the input of design heating and

cooling load data sufficient for the program to calculate hourly skin loads. Most of the information in this section is optional. The "building construction" data is not used to calculate thermal loads but rather to calculate the time lag of these loads.

(4) Section IV is input for space type data. A space type is an optional collection of zones of similar usage. The space type input may be used in lieu of zone input for items common to each zone. Any differences may be input later in zone data, which overrides space data. For example, five office zones may differ only in exposure and in that one has an additional internal gain. Only that one zone would need a separate input form, while the rest of the information for all five zones could be incorporated into one space type.

(5) Section V is zone data. Every thermal energy-using area of the building must be either within a zone or broken up into zones. On page one of this section, the information correlating zones, exposures and space types must be entered, but the zone input forms may be omitted for any zone whose data may be approximated by the program from the space type or building input.

(6) The first HVAC systems data is entered in Section VI of the input. The first entries of that portion of the input which may vary in alternate schemes and must be repeated for each scheme is also made in this section. It serves to identify those terminal (distribution) system loads which are met by a given primary energy conversion system - such as the cooling coil load of a terminal system being served by a chiller. This input also serves to identify which terminal systems serve which zones, which other primary systems use energy from this one, or vice-versa, and whether waste heat is available from this primary system. Most of the information in this section is mandatory.

(7) Section VII includes a description of each terminal system. Almost all of this section is optional. If the type of system is the only input given, the program will assume certain standard design parameters such as a 55°F. cold deck set-point.

(8) Section VIII includes a description of each primary system with its full load performance, mode of operation, part-load efficiencies, and auxiliaries.

(9) Section IX serves to design the output format. Here all the energy-using devices under consideration are grouped into "meters" and "submeters", named, and assigned to energy sources.

5.1.3.3 Program Output. Basic program output consists of: (1) a complete printout of input data for verification purposes; and (2) monthly and annual statements of energy usage and demand by energy source type.

5.1.3.3.1 A sample calculation in which the calculation is printed out for a selected day, hour, zone and scheme. By comparing the procedure and results to this longhand calculation, the user may thus verify, by spot-checking, the accuracy of the program.

5.1.3.3.2 Breakdowns of energy usage by load types as dictated by the user's selection of simulated meters and submeters.

5.1.3.3.3 Monthly total heat rejected from air-cooled or water-cooled primary refrigeration systems.

5.1.3.3.4 Hourly and/or monthly deficit or excess kwh for thermal balance isolated generation systems.

5.1.3.3.5 Hourly or daily energy usage for each meter.

5.1.3.4 Program Availability. The AXCESS Energy Analysis program is available for use by NAVFAC Engineering field divisions through the computer located at the Facility Systems Office, Port Hueneme, California. The AXCESS Energy Analysis program was released to investor-owned electric utilities through a nationwide series of user training programs beginning in November 1972. Each of these companies has been furnished with a program source deck, user's manual and all other materials necessary for implementation on its own computer. AXCESS is also available to these utilities through a remote service bureau. Inquiries should be addressed to:

Electric Energy Association
Ninety Park Avenue
New York, New York 10016
Phone: (212) 986-4154

5.1.4 ECUBE - ENERGY CONSERVATION USING BETTER ENGINEERING. ECUBE (Energy Conservation Using Better Engineering) is an integrated series of energy analysis computer programs available from the American Gas Association. It evaluates energy systems to determine which equipment most economically satisfies the energy requirements of a particular structure. The program weighs the effects of weather, orientation, usage profiles, lights, motors and varying ventilation requirements on the system and generates reports which indicate the amounts of energy the site will consume with alternative types of equipment. It also provides data on economic comparison, based on factors such as equipment characteristics, local weather profiles, interest charges, and fuel and electrical costs. The ECUBE program series was initially developed to satisfy three requirements: (1) to integrate design point calculations of peak thermal and electrical load for making a realistic estimate of the hourly, monthly and annual energy requirements of a building; (2) to determine the energy consumption of various types of systems; and (3) to compare the total owning and operating costs of the various systems being considered. The programs analyze energy systems, including diesel, gas reciprocating engines, gas turbines, all-electric configurations and conventional configurations such as chillers and boilers. In addition, the programs weigh the effects of differing zones that require varying physical conditions and schedules. The program series consists of three basic computer programs, each designed to do a particular segment of an analysis, starting with design data and culminating in an economic comparison.

5.1.4.1 Energy Requirements Program. The ECUBE series starts with an analysis of the energy requirements for a building. This is accomplished by simulating the building with a computerized model and using historical hourly weather data furnished by the user. This program takes design point values for seven components of thermal load and the base component electric load, and distributes them over each hour of the year in accordance with dry bulb and dew point temperature variations, solar and cloud variations, and building use and operation schedules for various types of operational days. The program will evaluate the effects of thermostat setback or periodic system shutdown, as well as the resulting thermal storage and lag effects. This feature permits the user to evaluate the change in energy requirements resulting from shutting the system off at night or on weekends, or of resetting the thermostat during periods of non-occupancy. The program also shows how often heating and cooling system limits are reached when operating with either of these features. Simultaneous heating and cooling characteristics of dual-duct or terminal reheat systems can be accommodated. The out-put of this portion consists of a tape with hourly values of thermal and electrical load; a print-out of peak heating, cooling, electrical and process loads each month; the time that they occurred; and the cumulative values of each load type. The complete Energy Requirements run is stored on magnetic tape for future use with the Equipment Selection and Energy Consumption Program.

5.1.4.2 Equipment Selection and Energy Consumption. This second program in the series is designed to simulate the operation of equipment as required to meet the hourly loads stored on magnetic tape developed by the Energy Requirements program. Up to four types of systems can be evaluated with each run. The operating characteristics of any type of generator set, chilller, boiler, or heater are input to the program. The print-out is a monthly summary of the gas, auxilliary fuel and electricity consumed; the peak electric demand; the number of operating hours of each generator and chiller; and an evaluation of thermal energy usage.

5.1.4.3 Economic Comparison Program. The Economic Comparison program helps the designer make the rational choices among various alternative solutions to the problem. Three categories of information are required; (1) project capital requirements; (2) project annual operating costs; and (3) details concerning the methods of financing and criteria for judging economic feasibility. Development of annual operating costs (maintenance, fuels, etc.), capital costs and financial criteria is the responsibility of the program user. The feasibility of alternative system designs is compared to a base or reference system. Many options are included in the programming to evaluate the financial particulars of each client so that the detailed methods of financing are properly weighed in the rate of return analysis. The Economic Comparison Program is not directly tied to the first and second parts, so it can also be used on other types of engineering projects which require concise financial evaluation.

5.1.4.4 Supplementary Programs. Three supplementary programs are also available for modeling solar conditions and weather conditions, and for the summation of loads when several building or parts of buildings are analyzed separately. The three programs are as follows:

5.1.4.4.1 Solar Distribution. This program accurately simulates the solar conditions that affect the structure being evaluated by utilizing the maximum solar value for the building and, through an allocation procedure, correctly distributes the solar loading by hour and direction.

5.1.4.4.2 Weather Check. The Weather Check program takes the weather data that the user develops or gets from the U. S. Weather Services and converts it in a format usable by the main portion of the program.

5.1.4.4.3 Summation of Energy Requirements. This program is used when loads from zones of one building or several buildings are to be input as one system into the Equipment Selection and Energy Consumption program. Sums of the heating, cooling, electrical and process heat loads are produced for all the buildings generated by the Energy Requirements program for each hour of the year.

5.1.4.5 Availability. Further information concerning the use and availability of the ECUBE program may be obtained from the American Gas Association. Inquiries should be directed as follows:

Mr. Kenneth Cuccinelli
Manager, Energy Systems and
Energy Systems Analysis
AMERICAN GAS ASSOCIATION
1515 Wilson Boulevard
Arlington, Virginia 22209

5.1.5 MACE - McDONNELL ANNUAL CONSUMPTION OF ENERGY. McDonnell Douglas Automation Company's Annual Consumption of Energy Program (MACE) is designed to estimate the total annual energy requirements of a building by making hour by hour loads analysis and system simulations. Since the majority of commercial buildings have internal activities and operations that are cyclic by week, time schedules controlling these activities and operations are designed to handle a weekly schedule. The user can specify the "hour on" and "hour off" for weekdays, Saturdays, and Sundays. The program allows the user to select any given month or months to be analyzed if he is interested only in the totals for a few selected months. The user can also select the analysis to be made for one, two, or four weeks (all days per month). If one week per month is selected, the analysis is made for the first week of the month and the same energy requirements are assumed for the remaining three weeks. If two weeks per month are selected, the first and third weeks are analyzed and the same energy requirements are assumed for the second and fourth weeks respectively. This feature allows the user to reduce calculation time with a possible loss in accuracy; but, still get meaningful monthly and yearly totals. There are basically three main sectins to the program each of which overlays the other.

(1) Input Section. This set of subprograms reads in the input model, does extensive error checking and converts the input model into an internal format designed for optimal execution of the loads and systems analysis. The internal input model can be checkpointed for future processing of the job.

(2) Loads Analysis and System Simulation Sections. This set of subprograms calculates the hour by hour loads and system simulation. The monthly energy usage and demand usage is saved for the economics analysis. The hourly space loads can be saved for future system analysis studies.

(3) Economics Section. This set of subprograms calculates the cost of energy using the energy rate schedules input by the user. Monthly and yearly energy usage totals and costs are printed.

The hourly load and system simulation procedures require hourly values of dry bulb temperature, wet bulb temperature, total cloud amount, cloud type, wind velocity, wind direction and barometric pressure. The basic source of weather data is from the National Climatic Center (see Section 5.1.2.8 for address). Weather data is available for major cities and many other localities for many past years. McDonnell Douglas Automation Company is building a library of weather data on a user requirement basis and currently has available ten year data for several cities. The selection of a particular year of data to be used for an energy analysis can be handled by consultations of the user with McDonnell Douglas Automation Company Engineers.

5.1.5.1 Program Input. Careful attention has been given to the input requirements of the MACE program. Insofar as possible, only data that is readily available to the user is required for the program. Most of the data may be taken directly from the building plans. The following data are required for input to the program:

(1) Identification of the job and other user comments.

(2) Specification of the weather data to be used.

(3) Designation of the months for which calculations are to be made and if one, two, or all weeks of these months will be computed.

(4) Specification of wall and roof type.

(5) Window description, including dimensions and type of glass; exterior shading features including setback, overhang and fins; and leakage factor of windows.

(6) Floor, partition and door dimensions, and "U" values.

(7) Specification of wall, roof, and window types, and number of windows that are to be assigned to each space.

(8) Master time schedules and loads that conform to these time schedules, such as: occupants, lights, appliances and miscellaneous energy usage.

(9) Designation of duplicate space numbers for spaces that are identical to those already described.

(10) Specification of system types and methods of control.

(11) Performance characteristics of components.

(12) Energy rate structures, including data on graduated rates and demand charges.

(13) Designation of printouts that are desired in addition to the monthly and annual energy consumption and costs.

5.1.5.2 Output. The type and frequency of output is controlled by the user at input time by the time scheduled print options. The following general categories of output are available to the user:

(1) Printout of the input model by day type categories which helps in checking the input for completeness and validity.

(2) Printout of hourly space and building loads for specified days.

(3) Printout of the hourly system simulation operating variable for specified days.

(4) Printout of the electrical usage and cost (including lights, miscellaneous, heating and cooling system and auxiliaries) on an hourly basis for specified days and monthly and yearly totals.

(5) Printout of fuel consumption and cost on an hourly basis for specified days and monthly and yearly totals.

5.1.5.3 Availability. Additional information can be obtained from:

McDonnell-Douglas Automation Company
Marketing Services
Box 516
St. Louis, Missouri

5.1.6 TRANE AIR CONDITIONING ECONOMICS (TRACE). Trane Air Conditioning Economics (TRACE) compares the economic impact of such building alternatives as architectural features, HVAC systems, HVAC equipment, scheduling and economic alternatives.* Trane Air Conditioning Economics evaluates up to four alternatives and compares the six possible combinations. Trane Air Conditioning Economics, simulates system operation for each zone of a building, each hour of the day to arrive at annual operating costs. A U. S. Weather Bureau tape provides data from the most recent in ten years and the computer reduces it to twelve days (one typical day for each month), representing a typical year. This creates a typical external load for a building. The weather tape also provides hourly information for an entire year on such items as dry bulb, wet bulb, dew point temperatures, wind velocity and cloud cover modifiers. This data is compiled and averaged for each month of the year to arrive at a typical 24 hour period for that month. Deviations are recorded from an average temperature for each hour. Weather tapes for 150 cities are available that give the hourly climatic conditions to be expected. TRACE consists of five major phases: load, design, system, equipment and economics, explained as follows:

5.1.6.1. Load Phase. In the load phase conventional load input data describing the building and its thermal-time characteristics are entered. Weather data from a U. S. Weather Bureau tape is fed in next to simulate actual weather conditions. Loads are calculated by zone by hour for a full year. To calculate solar loads, the program allows for such factors as location, orientation and altitude of the building. Transmission loads are found from heat transfer coefficients, square footage for load allocations and summer and winter design temperatures. Internal loads are calculated using input on lights, people and equipment. Ten standard schedules and ten optional schedules are available to describe the use of the building.

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*NAVFAC experience indicates that the economic analysis performed by this program does not conform to NAVFAC P-442.

This will provide the load diversity due to such items as occupancy, lights and equipment for any hour of the day. There is also a base utility schedule to account for energy used outside the conditioned spaces. The computer tracks this consumption by hour and adds to the cooling or heating energy consumptions for that hour so that accurate energy rates will be used. A typical example of base utility is outside lighting.

5.1.6.2 Design Phase. In the design phase, the type mechanical system to be used is described. The design phase receives input from the load phase and such items from the user as zone to system load assignment and amount of minimum outside air required. It also picks up block loads, peak loads by zone, and room design from the load phase. After calculating the system sensible heat ratios, the program uses a psychrometric repeat loop to calculate supply air dry bulb temperatures to each zone. It can then calculate the supply air quantities required to handle the load in each zone. The system supply air quantity is then found by adding the zone cfm's or using block loads.

5.1.6.3 System Simulation Phase. System simulation phase picks up the sensible and latent load by zone from the load phase and calculates return air quantities and temperatures. If applicable, this phase picks up any return air loads such as lights and roof. It also calculates the mixture temperature when mixed with outside air. Temperatures of outside air are derived from the weather tapes. The final step of system simulation is to calculate loads peculiar to the mechanical system itself. Most all types of air conditioning systems can be used for either the perimeter or interior. Included are high velocity variable air, low velocity variable air, double duct, multizone, terminal reheat, packaged terminal air conditioner, hydronic heat pump, fan-coil, induction, radiation and variable temperature constant volume. And nearly 60 types of cooling, heating, and air handling equipment can be used. The full and part load data mainly for TRANE equipment is stored on the equipment performance tape. Performance characteristics of other manufacturers' equipment can be substituted.

5.1.6.4 Equipment Simulation Phase. The equipment simulation phase uses output from the system simulation phase plus user input on the type of cooling and heating equipment, fans and pumps to determine the cooling load on the refrigeration equipment, heating load on the heating equipment, humidification load and supply and return fan air quantities. This allows full and part load efficiencies to be assigned to each piece of equipment that uses energy. Outdoor weather data modifies part load efficiencies. Using the equipment load by system from the system simulation phases, the equipment simulation phase generates monthly energy consumption by utility type.

5.1.6.5 Economic Analysis Phase. The economic analysis phase uses utility consumption from the equipment phase and user input such as utility rate structure. At this stage, the user also inputs installed cost and maintenance costs for the project. These costs must be analyzed carefully to give objective results. The economic phase then generates the economic comparisons of alternatives chosen. The output from TRACE compares the effects of the alternatives being considered. It is expressed in terms owners ordinarily use, such as present worth, pay back and return on investment. In addition, annualized owning and operating costs are part of each alternative output. This is beneficial to the institutional and government building owner.

5.1.6.7 Availability. TRACE is available through professional engineers. The local Trane commercial air conditioning office in each city is equipped to give more information.

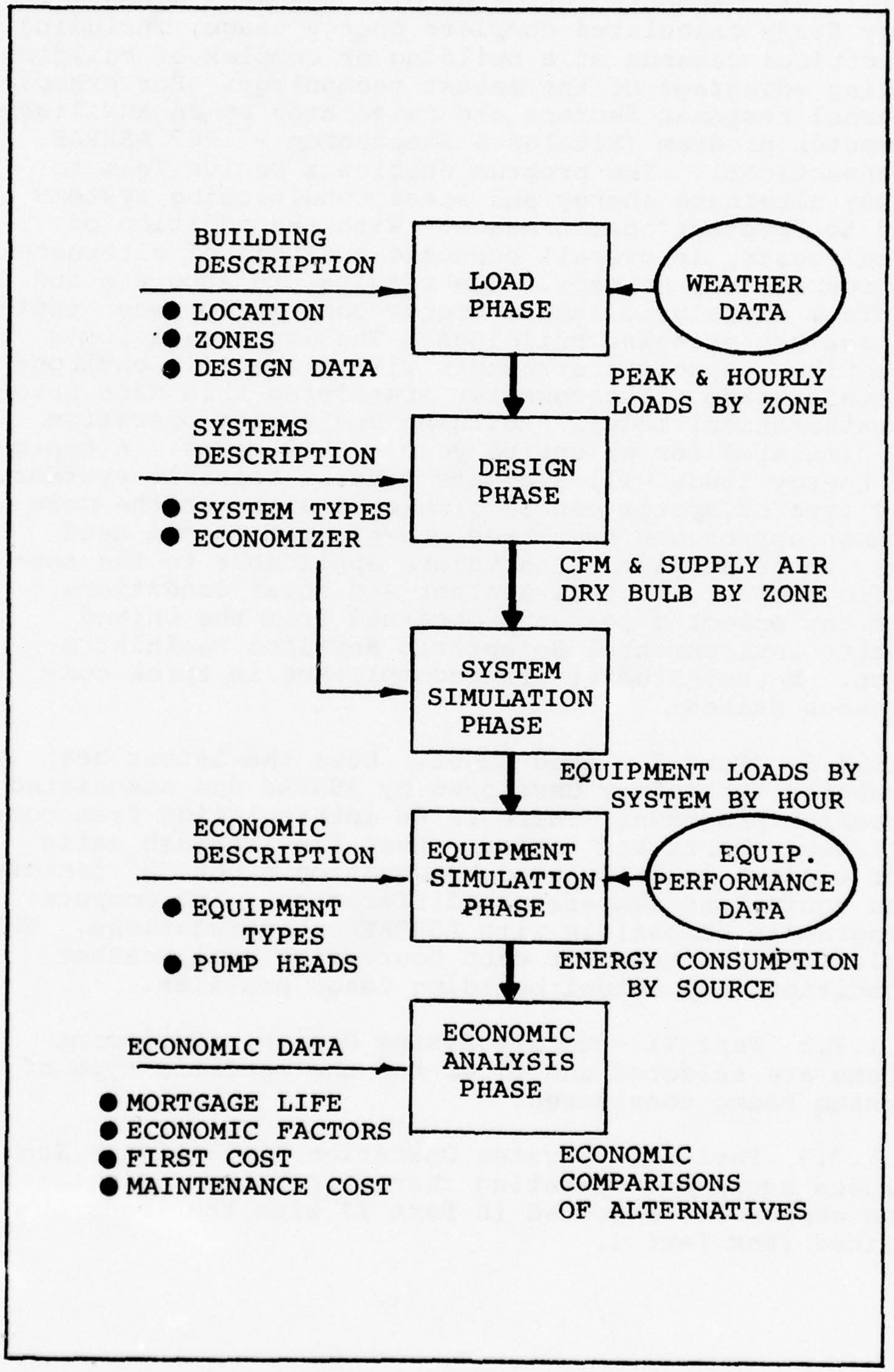

FIGURE 5-2

TRACE - CALCULATION FLOW CHART

5.1.7 WESTINGHOUSE ENERGY STUDY. The Westinghouse Energy Study calculates complete energy usage, including electrical demands of a building or complex of buildings, taking advantage of the latest technology. For example, thermal response factors are calculated by an auxiliary computer program (Mitalas & Stephenson - 1967 ASHRAE Transactions). The program enables a Design Team to study alternate energy and space conditioning systems and to "pretest" performance. With the addition of local costs, an overall economic analysis of alternate approaches may be made. The results are accurate and conform closely to actual energy consumption when tested against existing buildings. The input data forms describe a specific structure with a specific environmental system. The computer translates this data into a mathematical model. Building and system operation is simulated for an entire year - 8760 hours. A typical Energy Study will evaluate several feasible systems. Any type of system can be simulated although the more common approaches have been pre-programmed and need only to be supplied with values applicable to the specific project. Actual weather and solar conditions for the selected year are obtained from the United States Environmental Scientific Services Administration. Energy Studies are accomplished in three continuous phases:

5.1.7.1. Part I - Load Model. Uses the latest heat transfer technology developed by ASHRAE and associated research programs. There is no interpolation from complicated charts and tables. Heat flow through walls and windows is calculated by equation - not "U" factors and equivalent temperature differences. The computer program is compatible with ASHRAE recommendations. Calculations are made for each hour using real weather conditions and actual building usage profiles.

5.1.7.2 Part II - Supply System Design. Equipment items are selected and sized for the specific type of system being considered.

5.1.7.3 Part III - System Operation Simulation. Includes equipment operating characteristics. Simulates the components selected in Part II with the loads obtained from Part I.

5.1.7.4 Availability. Additional information and pricing details may be obtained from Westinghouse Major Projects and Urban Systems representatives who are located in major cities. Inquiries may be addressed to:

Westinghouse Electric Corporation
The Energy Utilization Project
Power Systems Planning
700 Braddock Avenue
East Pittsburgh, Penna. 15112

5.1.8 HEATING-COOLING CALCULATION (HCC-III). HCC-III is an advanced computerized procedure for calculating design heating and cooling loads for buildings in accordance with ASHRAE methodology. The program is unique in that it is based on a "down to earth" approach, having been developed by a group of outstanding design professionals for their own use. It thus not only incorporates those design aspects of most critical interest to the designer, but analyzes them with a thoroughness and degree of sophistication impossible before computerization. By combining ASHRAE techniques with engineering experience, the program avoids compromises with internationally recognized porcedures. As a design program, HCC-III is oriented to the design professional as a tool to develop the necessary information in a recognized manner and in such a way as to facilitate use of the output directly, without manual manipulation. Apart from design benefits, which are numerous, production costs per job have been reduced by application of the most advanced programming techniques.

5.1.8.1. Administrative Benefits.

(1) Standardization of calculation procedures (avoiding parochiolism, or idiosyncrasies of individual designers).

(2) Improvement of record keeping and calculation format.

(3) Utilization of lower echelon personnel for physical data takeoff, freeing design engineers for more significant decisions.

(4) Cost reduction of calculation procedures, with no sacrifice in thoroughness or quality.

5.1.8.2 Technical Benefits.

(1) Provide cooling load calculations on a 24 hour basis, comparing hourly totals for determination of true peak hour for each room, air handling system (zone), and for the entire building.

(2) Take into account thermal storage effect of building mass and contents on radiant components of both

transmission loads from the exterior and internal loads from lighting, people, and appliances.

(3) Consider all types of environmental surfaces affecting cooling and heating loads, and provide means to minimize input effort in directing their evaluation for individual spaces.

(4) Provide a means to analyze the effects of using a ceiling return air plenum system, including the reduced effect on room sensible loads and the increased effect on the air handling unit coil resulting from increased return air temperature.

(5) Calculate the shading effectiveness of any combination of window overhang and/or side fins in evaluating the solar components of glass loads.

(6) Consider the effect of varying glass types and glazing arrangements as well as interior shading devices, and the angle of exposure of glass to the sun, whether vertical, sloped, or a skylight.

(7) Take into account the varying effects on system loads and room CFM values of system type, whether variable volume, terminal reheat, or conventional mixed air.

(8) Calculate room CFM values (including Project altitude consideration) on either an input dehumidified temperature rise or a calculated apparatus dew point method, and sum them into designated air systems.

(9) Calculate ventilation air CFM values based on a variety of optional factors.

(10) Analyze thermodynamically the resultant air mixture entering and leaving the cooling coil, including the effects of supply fan location, use of return and/or outside air fans, coil bypass factor, estimated static pressures, and supply duct heat gain, for the purpose of determining maintained space humidity ratios and actual maximum cooling coil loads.

(11) Minimize input requirements by utilizing to the fullest possible extent the concept of "master"

data, with individual "over-ride" capabilities at more specific levels.

(12) Solar calculations take geographic factors into account including longitude and latitude, altitude, building orientation, atmospheric clearness and ground reflectivity.

(13) Minimize re-run effort in event of project modification by retaining in systems data file storage all input and result data; permitting re-input of only the modified areas, and recalculation and result analysis of only the parts and systems thus affected.

(14) Maximize program practicality by providing optional interface with available equipment selection programs for simultaneous output.

5. 1.8.3 Availability. HCC-III was developed by Automated Procedures for Engineering Consultants, Inc. (APEC). Access to the APEC program is restricted to APEC member firms. Information on membership, a descriptive abstract of the program, and access procedures may be obtained from:

Doris J. Wallace
APEC, INC.
Suite M-15, Grant-Deneau Tower
Dayton, Ohio 45402

Section 2. MINIMUM REQUIRED DESIGN PARAMETERS

5.2.1 INPUT. In this section, the types of information which can be input to computer programs for an energy system analysis are listed. The information is typical of that required for all computer programs and is dependent on the program used, the point in the design process at which the analysis is made, and the degree of accuracy desired in the results. Only information necessary for an energy analysis is shown. Information for the calculations of heating and cooling load is not shown although it is required in some cases.

5.2.2 BUILDING INFORMATION.

5.2.2.1 General.

(1) Area to be Air Conditioned

(2) Solar Area Percentages for Each Direction.

(3) Heat Storage Effect of Building and Hours for Spread.

5.2.2.2 Design Conditions for Heating and Cooling.

(1) Ambient Dry Bulb and Dew Point.

(2) Indoor Thermostat Settings for Dry Bulb and Dew Point.

5.2.2.3 Heating and Cooling Maximum Loads.

(1) Solar

(2) People

(3) Lights

(4) Equipment

(5) Miscellaneous

(6) Design Transmission Loads

5.2.2.4 Other Type Maximum Loads.

(1) Base Electric

(2) Process Heat with Type Firing

5.2.2.5 Operational Day Types and Schedules.

(1) Shut-off and Set Back Time Schedule for Heating, Cooling, Outside Air, Set Back

(2) Percent Variation Profile Followed by Each Load Type

(3) Months Each Day Type Occurs

(4) Days of Week Each Day Type Occurs

(5) Holidays

5.2.3 SYSTEM INFORMATION

5.2.3.1 Type.

5.2.3.2 Capacities for Heating and Cooling Systems.

5.2.3.3 Operation Schedules and Keys.

(1) Part Load Percent Variation Profiles - Used for describing part load performance for chillers, boilers, and generators by representing primary and auxillary fuel input curves, available recoverable heat curves, and accessory equipment input curves.

(2) Ambient Temperature Effect Profiles - Used for describing the ambient temperature effects on the rated capacities of chillers, generators, and cooling towers. Temperatures may be either ambient dry bulb, ambient wet bulb, or condenser water temperatures.

(3) Limitation Schedules

(4) Equipment Groupings and Operation Keys for Each System

5.2.3.4 Controls.

(1) Upper and Lower Limits for Economizer

(2) Humidification Reset

(3) Hot Deck Temperature Schedules

(4) Fan Operation Method

5.2.3.5 Air Flow.

(1) Total Supply Air

(2) Maximum Cold Deck and Hot Deck Air

(3) Maximum Outside Air

(4) Minimum Outside Air During Heating, Cooling, and Unoccupied Periods

(5) Minimum Percent for Variable Volume Systems

(6) Air Temperature Rise for Fans

(7) Primary and Secondary Supply Air Temperatures and Humidity Ratios

5.2.4 EQUIPMENT.

5.2.4.1 Chiller Characteristics.

(1) Type (direct/indirect fired absorption, steam turbine driven, etc.)

(2) Minimum Load for Starting Unit (base on manufacturers recommendation)

(3) Rated Output

(4) Maximum Primary Energy (Electricity, Gas, Oil, etc.) Input at Rated Output

(5) Maximum Recoverable Heat at Rated Output

(6) Maximum Auxiliary Energy (Electricity, Gas, Oil, etc.) Input at Rated Output

(7) Pilot Fuel Input for Dual Engines

(8) Accessory Loads with Type Accessory

(9) Part-Load Percent Profiles for Operation (See 5.2.3.3)

(10) Ambient Temperature Profiles for Operation (See 5.2.3.3)

5.2.4.2 Generator Characteristics.

(1) Type Drive

(2) Rated Output

(3) Maximum Primary Energy (Elec., Oil, Gas, etc.) Input at Rated Output

(4) Maximum Recoverable Heat at Rated Output

(5) Maximum Auxiliary Energy (Elec., Oil, Gas, etc.) Input at Rated Output

(6) Pilot Fuel Input for Dual Engines

(7) Accessory Loads with Type Accessory

(8) Part Load Percent Profile for Operation (See 5.2.3.3)

(9) Ambient Temperature Profiles for Operation (See 5.2.3.3)

5.2.4.3 Boiler, Process Heat Equipment, Heat Pumps Characteristics.

(1) Type

(2) Rated Output

(3) Energy Input at Rated Output

(4) Accessory Loads with Type Accessory

(5) Part-Load Percent Profiles for Operation (See 5.2.3.3)

(6) Ambient Temperature Profiles for Operation (See 5.2.3.3)

5.2.4.4 Cooling Tower.

(1) Fan Kilowatt

(2) Ambient Temperature Profiles

5.2.5 ENERGY FORM

5.2.5.1 Auxiliary Forms

(1) Amount per Unit

(2) Cost per Unit

(3) Heating Value of Fuel

5.2.5.2 Other Forms (Gas, Electricity, Steam, Chilled Water, Hot Water).

(1) Heating Value of Gas

(2) Charge - Monthly Minimum Fixed, Adjustment Rates

(3) Methods for Determining Charges

(4) Demand Rate Steps - Size, Method of Application, Cost

(5) Consumption Rate Steps - Size, Method of Application, Cost

(6) Energy Use Scheduling for Each System

5.2.6 ECONOMICS FOR EACH SYSTEM.

(1) Type Analysis (Follow NAVFAC P-442, Present Worth Method)

(2) Number of Years for Analysis

(3) Annual Costs for Energy Type

(4) Annual Costs for Maintenance, Operation, and Any Miscellaneous Costs

(5) Escalation Rates for Costs

(6) Initial Gross Investment with Percent Salvage

(7) Depreciation Methods

(8) Debt Amount with Interest Rate

(9) Equity Interest Rates

(10) Data for Reinvestments

Section 3. EXAMPLE - COMPUTER ANALYSIS FOR A NEW BUILDING

5.3.1 EXAMPLE BUILDING FOR COMPUTER SIMULATION. The selected example building for computer analysis, energy conservation and cost evaluation is a two-story rectangular building. A complete building description and input data are enclosed. The analysis is performed using Kling-Lindquist, Inc. heating and cooling load (HECOL) program in conjunction with Ross F. Meriwhether Energy System Analysis (ESA) programs. A description of HECOL is enclosed in this section. The analysis enables us to evaluate the effectiveness of each varying design parameter for energy conservation. The various parameters considered include:

(1) Orientation

(2) Enthalpy control

(3) Glazing

(4) Infiltration and ventilation rates

(5) Constant volume reheat system

(6) Variable volume system

(7) Double bundle heat recovery system

5.3.1.1 Run 1. The base system selected is a constant volume, reheat system with perimeter radiation and enthalpy control. Run 1 is made using single glass and no enthalpy control and with the building having the longest exposures with glass facing north and south. Run 1S is the same as Run 1 except this has enthalpy control.

5.3.1.2 Run 2. In Run 2 the building is rotated 90 degrees with glass exposed to east and west. The exposure with the lowest loads and energy cost is selected as the base building.

5.3.1.3 Run 3. Run 3 is made using the base building and base system except this has insulating glass.

5.3.1.4 Run 4. Run 4 is base building, base system and no glass.

5.3.1.5 Run 5. Run 5 has base building and base system with twice the infiltration and ventilation rate.

5.3.1.6 Run 6. Run 6 has base building with variable volume system, enthalpy control and perimeter radiation.

5.3.1.7 Run 7. Run 7 has base building and base system using double bundle machines and part enthalpy cycle, i.e., outdoor air is adjusted to meet the heating demand.

5.3.1.8 Table Summary. Tables 5-1 through 5-4 show the summary of all runs. Tables 5-2 and 5-3 show heating loads, cooling loads, energy usage, energy cost. Present worth of owning and operating cost and various methods of selection of alternatives is given in Table 5-4. Table 5-1 shows estimated investment cost based on various assumptions as stated in the table. A copy of the building drawing and important HECOL and ESA printouts are enclosed.

5.3.2 HECOL PROGRAM DESCRIPTION. Program "HECOL", code name for "Heating and Cooling Loads in Buildings", performs calculations of heating load in winter and cooling load in summer in buildings with specified design conditions. The calculation logics and procedures in HECOL follow closely the conventional practices in heating and cooling designs for buildings. The input data for the calculations can be divided into two groups:

(1) Design Constants. Criteria chosen by the user for the building concerned, such as materials of construction, temperature ranges, etc.

(2) Design Dimensions. Physical requirements of the building, such as room sizes, glass areas, number of people and equipment housed, etc.

The groups are so organized that minimum repetition and maximum utilization of the input data are achieved for the program. Alternate designs for the same building can be obtained with very little effort by merely changing a few of the design constants and submitting to the program for a rerun. By similar reasoning, refinement designs for the same building also can be

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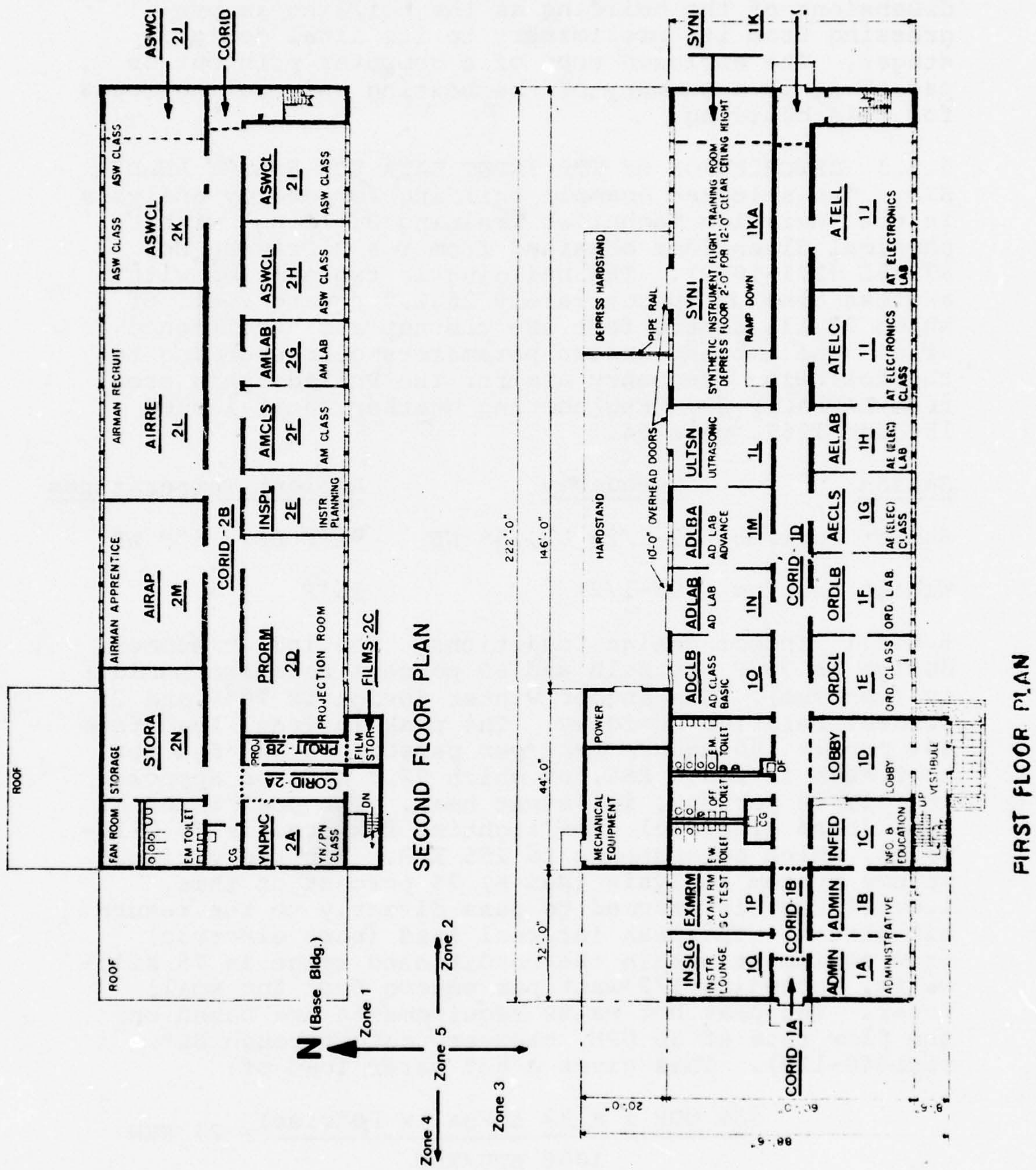


FIGURE 5-3

Example Building Dimensions Obtained From Y & D Dwg No **873591 (171-10-B)**	DATE 10/24/7 BY J.& C. SCALE 1" = 32' JOB NO. 1335-00

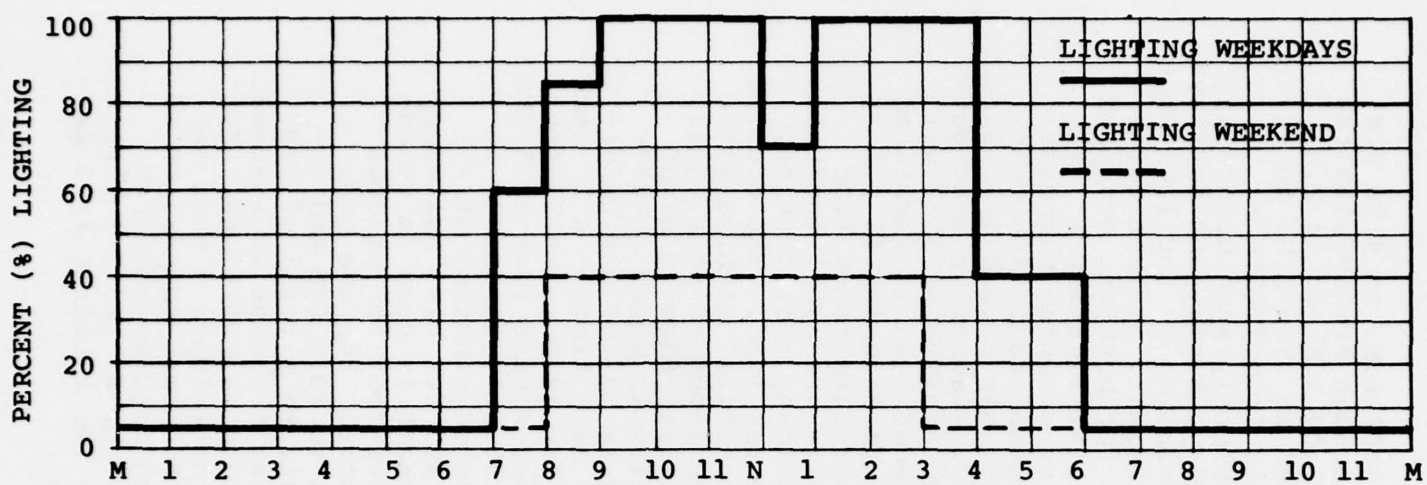
easily achieved by revising and updating the design dimensions of the building as the building is progressing from its preliminary to its final design stages. The enclosed copy of a computer printout on page 5-51 is a summary of the heating and cooling loads for this building.

5.3.3 DESCRIPTION OF THE INPUT DATA FOR ENERGY ANALYSIS. The selected example building for energy analysis is the "Aviation Technical Training Building" with physical dimensions obtained from Y & D Drawing No. 873591 (171-10-B). The building is two stories with a gross area of approximately 26,230 square feet, of which 23,136 square feet are the net air conditioned area. The ambient design parameters corresponding to the following frequency are for the Philadelphia area, from NAVFAC P-89, "Engineering Weather Data" issued 15 June 1967, page 54.

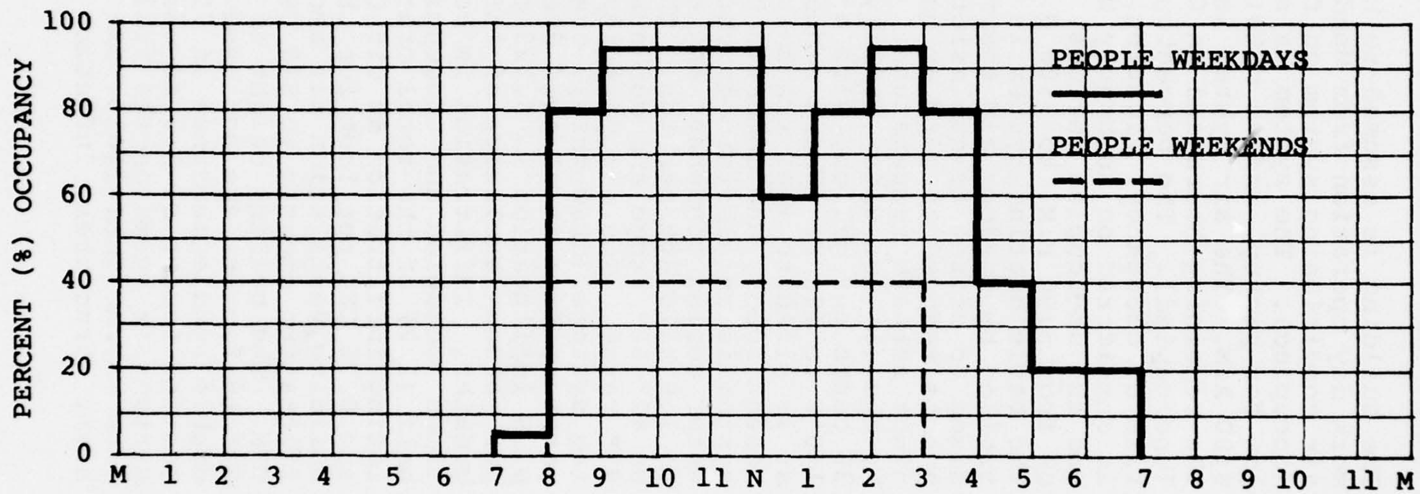
Season	Frequency	Ambient Temperatures
Summer Cooling	2-1/2% DB, 5% WB	91°F DB, 76°F WB
Winter Heating	97-1/2%	16°F

5.3.3.1 Indoor Design Conditions. The indoor summer design is 78°F dry bulb and 60 percent relative humidity (maximum). The indoor winter design is 70°F and 25 percent relative humidity. The peak internal load from 464 people (50 square feet per person) at 450 Btu per hour each is 208.8 KBH, of which 92.8 KBH, or approximately 44 percent, is latent heat. The peak internal load (base electric) from lighting fixtures is 75 kilowatts, which corresponds to 255 KBH. For the variable volume system analysis (Run 6) 25 percent of this, i.e. 64 KBH, is assumed to pass directly to the return air stream. The peak internal load (base electric) from equipment within the conditioned space is 75 kilowatts, including 1/2 watt per square foot for small power. The peak hot water requirements are based on the flow rate of 36 GPH, recovery rate through 80° rise(40-120). This gives a hot water load of:

$$\frac{(36 \text{ GPH} \times 8.33 \text{ lb/gal} \times 80°\text{rise})}{1000 \text{ BTU/KBH}} = 24 \text{ KBH}$$



TYPICAL DAY LIGHTING PROFILE



TYPICAL DAY OCCUPANCY PROFILE

FIGURE 5-4

The building is assumed to have 8:00 A.M. to 4:30 P.M. work day operation from Monday through Friday with the air conditioning system on from 7:00 A.M. to 5: P.M. (occupied). The system is started at 7:00 A.M. to allow for pickup, and no outside air from 7:00 A.M. to 8:00 A.M. The air system is completely shut off during the remaining hours and on Saturday, Sunday and holidays (unoccupied). The percentage variation in the load profile of occupancy, lighting, equipment, and hot water is considered on an hourly basis for both the occupied and the unoccupied periods. Partial lighting is left on from 5:00 P.M. to 7:00 P.M. for cleaning purposes. The basic heating system is perimeter radiation controlled by outdoor air thermostat, see Figure 5-5, and sized to offset transmission loss. Radiation heat output is reduced to zero at outside ambient of 55°F with night setback temperature. The heat load is furnished by one (1) hot water boiler, oil fired using #2 oil at 36 cents per gallon, with a temperature range from 160°F to 180°F. Where reheat coils are used, the boiler is sized to meet transmission loss, infiltration, reheat loads, and domestic hot water load. The cooling system is an electric driven centrifugal machine with water cooled condenser and energy input at peak equal to 1.0 kilowatt per ton. The chiller flow is based on 42°-62° range and condenser flow is based on 85°-95° range. The air distribution is by two air handling systems (draw-thru), one for each floor. A 3° temperature rise is allowed due to fan gain. Ventilation plus infiltration is 0.125 CFM per square foot or 5 CFM per person (exclusive of infiltration), whichever is greater. Infiltration is considered constant at all times (occupied and unoccupied) and for base building, half (1/2) of the total crack length is used in calculating infiltration at 0.5 CFM per foot for windows and 0.75 CFM per foot for the doors. For the base building,ventilation air amounts to 2371 CFM and infiltration amounts to 518 CFM, giving a total of 2889 CFM. The ceiling height of 8 feet 6 inches is assumed. Safety factors are not used in load calcualtions. For double bundle machines at 75 percent load, a heat rejection factor of 1.3 is used. Economic comparison of systems is based on the following figures obtained from NAVFAC INST 11010.55A: Life of 25 years for permanent structures, inflation rate for electricity 3 per-

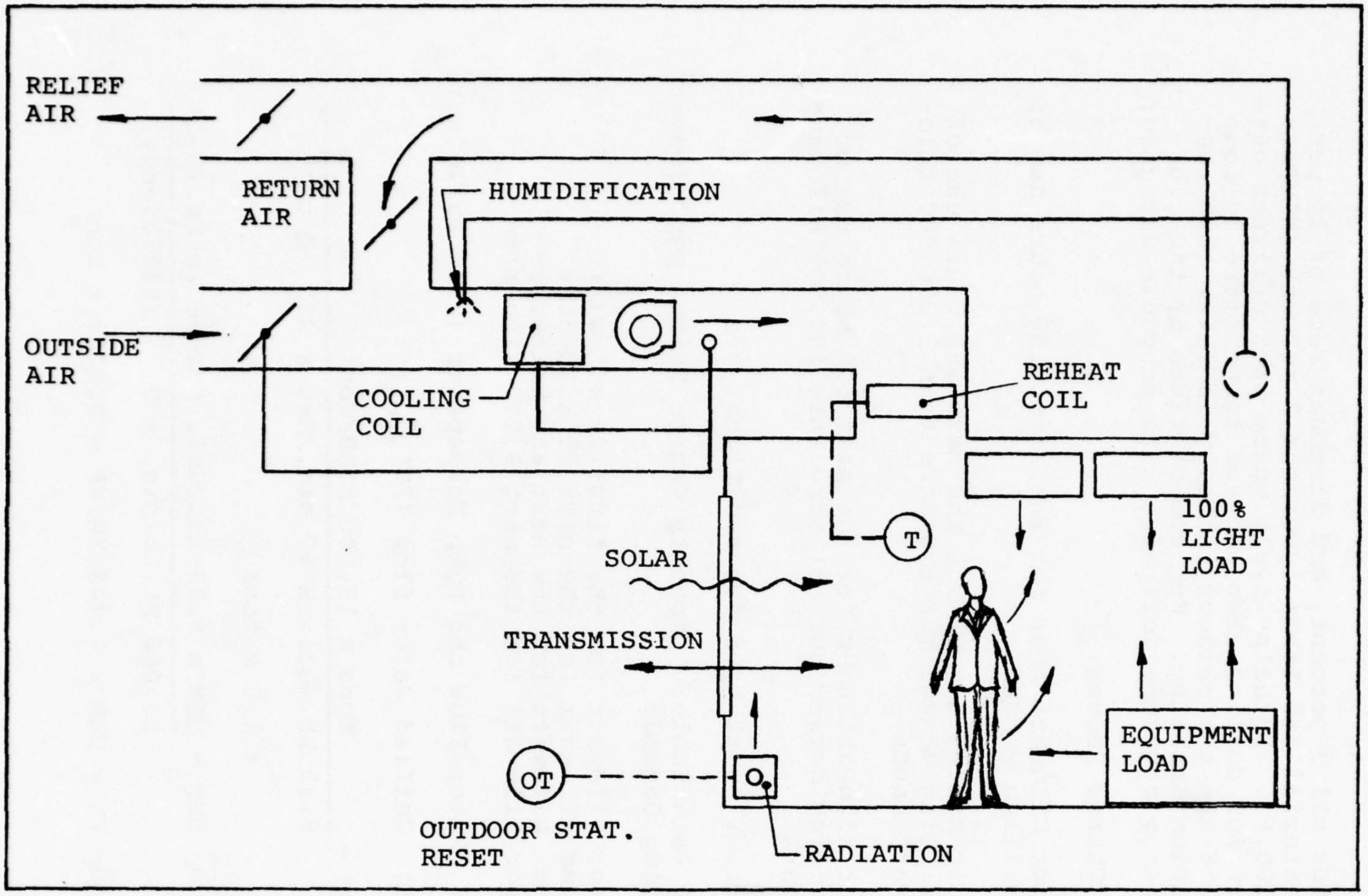


FIGURE 5-5

CONSTANT VOLUME WITH REHEAT AND PERIMETER RADIATION cent, for oil 9 percent, and discount rate of 10 percent, material and labor increases 3 percent (NAVFAC INST 4100.6). Enthalpy cycle where used utilizes outdoor air for cooling when ambient temperature is less than 60°F and the outdoor enthalpy is lower than the room design enthalpy. For computations of the electric energy cost the following rate schedule was used:

Energy Charge Prices:

1.66¢ per kilowatt hour for the first 150 hours use of billing demand

1.27¢ per kilowatt hour for the next 150 hours use of billing demand but not more than 1,200,000 kilowatt hours

.91¢ per kilowatt hour for the next 100 hours use of billing demand but not more than 500,000 kilowatt hours

.77¢ per kilowatt hour for additional use

Monthly Rate Table: Capacity Charge Prices per Kilowatt of Billing Demand:

$4.31 per kilowatt for the first 25 kilowatts
$2.70 per kilowatt for the next 25 kilowatts
$1.83 per kilowatt for the next 450 kilowatts
$1.66 per kilowatt for the next 4,500 kilowatts

5.3.3.2 Water Flow and Pump Horsepower (Base Building).

(1) Chilled water flow (20° Δ t)

$$\text{GPM} = \frac{\text{Tons} \times 12{,}000 \text{ BTUH/Ton}}{8.33 \text{ Lb./Gal.} \times 60 \text{ Min./Hr.} \times 20^\circ \ \Delta t}$$

$$= 1.2 \times \text{tons}$$

$$\text{Pump BHP} = \frac{\text{GPM} \times 8.33 \text{ Lb./Gal.} \times \text{Head in Ft.}(H_2O)}{33{,}000 \text{ Ft.Lb/Min.} \times 0.55 \text{ (Efficiency)}}$$

$$\text{Pump KW} = \text{BHP} \times 0.746 \text{ KW/HP} = 0.0169 \times \text{tons}$$

(40 foot head assumed includes 10 feet for chiller plus 10 feet for coil plus 10 feet for piping plus 10 feet for control valve.) Frictional head for other runs is taken as proportional to square of the flow. However, for large size job changing the pipe size should be considered.

(2) Condenser water flow ($10° \Delta t$)

GPM = Tons x 3 GPM/Ton = 3 x Tons

Similarly, Pump KW = 0.041 x Tons

(40 foot head assumed includes 15 feet for condenser plus 10 feet for tower static head plus 15 feet for piping and valves.) Frictional head for other runs is taken as proportional to square of the flow.

(3) Hot Water Pumps ($20° \Delta t$)

Radiation Pump KW:

$$\frac{\text{KBH(Trans)} \times 1000 \times 8.33 \times 40' \text{ (head)} \times 0.746}{8.33 \text{ Lb./Gal.} \times 60 \text{ Min./Hr.} \times 20° \ \Delta t \times 33000 \times 0.55}$$

= 0.00137 x KBH

Same value for reheat coils pump. For other runs revise frictional head proportional to the square of the flow rate.

(4) Fan Energy. (Base Building)

$$\text{KW} = \frac{\text{CFM} \times \text{Static Pressure Inches } (H_2O) \times 0.746}{6350 \times 0.60 \text{ (Efficiency)}}$$

= 0.00059 x CFM

Based on 3.0 inch static pressure (1.20 inches for coil plus 0.60 inches for filter plus 0.90 inches for ducts and fittings plus 0.15 inches diffuser plus 0.15 inches for reheat coil). Friction for other runs proportional to square of flow. Consider changing the duct size for buildings larger in size. For variable volume allow 1.0 inch additional for high pressure system.

(5) Cooling Tower Fan. (5 Horsepower assumed)

KW = Horsepower x 0.746 = 3.73 Kilowatts

5.3.3.4 Holidays. The calendar year is to be 1975, which begins on a Wednesday, and the following 8 holidays will be observed: January 1 - New Year's Day, May 26 - Memorial Day, July 4 - Independence Day, September 1 - Labor Day, October 13 - Columbus Day, October 27 - Veterans Day, November 27 - Thanksgiving Day, and December 25 - Christmas.

5.3.3.5 Runs. The following seven (7) runs were made to see the affect of variation in the design parameters and systems:

(a) Run 1. Loads with glass exposed to north and south; constant volume with reheat, perimeter radiation. Without enthalpy cycle. Reheat coils designed for 10° air temperature rise.

$U_w = 0.08$

$U_r = 0.05$

$U_f = 0.11$

$U_o = 0.31$ (overall coefficient of heat transfer)

22 percent single glass (percent glass corresponding to $U_o = 0.31$)

$U_{glass} = 1.13$, shading coefficient 0.55

Ventilation + infiltration = 0.125 CFM per square foot or 5 CFM per person (exclusive of infiltration)

(b) Run 1S. Same as Run 1, but with enthalpy cycle. In comparison with Run 1, the enthalpy cycle (1S) saves 82276 kilowatt hours of electrical energy, (see Table 5-3 on page 5-50) a highly refined source of energy. The enthalpy cycle (1S) uses less energy at approximately the same cost as Run 1 (Table 5-4), so the enthalpy cycle is used when the building is turned 90 degrees for Run 2.

(c) Run 2. Loads with glass exposed to east and west; single glass, with enthalpy cycle. Select lower of (1S) and (2) as base building. The energy consumption for Run 2,when compared with Run 1S,is increased approximately 3 percent and 18 percent for electricity and oil,respectively. This gives an overall increase of 10 percent in the annual energy usage so the base building is selected incorporating the enthalpy cycle with glass to the north and south for Run 3.

(d) Runs 3 to 7. Run 3 to Run 7 considers each design variable separately when applied to the base building (1S) for computer simulation to evaluate its effect on energy consumption. Results are summarized in Table 5-2 through Table 5-4.

(e) Run 3. Base building with insulating glass (retain same percentage of glass). $U_{glass} = 0.65$, shading coefficient 0.51. By changing to double glazing a saving in energy of 9 percent can be realized when compared with Run 1S.

(f) Run 4. Base building with no glass (0 percent glass). By complete elimination of glass a saving in energy of 20 percent can be realized when compared with Run 1S. This run has the lowest value of electrical energy consumption, total energy cost and total present worth of owning and operating cost. This design normally is not acceptable for architectural considerations.

(g) Run 5. Base building with double infiltration and ventilation rate. Increasing the infiltration and ventilation rate consumes 5 percent additional energy when compared with Run 1S. This run made only to display the effect of changing these parameters. Energy increase is low since there was lower outdoor air to start with.

(h) Run 6. Base building with variable volume, enthalpy cycle. Consideration of variable volume system reduces the energy consumption by 20 percent when compared with Run 1S. See Figure 5-6.

(i) Run 7. Base building with double bundle machines with part enthalpy cycle. Consideration of double bundle heat recovery machine reduces the energy consumption by 22 percent when compared with Run 1S.

(j) Run 8. For comparison purposes an additional run was made without limiting the building envelope U_o factor. The building was assumed to have been built exactly as shown on Y & D Drawing No. 873591 (171-10-B), a copy of which is included. 54 percent single glass as obtained from the drawing, $U_w = 0.30$, $U_r = 0.18$, and one-half air change per hour infiltration value is used for computer simulation of this run. The last run simulated using the existing building consumed 59 percent additional energy when compared with Run 1S.

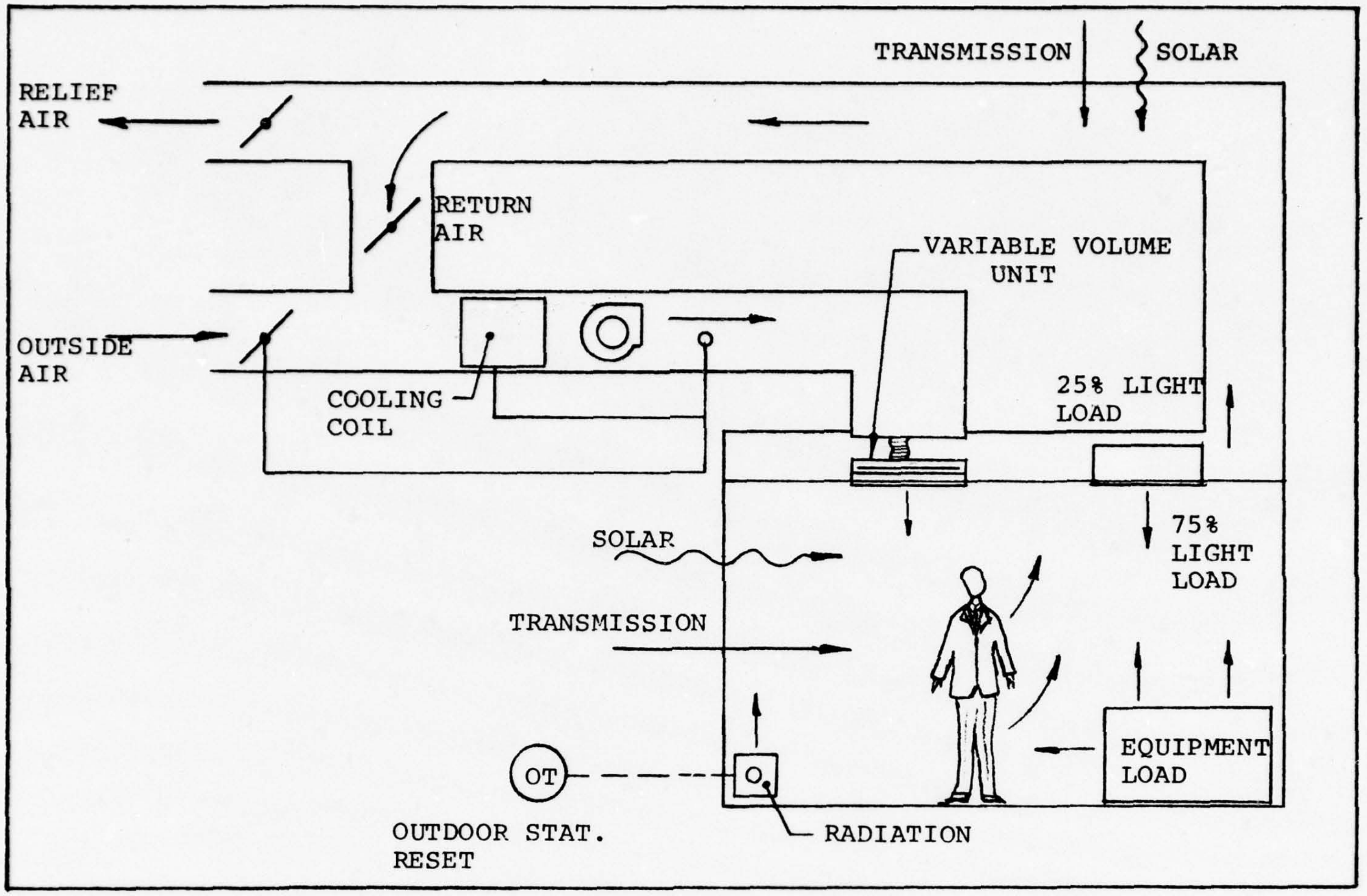


FIGURE 5-6

VARIABLE VOLUME SYSTEM WITH PERIMETER RADIATION

Table 5-1
Initial Investment
(Estimated HVAC Systems and Components)

	1E	2E	3E	4E
A. Central Chilled Water Plant @ $2,000/Ton constant volume reheat system and $1,675/ton for variable volume system (Air systems, cooling tower, heating coil, enthalpy control, piping, etc.)	$210,060	$203,580	$207,480	$203,000
B. Heating Hot water and humidification boiler, radiation	$ 32,080	$ 27,920	$ 26,620	$ 24,380
C. Insulation and Glazing				
Wall insulation	0	0	$ 3,680	0
Roof insulation	$ 5,070	$ 5,070	$ 25,340	$ 5,070
Glass - single @ $4.00/sq.ft. and double @ $6.50/sq.ft.	$ 17,470	$ 28,390	$ 17,470	$ 17,470
D. Miscellaneous Heat recovery wheels	-	-	-	$ 2,400
Total	$264,680	$264,960	$270,590	$252,320

Table 5-2
HECOL Load Summary for Existing Building (23,136 SQ.FT.)

Run No.	1E	2E	3E	4E
Peak Load (Hr/Month)	2pm/SEP	2pm/SEP	1pm/OCT	2pm/SEP
Lighting, MBH	255.0	255.0	255.0	255.0
Equipment, MBH	255.0	255.0	255.0	255.0
People, MBH (Total)	208.8	208.8	208.8	208.8
Solar, MBH	275.0	255.0	330.6	275.0
Transmission, MBH	87.3	68.4	16.2	87.3
Infilt. & Vent., MBH (Total)	176.1	176.1	176.1	176.1
Supply CFM	50,529	48,749	48,516	50,529
Total Tonnage	105.03	101.79	103.74	105.03
Winter Ventilation, MBH	138.3	138.3	138.3	138.3
Winter Trans. & Infilt., MBH	566.8	453.6	430.7	566.8

Table 5-3
Summary of Annual Energy Consumption For Existing Building

Run No.	1	2	3	4
Electricity, KWH	753,189	731,957	729,130	852,037
Electrical Cost	$20,918	$20,355	$20,309	$22,637
Oil, Gallons	24,736	19,685	20,773	21,885
Oil Cost	$ 8,905	$ 7,087	$ 7,478	$ 7,879
Total Energy Cost	$29,823	$27,441	$27,787	$30,515

TABLE 5-4
Summary of Energy System Analysis

	1	1S	2	3	4	5	6	7	8
1. Humidification Load MBH	0	352	334	347	285	353	336	323	482
2. Initial Investment Cost HVAC Systems $	203,900	211,770	215,970	209,600	181,130	228,740	178,110	217,250	264,689
3. Total Present Worth of Owning and Operating Cost $1,000	796,447	797,576	821,501	777,275	726,614	824,353	728,444	758,905	980,410
4. Selection Preference Based on 3	5	6	7	4	1	8	2	3	9
5. BTU/Yr./Sq.Ft. at the Building Boundary	168,254	164,838	180,128	150,231	131,951	172,453	132,630	128,610	261,860
6. Selection Preference Based on 5	6	5	8	4	2	7	3	1	9
7. BTU/Yr./Sq.Ft. Based on Raw Energy to Allow Utility Plant-Efficiency (30%)	415,075	383,338	404,015	367,385	341,518	394,160	344,763	355,309	522,590
8. Selection Preference Based on 7	8	5	7	4	1	6	2	3	9

NOTE: Item 5 = $\frac{\text{KWH} \times 3413 + \text{Gals.} \times 141{,}000}{23{,}136}$ Item 7 = $\frac{\frac{\text{KWH} \times 3413}{0.3} + \text{Gals.} \times 141{,}000}{23{,}136}$

KLING-LINDQUIST * NAVFAC ENERGY STUDY * RUN NO. 1 * JOB NO. 1355-00 * CARPENTER ZONE ALL

* ROOM * MEASURE	ROOM AREA SF	ROOM VOL CF	WALL AREA SF	GLASS AREA SF	NET WALL AREA SF	PCT GLASS TO WALL
	23136.	196558.	8134.	1753.	6381.	21.55
* SUMMER * SENSIBLE*	SENSIBLE HEAT BTUH	SUPPLY AIR CFM	DEHUMID AIR CFM	FRESH AIR CFM	SUPPLY AIR RATE CFM/SF	SUPPLY AIR CHANGE MIN
	763999.	(35370) 36,716 (Undiversified)	30756.	2371.	1.53	5.61
	FRESH AIR CHANGE MIN	SENSIBLE HEAT TONS	SENS HEAT RATE SF/TON	FRESH AIR HEAT BTUH		
	63.74	63.67	363.40	33289.00		
		GLASS HEAT BTUH	WALL HEAT BTUH	ROOF HEAT BTUH	FLOOR HEAT BTUH	
		17828.01	3205.74	10474.52	.00	
		PARTITION HEAT BTUH	LIGHT HEAT BTUH	MISCEL. HEAT BTUH	PEOPLE HEAT BTUH	
		3267.40	254866.17	254966.85	116000.00	
* SUMMER * LATENT	LATENT HEAT BTUH	LATENT HEAT TONS	FRESH AIR HEAT BTUH			
	102069.	8.51	42156.			
* SUMMER * TOTAL	TOTAL S+L HEAT BTUH	SENSIBLE HEAT RATIO	FRESH AIR PEN BTUH	FRESH AIR PEN TONS	TOT S+L+FAP HEAT TONS	TOT S+L+FAP RATE SF/TON
	866059.	.88	75445.	6.29	78.46	294.88
* SUMMER * SUBTOTAL	SKIN HEAT LOAD BTUH	SOLAR HEAT ONLY BTUH	NUMBER OF PEOPLE			
	127582.	96074.	464.			
* WINTER * HEAT LOAD	TRANSM HEAT BTUH	VENTIL HEAT BTUH	TOTAL HEAT BTUH			
	237358.	138277.	375635.			

NUMBER OF ROOMS IN BUILDING	ORIGINAL	39
	DUPLICATE	0
	TOTAL	39

BEST AVAILABLE COPY

EQUIPMENT ENERGY CONSUMPTION AND OPERATING DATA FOR

RUN1 NAVFAC ENERGY ANALYSIS.CONSTANT VOL.WITH REHEAT.PERIMETER RAD.N-S SING.GL.

SYSTEM 1 CONSTANT VOLUME WITH REHEAT PERIMETER RADIATION N-S SINGLE GLASS

FUEL AND POWER CONSUMPTION

SYS	GAS USAGE MCF	PEAK DAY GAS MCF	ELECTRIC USAGE KWH	PEAK ELEC DEMAND (30 MIN)	AUXILIARY FUEL USAGE
** JAN **					
1	0.	0.	57918.	232.	1711.
** FEB **					
1	0.	0.	53613.	248.	1580.
** MAR **					
1	0.	0.	61106.	249.	1074.
** APR **					
1	0.	0.	62990.	250.	709.
** MAY **					
1	0.	0.	60886.	250.	822.
** JUN **					
1	0.	0.	61126.	252.	647.
** JUL **					
1	0.	0.	63942.	251.	391.
** AUG **					
1	0.	0.	61723.	250.	96.
** SEP **					
1	0.	0.	61132.	251.	122.
** OCT **					
1	0.	0.	60785.	250.	672.
** NOV **					
1	0.	0.	54536.	249.	891.
** DEC **					
1	0.	0.	57306.	244.	1537.
** ANN **					
1	0.	0.	717062.	252.	10251.

MONTHLY AND ANNUAL UTILITY COST FOR

RUN1 CONSTANT VOL W.REHEAT.PERIMETER RAD.N-S-SINGLE-GLASS

	ELECTRIC COST, $	AVER RATE C/KWH	UNIT COST C/SQFT	GAS COST, $	AVER RATE C/MCF	UNIT COST C/SQFT	AUX FUEL COST, $	AVER RATE C/UNIT	UNIT COST C/SQFT	CHILLED WATER COST, $	AVER RATE C/TNHR	UNIT COST C/SQFT	STEAM OR HOT WATER COST, $	AVER RATE $/MMB	UNI COS C/SQF
SYSTEM 1 RUN1 CONSTANT VOL W/REHT															
JAN	1549.	2.675	6.696	0.	.00	.000	616.	36.00	2.662						
FEB	1472.	2.746	6.364	0.	.00	.000	569.	36.00	2.459						
MAR	1606.	2.629	6.942	0.	.00	.000	387.	36.00	1.671						
APR	1640.	2.693	7.088	0.	.00	.000	255.	36.00	1.103						
MAY	1602.	2.632	6.925	0.	.00	.000	296.	36.00	1.279						
JUN	1607.	2.628	6.944	0.	.00	.000	233.	36.00	1.007						
JUL	1657.	2.591	7.161	0.	.00	.000	141.	36.00	.608						
AUG	1617.	2.620	6.990	0.	.00	.000	35.	36.00	.149						
SEP	1607.	2.628	6.944	0.	.00	.000	44.	36.00	.190						
OCT	1600.	2.633	6.918	0.	.00	.000	242.	36.00	1.046						
NOV	1489.	2.730	6.435	0.	.00	.000	321.	36.00	1.386						
DEC	1538.	2.684	6.649	0.	.00	.000	553.	36.00	2.392						
ANN	18985.	2.648	82.058	0.	.00	.000	3691.	36.00	15.952						

TOTAL UTILITY COST

NO	NAME	$
1	RUN1 CONSTANT VOL W/REHT	22676.

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* NAVFAC * RUN NO 15* CONST VOL-REHT * SINGLE,NORTH-SOUTH GLASS * DATE 101674 PAGE 37

KLING-LINDQUIST * NAVFAC ENERGY STUDY * RUN NO. 1 * JOB NO. 1355-00 * CARPENTER ZONE ALL

* ROOM * MEASURE	ROOM AREA SF	ROOM VOL CF	WALL AREA SF	GLASS AREA SF	NET WALL AREA SF	PCT GLASS TO WALL
	23136.	198558.	8139.	1755.	6381.	21.55
* SUMMER * SENSIBLE	SENSIBLE HEAT BTUH	SUPPLY AIR CFM	DEHUMID AIR CFM	FRESH AIR CFM	SUPPLY AIR RATE CFM/SF	SUPPLY AIR CHANGE MIN
	765990.	(35370) 36,716 (Undiversified)	30756.	2371.	1.55	5.61
	FRESH AIR CHANGE MIN	SENSIBLE HEAT TONS	SENS HEAT RATE SF/TON	FRESH AIR HEAT BTUH		
	65.74	63.67	363.40	33289.00		
		GLASS HEAT BTUH	WALL HEAT BTUH	ROOF HEAT BTUH	FLOOR HEAT BTUH	
		17820.01	3205.74	10474.52	.00	
		PARTITION HEAT BTUH	LIGHT HEAT BTUH	MISCEL. HEAT BTUH	PEOPLE HEAT BTUH	
		3267.40	254866.17	254966.85	116000.00	
* SUMMER * LATENT	LATENT HEAT BTUH	LATENT HEAT TONS	FRESH AIR HEAT BTUH			
	102969.	6.51	42156.			
* SUMMER * TOTAL	TOTAL S+L HEAT BTUH	SENSIBLE HEAT RATIO	FRESH AIR PEN BTUH	FRESH AIR PEN TONS	TOT S+L+FAP HEAT TONS	TOT S+L+FAP RATE SF/TON
	868059.	.88	75445.	6.29	78.46	294.88
* SUMMER * SUBTOTAL	SKIN HEAT LOAD BTUH	SOLAR HEAT ONLY BTUH	NUMBER OF PEOPLE			
	127582.	96074.	464.			
* WINTER * HEAT LOAD	TRANSM HEAT BTUH	VENTIL HEAT BTUH	TOTAL HEAT BTUH			
	237358.	138277.	375635.			

NUMBER OF ROOMS IN BUILDING ORIGINAL 39
DUPLICATE 0
TOTAL 39

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EQUIPMENT ENERGY CONSUMPTION AND OPERATING DATA FOR

MUNIS NAVFAC ENERGY ANALYSIS.CONSTANT VOL.W/REHEAT.PERIMETER RAD.N=S SINGL GL.

SYSTEM 1S CONSTANT VOLUME WITH REHEAT PERIMETER RADIATION N=S SINGLE GLASS

FUEL AND POWER CONSUMPTION

SYS	GAS USAGE MCF	PEAK DAY GAS MCF	ELECTRIC USAGE KWH	PEAK ELEC DEMAND (30 MIN)	AUXILIARY FUEL USAGE
** JAN **					
1	0.	0.	45740.	165.	1988.
** FEB **					
1	0.	0.	41494.	180.	1951.
** MAR **					
1	0.	0.	47318.	253.	1343.
** APR **					
1	0.	0.	51852.	256.	795.
** MAY **					
1	0.	0.	58457.	250.	858.
** JUN **					
1	0.	0.	61060.	252.	686.
** JUL **					
1	0.	0.	63942.	251.	407.
** AUG **					
1	0.	0.	61723.	250.	96.
** SEP **					
1	0.	0.	59508.	251.	144.
** OCT **					
1	0.	0.	51913.	255.	796.
** NOV **					
1	0.	0.	45329.	249.	969.
** DEC **					
1	0.	0.	46451.	242.	1650.
** ANN **					
1	0.	0.	634786.	256.	11682.

MONTHLY AND ANNUAL UTILITY COST FOR

RUN 15 CONSTANT VOL W.REHEAT.PERIMETER RAD.N-S SINGLE GLASS

	ELECTRIC COST, $	AVER RATE C/KWH	UNIT COST C/SQFT	GAS COST, $	AVER RATE C/MCF	UNIT COST C/SQFT	AUX FUEL COST, $	AVER RATE C/UNIT	UNIT COST C/SQFT	CHILLED WATER COST, $	AVER RATE C/TNHR	UNIT COST C/SQFT	STEAM OR HOT WATER COST, $	AVER RATE $/MMB	UNIT COST C/SQFT
SYSTEM 1 RUN1 CONSTANT VOL W/REHT															
JAN	1332.	2.912	5.757	0.	.00	.000	716.	36.00	3.093						
FEB	1256.	3.027	5.429	0.	.00	.000	702.	36.00	3.036						
MAR	1364.	2.863	5.897	0.	.00	.000	483.	36.00	2.090						
APR	1458.	2.812	6.302	0.	.00	.000	286.	36.00	1.237						
MAY	1559.	2.667	6.750	0.	.00	.000	309.	36.00	1.335						
JUN	1605.	2.629	6.939	0.	.00	.000	247.	36.00	1.067						
JUL	1657.	2.591	7.161	0.	.00	.000	147.	36.00	.635						
AUG	1617.	2.620	6.990	0.	.00	.000	35.	36.00	.149						
SEP	1576.	2.651	6.819	0.	.00	.000	52.	36.00	.224						
OCT	1455.	2.802	6.288	0.	.00	.000	287.	36.00	1.239						
NOV	1325.	2.922	5.725	0.	.00	.000	349.	36.00	1.508						
DEC	1345.	2.695	5.812	0.	.00	.000	594.	36.00	2.567						
ANN	17550.	2.765	75.856	0.	.00	.000	4206.	36.00	18.179						

TOTAL UTILITY COST

NO	NAME	$
1	RUN1 CONSTANT VOL W/REHT	21756.

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KLING-LINDQUIST * NAVFAC ENERGY STUDY * RUN NO. 2 * JOB NO. 1335-00 * CARPENTER ZONE ALL

* ROOM * MEASURE	ROOM AREA SF 23136.	ROOM VOL CF 198558.	WALL AREA SF 8134.	GLASS AREA SF 1753.	NET WALL AREA SF 6381.	PCT GLASS TO WALL 21.55
* SUMMER * SENSIBLE	SENSIBLE HEAT BTUH 790384.	SUPPLY AIR CFM (36592.) 39,444 (Undiversified)	DEHUMID AIR CFM 31819.	FRESH AIR CFM 2371.	SUPPLY AIR RATE CFM/SF 1.58	SUPPLY AIR CHANGE MIN 5.43
	FRESH AIR CHANGE MIN 83.74	SENSIBLE HEAT TONS 65.87	SENS HEAT RATE SF/TON 351.26	FRESH AIR HEAT BTUH 33289.00		
		GLASS HEAT BTUH 21789.79	WALL HEAT BTUH 6941.27	ROOF HEAT BTUH 23686.93	FLOOR HEAT BTUH .00	
		PARTITION HEAT BTUH 3267.40	LIGHT HEAT BTUH 254866.17	MISCEL. HEAT BTUH 254966.85	PEOPLE HEAT BTUH 116000.00	
* SUMMER * LATENT	LATENT HEAT BTUH 102069.	LATENT HEAT TONS 8.51	FRESH AIR HEAT BTUH 42156.			
* SUMMER * TOTAL	TOTAL S+L HEAT BTUH 892453.	SENSIBLE HEAT RATIO .89	FRESH AIR PEN BTUH 75445.	FRESH AIR PEN TONS 6.29	TOT S+L+FAP HEAT TONS 80.66	TOT S+L+FAP RATE SF/TON 286.84
* SUMMER * SUBTOTAL	SKIN HEAT LOAD BTUH 153977.	SOLAR HEAT ONLY BTUH 101559.	NUMBER OF PEOPLE 464.			
* WINTER * HEAT LOAD	TRANSM HEAT BTUH 237358.	VENTIL HEAT BTUH 138277.	TOTAL HEAT BTUH 375635.			

NUMBER OF ROOMS IN BUILDING ORIGINAL 39
DUPLICATE 0
TOTAL 39

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EQUIPMENT ENERGY CONSUMPTION AND OPERATING DATA FOR

RUN2 NAVFAC ENERGY ANALYSIS CONSTANT VOL WITH REHEAT PERIMETER RAD E-W SING GL

SYSTEM 1 CONSTANT VOLUME WITH REHEAT PERIMETER RADIATION E-W SINGLE GLASS

FUEL AND POWER CONSUMPTION

SYS	GAS USAGE MCF	PEAK DAY GAS MCF	ELECTRIC USAGE KWH	PEAK ELEC DEMAND (30 MIN)	AUXILIARY FUEL USAGE
** JAN **					
1	0.	0.	46892.	170.	1959.
** FEB **					
1	0.	0.	42542.	186.	1892.
** MAR **					
1	0.	0.	48110.	258.	1486.
** APR **					
1	0.	0.	52650.	258.	1173.
** MAY **					
1	0.	0.	60157.	258.	860.
** JUN **					
1	0.	0.	62803.	259.	689.
** JUL **					
1	0.	0.	65677.	259.	420.
** AUG **					
1	0.	0.	63525.	259.	711.
** SEP **					
1	0.	0.	61200.	259.	790.
** OCT **					
1	0.	0.	52928.	258.	1076.
** NOV **					
1	0.	0.	46334.	257.	1113.
** DEC **					
1	0.	0.	47619.	255.	1643.
** ANN **					
1	0.	0.	650437.	259.	13812.

MONTHLY AND ANNUAL UTILITY COST FOR

RUN2 CONSTANT VOL W,REHEAT PER,RADIATION E-W SINGLE GLASS

	ELECTRIC COST, $	AVER RATE C/KWH	UNIT COST C/SQFT	GAS COST, $	AVER RATE C/MCF	UNIT COST C/SQFT	AUX FUEL COST, $	AVER RATE C/UNIT	UNIT COST C/SQFT	CHILLED WATER COST, $	AVER RATE C/TNHR	UNIT COST C/SQFT	STEAM OR HOT WATER COST, $	AVER RATE $/MMB	UNI COS C/SQF
SYSTEM 1 RUN2 CONSTANT VOL W/REHT															
JAN	1364.	2.910	5.897	0.	.00	.000	705.	36.00	3.048						
FEB	1287.	3.024	5.561	0.	.00	.000	681.	36.00	2.944						
MAR	1386.	2.681	5.991	0.	.00	.000	535.	36.00	2.312						
APR	1467.	2.787	6.341	0.	.00	.000	422.	36.00	1.825						
MAY	1601.	2.662	6.920	0.	.00	.000	310.	36.00	1.338						
JUN	1648.	2.625	7.125	0.	.00	.000	248.	36.00	1.072						
JUL	1700.	2.588	7.346	0.	.00	.000	151.	36.00	.654						
AUG	1661.	2.615	7.180	0.	.00	.000	256.	36.00	1.106						
SEP	1620.	2.647	7.001	0.	.00	.000	284.	36.00	1.229						
OCT	1472.	2.781	6.363	0.	.00	.000	387.	36.00	1.674						
NOV	1354.	2.923	5.854	0.	.00	.000	401.	36.00	1.732						
DEC	1377.	2.692	5.953	0.	.00	.000	591.	36.00	2.557						
ANN	17938.	2.758	77.533	0.	.00	.000	4972.	36.00	21.492						

TOTAL UTILITY COST

NO	NAME	$
1	RUN2 CONSTANT VOL W/REHT	22910.

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NAVFAC   UN NO 3 # CONST VOL-REHT # DOUBLE;NORTH-SOUTH GLASS #   DATE 101674   PAGE 37

KLING-LINDQUIST # NAVFAC ENERGY STUDY # RUN NO. 3 # JOB NO. 1335-00 # CARPENTER   ZONE ALL

# ROOM # MEASURE	ROOM AREA SF	ROOM VOL CF	WALL AREA SF	GLASS AREA SF	NET WALL AREA SF	PCT GLASS TO WALL
	23130.	198558.	8134.	1753.	6381.	21.55
# SUMMER # SENSIBLE	SENSIBLE HEAT BTUH	SUPPLY AIR CFM	DEHUMID AIR CFM	FRESH AIR CFM	SUPPLY AIR RATE CFM/SF	SUPPLY AIR CHANGE MIN
	749433.	(34696.) 36,022 (Undiversified)	30170.	2371.	1.50	5.72
	FRESH AIR CHANGE MIN	SENSIBLE HEAT TONS	SENS HEAT RATE SF/TON	FRESH AIR HEAT BTUH		
	83.74	62.45	370.46	33289.00		
		GLASS HEAT BTUH	WALL HEAT BTUH	ROOF HEAT BTUH	FLOOR HEAT BTUH	
		10255.05	3205.74	10474.52	.00	
		PARTITION HEAT BTUH	LIGHT HEAT BTUH	MISCEL. HEAT BTUH	PEOPLE HEAT BTUH	
		3267.40	254866.17	254966.85	116000.00	
# SUMMER # LATENT	LATENT HEAT BTUH	LATENT HEAT TONS	FRESH AIR HEAT BTUH			
	102069.	8.51	42156.			
# SUMMER # TOTAL	TOTAL S+L HEAT BTUH	SENSIBLE HEAT RATIO	FRESH AIR PEN BTUH	FRESH AIR PEN TONS	TOT S+L+FAP HEAT TONS	TOT S+L+FAP RATE SF/TON
	851502.	.88	75445.	6.29	77.25	299.51
# SUMMER # SUBTOTAL	SKIN HEAT LOAD BTUH	SOLAR HEAT ONLY BTUH	NUMBER OF PEOPLE			
	113022.	89086.	464.			
# WINTER # HEAT LOAD	TRANSM HEAT BTUH	VENTIL HEAT BTUH	TOTAL HEAT BTUH			
	191919.	138277.	330196.			

NUMBER OF ROOMS IN BUILDING ORIGINAL 39
DUPLICATE 0
TOTAL 39

BEST AVAILABLE COPY

EQUIPMENT ENERGY CONSUMPTION AND OPERATING DATA FOR

RUN3 NAVFAC ENERGY ANALYSIS BASE BLDG CONS VOL REHEAT PER RAD N-8 INSULATING GL

SYSTEM 1 CONSTANT VOLUME WITH REHEAT PERIMETER RADIATION N-8 INSULATING GLASS

FUEL AND POWER CONSUMPTION

SYS	GAS USAGE MCF	PEAK DAY GAS MCF	ELECTRIC USAGE KWH	PEAK ELEC DEMAND (30 MIN)	AUXILIARY FUEL USAGE
** JAN **					
1	0.	0.	45511.	165.	1607.
** FEB **					
1	0.	0.	41285.	179.	1573.
** MAR **					
1	0.	0.	46996.	250.	1084.
** APR **					
1	0.	0.	51486.	254.	635.
** MAY **					
1	0.	0.	58095.	248.	657.
** JUN **					
1	0.	0.	60650.	249.	534.
** JUL **					
1	0.	0.	63512.	249.	316.
** AUG **					
1	0.	0.	61339.	248.	95.
** SEP **					
1	0.	0.	59140.	249.	129.
** OCT **					
1	0.	0.	51582.	253.	630.
** NOV **					
1	0.	0.	45046.	247.	785.
** DEC **					
1	0.	0.	46234.	243.	1335.
** ANN **					
1	0.	0.	630874.	254.	9380.

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RU.3 BASE BLDG CONS VOL REHEAT PER RADIA INSUL GLASS

	ELECTRIC COST, $	AVER RATE C/KWH	UNIT COST C/SQFT	GAS COST, $	AVER RATE C/MCF	UNIT COST C/SQFT	AUX FUEL COST, $	AVER RATE C/UNIT	UNIT COST C/SQFT	CHILLED WATER COST, $	AVER RATE C/TNHR	UNIT COST C/SQFT	STEAM OR HOT WATER COST, $	AVER RATE $/MMB	UNIT COST C/SQFT
SYSTEM 1 RUNS CONSTANT VOL W/REHE															
JAN	1525.	2.906	5.717	0.	.00	.000	579.	36.00	2.501						
FEB	1247.	3.021	5.391	0.	.00	.000	566.	36.00	2.448						
MAR	1353.	2.880	5.850	0.	.00	.000	390.	36.00	1.687						
APR	1451.	2.817	6.270	0.	.00	.000	229.	36.00	.988						
MAY	1547.	2.665	6.688	0.	.00	.000	237.	36.00	1.022						
JUN	1593.	2.626	6.885	0.	.00	.000	192.	36.00	.831						
JUL	1644.	2.589	7.106	0.	.00	.000	114.	36.00	.492						
AUG	1605.	2.617	6.938	0.	.00	.000	34.	36.00	.148						
SEP	1566.	2.648	6.769	0.	.00	.000	46.	36.00	.201						
OCT	1448.	2.607	6.259	0.	.00	.000	227.	36.00	.980						
NOV	1314.	2.918	5.681	0.	.00	.000	283.	36.00	1.221						
DEC	1336.	2.889	5.773	0.	.00	.000	481.	36.00	2.077						
ANN	17428.	2.762	75.327	0.	.00	.000	3377.	36.00	14.595						

TOTAL UTILITY COST

NO	NAME	$
	RUNS CONSTANT VOL W/REHE	20804.

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BLING-LI JST * NAVFAC ENERGY STUDY * RUN NO. 4 * JOB NO. 1334 * CARPENTER ZONE ALL

BEST AVAILABLE COPY

* ROOM * MEASURE	ROOM AREA SF	ROOM VOL CF	WALL AREA SF	GLASS AREA SF	NET WALL AREA SF	PCT GLASS TO WALL
	23136.	198558.	8134.	70.	8064.	.86
* SUMMER * SENSIBLE	SENSIBLE HEAT BTUH	SUPPLY AIR CFM	DEHUMID AIR CFM	FRESH AIR CFM	SUPPLY AIR RATE CFM/SF	SUPPLY AIR CHANGE MIN
	666450.	(30854.) 31,072 (Undiversified)	26830.	2371.	1.33	6.44
	FRESH AIR CHANGE MIN	SENSIBLE HEAT TONS	SENS HEAT RATE SF/TON	FRESH AIR HEAT BTUH		
	83.74	55.54	416.58	33289.00		
		GLASS HEAT BTUH	WALL HEAT BTUH	ROOF HEAT BTUH	FLOOR HEAT BTUH	
		870.10	6024.66	23686.93	.00	
		PARTITION HEAT BTUH	LIGHT HEAT BTUH	MISCEL. HEAT BTUH	PEOPLE HEAT BTUH	
		3267.40	254866.17	254966.85	116000.00	
* SUMMER * LATENT	LATENT HEAT BTUH	LATENT HEAT TONS	FRESH AIR HEAT BTUH			
	96018.	8.00	42156.			
* SUMMER * TOTAL	TOTAL S+L HEAT BTUH	SENSIBLE HEAT RATIO	FRESH AIR PEN BTUH	FRESH AIR PEN TONS	TOT S+L+FAP HEAT TONS	TOT S+L+FAP RATE SF/TON
	762468.	.87	75445.	6.29	69.83	331.34
* SUMMER * SUBTOTAL	SKIN HEAT LOAD BTUH	SOLAR HEAT ONLY BTUH	NUMBER OF PEOPLE			
	34814.	4233.	464.			
* WINTER * HEAT LOAD	TRANSM HEAT BTUH	VENTIL HEAT BTUH	TOTAL HEAT BTUH			
	122123.	138277.	260400.			

NUMBER OF ROOMS IN BUILDING	ORIGINAL	39
	DUPLICATE	0
	TOTAL	39

EQUIPMENT ENERGY CONSUMPTION AND OPERATING DATA FOR

RUN4 NAVFAC ENERGY ANALYSIS BASE BLDG CONS VOL REHEAT PER RADIATION NO GLASS

SYSTEM 1 CONSTANT VOLUME WITH REHEAT PERIMETER RADIATION NO GLASS N=8 GLASS

FUEL AND POWER CONSUMPTION

SYS	GAS USAGE MCF	PEAK DAY GAS MCF	ELECTRIC USAGE KWH	PEAK ELEC DEMAND (30 MIN)	AUXILIARY FUEL USAGE
** JAN **					
1	0.	0.	44546.	161.	1110.
** FEB **					
1	0.	0.	40408.	176.	1078.
** MAR **					
1	0.	0.	45454.	236.	831.
** APR **					
1	0.	0.	49601.	236.	610.
** MAY **					
1	0.	0.	55803.	236.	327.
** JUN **					
1	0.	0.	58183.	237.	256.
** JUL **					
1	0.	0.	60901.	237.	146.
** AUG **					
1	0.	0.	58926.	237.	248.
** SEP **					
1	0.	0.	56729.	242.	250.
** OCT **					
1	0.	0.	49542.	238.	535.
** NOV **					
1	0.	0.	43485.	235.	600.
** DEC **					
1	0.	0.	45254.	236.	925.
** ANN **					
1	0.	0.	608832.	242.	6914.

BEST AVAILABLE COPY

MONTHLY AND ANNUAL UTILITY COST FOR

RUN4 BASE BLDG CONS VOL REHEAT PER RAD NO GLASS

	ELECTRIC COST, $	AVER RATE C/KWH	UNIT COST C/SQFT	GAS COST, $	AVER RATE C/MCF	UNIT COST C/SQFT	AUX FUEL COST, $	AVER RATE C/UNIT	UNIT COST C/SQFT	CHILLED WATER COST, $	AVER RATE C/TNHR	UNIT COST C/SQFT	STEAM OR HOT WATER COST, $	AVER RATE $/MMB	UNIT COST C/SQFT
SYSTEM 1 RUN4 CONSTANT VOL W/REHT															
JAN	1294.	2.904	5.591	0.	.00	.000	400.	36.00	1.727						
FEB	1220.	3.017	5.272	0.	.00	.000	388.	36.00	1.677						
MAR	1310.	2.882	5.661	0.	.00	.000	299.	36.00	1.293						
APR	1384.	2.790	5.981	0.	.00	.000	220.	36.00	.949						
MAY	1495.	2.678	6.460	0.	.00	.000	118.	36.00	.509						
JUN	1537.	2.642	6.643	0.	.00	.000	92.	36.00	.398						
JUL	1586.	2.603	6.853	0.	.00	.000	53.	36.00	.227						
AUG	1550.	2.631	6.701	0.	.00	.000	89.	36.00	.386						
SEP	1511.	2.664	6.531	0.	.00	.000	90.	36.00	.389						
OCT	1383.	2.791	5.977	0.	.00	.000	193.	36.00	.832						
NOV	1275.	2.931	5.509	0.	.00	.000	216.	36.00	.934						
DEC	1306.	2.886	5.646	0.	.00	.000	335.	36.00	1.439						
ANN	16849.	2.767	72.826	0.	.00	.000	2490.	36.00	10.761						

TOTAL UTILITY COST

NO	NAME	$
1	RUN4 CONSTANT VOL W/REHT	19339.

BEST AVAILABLE COPY

KLING-LINDQUIST * NAVFAC ENERGY STUDY * RUN NO. 5 * JOB NO. 1335-00 * CARPENTER ZONE ALL

* ROOM * MEASURE	ROOM AREA SF	ROOM VOL CF	WALL AREA SF	GLASS AREA SF	NET WALL AREA SF	PCT GLASS TO WALL
	23136.	198558.	8134.	1753.	6381.	21.55
* SUMMER * SENSIBLE	SENSIBLE HEAT BTUH	SUPPLY AIR CFM	DEHUMID AIR CFM	FRESH AIR CFM	SUPPLY AIR RATE CFM/SF	SUPPLY AIR CHANGE MIN
	771295.	(35708.) 37,054 (Undiversified)	31051.	4790.	1.54	5.56
	FRESH AIR CHANGE MIN	SENSIBLE HEAT TONS	SENS HEAT RATE SF/TON	FRESH AIR HEAT BTUH		
	41.80	64.27	359.96	66686.00		
		GLASS HEAT BTUH	WALL HEAT BTUH	ROOF HEAT BTUH	FLOOR HEAT BTUH	
		17828.01	3205.74	10474.52	.00	
		PARTITION HEAT BTUH	LIGHT HEAT BTUH	MISCEL. HEAT BTUH	PEOPLE HEAT BTUH	
		3267.40	254866.17	254966.85	116000.00	
* SUMMER * LATENT	LATENT HEAT BTUH	LATENT HEAT TONS	FRESH AIR HEAT BTUH			
	111332.	9.28	84445.			
* SUMMER * TOTAL	TOTAL S+L HEAT BTUH	SENSIBLE HEAT RATIO	FRESH AIR PEN BTUH	FRESH AIR PEN TONS	TOT S+L+FAP HEAT TONS	TOT S+L+FAP RATE SF/TON
	882627.	.87	151131.	12.59	86.15	268.57
* SUMMER * SUBTOTAL	SKIN HEAT LOAD BTUH	SOLAR HEAT ONLY BTUH	NUMBER OF PEOPLE			
	127582.	96074.	464.			
* WINTER * HEAT LOAD	TRANSM HEAT BTUH	VENTIL HEAT BTUH	TOTAL HEAT BTUH			
	267683.	277020.	544703.			

NUMBER OF ROOMS IN BUILDING ORIGINAL 39
DUPLICATE 0
TOTAL 39

EQUIPMENT ENERGY CONSUMPTION AND OPERATING DATA FOR

RH5R NAVFAC ENERGY ANALYSIS BASE BLDG CONS VOL REHT PER RAD DOUBLE INFIL=VENTIL

SYSTEM 1 CONSTANT VOLUME WITH REHEAT PERIMETER RADIATION SINGLE GLASS N=3 GLASS

FUEL AND POWER CONSUMPTION

SYS	GAS USAGE MCF	PEAK DAY GAS MCF	ELECTRIC USAGE KWH	PEAK ELEC DEMAND (30 MIN)	AUXILIARY FUEL USAGE
** JAN **					
1	0.	0.	45858.	165.	2099.
** FEB **					
1	0.	0.	41605.	182.	2054.
** MAR **					
1	0.	0.	47668.	262.	1450.
** APR **					
1	0.	0.	52359.	263.	916.
** MAY **					
1	0.	0.	59283.	260.	941.
** JUN **					
1	0.	0.	62052.	261.	765.
** JUL **					
1	0.	0.	65976.	261.	530.
** AUG **					
1	0.	0.	63724.	260.	96.
** SEP **					
1	0.	0.	61369.	260.	180.
** OCT **					
1	0.	0.	52354.	262.	858.
** NOV **					
1	0.	0.	45295.	254.	1019.
** DEC **					
1	0.	0.	46563.	248.	1800.
** ANN **					
1	0.	0.	644105.	263.	12706.

BEST AVAILABLE COPY

MONTHLY, AND ANNUAL UTILITY COST FOR

RNSR BASE BLDG CONS VOL REHT PER RAD DOUBLE INFIL-VENTIL

	ELECTRIC COST, $	AVER RATE C/KWH	UNIT COST C/SQFT	GAS COST, $	AVER RATE C/MCF	UNIT COST C/SQFT	AUX FUEL COST, $	AVER RATE C/UNIT	UNIT COST C/SQFT	CHILLED WATER COST, $	AVER RATE C/TNHR	UNIT COST C/SQFT	STEAM OR HOT WATER COST, $	AVER RATE $/MMB	UNI COS C/SQF
SYSTEM 1	RNSR CONSTANT VOL W/REHT														
JAN	1349.	2.942	5.832	0.	.00	.000	756.	36.00	3.266						
FEB	1273.	3.060	5.503	0.	.00	.000	739.	36.00	3.196						
MAR	1386.	2.907	5.990	0.	.00	.000	522.	36.00	2.256						
APR	1474.	2.815	6.370	0.	.00	.000	330.	36.00	1.425						
MAY	1589.	2.680	6.868	0.	.00	.000	339.	36.00	1.464						
JUN	1638.	2.640	7.081	0.	.00	.000	275.	36.00	1.187						
JUL	1708.	2.589	7.384	0.	.00	.000	191.	36.00	.825						
AUG	1668.	2.618	7.210	0.	.00	.000	35.	36.00	.149						
SEP	1626.	2.650	7.029	0.	.00	.000	65.	36.00	.280						
OCT	1469.	2.607	6.351	0.	.00	.000	309.	36.00	1.335						
NOV	1339.	2.957	5.788	0.	.00	.000	367.	36.00	1.586						
DEC	1362.	2.925	5.886	0.	.00	.000	648.	36.00	2.801						
ANN	17883.	2.776	77.293	0.	.00	.000	4574.	36.00	19.771						

TOTAL UTILITY COST

NO	NAME	$
1	RNSR CONSTANT VOL W/REHT	22457.

BEST AVAILABLE COPY

KLING-LINDQUIST * NAVFAC ENERGY STUDY * RUN NO. 6 * JOB NO. 1335-00 * CARPENTER ZONE ALL

* ROOM * MEASURE	ROOM AREA SF 23136.	ROOM VOL CF 198558.	WALL AREA SF 8134.	GLASS AREA SF 1753.	NET WALL AREA SF 6381.	PCT GLASS TO WALL 21.55
* SUMMER * SENSIBLE	SENSIBLE HEAT BTUH 693549.	SUPPLY AIR CFM 32109.	DEHUMID AIR CFM 27921.	FRESH AIR CFM 2371.	SUPPLY AIR RATE CFM/SF 1.39	SUPPLY AIR CHANGE MIN 6.18
	FRESH AIR CHANGE MIN 83.74	SENSIBLE HEAT TONS 57.80	SENS HEAT RATE SF/TON 400.31	FRESH AIR HEAT BTUH 33289.00		
		GLASS HEAT BTUH 9904.45	WALL HEAT BTUH 1914.70	ROOF HEAT BTUH 1133.03	FLOOR HEAT BTUH .00	
		PARTITION HEAT BTUH 3267.40	LIGHT HEAT BTUH 191149.62	MISCEL. HEAT BTUH 254936.85	PEOPLE HEAT BTUH 116000.00	
* SUMMER * LATENT	LATENT HEAT BTUH 102069.	LATENT HEAT TONS 8.51	FRESH AIR HEAT BTUH 42156.			
* SUMMER * TOTAL	TOTAL S+L HEAT BTUH 795618.	SENSIBLE HEAT RATIO .87	FRESH AIR PEN BTUH 75445.	FRESH AIR PEN TONS 6.29	TOT S+L+FAP HEAT TONS 72.59	TOT S+L+FAP RATE SF/TON 318.73
* SUMMER * SUBTOTAL	SKIN HEAT LOAD BTUH 120858.	SOLAR HEAT ONLY BTUH 107905.	NUMBER OF PEOPLE 464.			
* WINTER * HEAT LOAD	TRANSM HEAT BTUH 212853.	VENTIL HEAT BTUH 138277.	TOTAL HEAT BTUH 351130.			

NUMBER OF ROOMS IN BUILDING	ORIGINAL	39
	DUPLICATE	0
	TOTAL	39

EQUIPMENT ENERGY CONSUMPTION AND OPERATING DATA FOR

RN6R NAVFAC ENERGY ANALYSIS BASE BLDG VARIABLE VOLUME WITH PERIMETER RADIATION

SYSTEM 1 BASE BUILDING VARIABLE VOLUME PERIMETER RADIATION

FUEL AND POWER CONSUMPTION

SYS	GAS USAGE MCF	PEAK DAY GAS MCF	ELECTRIC USAGE KWH	PEAK ELEC DEMAND (30 MIN)	AUXILIARY FUEL USAGE
** JAN **					
1	0.	0.	45970.	168.	1243.
** FEB **					
1	0.	0.	41799.	184.	1210.
** MAR **					
1	0.	0.	47406.	258.	958.
** APR **					
1	0.	0.	51672.	255.	576.
** MAY **					
1	0.	0.	52405.	250.	189.
** JUN **					
1	0.	0.	55409.	250.	116.
** JUL **					
1	0.	0.	60242.	251.	98.
** AUG **					
1	0.	0.	61750.	250.	96.
** SEP **					
1	0.	0.	59541.	257.	122.
** OCT **					
1	0.	0.	50735.	255.	506.
** NOV **					
1	0.	0.	43853.	253.	668.
** DEC **					
1	0.	0.	45505.	215.	1063.
** ANN **					
1	0.	0.	616286.	258.	6845.

BEST AVAILABLE COPY

MONTHLY AND ANNUAL UTILITY COST FOR

RNGR BASE BLDG VARIABLE VOLUME WITH PERIMETER RADIATION

	ELECTRIC COST, $	AVER RATE C/KWH	UNIT COST C/SQFT	GAS COST, $	AVER RATE C/MCF	UNIT COST C/SQFT	AUX FUEL COST, $	AVER RATE C/UNIT	UNIT COST C/SQFT	CHILLED WATER COST, $	AVER RATE C/TNHR	UNIT COST C/SQFT	STEAM OR HOT WATER COST, $	AVER RATE $/MMB	UNIT COST C/SQFT
SYSTEM 1 RNGR BASE BLG VAR VOL															
JAN	1344.	2.925	5.8[illegible]	0.	.00	.000	447.	36.00	1.934						
FEB	1270.	3.038	5.4[illegible]	0.	.00	.000	436.	36.00	1.883						
MAR	1374.	2.899	5.9[illegible]	0.	.00	.000	345.	36.00	1.491						
APR	1446.	2.799	6.251	0.	.00	.000	207.	36.00	.896						
MAY	1459.	2.785	6.308	0.	.00	.000	68.	36.00	.294						
JUN	1513.	2.731	6.539	0.	.00	.000	42.	36.00	.180						
JUL	1599.	2.655	6.912	0.	.00	.000	35.	36.00	.152						
AUG	1626.	2.633	7.029	0.	.00	.000	35.	36.00	.149						
SEP	1587.	2.665	6.858	0.	.00	.000	44.	36.00	.190						
OCT	1429.	2.818	6.179	0.	.00	.000	182.	36.00	.787						
NOV	1307.	2.980	5.648	0.	.00	.000	240.	36.00	1.039						
DEC	1336.	2.936	5.775	0.	.00	.000	383.	36.00	1.654						
ANN	[illegible]2[illegible]	2.806	74.740	0.	.00	.000	2464.	36.00	1[illegible].651						

[illegible]TAL UTILITY COST

NO	NAME	$
1	RNGR BASE BLG VAR VOL	19756.

BEST AVAILABLE COPY

KLING-LINDQUIST * NAVFAC ENERGY STUDY * RUN NO. 1 * JOB NO. 1335-00 * CARPENTER ZONE ALL

* ROOM * MEASURE	ROOM AREA SF	ROOM VOL CF	WALL AREA SF	GLASS AREA SF	NET WALL AREA SF	PCT GLASS TO WALL
	23136.	198558.	8134.	1753.	6381.	21.55
* SUMMER * SENSIBLE	SENSIBLE HEAT BTUH	SUPPLY AIR CFM	DEHUMID AIR CFM	FRESH AIR CFM	SUPPLY AIR RATE CFM/SF	SUPPLY AIR CHANGE MIN
	763990.	(35370) 36,716 (Undiversified)	30756.	2371.	1.53	5.61
	FRESH AIR CHANGE MIN	SENSIBLE HEAT TONS	SENS HEAT RATE SF/TON	FRESH AIR HEAT BTUH		
	85.74	63.67	363.40	33289.00		
		GLASS HEAT BTUH	WALL HEAT BTUH	ROOF HEAT BTUH	FLOOR HEAT BTUH	
		17828.01	3205.74	10474.52	.00	
		PARTITION HEAT BTUH	LIGHT HEAT BTUH	MISCEL. HEAT BTUH	PEOPLE HEAT BTUH	
		3267.40	254866.17	254966.85	116000.00	
* SUMMER * LATENT	LATENT HEAT BTUH	LATENT HEAT TONS	FRESH AIR HEAT BTUH			
	102069.	8.51	42156.			
* SUMMER * TOTAL	TOTAL S+L HEAT BTUH	SENSIBLE HEAT RATIO	FRESH AIR PEN BTUH	FRESH AIR PEN TONS	TOT S+L+FAP HEAT TONS	TOT S+L+FAP RATE SF/TON
	866059.	.88	75445.	6.29	78.46	294.88
* SUMMER * SUBTOTAL	SKIN HEAT LOAD BTUH	SOLAR HEAT ONLY BTUH	NUMBER OF PEOPLE			
	127582.	96074.	464.			
* WINTER * HEAT LOAD	TRANSM HEAT BTUH	VENTIL HEAT BTUH	TOTAL HEAT BTUH			
	237358.	138277.	375635.			

NUMBER OF ROOMS IN BUILDING	ORIGINAL	39
	DUPLICATE	0
	TOTAL	39

BEST AVAILABLE COPY

EQUIPMENT ENERGY CONSUMPTION AND OPERATING DATA FOR

RN7S NAVFAC ENERGY ANALYSIS BASE BLDG CONS VOL REHEAT PER RAD DOUBLE BUNDLE CON

SYSTEM 1 BASE BLDG CONS VOL WITH REHEAT PER RADIATION DOUBLE BUNDLE MACHINES SINGLE GL

FUEL AND POWER CONSUMPTION

SYS	GAS USAGE MCF	PEAK DAY GAS MCF	ELECTRIC USAGE KWH	PEAK ELEC DEMAND (30 MIN)	AUXILIARY FUEL USAGE
** JAN **					
1	0.	0.	50078.	220.	979.
** FEB **					
1	0.	0.	45279.	222.	968.
** MAR **					
1	0.	0.	50445.	249.	800.
** APR **					
1	0.	0.	54490.	249.	456.
** MAY **					
1	0.	0.	59015.	250.	86.
** JUN **					
1	0.	0.	60929.	251.	29.
** JUL **					
1	0.	0.	63857.	251.	23.
** AUG **					
1	0.	0.	61709.	250.	26.
** SEP **					
1	0.	0.	59920.	251.	48.
** OCT **					
1	0.	0.	54029.	249.	375.
** NOV **					
1	0.	0.	48488.	249.	518.
** DEC **					
1	0.	0.	50367.	243.	853.
** ANN **					
1	0.	0.	658606.	251.	5161.

BEST AVAILABLE COPY

MONTHLY AND ANNUAL UTILITY COST FOR

RN7S BASE BLDG CONS VOL REHEAT PER RAD DOUBLE BUNDLE COND

	ELECTRIC COST, $	AVER RATE C/KWH	UNIT COST C/SQFT	GAS COST, $	AVER RATE C/MCF	UNIT COST C/SQFT	AUX FUEL COST, $	AVER RATE C/UNIT	UNIT COST C/SQFT	CHILLED WATER COST, $	AVER RATE C/TNHR	UNIT COST C/SQFT	STEAM OR HOT WATER COST, $	AVER RATE $/MMB	UNIT COST C/SQFT
SYSTEM 1 RN7S BASE BLDG DBL BNDL															
JAN	1408.	2.811	6.064	0.	.00	.000	352.	36.00	1.523						
FEB	1322.	2.920	5.714	0.	.00	.000	348.	36.00	1.506						
MAR	1414.	2.803	6.112	0.	.00	.000	288.	36.00	1.245						
APR	1486.	2.728	6.425	0.	.00	.000	164.	36.00	.710						
MAY	1567.	2.656	6.774	0.	.00	.000	31.	36.00	.134						
JUN	1601.	2.628	6.921	0.	.00	.000	10.	36.00	.045						
JUL	1654.	2.590	7.147	0.	.00	.000	8.	36.00	.036						
AUG	1615.	2.618	6.982	0.	.00	.000	9.	36.00	.040						
SEP	1583.	2.642	6.844	0.	.00	.000	17.	36.00	.075						
OCT	1478.	2.736	6.389	0.	.00	.000	135.	36.00	.584						
NOV	[illegible]	2.644	5.961	0.	.00	.000	186.	36.00	.806						
DEC	[illegible]	2.605	6.196	0.	.00	.000	307.	36.00	1.327						
ANN	17921.	2.721	77.459	0.	.00	.000	1858.	36.00	8.036						

TOTAL UTILITY COST

NO	NAME	$
	RN7S BASE BLDG DBL BNDL	19779.

BEST AVAILABLE COPY

KLING-LINDQUIST * NAVFAC ENERGY STUDY * RUN NO. 1 * JOB NO. 1335-00 * CARPENTER ZONE ALL

* ROOM * MEASURE	ROOM AREA SF 23136.	ROOM VOL CF 198558.	WALL AREA SF 8164.	GLASS AREA SF 4368.	NET WALL AREA SF 3796.	PCT GLASS TO WALL 53.50
* SUMMER * SENSIBLE	SENSIBLE HEAT BTUH 1012505.	SUPPLY AIR CFM (46866.) 50,529 (Undiversified)	DEHUMID AIR CFM 40753.	FRESH AIR CFM 2371.	SUPPLY AIR RATE CFM/SF 2.03	SUPPLY AIR CHANGE MIN 4.24
	FRESH AIR CHANGE MIN 65.74	SENSIBLE HEAT TONS 84.36	SENS HEAT RATE SF/TON 274.26	FRESH AIR HEAT BTUH 33289.00		
		GLASS HEAT BTUH 44422.56	WALL HEAT BTUH 5166.69	ROOF HEAT BTUH 37708.26	FLOOR HEAT BTUH .00	
		PARTITION HEAT BTUH 3267.40	LIGHT HEAT BTUH 254866.17	MISCEL. HEAT BTUH 254966.85	PEOPLE HEAT BTUH 116000.00	
* SUMMER * LATENT	LATENT HEAT BTUH 134257.	LATENT HEAT TONS 11.19	FRESH AIR HEAT BTUH 80456.			
* SUMMER * TOTAL	TOTAL S+L HEAT BTUH 1146560.	SENSIBLE HEAT RATIO .88	FRESH AIR PEN BTUH 113745.	FRESH AIR PEN TONS 9.48	TOT S+L+FAP HEAT TONS 105.03	TOT S+L+FAP RATE SF/TON 220.29
* SUMMER * SUBTOTAL	SKIN HEAT LOAD BTUH 362281.	SOLAR HEAT ONLY BTUH 274983.	NUMBER OF PEOPLE 464.			
* WINTER * HEAT LOAD	TRANSM HEAT BTUH 566820.	VENTIL HEAT BTUH 138277.	TOTAL HEAT BTUH 705097.			

NUMBER OF ROOMS IN BUILDING	ORIGINAL	39
	DUPLICATE	0
	TOTAL	39

BEST AVAILABLE COPY

EQUIPMENT ENERGY CONSUMPTION AND OPERATING DATA FOR

RUN 8 NAVFAC ENERGY ANALYSIS.CONSTANT VOL.WITH REHEAT.PERIMETER RAD.N-S SING.GL.

SYSTEM 1 CONSTANT VOLUME WITH REHEAT.PERIMETER RADIATION N-S SINGLE GLASS EXISTNG BLDG

FUEL AND POWER CONSUMPTION

SYS	GAS USAGE MCF	PEAK DAY GAS MCF	ELECTRIC USAGE KWH	PEAK ELEC DEMAND (30 MIN)	AUXILIARY FUEL USAGE
** JAN **					
1	0.	0.	53015.	195.	4120.
** FEB **					
1	0.	0.	48121.	219.	4017.
** MAR **					
1	0.	0.	55431.	316.	2725.
** APR **					
1	0.	0.	61686.	316.	1638.
** MAY **					
1	0.	0.	69027.	313.	1749.
** JUN **					
1	0.	0.	72845.	315.	1484.
** JUL **					
1	0.	0.	77660.	315.	1284.
** AUG **					
1	0.	0.	74572.	312.	124.
** SEP **					
1	0.	0.	71927.	313.	240.
** OCT **					
1	0.	0.	61702.	314.	1513.
** NOV **					
1	0.	0.	53469.	311.	1959.
** DEC **					
1	0.	0.	53733.	298.	3884.
** ANN **					
1	0.	0.	753189.	316.	24736.

MONTHLY AND ANNUAL UTILITY COST FOR

RUN 5 CONSTANT VOL W.REHEAT PER.RADIATION SINGLE GLASS N=S

	ELECTRIC COST, $	AVER RATE C/KWH	UNIT COST C/SQFT	GAS COST, $	AVER RATE C/MCF	UNIT COST C/SQFT	AUX FUEL COST, $	AVER RATE C/UNIT	UNIT COST C/SQFT	CHILLED WATER COST, $	AVER RATE C/TNHR	UNIT COST C/SQFT	STEAM OR HOT WATER COST, $	AVER RATE $/MMB	UN CO C/SQ
SYSTEM 1	RH1E CONSTANT VOL W/REHT														
JAN	1569.	2.959	.678	0.	.00	.000	1483.	36.00	.641						
FEB	1478.	3.071	.639	0.	.00	.000	1446.	36.00	.625						
MAR	1616.	2.915	.698	0.	.00	.000	981.	36.00	.424						
APR	1728.	2.801	.747	0.	.00	.000	590.	36.00	.255						
MAY	1855.	2.687	.802	0.	.00	.000	630.	36.00	.272						
JUN	1923.	2.639	.831	0.	.00	.000	534.	36.00	.231						
JUL	2009.	2.586	.868	0.	.00	.000	462.	36.00	.200						
AUG	1954.	2.620	.844	0.	.00	.000	45.	36.00	.019						
SEP	1906.	2.650	.824	0.	.00	.000	86.	36.00	.037						
OCT	1724.	2.794	.745	0.	.00	.000	545.	36.00	.235						
NOV	1577.	2.949	.682	0.	.00	.000	705.	36.00	.305						
DEC	1581.	2.943	.684	0.	.00	.000	1398.	36.00	.604						
ANN	20918.	2.777	9.041	0.	.00	.000	8905.	36.00	3.849						

TOTAL UTILITY COST

NO	NAME	$
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CHAPTER 6. ECONOMIC EVALUATION

6.1 ALTERNATIVE SYSTEMS. Alternative fuel sources and alternative environmental systems for buildings shall be evaluated on the basis of present worth derived from life cycle cost analysis as outlined in Economic Analysis Handbook P-442 and NAVFACINST 11010.55A. It is important that all reasonable alternatives be explored in adequate depth to yield a defensible conclusion regarding the most cost-effective design. This usually will require realistic estimates of annual energy consumption. For buildings of greater than 10,000 square feet if both heated and cooled or 40,000 square feet if heated only, the energy estimates should be provided by computer simulation of outdoor and indoor conditions as discussed in Chapters 1 and 5 of this manual. For smaller buildings heating energy may be based on a degree days approach and cooling energy estimates may be based on equivalent full load hours of operation or bin system as discussed in Chapter 1, Section 7.

6.2 FACTORS AND LIFE EXPECTANCY. Factors for discount and inflation rates shall be those given in NAVFACINST 11010.55A as revised from time to time unless local conditions warrant departure from these specified values. Service life of systems components shall be the following:

YEARS	COMPONENTS
10	Window air conditioners, split packages, heat pumps, electric coils, electric packaged steam generator.
15	Roof-top single and multi-zone, reciprocating compressor 25 tons or less, air-cooled condensing unit, evaporative coolers, outdoor insulation - pipe, duct and equipment covering outdoor; protective coating, radiant electric heater, electric radiant panel, heat recovery units (air to air), chiller packaged - reciprocating, air-cooled condenser, evapprative condenser; cooling towers up to 100

YEARS	COMPONENTS
	tons, DX coils, copper fin spray coil, steam and water coils, air washer galvanized, piping - hot water untreated, steam and condensate, oil burner, unit heater - hot water or steam, radiant gas heater, radiant panels - hot water or steam, fans - wall mounted and roof mounted ventilating, water heater - gas, oil and electric.
20	Runaround system, chiller packages - centrifugal and absorption type, cooling tower 100 tons and larger (wood), terminal induction units, water cooled condensers, fan coil units, control valves, aluminum intake louvers, indoor insulation - pipe, duct and equipment, circulating pump - pipe mounted, sump and well and condensate receiver, heat exchanger - steam or hot water - duct heater - gas, fans - forward curved, B.I. (air-foil), air handling units - factory packaged or field designed; reciprocating diesel engine, fuel oil storage tank (steel or fiberglass), transformer - dry, instrument - pipe and duct mounted.
25	Insulation (walls, roof), sheet metal duct, diffusers, grilles and registers, expansion tank, piping - refrigeration, chilled water, circulating pump - base mounted, boilers, reciprocating gas engine, oil transformer, electric control center, control air compressor.
	Pipe - hot water treated and steam, turbine - liquid fuel, gas or steam.

6.3 SELECTION OF ENERGY CONSERVATIVE SYSTEM. In cases where the lowest present worth results in the most energy conservative system that system shall be specified.

CHAPTER 7. ELECTRICAL SYSTEMS

7.1 LIGHTING DESIGN REQUIREMENTS. The design of interior, exterior and sports lighting shall be in accordance with fundamentals and recommendations of the IES Lighting Handbook, published by the Illuminating Engineering Society, subject to the modifications and clarifications for implementing these criteria as noted in paragraphs 7.2 and 7.3.

7.2 LIGHTING INTENSITIES FOR FACILITIES. Maintained lighting intensities shall be those specified in Tables 7-1, 7-2, 7-3 and 7-4. Lighting intensities for other occupancies shall conform to the intensities established in the current edition of the IES Lighting Handbook. The IES recommended intensities are the illuminations required for specific visual tasks, and may be provided by the general illumination in those areas where lower intensities are required. However, the IES recommended intensities are not necessarily to be considered as general illumination intensities for specific areas. The intensity of the general illumination for any area shall not exceed 75 footcandles maintained. If a higher intensity is required for a particular task, it will be achieved by supplementing the general illumination with localized (supplementary) lighting for the particular task, Figure 7-1. The ratios between general and supplementary (local) illumination shall be at least those recommended by IES. Supplementary lighting, where required, will normally be provided by the user of the facility. However, special power requirements for such supplementary lighting shall be established in the design phase.

(1) Environmental Factors. The finish and color of surrounding surfaces and the surfaces of equipment and furniture shall be selected to reduce glare, increase light utilization and obtain an acceptable brightness balance. Lighting equipment and layout shall be coordinated with other facilities to prevent interferences and to promote good appearance.

(2) Medical and Dental Facility Illumination. Lighting intensities for medical and dental facilities shall conform to the IES recommendations with the exceptions listed in Table 7-1 below:

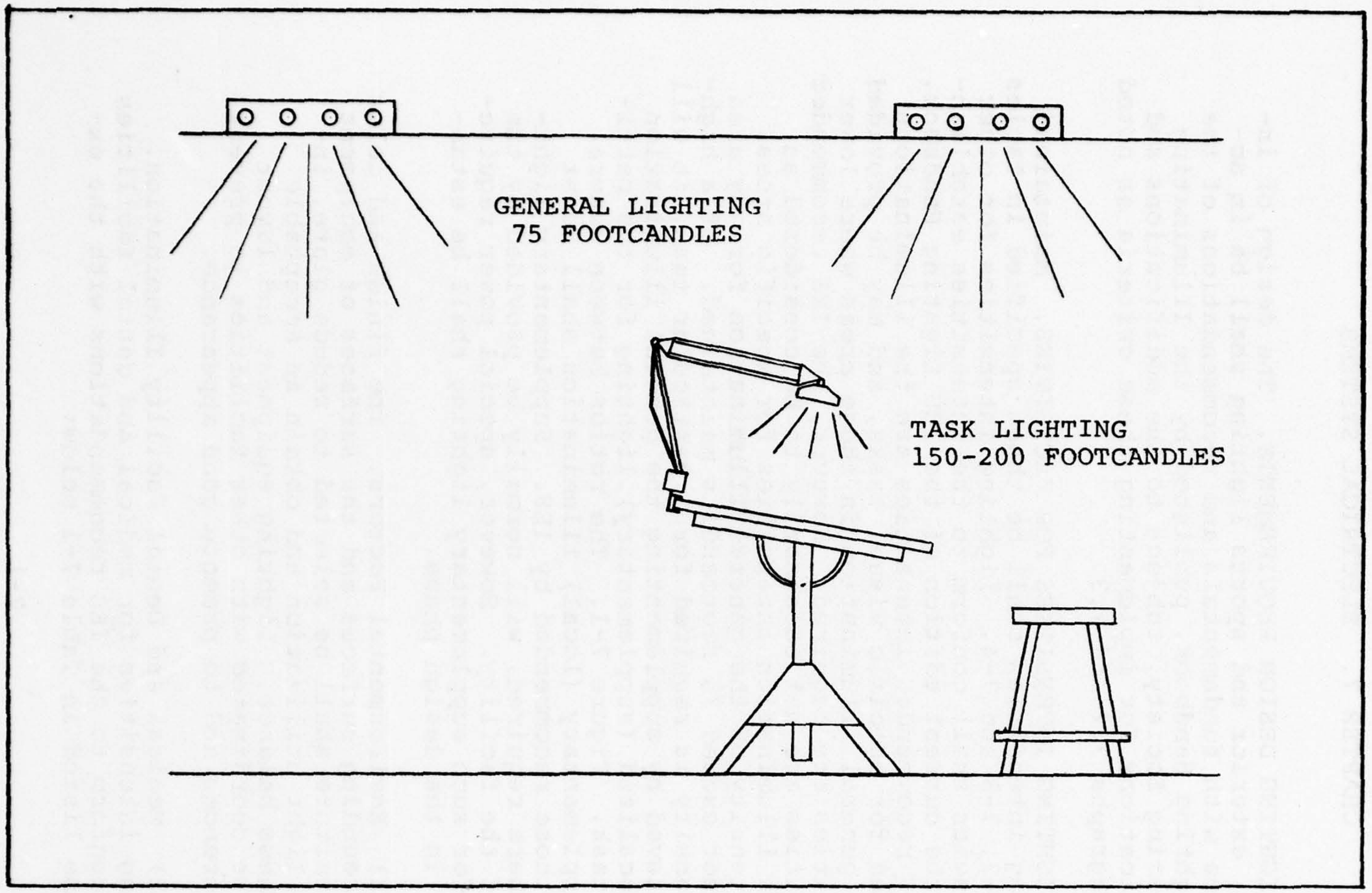

FIGURE 7-1

GENERAL VERSUS TASK LIGHTING

TABLE 7-1

Illumination in Medical and Dental Facilities

Area	Footcandle Intensity[1]
Anesthesia and Preparation Room	30
Control Sterile Supply	30
Fracture Room	50
Nurseries	30
Sewing Room	30
Exits, at Floor	5
Corridors	20

1 Supplementary lighting, where needed for rooms and spaces in medical and dental facilities, will be prescribed by the Surgeon General of the using Service.

(3) Office Building Illumination. The general illumination level in office building spaces shall not exceed the rates listed in Table 7-2 as measured 2 feet 6 inches from the floor. In designing to a specific level for the room size and configuration, flexibility is lost in large open areas and additional fixtures will be required if the design provides for future partitioning, Figure 7-2. Light colored walls, ceilings and carpets contribute to higher lighting values at lower wattage. In determining the type of light fixture and lamp source prime consideration should be given to the 1 x 4 and 2 x 4 fluorescent fixtures with high lumens per watt, 40 watt rapid start lamps. Where ceiling conditions prohibit the above source, the 40 watt "U" lamp should be utilized. Lenses should be prismatic acrylic with good high angle brightness cutoff. Tinted lenses are to be avoided. Position luminaires to avoid direct and reflected glare zones, Figure 7-3.

TABLE 7-2

Illumination in Office Buildings

Area	Footcandle Intensity
Accounting Rooms	75
Auditoriums	20
Cafeterias	25
Computer Rooms	50
Conference Rooms	30
Corridors	15
Drafting Rooms	75
Elevator Machine Rooms	15
Emergency Generator Rooms	15
Garage Entrance	30
Garage Driving and Parking	5
General Office Space	70
Janitor's Closets	5
Kitchens	70
Lobbies	15
Lounges	15
Mechanical Rooms	15
Parking Lots	0.5
Stairways	20
Storage Rooms	5
Switchgear Rooms	15
Toilets	20
Transformer Vaults	15

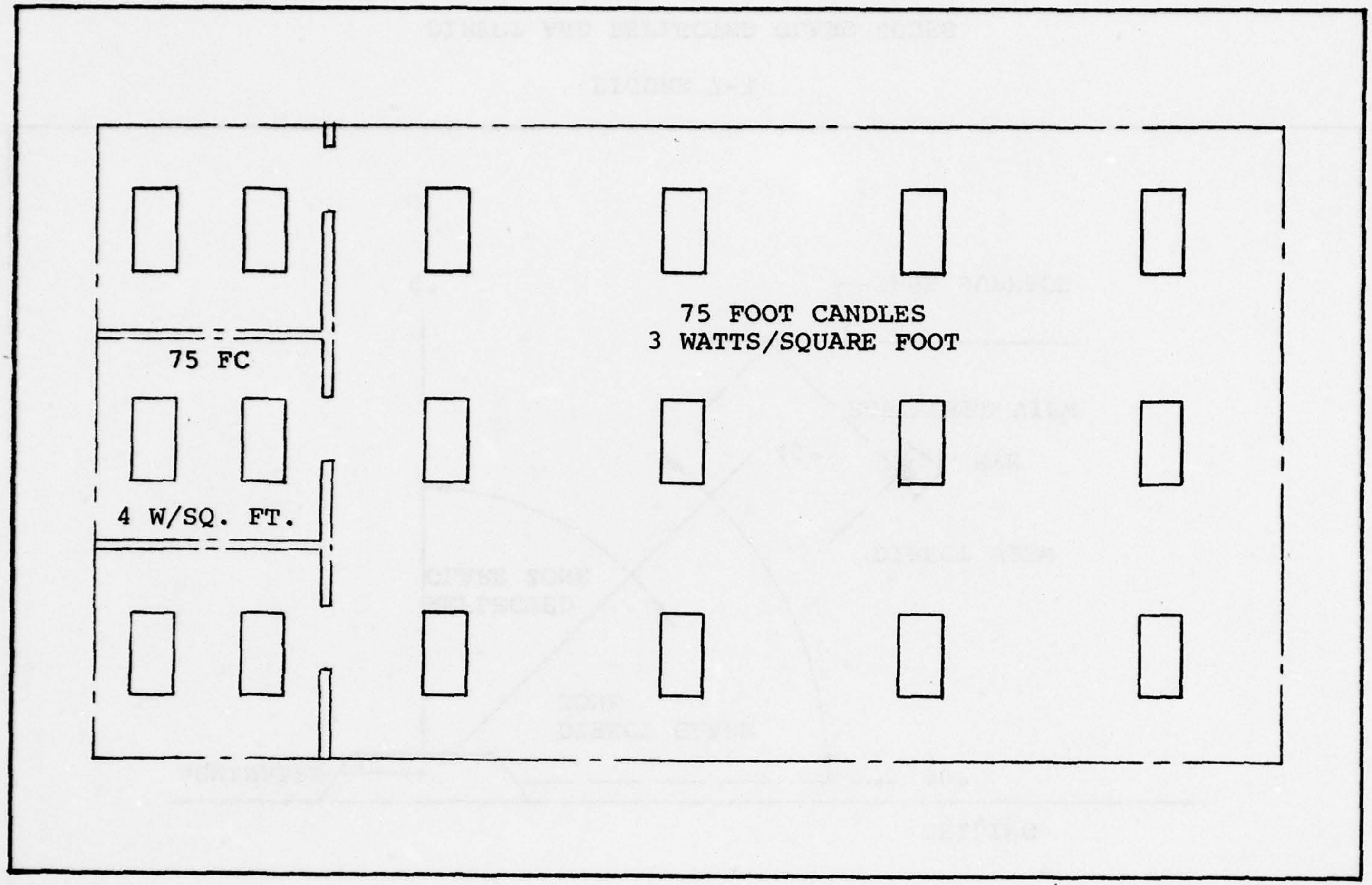

FIGURE 7-2

LARGE VERSUS SMALL AREA LIGHTING ADVANTAGES

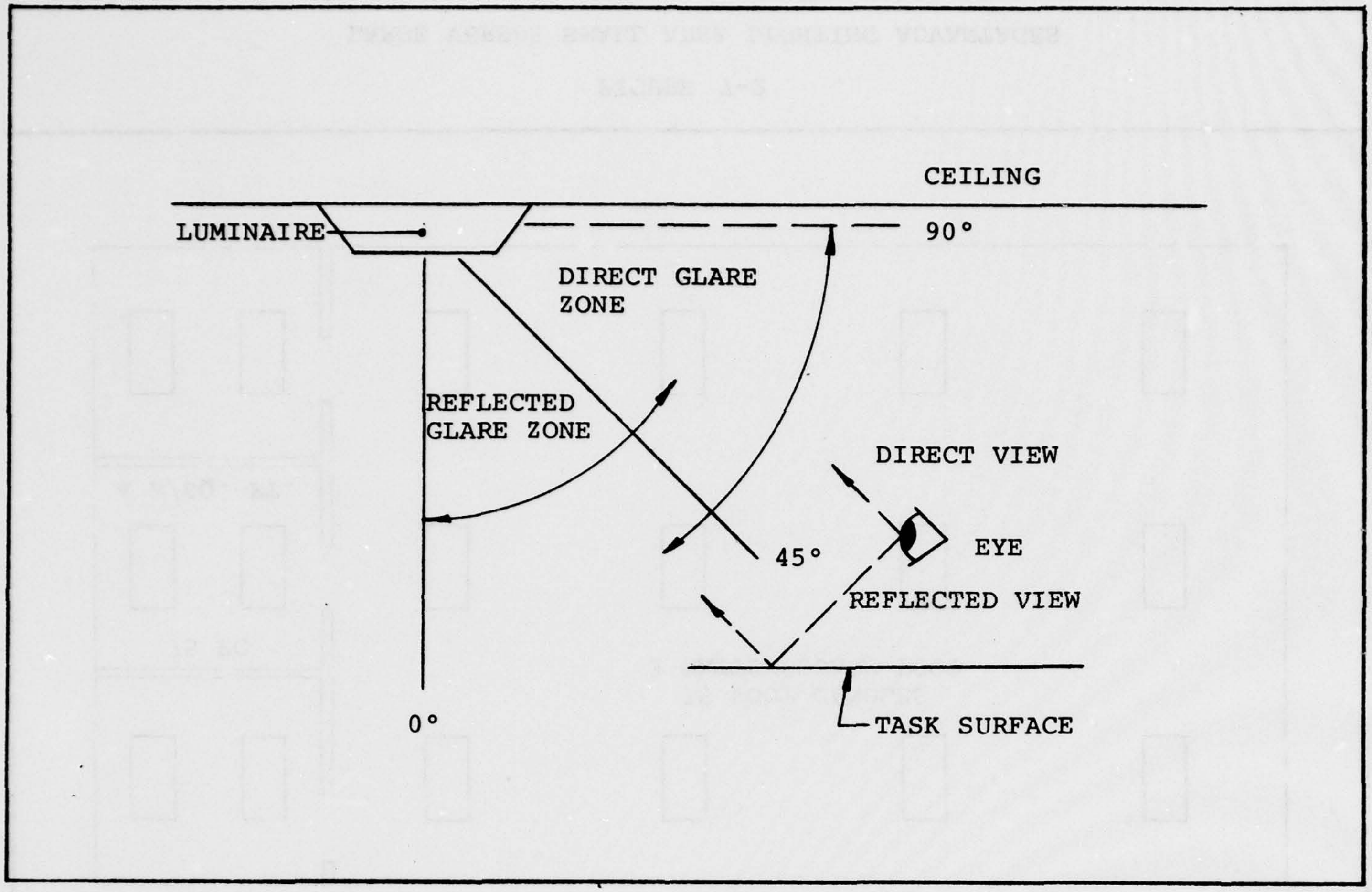

FIGURE 7-3

DIRECT AND REFLECTED GLARE ZONES

Note: Design for maintained footcandles frequently assumes a .75 maintenance factor (fixture cleaning and relamping annually). Where better maintenance program occurs, reduced quantities of fixtures could be used to achieve the same footcandles.

7.2.1 Discreet Lighting. In areas requiring aesthetic or discreet lighting treatment, consideration should be given to round fluorescent fixtures, using 30 or 40 watt straight lamps, 40 watt "U" lamps or multi-vapor or warm deluxe white mercury sources in suitable fixtures. Incandescent sources should be avoided, except in normally unoccupied spaces. Where no other source will suit a specific application except incandescent, then quartz iodine lamps should be considered or incandescent "A" lamps in lieu of PAR or R lamps. The recently developed highly efficient high pressure sodium lamps now being produced in lower wattages may find application for interior use in high ceiling areas, Figure 7-4. However, due to its characteristic color its use and application should be carefully studied prior to specifying. Provision for selective control of lighting levels should be considered. This may be accomplished in design in the following manner:

(a) Inboard-outboard switching of 2 x 4 fluorescents allows two levels of light as needed, Figure 7-5.

(b) Alternate switching of fixtures may accomplish the same result if fixture spacing is not excessive, Figure 7-5.

(c) As much local switching as possible should be provided so that unoccupied spaces may be switched off.

(d) Where panel switching is necessary it should be done by alternate rows or fixtures.

(e) Perimeter lighting adjacent to glass areas should be controllable to take advantage of supplemental daylight.

(f) The use of SCR dimming should be applied wherever economically feasible.

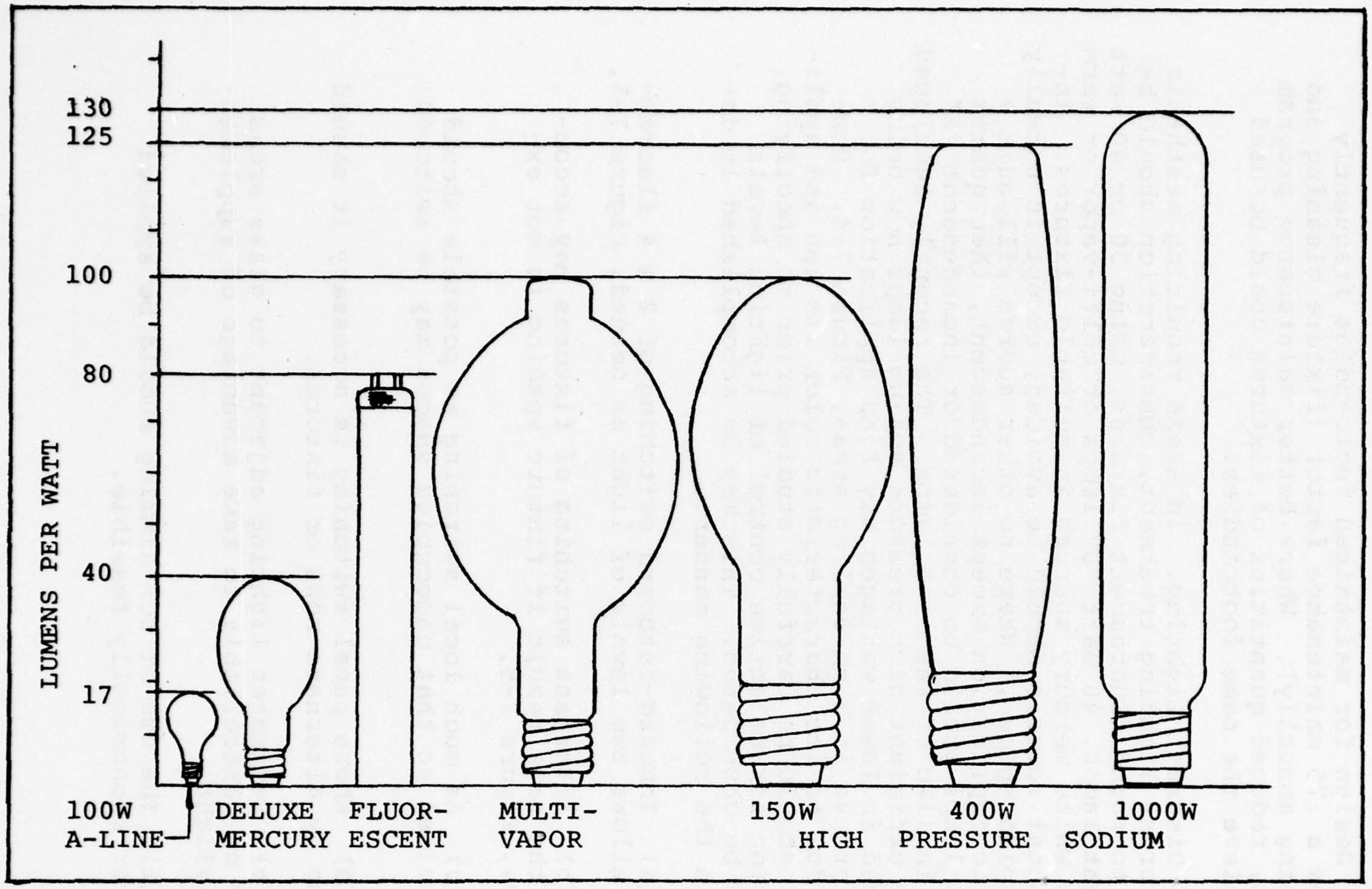

FIGURE 7-4

LIGHT OUTPUT IN LUMENS PER WATT FOR VARIOUS SOURCES

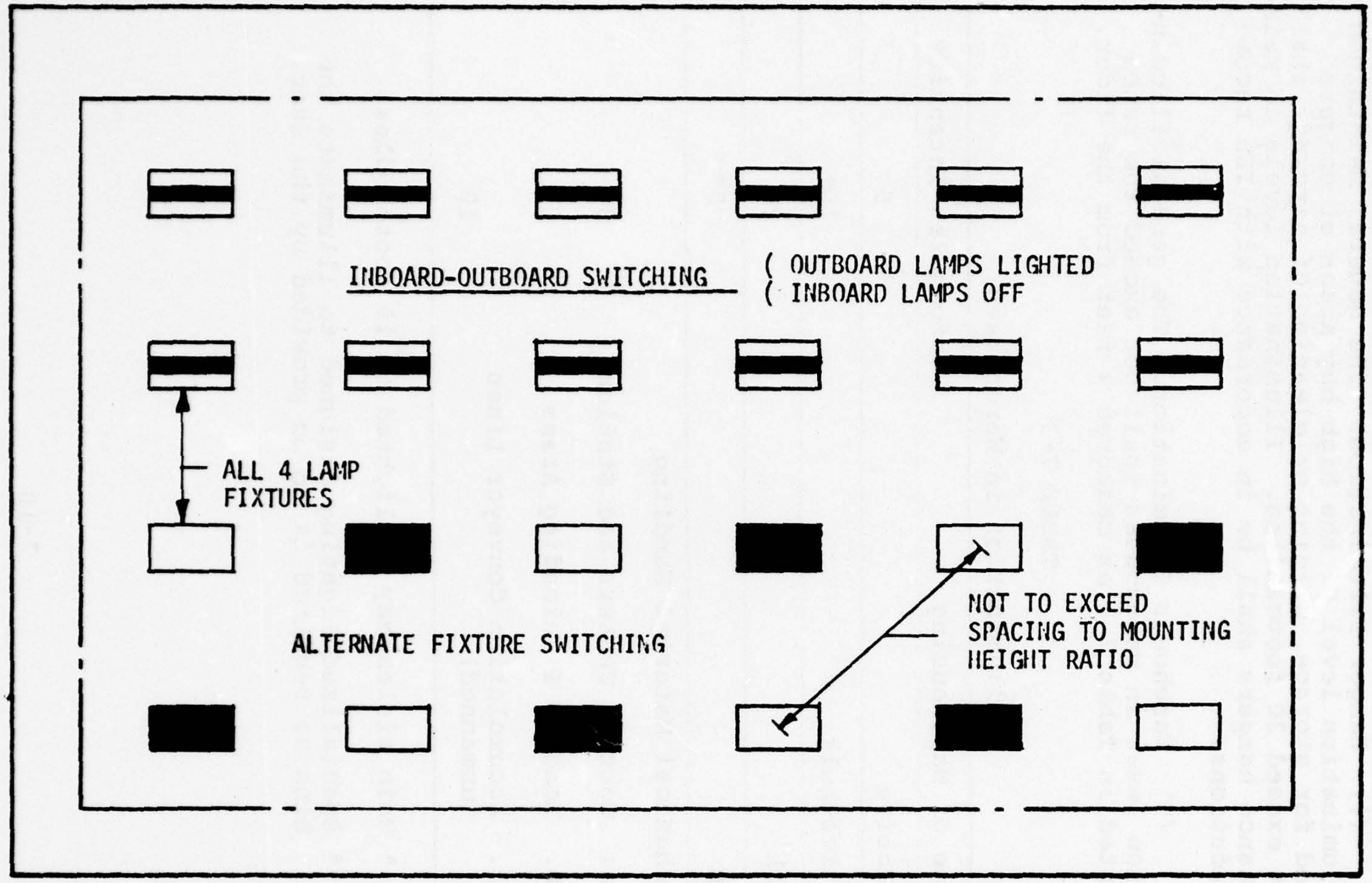

FIGURE 7-5

SELECTIVE SWITCHING SCHEMES

(4) Hanger Illumination. The general maintained illumination level of the high bay areas of hangers used for storage, deicing or cleaning of aircraft shall not exceed 30 footcandles. Illumination levels in maintenance hangers shall be in accordance with IES recommendations.

(5) Warehouse Illumination. The general illumination level in warehouses shall not exceed the rates listed in Table 7-3 as measured 4 feet from the floor.

TABLE 7-3

Illumination in Warehouses

Type of Warehousing	Footcandles Intensity
Inactive	5
Active-bulk	10*
Rack	20
Bin	5**
Mechanical Material Handling	
a. Control Centers and Stations	30
b. Loading & Unloading Areas	20
c. Accumulation Conveyor Lines (unmanned)	10

* Main aisles may be lighted to 15 footcandles.

** Specialized lighting designed to illuminate the bins as required is to be provided by the user.

(6) Exterior Sports Illumination. Outdoor sports lighting shall conform to the classifications stated in the IES Lighting Handbook as listed in Table 7-4.

TABLE 7-4

IES Sports Classifications

Sport	IES Classification
Baseball	Municipal and Semi-professional
Softball	Industrial League
Football	Class III or IV
Other	Recreational

(7) Special Facility Illumination. Where fluorescent or mercury vapor lighting is prohibited and the indicated intensity exceeds 50 footcandles, the general lighting system should be designed for incandescent lighting of 20 footcandles as a practical limit with supplementary incandescent lighting being used for specific tasks where higher lighting levels are required.

(8) Air Handling Luminaires. Where general lighting loads are 2.5 watts per square foot or greater in air conditioned areas, an integrated air conditioning and lighting system is required and lighting fixtures shall conform to the necessary requirements.

7.3 STREET, AREA AND SECURITY LIGHTING.

(1) Street and Area Lighting. Streets, parking areas, and walks shall be lighted to provide safe vehicular and pedestrian circulation. Lights shall be provided at street intersections and between intersections at a spacing of approximately 150-200 feet. Walks not adjacent to streets, and steps in public walks, shall be separately lighted.

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(2) Security Lighting. Since most security lighting must meet specialized requirements, such lighting will be designed to meet the users' needs.

(3) Facade floodlighting and tree and garden lighting should not be used. Street, parking lot, walkway and access road lighting should be accomplished only with efficient sources such as high pressure sodium, fluorescent or mercury vapor, Figure 7-4. Where instantaneous start and restrike are not required, security lighting using the above sources should be applied. Wherever possible exterior lighting should be provided with automatic control as well as manual. The use of photocells only is not recommended. Time clocks or time clock-photocell combinations should be employed.

7.4 SYSTEM CHARACTERISTICS. System characteristics shall be selected to provide for most efficient and economical distribution of energy.

(1) Voltages. Voltages selected shall be of the highest order consistent with the load served. Single phase 120/240 or three phase 120/208 volts shall generally be used to serve combined incandescent and fluorescent or high intensity discharge lighting loads, and small power loads. Where economically feasible and safe, a three phase 277/480 volt system shall be used.

(2) In each building having one or more electric services aggregating 600 amperes or more where more than 50 percent of the load may be served by 277/480 volts, the service voltage shall be nominally 277/480 volts, 3 phase, 4 wire. The lighting system shall be designed to operate at 277 volts or higher, and all motors 1/2 horsepower and above shall be served at 480 volts. For small appliance loads, convenience outlets and other loads requiring other than 277/480 volts, use transformers to serve these loads at their required voltage.

7.4.1 Design Guidance.

(1) The design analysis of electrical systems shall be made in accordance with good design procedures based on the conservation of energy and shall show all

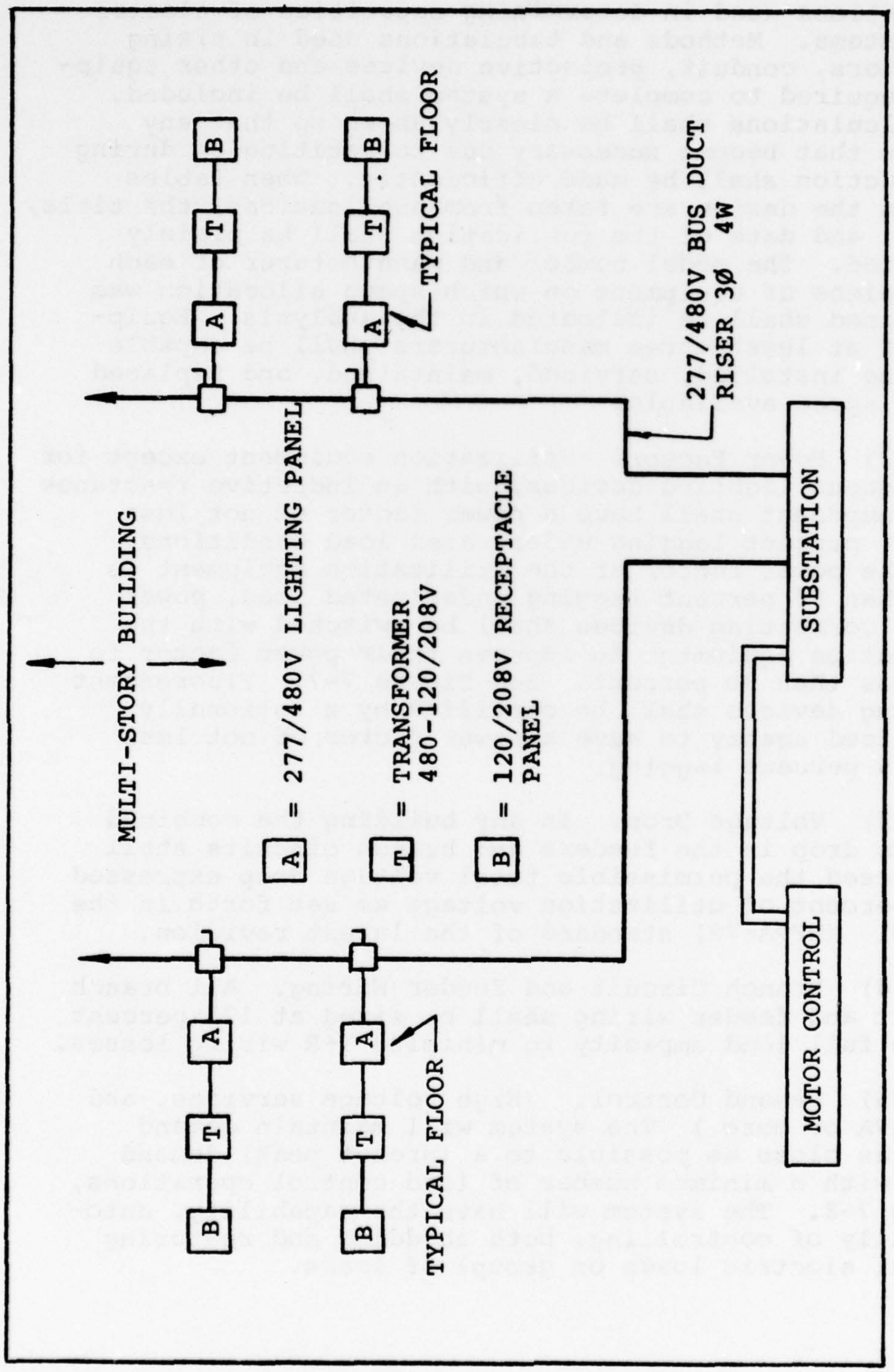

FIGURE 7-6

277/480 VOLT DISTRIBUTION SYSTEM

calculations used in determining capacities of electrical systems. Methods and tabulations used in sizing conductors, conduit, protective devices and other equipment required to complete a system shall be included. All calculations shall be clearly shown so that any changes that become necessary due to resiting or during construction shall be made efficiently. When tables used in the design are taken from publications, the title, source, and date of the publication shall be plainly indicated. The model number and manufacturer of each major piece of equipment on which space allocation was determined shall be indicated in the analysis. Equipment of at least three manufacturers shall be capable of being installed, serviced, maintained, and replaced in the space available.

(2) Power Factor. Utilization equipment except for fluorescent lighting devices, with an inductive reactance load component shall have a power factor of not less than 90 percent lagging under rated load conditions. When the power factor of the utilization equipment is less than 90 percent lagging under rated load, power factor correcting devices shall be switched with the utilization equipment to improve their power factor to not less than 90 percent. See Figure 7-7. Fluorescent lighting devices shall be certified by a nationally recognized agency to have a power factor of not less than 95 percent lagging.

(3) Voltage Drop. In any building the combined voltage drop in the feeders and branch circuits shall not exceed the permissible total voltage drop expressed as a percent of utilization voltage as set forth in the ANSI-C1 (NFPA-70) standard of the latest revision.

(4) Branch Circuit and Feeder Wiring. All branch circuit and feeder wiring shall be sized at 125 percent of the full load ampacity to minimize I^2R wiring losses.

(5) Demand Control. (High voltage services, and 2000 KVA or more.) The system will maintain demand peaks as close as possible to a (preset peak) demand limit with a minimum number of load control operations, Figure 7-8. The system will have the capability, automatically of controlling, both shedding and restoring several electric loads or groups of loads.

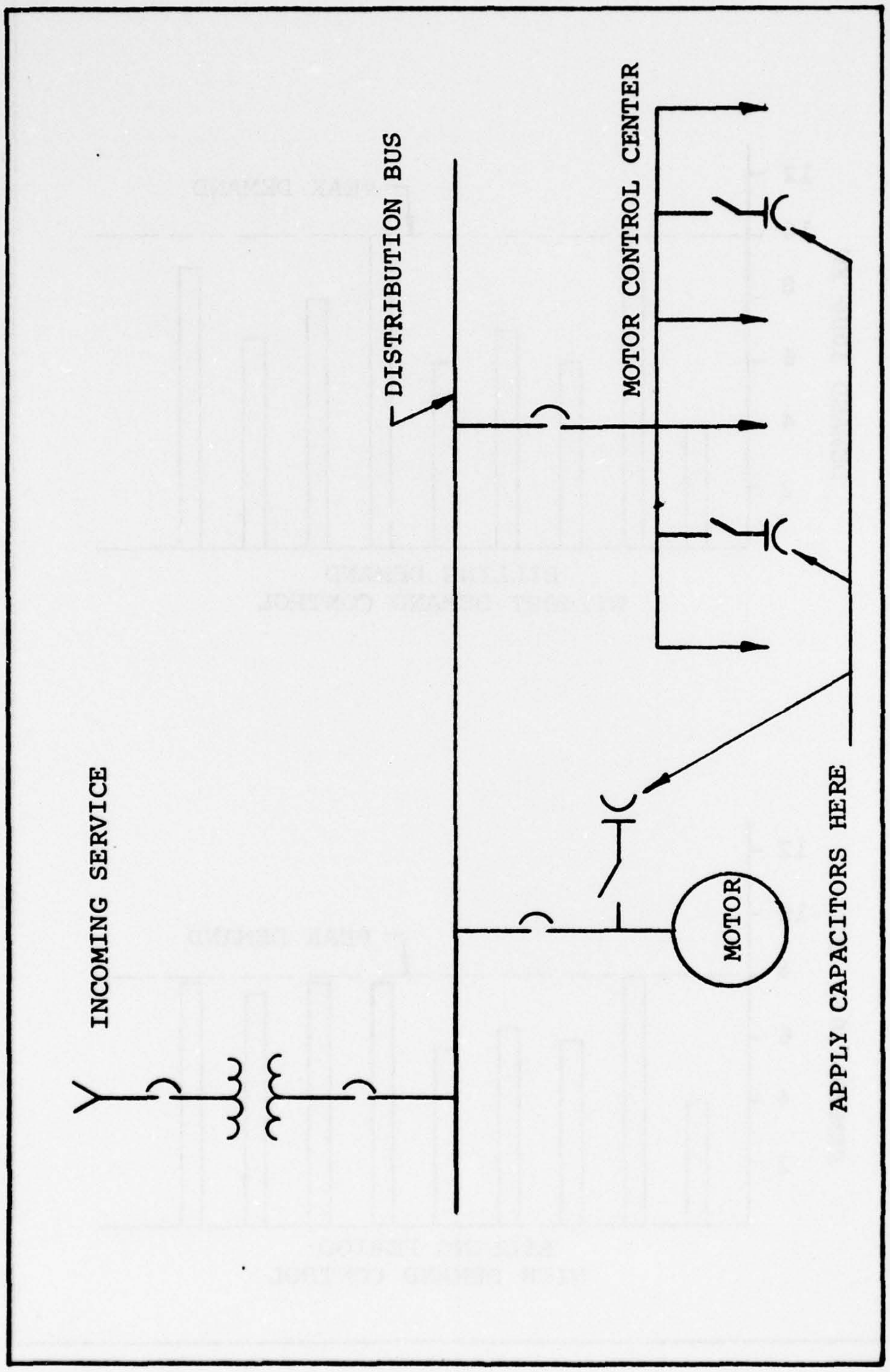

FIGURE 7-7

APPLICATION OF POWER FACTOR CORRECTION

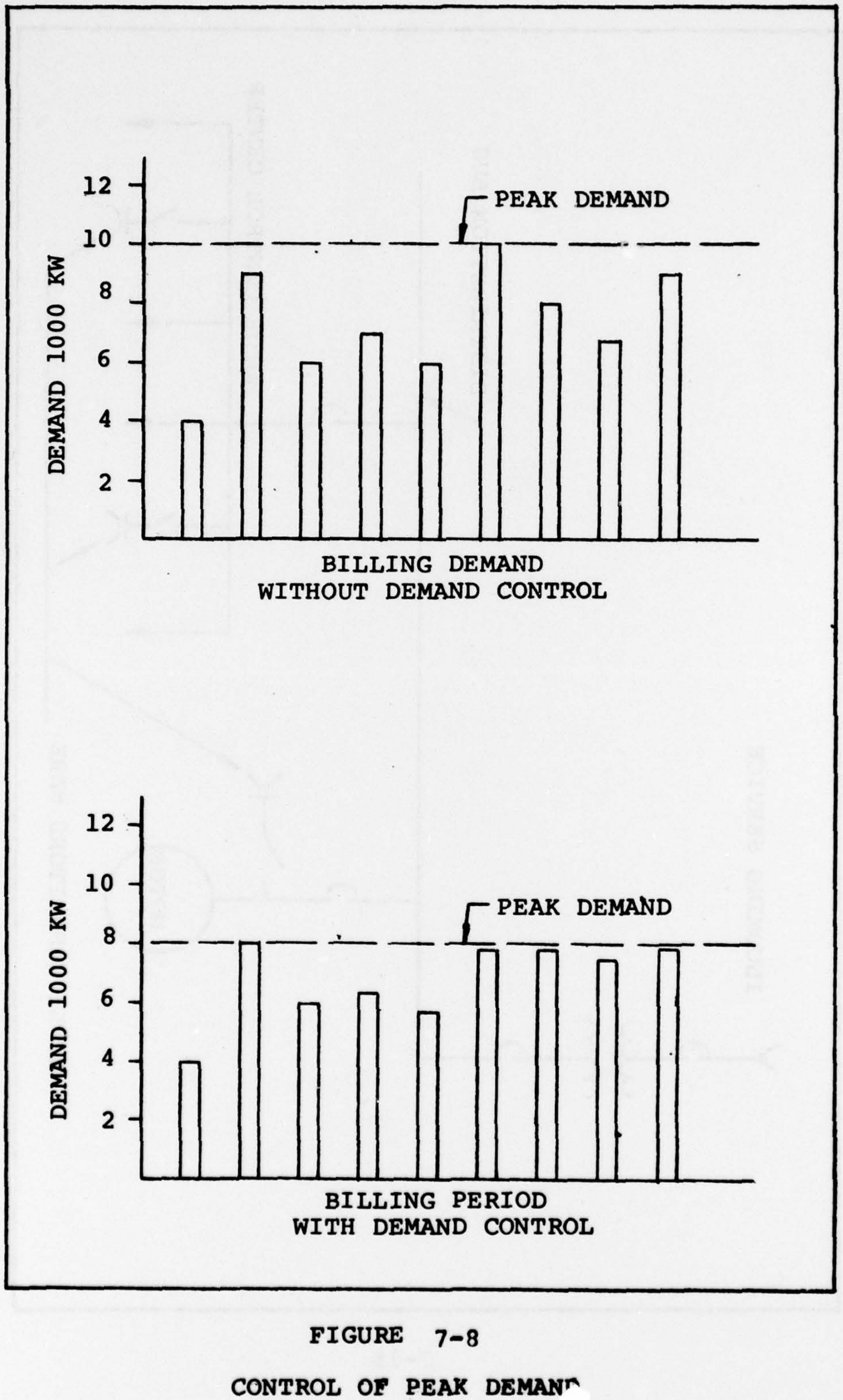

FIGURE 7-8

CONTROL OF PEAK DEMAND

7.4.2 Additional Measures. Additional energy conservation measures which should be considered:

(1) Select lighting fixtures of high efficacy (high coefficients of utilization).

(2) Lower ceiling or mounting heights (to 8'-0") to increase footcandle levels with less wattage.

(3) Use heat pumps in place of resistance heating.

(4) Make use of high voltage distribution if electrode type boilers are considered.

(5) Hot water storage or water could be heated during off peak hours (demand control).

CHAPTER 8. MISCELLANEOUS ITEMS

Section 1. SYSTEM OPERATION, MAINTENANCE AND BALANCING

8.1.1 ACCEPTANCE CRITERIA. Prior to acceptance of a completed building by the Government ceritifcation should be received from the Contractor that all building systems have been balanced, tested and all systems are operating as designed. Specifications should require the Contractor to submit a set of operating instructions describing the function of each piece of equipment, its psychrometrics, normal operating temperatures seasonally, and for pumps, normal pressures, flow and temperatures. Instructions should also include any changeover operation, periodic replacement, lubricating data, and any deviation in the design from standard practices.

8.1.2 BALANCING. Certification of balancing should include copies of final data on the flow balance of air and hydronic systems. The Government must be assured that all results are within specified tolerance and corrections have been made for any excessive deviations. In striving for an efficient, economical building operation, the staff is at a disadvantage from the beginning if this type of information is not furnished. The initial balance records should become the base data to which any future abnormal readings are compared in searching for system malfunctions. The staff will improve the methods of operation as they gain experience with the building, and this process will be accelerated by the early understanding of equipment performances.

8.1.3 AIR SYSTEMS. There are a number of ways energy is wasted in the operation of air systems, some from lack of interest, others from lack of instruction.

(1) Outside Air Intake. This quantity is set by calculation on a square foot basis or some legal or code air change rate. This should be set during air balancing either by damper stops or orifices and never exceeded. During precooling or preheating cycles when the building is not occupied, intake dampers are frequently left closed temporarily. This will conserve the added energy of heating or cooling outside

air until occupancy. If this cycle is used, do not operate toilet or other exhaust systems until outside air dampers are open. Do not operate exhaust fans on winter nights without supply air systems. All exhausted air must then infiltrate into the buildings. Do not operate supply systems without refrigeration or exhaust fans during high wet bulb summer nights for "cooling". This saturates the building with moisture which takes hours to remove the next day.

(2) Fan Air Quantities. Neither supply or exhaust fans should be operated above the specified air delivery. Pressure rises as the square of air delivery increases and horsepower as the cube. This is a waste of energy in fan horsepower, without considering the possibility of overpulling on outside air or infiltration of outside air from too much exhaust.

(3) Dewpoint Control. Moisture gain during the cooling season is usually calculated on the basis of the number of occupants plus some infiltration and interior gains. Air dewpoint at the dehumidifier coil discharge should be below the maintained room dewpoint only the exact number of degrees necessary to hold the room dewpoint constant. If this is specified at 50 percent relative humidity, any reduction in moisture content below 50 percent is an excessive removal of latent tons. Since the airside coil surface is fixed, the chilled water temperature entering can be raised slightly to compensate. This will reduce the amount of latent cooling as well as kilowatt input to the compressor.

(4) Calibration and Setting of Thermostats. Much energy can be wasted through inaccurate calibration and random setting of thermostats. In calibrating, room temperature should be measured with a certified, mercury etched stem laboratory thermometer. Unless carefully done a three degree error is not uncommon. The best defense against random settings is a locked cover, inside adjusted thermostat.

(5) Perimeter Radiation Systems. Perimeter radiation systems should be controlled by means of an outside master thermostat resetting water temperatures

on a precalculated schedule. Separate zoning of south with a solar override is desirable. Exact temperature schedules can only be devised after some experience with the building, but careful observation is needed to prevent override and subsequent removal of radiator heat by the air system.

(6) An important means of continuously monitoring performance is a daily log. Whether done electronically or manually, knowledge of the degree of excellence in operation of each system will contribute to a well managed plant. Need for maintenance or replacement is immediately obvious, as is any malfunction of control systems. This includes observation of outdoor conditions every few hours and thoughtful attention to the response of all equipment to change.

Section 2. DOMESTIC AND SANITARY WATER SYSTEMS

8.2.1 SERVICE WATER HEATING. Hot water for domestic and sanitary purposes should be generated and delivered in a manner conducive to saving heat energy. Whenever possible, heat recovery should be used for preheating, partial heating, or the total heating of the domestic hot water.

8.2.1.1 Water Heaters, Storage Tanks and Boilers. Automatic electric storage water heaters should comply with Section 4.3 of ANSI-C72.1-72. Gas and oil fired automatic storage water heaters should comply with either Section 2.7 of Volume I of ANSI-Z21.10.1 or Section 2.7 of Volume III of ANSI-Z21.10.3. Domestic water heating requirements shall comply with NAVFACINST 4100.4. Also domestic water heating systems serving more than four single family living units should not use electric resistance heating unless (1) electricity is used as the principle means for heating the building, or (2) point of use heaters are proven to be more energy-efficient. Fossil fuel fired service water heaters with input heating capacities greater than 250,000 Btu per hour that provide space heating and/or water heating should be equipped with off-cycle air flow heat loss control devices. These devices should be safety listed as a part of the appliance. The minimum overall system efficiency for boilers, regardless of usage, should be no less than 65 percent based on total annual energy use.

8.2.1.2 Insulation. All unfired service hot water storage tanks and piping containing heated water should be insulated. Heat loss for above ground piping and storage tanks should be limited to 25 Btu per hour per square foot and 35 Btu per hour per square foot for underground piping and recirculating systems.

8.2.1.3 Temperature Control. Service water heating systems should be equipped with automatic temperature controls capable of adjustment to the lowest acceptable temperature setting for intended use (see Table 8-1). Unless conditions demand otherwise, water should not be heated and stored at a higher than utilization temperature. Where special functions, such as dishwashing,

TABLE 8-1

Hot Water Temperature Based on Utilization

Use	Temperature °F.
Lavatory	
Handwashing	105
Shaving	115
Showers and Tubs	110
Therapeutic Baths	110
Commercial and Institutional Dishwashing	
Wash	140
Sanitizing Rinse	180
Commercial and Institutional Laundry	180
Residential Dishwashing and Laundry	140
Surgical Scrubbing	110

require higher temperature water, provide a local booster rather than heat all of the building hot water to meet the localized condition.

8.2.1.4 Pump Operation. Circulating hot water systems should be so arranged that the pump energy consumption is low consistent with correct operation of the pump. Circulating systems should be so arranged that the circulating pump(s) can be conveniently turned off when the portion of or the entire building served by that pump is not in use.

8.2.1.5 Conservation of Hot Water. Showers used for other than safety reasons should be equipped with flow control devices to limit flow to 3 GPM. Flow control devices should also be given consideration for installation on lavatory faucets. Mixing valves at point of use of showers and lavatories should be considered to limit the maximum use temperature. Consideration should be given to recovery of heat from laundry waste water to preheat and/or heat service water. Rejected heat from refrigeration equipment, gas or diesel engine driven equipment can be recovered to heat service water where applicable heat can be recovered from steam condensate.

8.2.1.6 Solar Heat. Use of solar heat to furnish domestic hot water requirements must be considered if economically justified.

Section 3. ENERGY CONSERVATION BY OPTIMIZATION OF CONTROLS

8.3.1 ROOM THERMOSTAT LOCATIONS. The basic and most obvious controlling device is the room thermostat. Location is very important. Room thermostats should be located such that drafts from doors, drafts from loose windows, or the effects of cold walls do not cause erroneous sensing of the actual space temperature. The result of poor locations will cause the space temperature to be raised to compensate for low readings, an obvious waste of energy.

8.3.2 ZONE CONTROL. Using one thermostat to control several different offices or spaces should also be avoided. Unless the loads are always the same, which is the rare exception, one of the spaces is going to be overheated or overcooled, again wasting energy.

8.3.3 INTERMITTENT OCCUPANCY CONTROL. Classrooms, conference rooms, cafeterias, and other areas which have intermittent occupancy should have occupied-unoccupied switches. These switches should function to eliminate conditioning of these spaces when the rooms are not being used.

8.3.4 INTERIOR ZONE CONTROL. The space thermostats for interior zones (these zones normally require cooling all year around) should have their settings kept at 78°, whereas the exterior zones which are affected by external temperature changes can be set at 78° for summer operation and 70° for winter operation. This is recommended provided that discomfort is not felt when passing from an exterior zone or vice versa.

8.3.5 PERIMETER RADIATION ENERGY SAVINGS. Perimeter heating system control should have a daytime and a higher nighttime reset schedule (see Figure 8-1). If the perimeter radiation is supplying more heat in the space than is being lost through outside walls during the daytime, the air systems supplying those areas will be in a cooling mode of operation. During occupied periods, there is excess internal heat gain produced by internal loads (people, lighting, machinery),

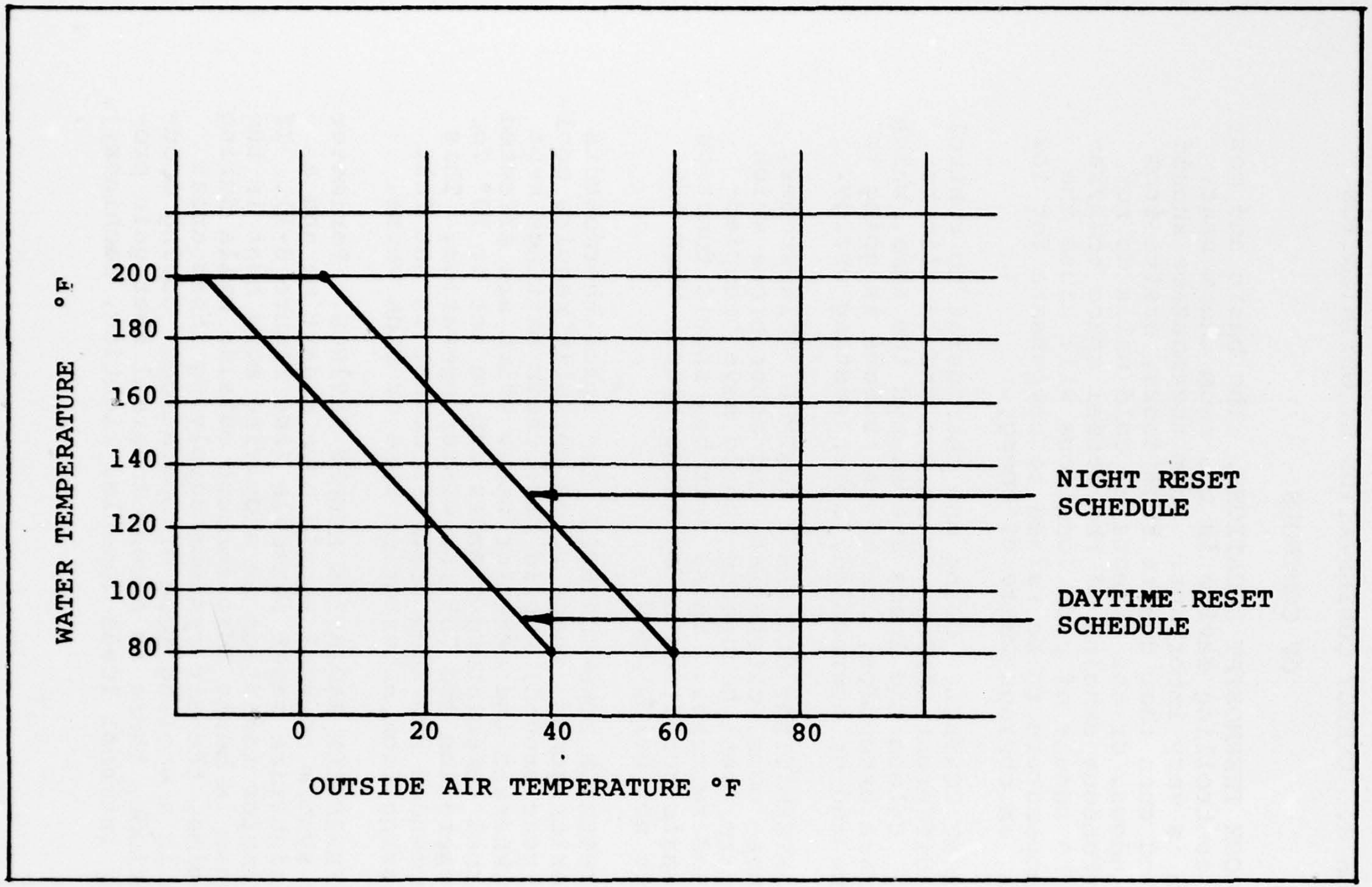

FIGURE 8-1

REDUCED DAYTIME RESET SCHEDULE

and the perimeter radiation systems are designed for an absence of these loads.

8.3.6 CENTRALIZED CONTROL SYSTEMS. Figures 8-2 and 8-3. Consider central control systems to provide the following:

(1) Gathering information, such as temperatures, pressures, relative humidity readings, electrical demand, and other data required for efficient building operation.

(2) Displaying the information digitally to provide the operator with accurate readings anywhere in the building.

(3) Alarming is an important function. The operator is notified immediately when controllers drift off of their set points causing excessive high or low air temperature, and the alarming is on a 24 hours basis.

(4) Adjustments to controller settings, damper positions, discharge air control points can be made quickly, accurately and efficiently.

(5) Conservation of manpower otherwise required for starting or stopping fans or taking temperature readings throughout the building.

(6) Scheduling maintenance for equipment based on actual run-times rather than on a calendar schedule.

8.3.7 RUN-TIME ENERGY SAVINGS. Reducing the run-time of a 40 horsepower fan an average of 15 minutes per day could yield 1939.6 kilowatt savings over a period of a year. Calculated as follows:

(Number of extra hours run-time per day) x (days per week) x (52 wks per year) x (horsepower of fan) x (.746 kilowatts per horsepower) = (.25)x(5)x(52)x(40)x (.746) = 1939.6

For the fan which operates over the period of a year using 69,825.6 kilowatts, this represents approximately 3 percent in electric savings alone. Combining

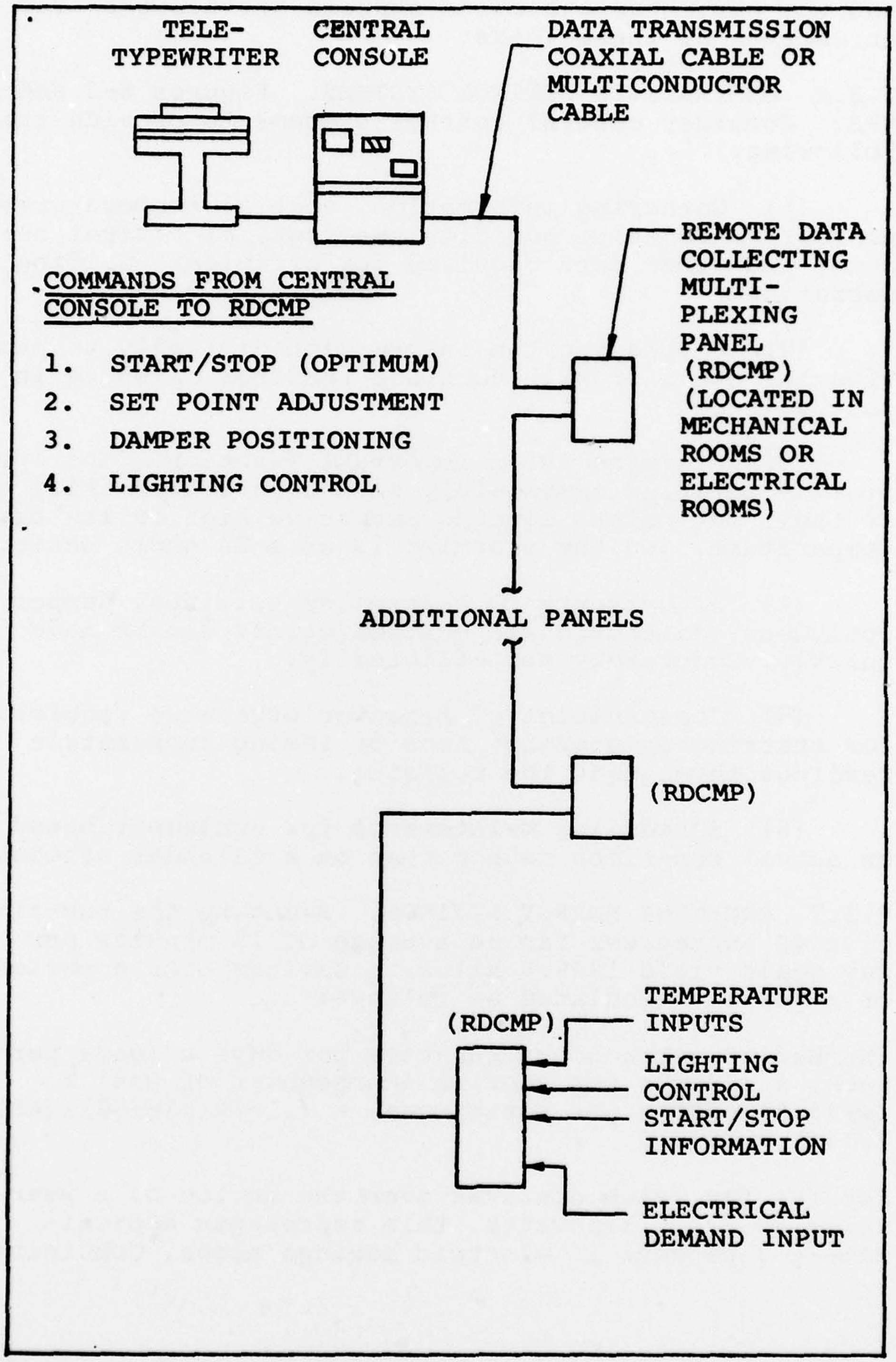

FIGURE 8-2

TYPICAL CENTRALIZED CONTROL SYSTEM LAYOUT

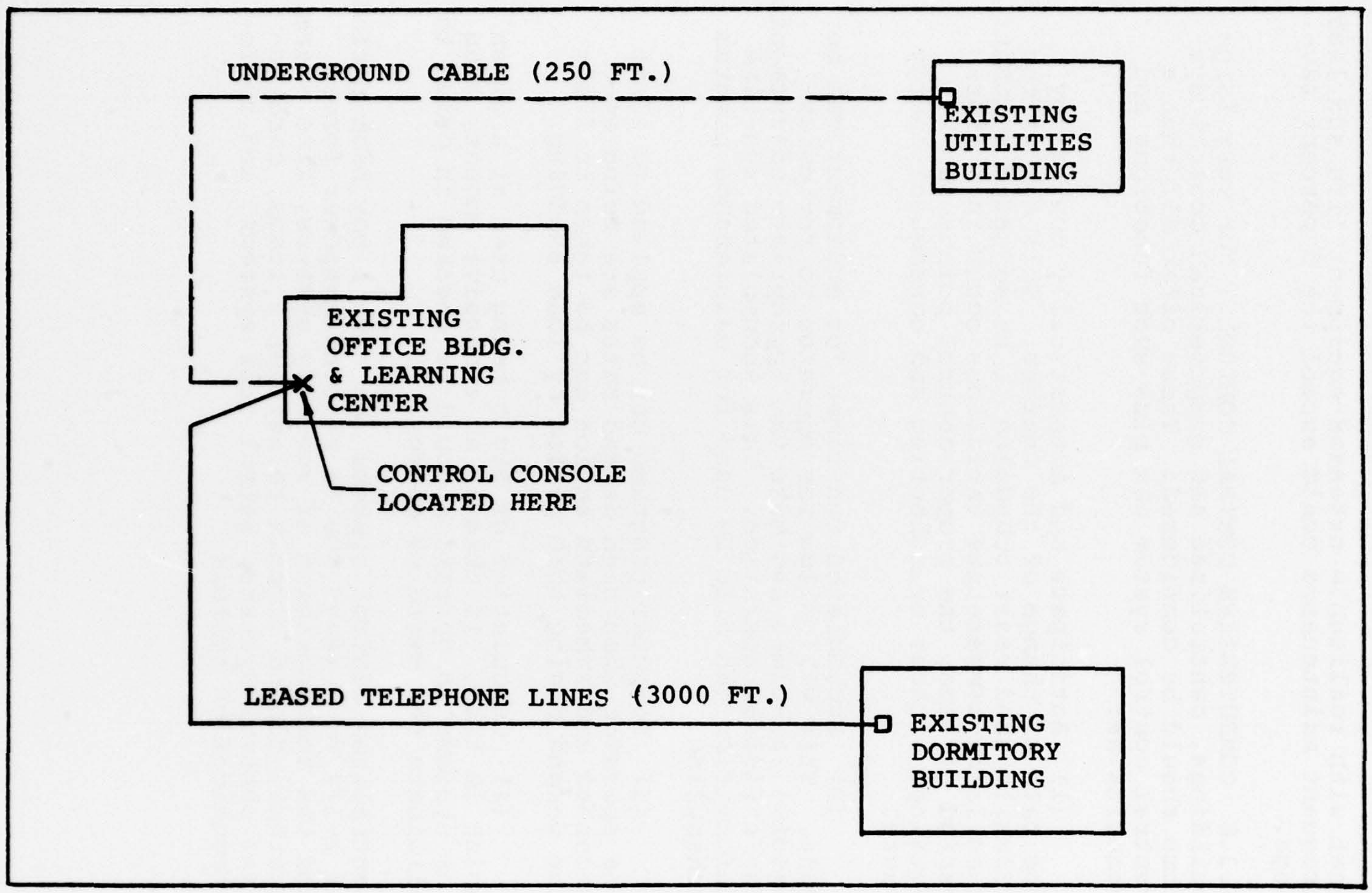

FIGURE 8-3

TYPICAL MULTI-BUILDING CONTROL FROM A SINGLE CONTROL CONSOLE that with realizable extended equipment life and less frequent maintenance could exceed the 3 percent savings.

8.3.8 COMPUTERIZED CENTRAL CONTROL. For very large buildings, centralized and computerized control systems should be considered. These offer all that a central control system can plus such functions and benefits as:

(1) Anticipate and automatically control by the rate of change of the weather. This allows the normally used reset schedules to be adjusted so that when large temperature variations occur in a short period of time, the proper control points can be obtained without overshooting and consequent energy waste.

(2) Accumulated run-times for equipment can be made. This will allow the operator to review excessive run-times and make the appropriate adjustments for efficient operation. This accumulated run-time information can also be used for maintenance program scheduling.

(3) Computer programs can be applied to alert the operator when high demand rates are being approached and immediate action can be taken to limit the demand during this period by load shedding.

(4) Calculation of Btu's being used at a given point in time. If this is an excessive amount, then the systems in operation could be checked in order to eliminate the waste of energy.

Centralized control systems can have a pay back period of 2-1/2 to 3 years due to reduced manpower forces and the incorporation of security systems, fire alarm systems, closed circuit television systems, card access control systems, patrol tour systems, and audio-communication systems.