TM-1436
Appendices X - XII

MANUFACTURER EVALUATION
OF
COMPUTER ROOM DX AC EQUIPMENT

Liebert Corporation

Environmental Systems

Energy Analysis

5/16/86

For
FERMI LABS
STEVE KRSTULOVICH

15 TON UNIT

Prepared by
ZONATHERM PRODUCTS
MICHAEL BORDENET

100%

Liebert Corporation

1050 Dearborn Drive, Columbus, Ohio 43229

5/16/86
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Environmental Energy Analysis
For **FERMI LAB**

Pre-report summary

The energy cost of 1 FH 192G (60 Hz) units for one year is $ **14375**

Design Criteria

Region Chicago, Illinois
Room Drybulb.................. 72 F
Room Relative Humidity....... 50 %
Main Fan Motor............. 3.00 HP
Number of Units............... 1
Design Ambient................ 95 F

Model Unit..........FH 192G (60 Hz)
Air Flow per Unit.......... 8400 CFM
No. of Compressors per Unit.. 2
Glycol Pump Motor........... 2.00 HP
Drycooler Fan Motor......... 2.25 HP

Performance and Operation Breakdown

Temperature Range (F)	min-35	35-39	40-44	45-49	50-54	55-59	60-64
Mean Coincident WB (F)	30.0	34.0	38.0	43.0	47.0	52.0	57.0
Total Room Load(KBTUH)	164.2	164.2	164.2	164.2	164.2	164.2	164.2
Sens Room Load (KBTUH)	156.4	156.4	156.4	156.4	156.4	156.4	156.4
Total Cool/Unit(KBTUH)	188.7	188.7	188.7	188.7	188.7	188.7	188.7
Sens Cool/Unit (KBTUH)	167.2	167.2	167.2	167.2	167.2	167.2	167.2
Evaporator Temp (F)	43.4	43.4	43.4	43.4	43.4	43.4	43.4
Condensing Temp (F)	105.0	105.0	105.0	105.0	105.0	105.0	105.0
Humidifier Cap (lb/hr)	17.4	17.4	17.4	17.4	17.4	17.4	17.4

Breakdown of Energy Use

		min-35	35-39	40-44	45-49	50-54	55-59	60-64
Fan Motor	Max (KW)	2.2	2.2	2.2	2.2	2.2	2.2	2.2
	x %Oper	100.0	100.0	100.0	100.0	100.0	100.0	100.0
	= Actual (KW/H)	2.2	2.2	2.2	2.2	2.2	2.2	2.2
Compressor 1	Max (KW)	6.2	6.2	6.2	6.2	6.2	6.2	6.2
	x %Oper	100.0	100.0	100.0	100.0	100.0	100.0	100.0
	= Actual (KW/H)	6.2	6.2	6.2	6.2	6.2	6.2	6.2
Compressor 2	Max (KW)	6.2	6.2	6.2	6.2	6.2	6.2	6.2
	x %Oper	83.8	83.8	83.8	83.8	83.8	83.8	83.8
	= Actual (KW/H)	5.2	5.2	5.2	5.2	5.2	5.2	5.2
Humidifier	Max (KW)	6.4	6.4	6.4	6.4	6.4	6.4	6.4
	x %Oper	59.6	59.6	59.6	59.6	59.6	59.6	59.6
	= Actual (KW/H)	3.8	3.8	3.8	3.8	3.8	3.8	3.8
Reheat	Max (KW)	25.0	25.0	25.0	25.0	25.0	25.0	25.0
	x %Oper	0.0	0.0	0.0	0.0	0.0	0.0	0.0
	= Actual (KW/H)	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Glycol Pump	Max (KW)	3.2	3.2	3.2	3.2	3.2	3.2	3.2
& Drycooler	x %Oper	100.0	100.0	100.0	100.0	100.0	100.0	100.0
	= Actual (KW/H)	3.2	3.2	3.2	3.2	3.2	3.2	3.2
Total KW/Hour.........		20.5	20.5	20.5	20.5	20.5	20.5	20.5
x Hours/Year........		2521	725	572	565	581	604	700
x Cost/KWHR		$0.080	$0.080	$0.080	$0.080	$0.080	$0.080	$0.080
		=====	=====	=====	=====	=====	=====	=====
= Cost/Unit		$ 4141	$ 1191	$ 940	$ 928	$ 954	$ 992	$ 1150

These ratios reflect the opinion of Liebert Corporation engineers based on available published catalog information.

Liebert Corporation

1050 Dearborn Drive, Columbus, Ohio 43229

Environmental Energy Analysis
For **FERMI LAB**

Performance and Operation Breakdown (cont.)

Temperature Range (F)	65-69	70-74	75-79	80-84	85-89	90-max
Mean Coincident WB (F)	61.0	64.0	67.0	70.0	72.0	74.0
Total Cool Load(KBTUH)	164.2	164.2	164.2	164.2	164.2	164.2
Sens Cool Load (KBTUH)	156.4	156.4	156.4	156.4	156.4	156.4
Total Cool/Unit(KBTUH)	188.7	183.5	178.4	173.1	167.8	162.2
Sens Cool/Unit (KBTUH)	167.2	165.1	163.0	160.8	158.6	156.4
Evaporator Temp (F)	43.4	43.9	44.3	44.8	45.3	45.8
Condensing Temp (F)	105.0	110.0	115.0	120.0	125.0	130.0
Humidifier Cap (lb/hr)	17.4	17.4	17.4	17.4	17.4	17.4

Breakdown of Energy Use (cont.)

Fan Motor Max (KW)	2.2	2.2	2.2	2.2	2.2	2.2
x %Oper	100.0	100.0	100.0	100.0	100.0	100.0
= Actual (KW/H)	2.2	2.2	2.2	2.2	2.2	2.2
Compressor 1 Max (KW)	6.2	6.5	6.8	7.1	7.4	7.7
x %Oper	100.0	100.0	100.0	100.0	100.0	100.0
= Actual (KW/H)	6.2	6.5	6.8	7.1	7.4	7.7
Compressor 2 Max (KW)	6.2	6.5	6.8	7.1	7.4	7.7
x %Oper	83.8	86.8	89.9	93.1	96.5	100.0
= Actual (KW/H)	5.2	5.6	6.1	6.6	7.2	7.7
Humidifier Max (KW)	6.4	6.4	6.4	6.4	6.4	6.4
x %Oper	59.6	48.1	35.4	21.3	5.8	0.0
= Actual (KW/H)	3.8	3.1	2.3	1.4	0.4	0.0
Reheat Max (KW)	25.0	25.0	25.0	25.0	25.0	25.0
x %Oper	0.0	0.0	0.0	0.0	0.0	0.0
= Actual (KW/H)	0.0	0.0	0.0	0.0	0.0	0.0
Glycol Pump & Drycooler Max (KW)	3.2	3.2	3.2	3.2	3.2	3.2
x %Oper	100.0	100.0	100.0	100.0	100.0	100.0
= Actual (KW/H)	3.2	3.2	3.2	3.2	3.2	3.2
Total KW/Hour..........	20.5	20.6	20.6	20.5	20.4	20.8
x Hours/Year.........	759	681	487	324	165	64
x Cost/KWHR	$0.080	$0.080	$0.080	$0.080	$0.080	$0.080
= Cost/Unit	$ 1247	$ 1122	$ 802	$ 532	$ 269	$ 107

Cost Summary

Unit energy cost per year .... $ 14375
Energy cost for 1 FH 192G unit(s) per year ...... $ 14375

Ten year projected energy cost .... $ 189475
Present value of ten year energy cost .... $ 122370

Note: This projection is based on 6% annual inflation and a 10% interest rate.

These ratios reflect the opinion of Liebert Corporation engineers based on available published catalog information.

Liebert Corporation

Environmental Systems

Energy Analysis

5/16/86

For
FERMI LABS
STEVE KRSTULOVICH

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1050 Dearborn Drive, Columbus, Ohio 43229

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Page: 1

Stepped Operation Environmental Energy Analysis
For **FERMI LABS**

Pre-report summary
The energy cost of 1 FE 192GU (60 Hz) units for one year is $ **9398**

Design Criteria
Region Chicago, Illinois
Room Drybulb.................. 72 F
Room Relative Humidity....... 50 %
Main Fan Motor............ 5.00 HP
Number of Units............... 1
Design Ambient............... 95 F

Model Unit..........FE 192GU (60 Hz)
Air Flow per Unit.......... 8400 CFM
No. of Stages per Unit 4
Glycol Pump Motor.......... 3.00 HP
Drycooler Fan Motor......... 2.25 HP

Performance and Operation Breakdown

Temperature Range (F)	min-35	35-39	40-44	45-49	50-54	55-59	60-64
Mean Coincident WB (F)	30.0	34.0	38.0	43.0	47.0	52.0	57.0
Humidifier Cap (lb/hr)	17.4	17.4	17.4	17.4	17.4	17.4	17.4
Total Room Load(KBTUH)	164.2	164.2	164.2	164.2	164.2	164.2	164.2
Sens Room Load (KBTUH)	156.4	156.4	156.4	156.4	156.4	156.4	156.4
Step 1 Totl Cap(KBTUH)	54.0	54.0	55.5	56.6	57.4	60.5	63.1
Sens Cap(KBTUH)	48.8	49.5	51.4	53.0	54.1	60.5	63.1
Step 2 Totl Cap(KBTUH)	107.9	108.1	110.9	113.2	114.8	121.1	126.2
Sens Cap(KBTUH)	97.6	99.0	102.8	106.1	108.3	121.1	126.2
Step 3 Totl Cap(KBTUH)	137.4	137.7	140.9	143.7	146.2	150.6	155.3
Sens Cap(KBTUH)	120.7	121.7	125.6	129.1	132.2	140.2	145.1
Step 4 Totl Cap(KBTUH)	167.0	167.4	170.9	174.2	177.7	180.0	184.5
Sens Cap(KBTUH)	143.8	144.4	148.4	152.0	156.2	159.2	164.0
% Load Free-Cooled	**94.4**	**86.5**	**70.8**	**55.1**	**39.3**	**23.6**	**7.9**
Econ-o-Coil Cap(KBTUH)	**147.6**	**135.3**	**110.7**	**86.1**	**61.5**	**36.9**	**12.3**

Breakdown of Energy Use

		min-35	35-39	40-44	45-49	50-54	55-59	60-64
Fan Motor	Max (KW)	3.7	3.7	3.7	3.7	3.7	3.7	3.7
	x %Oper	100.0	100.0	100.0	100.0	100.0	100.0	100.0
	= Actual (KW/H)	3.7	3.7	3.7	3.7	3.7	3.7	3.7
Step 1	Max (KW)	2.4	2.4	2.3	2.3	2.2	2.2	2.2
	x %Oper	18.0	42.6	88.9	67.5	24.7	2.6	0.0
	= Actual (KW/H)	0.4	1.0	2.0	1.6	0.5	0.1	0.0
Step 2	Max (KW)	4.8	4.8	4.6	4.6	4.4	4.4	4.4
	x %Oper	0.0	0.0	0.0	32.5	75.3	97.4	5.3
	= Actual (KW/H)	0.0	0.0	0.0	1.5	3.3	4.3	0.2
Step 3	Max (KW)	8.4	8.4	8.3	8.4	8.2	8.3	8.2
	x %Oper	0.0	0.0	0.0	0.0	0.0	0.0	94.7
	= Actual (KW/H)	0.0	0.0	0.0	0.0	0.0	0.0	7.8
Step 4	Max (KW)	12.0	12.0	12.0	12.2	12.0	12.2	12.0
	x %Oper	0.0	0.0	0.0	0.0	0.0	0.0	0.0
	= Actual (KW/H)	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Humidifier	Max (KW)	6.4	6.4	6.4	6.4	6.4	6.4	6.4
	x %Oper	0.0	0.0	0.0	0.0	0.0	0.0	11.1
	= Actual (KW/H)	0.0	0.0	0.0	0.0	0.0	0.0	0.7
Reheat	Max (KW)	25.0	25.0	25.0	25.0	25.0	25.0	25.0
	x %Oper	0.0	0.0	0.0	0.0	0.0	0.0	0.0
	= Actual (KW/H)	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Glycol Pump & Drycooler	Max (KW)	3.9	3.9	3.9	3.9	3.9	3.9	3.9
	x %Oper	100.0	100.0	100.0	100.0	100.0	100.0	100.0
	= Actual (KW/H)	3.9	3.9	3.9	3.9	3.9	3.9	3.9
Total KW/Hour.........		8.1	8.7	9.7	10.7	11.5	12.0	16.4
x Hours/Year........		2521	725	572	565	581	604	700
x Cost/KWHR		$0.080	$0.080	$0.080	$0.080	$0.080	$0.080	$0.080
		=====	=====	=====	=====	=====	=====	=====
= Cost/Unit		$ 1629	$ 503	$ 443	$ 483	$ 535	$ 579	$ 916

These ratios reflect the opinion of Liebert Corporation engineers based on available published catalog information.

Liebert Corporation

1050 Dearborn Drive, Columbus, Ohio 43229

Stepped Operation Environmental Energy Analysis
For **FERMI LABS**

Performance and Operation Breakdown (cont.)

Temperature Range (F)	65-69	70-74	75-79	80-84	85-89	90-max
Mean Coincident WB (F)	61.0	64.0	67.0	70.0	72.0	74.0
Humidifier Cap (lb/hr)	17.4	17.4	17.4	17.4	17.4	17.4
Total Cool Load(KBTUH)	164.2	164.2	164.2	164.2	164.2	164.2
Sens Cool Load (KBTUH)	156.4	156.4	156.4	156.4	156.4	156.4
Step 1 Totl Cap(KBTUH)	61.9	60.5	59.1	57.7	56.4	55.0
Sens Cap(KBTUH)	61.9	60.5	59.1	57.7	56.4	55.0
Step 2 Totl Cap(KBTUH)	123.7	121.0	118.3	115.5	112.8	110.0
Sens Cap(KBTUH)	123.7	121.0	118.3	115.5	112.8	110.0
Step 3 Totl Cap(KBTUH)	154.6	151.0	147.3	143.6	139.9	134.9
Sens Cap(KBTUH)	144.8	142.5	140.2	137.9	135.6	132.7
Step 4 Totl Cap(KBTUH)	185.5	181.0	176.4	171.7	167.0	159.9
Sens Cap(KBTUH)	165.9	164.0	162.2	160.3	158.3	155.4
% Load Free-Cooled	0%	0%	0%	0%	0%	0%
Econ-o-Coil Cap(KBTUH)	0.0	0.0	0.0	0.0	0.0	0.0

Breakdown of Energy Use (cont.)

		65-69	70-74	75-79	80-84	85-89	90-max
Fan Motor	Max (KW)	3.7	3.7	3.7	3.7	3.7	3.7
	x %Oper	100.0	100.0	100.0	100.0	100.0	100.0
	= Actual (KW/H)	3.7	3.7	3.7	3.7	3.7	3.7
Step 1	Max (KW)	2.6	2.8	3.1	3.3	3.6	3.8
	x %Oper	0.0	0.0	0.0	0.0	0.0	0.0
	= Actual (KW/H)	0.0	0.0	0.0	0.0	0.0	0.0
Step 2	Max (KW)	5.2	5.6	6.2	6.6	7.2	7.6
	x %Oper	0.0	0.0	0.0	0.0	0.0	0.0
	= Actual (KW/H)	0.0	0.0	0.0	0.0	0.0	0.0
Step 3	Max (KW)	8.7	9.3	10.0	10.5	11.2	11.7
	x %Oper	45.0	35.3	26.4	17.4	8.4	0.0
	= Actual (KW/H)	3.9	3.3	2.6	1.8	0.9	0.0
Step 4	Max (KW)	12.2	13.0	13.6	14.4	15.0	15.6
	x %Oper	55.0	64.7	73.6	82.6	91.6	100.0
	= Actual (KW/H)	6.7	8.4	10.0	11.9	13.7	15.6
Humidifier	Max (KW)	6.4	6.4	6.4	6.4	6.4	6.4
	x %Oper	43.2	36.2	26.4	15.2	3.0	0.0
	= Actual (KW/H)	2.8	2.3	1.7	1.0	0.2	0.0
Reheat	Max (KW)	25.0	25.0	25.0	25.0	25.0	25.0
	x %Oper	0.0	0.0	0.0	0.0	0.0	0.0
	= Actual (KW/H)	0.0	0.0	0.0	0.0	0.0	0.0
Glycol Pump & Drycooler	Max (KW)	3.9	3.9	3.9	3.9	3.9	3.9
	x %Oper	100.0	100.0	100.0	100.0	100.0	100.0
	= Actual (KW/H)	3.9	3.9	3.9	3.9	3.9	3.9
Total KW/Hour..........		21.0	21.7	22.0	22.3	22.5	23.2
x Hours/Year.........		759	681	487	324	165	64
x Cost/KWHR		$0.080	$0.080	$0.080	$0.080	$0.080	$0.080
		=====	=====	=====	=====	=====	=====
= Cost/Unit		$ 1277	$ 1180	$ 857	$ 579	$ 297	$ 119

Cost Summary

Unit energy cost per year.......................... $ 9398
Energy cost for 1 FE 192GU units per year......... $ 9398
Ten year projected energy cost..................... $ 123872
Present value of ten year energy cost............... $ 80001

Note: This projection is based on 6% annual inflation and a 10% interest rate.

These ratios reflect the opinion of Liebert Corporation engineers based on available published catalog information.

DOE 6430.1
CHAPTER XIII
ENERGY CONSERVATION GUIDELINES

CHAPTER XIII

ENERGY CONSERVATION AND USE OF RENEWABLE ENERGY SOURCES

1. COVERAGE.

   a. These criteria are particularly oriented to the Department's new buildings and building additions, their operating systems and energy using equipment. They shall be applied in the planning and design of such facilities with the objective of minimizing consumption of nonrenewable energy on a life cycle cost effective basis. Companion use shall be made of energy conservation-related design criteria in Chapters IV, V, and VI of this Order. For purposes of this chapter the term building shall be interpreted as new building and building additions, unless otherwise stated.

   b. The objective of minimizing consumption of nonrenewable energy on a life cycle cost effective basis also shall be applied in the planning and design of building and building systems alteration projects and other energy-using facilities (such as new central utilities plants, utility distribution systems, and exterior lighting systems).

   c. It is also necessary that buildings or other structures acquired by the Department, or by contractors or subcontractors for the Department, are energy efficient. These include preengineered metal buildings, in-plant fabricated modular/relocatable buildings, trailer units, and other buildings that may be acquired. The building envelope thermal transmittance values criteria in paragraph 6b and other applicable energy conservation criteria for mechanical-electrical systems identified in paragraph 6c shall be considered minimum criteria to be applied. These transmittance values criteria shall be reflected, to the maximum extent feasible, in specifications that are developed and applied in such facility acquisitions.

   d. This chapter contains a substantial amount of information that is deemed necessary to assure full recognition within the Department, and by contractors and subcontractors, of the Federal law, Executive order, and Federal regulation requirements as related to energy conservation and use of renewable energy sources in the Department's facilities.

2. FEDERAL LAW, EXECUTIVE ORDERS, REGULATIONS, AND DEPARTMENTAL DIRECTIVES.

   a. Title V of Public Law 95-619, "National Energy Conservation Policy Act," of 11-9-78, which states in part:

      (1) "It is the policy of the United States that the Federal Government has the opportunity and responsibility, with the participation of industry to further develop, demonstrate, and promote the use of

energy conservation, solar heating and cooling, and other renewable energy sources in Federal buildings." (section 542, part 3 Title V.)

(2) "All new Federal buildings shall be life cycle cost effective...." "In the design of new Federal buildings, cost evaluations shall be made on the basis of life cycle cost rather than initial cost." (section 545(b), part 3 Title V.)

(3) "The term 'Federal Building' means any building, structure, or facility which is constructed, renovated, or leased, or purchased in whole or in part for use by the United States, and which includes a heating system, a cooling system, or both." (section 544, part 3 Title V.)

b. Executive Order (E.O.) 12003, "Relating to Energy Policy and Conservation," of 7-20-77, amending E.O. 11912, "Delegations of Authorities Relating to Energy Policy and Conservation," of 4-13-76, which established the 45 percent energy-use reduction goal for new Federal buildings.

c. Subpart C, "Guidelines for Buildings Plans," of 10 CFR part 436, which requires in Section 436.51, "Design Program for New Federal Buildings":

(1) That each Federal agency shall provide in its buildings plan for the metering of building energy use in new Federal buildings; analysis of at least two alternative building designs, at least one of which includes a renewable energy system; and selection of a building design which minimizes total life cycle costs as measured in accordance with subpart A of 10 CFR part 436.

(2) That the design goal for a new Federal building shall be set by building category at the rate of building energy consumption equivalent to a reduction of 45 percent in average energy use per gross square foot of floor area in FY 1985, from the average building energy use per gross square foot of floor area of a representative Federal building of that category in FY 1975.

d. Subpart A, "Methodologies and Procedures for Life Cycle Cost Analyses," of 10 CFR part 436, which establishes a methodology for estimating and comparing the life cycle costs of Federal buildings and for determining life cycle cost-effectiveness. The methodology evaluates the economic consequences of investments in alternative building systems (energy conservation measures including renewable energy systems for existing buildings, and energy-saving building systems including renewable energy systems for new buildings).

e. DOE 4330.3, FUELS AND ENERGY USE POLICY, of 10-22-80, states in part, that "It is the policy of the Department to --- maximize the use of noncritical fuels, such as coal and solar, and minimize the use of the critical fuels, petroleum and natural gas, by discontinuation of the use of natural gas and petroleum in new facilities; --- (by) the increased use of residual (waste) energy; (by) emphasis on the use of new and advanced energy technologies; ---." See DOE 4330.3 for all requirements related to fuels and energy use, including controls on the use of electric resistance space heating in new facilities.

f. DOE 4330.2A, IN-HOUSE ENERGY MANAGEMENT PROGRAM, of 2-16-82, states that "It is the policy of DOE to promote efficient and economical use of energy in all DOE-owned or -leased facilities, including buildings and energy conversion and distribution systems, and DOE-owned or -operated vehicles and equipment implementing an In-House Energy Management Program for DOE," with an objective of "assuring energy efficient design of new facilities."

3. TIMING OF EVALUATIONS AND SELECTIONS OF ENERGY CONSERVATION FEATURES AND ENERGY SUPPLY SOURCES.

a. Integral elements of the facility planning and design process are evaluations and selections of energy conservation features and energy supply sources. To assure that the necessary energy conservation features are included in the overall project requirements, and associated cost are included in the official cost estimate prior to project authorization, evaluations and selections of features on the basis of life cycle cost effectiveness need to be made during:

(1) The conceptual design phase, for line item construction projects proposed for full project authorization in annual budget requests after performance of conceptual design. Where all such energy conservation and energy supply features cannot be identified during the conceptual design phase, suitable allowances will need to be included in the project authorization funding requests to assure the achievement of an energy-efficient facility in the follow-on design phases.

(2) The conceptual design phase, or in preliminary (Title I) or detailed (Title II) design prior to full project authorization for line item construction projects that are to initially receive partial authorization (e.g., for architect-engineer work only, prior to full project authorization).

(3) The planning phase (e.g., conceptual design or other project planning), or in preliminary or detailed design prior to full project authorization, for contingency-type projects.

(4) The planning phase, or in preliminary or detailed design prior to full project authorization, for general plant projects.

b. Evaluations of energy conservation and supply alternatives will usually be required through the preliminary design and into the final design phase. These evaluations usually include new or updated life cycle cost analyses, and often result in different systems being selected than originally conceptualized. Therefore, careful planning is required during the conceptual phase to ensure that the conceptual cost estimates provide contingencies to cover the changes in selected systems that are likely to occur during the design process.

c. Fuels and energy selections shall be in conformance with DOE 4330.3, FUELS AND ENERGY USE POLICY, of 10-22-80.

4. LIFE CYCLE COST ANALYSES.

a. Life cycle costing involves a systematic comparison of investment decisions using a discount factor to calculate the present worth of future benefits and costs. Prescribed "Methodology and Procedures for Life Cycle Cost Analyses," promulgated by the Assistant Secretary for Conservation and Renewable Energy, as a final rule in the Federal Register of 1-23-80 (subpart A to 10 CFR part 436), as amended, shall be utilized.

b. The latest edition of the "Life Cycle Costing Manual for Federal Energy Management Programs," NBS Handbook 135, shall be used, as supplemented with additional DOE-specific requirements developed and promulgated by the In-House Energy Management Branch, Office of Project and Facilities Management, Directorate of Administration, at DOE Headquarters. Single copies of the Handbook may be obtained from the Department's Technical Information Center, P.O. Box 62, Oak Ridge, TN, 37830.

5. USE OF COMPUTER OR OTHER ENERGY ANALYSIS TECHNIQUES.

a. For most of the Department's buildings, a suitable type of computer analysis (dynamic or static) technique or other automated analysis technique (such as the use of programmable calculators), or combinations thereof, shall be used to evaluate energy conservation alternatives. Such computer techniques shall also be used to develop energy-efficient building design concepts and determine the design energy consumption (i.e., estimate of building energy use). The term, "building energy use," means energy use that is principally for heating, ventilation, cooling, domestic hot water, and lighting. Exception to the use of these techniques may be taken for small buildings such as small utility-type buildings, or other buildings with relatively low projected building energy use, where manual analysis methods may be adequate. In general, such exceptions could be taken when building energy use is not expected to exceed 500 million Btu per year (apply conversion values for electricity of 3,412 Btu/kilowatt hour and 1,000 Btu/pound for steam).

b. Analysis techniques in general declining order of sophistication and analysis capabilities, and detail of input data required, are briefly described below:

(1) Dynamic computer analysis techniques, such as DOE 2 (and DOE 2.1) or BLAST, using mainframe CPU (central processing units)/time-sharing computer systems. These techniques provide capability for hour-to-hour load and energy consumption analysis over a full-year study period (see paragraph c, below, for additional information).

(2) Static computer analysis techniques, using mainframe CPU/time-sharing computer systems. These techniques utilize load estimate inputs for peak load periods and generally provide capability for 3-hour interval energy consumption analysis over a full-year study period.

(3) Other static analysis techniques include:

(a) Use of smaller, "minicomputer," systems (present generation not capable of handling the more detailed programs).

(b) Use of simpler, "microcomputer," systems (lower range machine capability than minicomputers, but much faster and more capable than programmable calculators).

(c) Use of programmable calculators in table-top systems including a printer and magnetic card reader, with commercially available programs from calculator manufacturers, or others.

(d) Use of manual analysis methods (generally requiring only a 4-function calculator).

c. It is not within the scope of these criteria to dictate the specific automated analysis technique to be used, as this will need to be determined on a case-by-case basis for each building. However, since dynamic computer analysis techniques, such as DOE-2, currently provide the greatest flexibility and analysis capability, it is recommended that such dynamic techniques be used for the larger, more energy-consuming, buildings. Departmental elements having first-line responsibilities for the design of facilities should assure that capability for use of dynamic techniques, such as DOE 2, is available within their DOE or operating contractor organizations and application of these techniques is made, where feasible. All buildings and building additions of greater than 30,000 gross square feet in size shall be considered for dynamic analysis applications. The magnitude of projected building energy use, type of building occupancy, and opportunities for reducing energy use from non-renewable energy sources on a life cycle cost-effective basis, normally determine if these techniques should be used. There are, generally, four basic program phases in dynamic computer analysis and the related design process in developing energy-efficient building concepts.

(1) <u>Determine the building design heating and cooling loads based upon multiple architectural, mechanical, and electrical systems combinations</u>.

(2) Based on weather data, determine the annual thermal loads for phase (1), above.

(3) Simulate the operation of various mechanical-electrical environmental systems to the thermal load analysis in phase (2), above, to determine hourly, monthly, and annual energy consumption.

(4) Perform life cycle cost analysis for the systems evaluated in phases (1) and (3), above. (See paragraph 4b, above.)

d. Effective application of dynamic analysis techniques, and the more sophisticated static analysis techniques, may often require the type of design detail and other informational input that is developed in the preliminary (Title I) design phase, and sometimes continuing into the detailed (Title II) design phase. Where this is the case, other automated analysis techniques, requiring commensurately less informational input, should be used in the project planning phase for the larger building projects with the objective of identifying the principal energy conservation features. At a minimum, the use of programmable calculator techniques is recommended during the project planning phase for all but the very small building projects, where manual analysis techniques may be adequate. The use of mini- or micro-computer analysis techniques is recommended for other building projects, and these may also be adequate for use during final design for some of the less-complex, less energy-using buildings. However, where the capability exists to use the more sophisticated techniques, such as DOE 2, they should be used wherever feasible. For the larger building projects, where such capability has not yet been developed or is not available from architect-engineer firms in the area, the lesser sophisticated computer analysis techniques should at least be used to assure as energy-efficient designs as practicable.

e. The inside temperatures to be used for building energy consumption analyses shall conform to Federal Property Management Regulations (FPMR), 41 CFR, Chapter 101, Subchapter D, Section 101-20.116 Inside Operating Temperature Requirements, except where the use of other less stringent inside operating temperatures is justified.

6. ENERGY CONSERVATION FEATURES FOR BUILDINGS.

a. Evaluation and Selection of Energy Conservation Features.

(1) Energy conservation shall be given its full share of attention in the planning and design, or acquisition, of DOE buildings. While the basic programmatic or operating requirements must be the principal "driving force" in the development of the building concept and its design, the Federally-mandated requirements to maximize energy conservation on a life cycle cost-effective basis, and with regard to the 45 percent energy-use reduction goal must also be satisfied. Incorporation of energy conservation features on the basis of their life cycle cost-effectiveness will result in some additional

first-costs for building projects. For a typical building project, present indications are that the additional construction costs (first costs) can be on the order of 3-5 percent, or more, depending upon the types and numbers of energy conservation features that are determined to be cost-effective and included in the project. When renewable energy systems, such as active solar systems, are determined to be cost-effective, and included in the project, there will be additional construction costs. From experience, to date, there are indications that these additional construction costs could be on the order of 5-10 percent, or more, for typical building applications.

(2) The additional first-costs for building projects or additional acquisition costs for other buildings such as pre-engineered metal buildings or in-plant fabricated modular buildings, from an energy-use efficiency standpoint, need to be included in the total project cost estimate. Cost allowances also need to be included in the cost estimate for the performance of energy analysis and life-cycle costing evaluations and for contingencies, the same as for other project elements.

(3) Insulating characteristics of the building envelope are of paramount importance with relation to energy conservation for building heating and cooling, and are an integral and dependent element of the architectural and structural building design concept. Therefore, development of basic insulation characteristics shall be considered a prerequisite to follow on evaluation of other energy conservation alternatives on the basis of their life cycle cost-effectiveness. The basic criteria to be applied in architectural and structural planning and design are contained in paragraph b, below and in paragraph 11(1), Chapter IV of this Order.

(4) Evaluation of other energy conservation features (i.e., energy-related building components and systems) shall be based on application of life cycle costing methodologies. Care should be taken to assure that the combination selected will best meet the minimum life cycle cost objective to maximize the net dollar benefits, comparing total energy conservation costs with total energy cost savings. Meeting the objectives is not, usually, only a matter of selecting energy conservation features that are determined individually to be life cycle cost-effective, because of the interdependence that may exist among the different features.

(5) The 45 percent building energy-use reduction goal, as described in paragraph 9 below, shall be applied in planning and designing each building project, in the planning and acquisition of each new pre-engineered metal building and in-plant fabricated modular building, and in the planning and acquisition of other semi-permanent or temporary facilities to the extent that technical specifications can be applied in their acquisition.

(2) Special attention shall be given to providing energy-efficient central air conditioning units for trailer units or other small buildings or facilities, where cooling is required. The seasonal energy efficiency ratios to be used in specifying equipment below 65,000 Btu/hour in capacity should not be less than 8 Btu/Watt-hour. By definition, "seasonal energy efficiency ratio" (SEER) means the total cooling of a central air conditioner in Btu during its normal annual usage period for cooling, divided by the total electrical energy input in watt-hours during the same period.

(3) Utilization of waste heat and waste heat recovery systems shall be evaluated for building projects, central utilities plants, and for other significant heat-producing process equipment and facility projects. Wherever technically feasible and life cycle cost-effective, heat recovery systems shall be incorporated.

(4) Utilization of cooling energy storage shall be evaluated for building projects, as appropriate. Chilled water storage based on a daily or weekly cycle can significantly reduce initial plant equipment, maintenance, and energy costs (particularly electrical demand charges). Wherever technically feasible and life cycle cost-effective, such energy storage systems should also be evaluated as an emergency supplemental water supply source for fire suppression.

7. ENERGY CONSERVATION FEATURES FOR OTHER PROJECTS. Energy conservation features that are life cycle cost effective shall be included in other facility projects, such as renovations of building, area, and site utility systems and central utilities plants.

8. USE OF RENEWABLE ENERGY SYSTEMS. Subsection (c) of Section 436.51, "Design Program for New Federal Buildings," 10 CFR part 436, "Federal Energy Management and Planning Programs," requires that "Each Federal agency shall plan to install one or more active or passive solar or other renewable energy systems to provide energy for building energy use unless the Federal agency states in its annual report that such a system would not minimize total life cycle costs...." Subsection (a)(3) requires that each Federal agency shall provide, in its Buildings Plan, "For analysis of at least two alternative building designs under Subpart A of this part at least one of which includes a renewable energy system...." Consistent with these regulations, the criteria for the Department's new buildings, building additions, and alterations are as follows:

a. Active Solar Systems. The application of active solar systems shall be evaluated for building projects. These are solar heating and/or cooling systems in which thermal storage devices other than the building mass are used and where thermal energy is transferred in a completely regulated way by pumps or fans. Active solar systems shall be provided wherever they are determined to be technically feasible, life cycle cost effective, and where the total project cost will not exceed applicable statutory limits.

features. Passive solar applications are also interrelated with or interdependent on such design parameters and requirements as building size, climatic conditions, building functions, levels of internally-generated thermal loads (mechanical-electrical systems, equipment, occupancy, and so forth) and space environmental requirements for occupants and functions.

(4) A source of guidance in the evaluation and design for passive solar applications is the two-volume DOE "Passive Solar Design Handbook": Volume One, "Passive Solar Design Concepts" of 3-80, DOE/CS-0127/1 and Volume Two, "Passive Solar Design Analysis" of 1-80, DOE/CS-0127/2. Single copies, or a set, may be obtained by request to the Department's Technical Information Center, P.O. Box 62, Oak Ridge, TN 37830.

c. Other Renewable Systems. The opportunity for application of other renewable energy sources in facility projects (photovoltaics or other renewable energy applications) should be evaluated on a case-by-case basis. Evaluations and selections should be based on the application of life cycle costing methodologies.

9. ENERGY-USE REDUCTION GOALS FOR NEW DOE BUILDINGS.

a. Application of the 45 Percent Reduction Goal.

(1) Executive Order (E.O.) 12003 and Federal Regulations, 10 CFR part 436 established two energy use reduction goals for owned and leased new Federal buildings. Both goal's "average energy use" is computed on the basis of annual use per gross square foot of floor area.

(a) The individual new building design goal of a 45 percent reduction in average building energy use over the average energy use of a representative Federal building of the same category completed prior to FY 1975.

(b) The overall Department "end goal" of 45 percent reduction in average energy use for all new buildings in FY 1985 over the average energy use for all buildings of the Department in FY 1975.

(2) For application purposes, a building shall be considered new if either of the following apply:

(a) Construction was not complete prior to November 9, 1978, and design could be feasibly modified after November 14, 1979; or

(b) Design started after November 9, 1979.

(3) For application purposes, a leased building is included if construction had not started prior to July 20, 1977.

In determining if the design can be feasibly modified, such factors as schedule effects and cost limitations shall be considered.

(4) It is important to keep in mind that the Department's "end goal" is a 45 percent reduction in average energy use per gross square foot for the total of all its new buildings in Fy 1985, from the average energy use per gross square foot of existing buildings in FY 1975. To achieve the "end goal" the average energy use reduction of all new buildings must be equal to 45 percent. This can only be met if all new buildings equal or exceed the individual building goal, or new buildings exceeding the 45 percent goal offset those that do not meet the individual buildings goal.

(5) All new DOE buildings will not have the same opportunities for reducing building energy use. It may not be possible for some buildings to achieve the 45 percent reduction goal, such as warehouse/storage facilities with minimal heating and lighting requirements and no cooling requirements. If DOE is to fully achieve the reduction goal for all of its new buildings, special attention needs to be given to meeting the goal for the energy-using buildings having large potentials for energy-use reduction, and to exceed the goal for these facilities, wherever feasible.

(6) Subsection (a)(3) of section 436.51, 10 CFR part 436, requires each Federal agency to provide in its Buildings Plan "For analysis of at least two alternative building designs using Life Cycle Cost Analysis, at least one of which includes a renewable energy system. Both alternative designs must be consistent with budget limitations and basic requirements for heating, ventilation, cooling, lighting, domestic hot water, and functional purposes."

(a) This requirement shall not be construed as requiring two completely different overall building design concepts for all new DOE buildings. The Final Rule on this subject, as published in the Federal Register of 11-14-79, contains the following clarifying information and guidance.

"The purpose of requiring analysis of more than one design is to promote a minimum level of exploration of design alternatives. Section 436.51 should be interpreted broadly. There should at least be a comparison to two or more component designs within a common approach. For more complex designs, DOE encourages, but will not mandate, a comparison of two completely different overall designs which could also have component design variances. Each agency should analyze reasonable alternatives since each agency has the ultimate responsibility in achieving the most 'cost-effective' design consistent with the energy reduction goal of 45 percent."

(b) Therefore, in applying this requirement the fundamental objective will be to evaluate alternative building systems and other energy-related features (components) and make selections from the standpoint of life cycle cost-effectiveness. The term, "building system," as defined in 10 CFR part 436, "means any part of the structure of a Federal building significantly affecting building energy use, or any energy using system contributing to building energy use."

1 Where a single overall building design concept, will not be expected to achieve the 45 percent reduction goal, an alternate concept may need to be evaluated. However, engineering judgment is needed here. For example, small buildings, or other buildings with low projected energy consumption and having limited opportunity for significant improvement in energy-use, the additional cost for making alternate design studies may often exceed the benefits to be achieved. For larger energy-using buildings or building additions having greater opportunities for significant improvements in energy efficiency, evaluation of alternate overall building design concepts may need to be made.

2 For the larger or more complex building projects, alternate overall building design concepts may often be evaluated for reasons other than for energy conservation specifically, such as to develop the best concept for satisfying programmatic or operating needs, which is the fundamental requirement in the planning and design or acquisition, of any building. In the evaluation of such alternate concepts, energy conservation shall be given its share of attention. Where energy conservation can be the discriminating factor between programmatic or operating needs options, it shall be applied in the option selection.

b. Estimating Energy Use of a Representative Building in FY 1975. For application of the 45 percent reduction goal to a new building, a reasonably accurate estimate of the building energy use for a representative building of the same category (and in the same general climatic region) in FY 1975, is needed. In many cases, accurate data will not be available, and a "best-judgment" approach will need to be taken. However, where computer or other automated analysis techniques are being used for energy analysis of the new building, during planning or design, a "simulation" of energy use for this building, as if it had been constructed in early-to-mid 1970's, may be a feasible method. This and other possible approaches are described below:

(1) By means of an energy audit of an existing building of that category on the particular site, or in the general area, that was completed in the FY 1970 - FY 1975 time period.

(2) From metered energy use of such existing buildings, either onsite or in the general area, when available.

(3) By a simulated energy-use analysis of the new building that is under planning or design, by applying the design and construction standards and criteria that were in use, by the Department or architect-engineer firms in the general locality during the FY 1970 - FY 1975 time period. (The then-applicable building envelope insulation standards; building design temperature criteria; interior illumination levels; building fenestration criteria; types of heating, ventilating, and air conditioning systems used; building operation features, such as HVAC controls, lighting controls). In making comparisons of the estimated building energy use of a new building with respect to the 45 percent reduction goal, exclude any estimate of energy to be supplied from a renewable energy source.

10. ESTIMATES OF ADDITIONAL CONSTRUCTION COSTS FOR NEW DOE BUILDINGS.

a. Federal Regulations in 10 CFR part 436, require that estimates be made and documented of the additional construction cost attributable to incorporation of energy conservation systems into new building designs, in order to achieve the 45 percent goal. The regulations in 10 CFR part 436 further require estimates of the related energy cost savings over the projected useful facility life. The term 'alternative building system' means a primarily energy-saving building system, including a renewable energy system, for consideration as part of the design for a new Federal building.

b. The above regulation provisions can be met by properly structuring the life cycle cost analysis required and described in this chapter.

c. Estimates of additional construction costs and energy cost savings for new DOE buildings are to be included in the Energy Conservation Report for the project. See paragraph 14, below.

11. BUILDING ENERGY PERFORMANCE STANDARDS.

a. The Department of Energy has responsibilities for the development and promulgation of "energy performance standards" for new commercial and residential buildings. Proposed Rulemaking (subpart A of 10 CFR part 435) was published in the Federal Register of 11-28-79. Section 306 of the "Energy Conservation Standards for New Buildings Act of 1976," requires that Federal agencies assure that new Federal buildings meet or exceed the applicable performance standards when they are promulgated for use. This requirement is also stated in section 436.52(b), subpart C of 10 CFR part 436.

b. When promulgated, applicable building performance standards for Federal buildings shall not be construed as limiting further reduction in energy use of the Department's new buildings where such further reduction can be achieved on the basis of life cycle cost analysis. The "energy performance standards" do not relate to the 45 percent building energy reduction goal for new Federal buildings. When the energy performance standards are promulgated for use, the lesser energy-use value, between the applicable energy performance standard and the energy use value necessary to achieve the 45 percent reduction goal, shall be applied in the planning and design, or acquisition, of new DOE buildings.

12. ENERGY MANAGEMENT SYSTEMS AND DEVICES.

a. Energy management systems and devices are characterized by their ability to control energy consuming systems or equipment. Examples of simple devices are manual valves and electrical switches. An example of a simple automatic energy management device is a single set-point thermostat. Moving to higher levels of complexity, technology, and methodology, a typical example would be a computer-based energy monitoring and control system (EMCS).

b. The application of energy management systems and devices will often directly interface with the architectural, mechanical and electrical design requirements for a building. They also interface with telecommunications system design requirements where telecommunication circuits are used both within the building and from building-to-building on the site. Close coordination needs to be maintained among all affected design disciplines, during the planning and design of energy management systems.

c. Energy monitoring and control systems (EMCS) usually consist of a central computing system with peripheral equipment, data transmission media, field interface devices, multiplex panels, necessary interfacing controls, and sensors. Each field interface device will generally contain a microcomputer that performs certain local control functions in a stand-alone mode of operation.

(1) Criteria and methodology for the design of an EMCS should be obtained from U.S. Army Corps of Engineers Technical Manual, TM 5-815-2, "Energy Monitoring and Control Systems."

(2) Guidance for preparation of design and procurement documents should be obtained from the following U.S. Army Corps of Engineers Guide Specifications.

(a) CEGS-13947, "Large Energy Monitoring and Control Systems."

(b) CEGS-13948, "Medium Energy Monitoring and Control Systems."

(c) CEGS-13949, "Small Energy Monitoring and Control Systems."

(d) CEGS-13950, "Micro Energy Monitoring and Control Systems."

(3) A methodology for estimating the economics and energy conservation performance of an EMCS may be obtained by adapting the guidance in Navy Civil Engineering Laboratory document, "EMCS Economic Analysis Guideline," PO No. 78 MR 423, to be consistent with the "Life Cycle Cost Analysis," of this Chapter XIII.

13. ENERGY METERING.

a. The intent of these metering criteria is to:

(1) assure that all significant utilization of energy in all of the Department's new owned and leased buildings and facilities is metered in accordance with 10 CFR part 436;

(2) provide submetering for process and production energy usage within the Department's new buildings and facilities;

(3) provide a means for validation of the Department's progress toward goals for effective energy efficient design in new buildings (see paragraph 2c(2));

(4) assure that the Department's new buildings and facilities are provided with sufficient metering to facilitate compliance with the energy usage reporting requirements of 10 CFR 436 and DOE 4330.2A.

(5) measure the effectiveness of corrective and energy conserving actions taken during the operation of a building; and

(6) identify capabilities for emergency energy use reduction during periods of shortages and curtailments.

b. Permanent metering shall be provided for each type of energy supplied to and consumed by the Department's owned and leased new buildings and facilities with the following exemptions:

(1) Permanent metering of the energy supplies to small buildings and facilities, and other buildings and facilities having relatively low total energy usage is not required. This exemption may generally be appropriate for a new temporary office facility, new perimeter guard station buildings, small storage and utility buildings, and other facilities where the individual total energy usage is not expected to exceed 500 million Btu per year. (Apply conversion values of 3,412 Btu/kilowatt hour for electricity and 1,000 Btu/pound for steam.)

(2) Normally, permanent metering is not required for a type of energy supply that is estimated to supply 10 percent, or less, of the total energy input to the building.

(3) Metering of energy supplied from a renewable energy source or from waste heat and waste heat recovery generally will not be required. However, where energy metering is feasible and if the information

gained is to be used to evaluate the effectiveness of the renewable energy systems or controls, permanent metering or features for ease of temporary metering should be provided.

c. Permanent submetering shall be provided for each type of process and production energy consumed in the Department's owned and leased new buildings and facilities except where the cost of providing the submetering becomes excessive compared to the management benefits gained. When permanent submetering is not provided, a discussion of the rationale for the exclusion shall be included in the Energy Conservation Report documentation. (See paragraph 14a(4).)

(1) Process energy means energy for production and research processes and does not include building energy. Building energy means energy used principally for heating, ventilating, cooling, domestic hot water and lighting.

(2) Generally, where the total process use is only about 10 percent or less of the total energy input to the building that total energy input may be considered as "building energy use." Conversely, where the total process energy use is about 90 percent or more of the total energy input to the building that total energy input may be considered as "process energy use."

d. In order to comply with the provisions of 10 CFR part 436 and DOE 4330.2A, the Department reports energy consumption in its buildings and facilities, establishes consumption goals, and develops 10-year Plans for "buildings energy" and "metered process energy" consumption. The metering features incorporated into the design will predetermine the manner in which energy consumption will be reported for new buildings or facilities as follows:

(1) If a facility has submetering to separate building energy and process energy usage within the facility, then;

(a) the building energy and the process energy will be separately reported as "building energy" consumption and "metered process energy" consumption in the Energy Conservation Report documentation and the Quarterly Energy Conservation Performance Report required by DOE 4330.2A.

(b) the development of the estimated total annual energy consumption that will be included in the Energy Conservation Report documentation (see paragraph 14) must be separated into building energy and metered process energy corresponding to the meters and submeters incorporated in the facility. The separation by energy use category is necessary for the reporting and subsequent validation of the Department's progress toward goals for energy efficient design in new buildings.

(2) If a building or facility is not provided with submetering for process or production energy usage, then:

(a) those buildings or facilities with predominant building energy usage will have the entire energy usage reported as building energy in the Energy Conservation Report documentation and the Quarterly Energy Conservation Performance Report (QECPR) required by DOE 4330.2A;

(b) those buildings or facilities with predominant process energy usage will have the entire energy usage reported as metered process energy in the Energy Conservation Report and the QECPR; and

(c) The decision not to provide submetering and the subsequent determination to report the entire energy usage of a building or facility as all building or all metered processes energy must be made by the appropriate DOE Energy Coordinator and an explanation of the decision and determination must be included in the Energy Conservation Report documentation.

e. Energy metering requirements need to be established prior to establishing the official project cost estimate for authorization to assure that the capital costs are properly included in the total project cost estimate.

f. In the selection of metering devices, proper consideration shall be given to compatibility for use with an existing or projected energy monitoring and control system (EMCS).

14. DOCUMENTATION.

a. Energy Conservation Report. An energy conservation report (summary evaluation) shall be developed for each new building, building addition, appropriate building alteration, and other energy-using projects. The initial report, covering such data and information as can be developed during the project planning phase, shall be included as a part of the appropriate project planning documents (conceptual design reports or other project planning documents). These initial analyses shall be updated at the end of preliminary (Title I) design and included as a part of the appropriate design documents (updated conceptual design reports, Title I design reports, or other Title I design documentation). They shall be further updated as a part of Title II design documentation, when final selections of energy conservation features or renewable energy sources are not made until the Title II design phase.

(1) Analyses for building or building addition projects shall include:

(a) Identification of methods used for building energy consumption analyses. This analysis includes loads and building systems analysis (computer dynamic analysis, other computer analysis, use of programmable calculator, or manual calculations).

(b) Methodology of life cycle costing analysis for the evaluation and comparison of energy conservation alternatives and use of renewable energy sources (computer dynamic analysis, other computer analysis, use of programmable calculator, or manual calculations).

(c) Description of the major energy conservation features selected, such as building envelope U values (or R values), type of fenestration and percent of gross wall area, type of air handling system, reheat systems, automatic system control features, central supervisory and control features, lighting levels and controls, and so forth.

(d) Results, including backup data (or identify source for obtaining data on an as-requested basis) of life cycle cost analyses of active or passive solar system applications or other renewable energy source applications, as appropriate.

(e) Discussion of evaluations made of nonrenewable energy supply alternatives, and basis for selection(s). Discussion shall include determination of conformance with the Department's fuels and energy use policy (DOE 4330.3).

(f) Estimates of total energy input to the building (see paragraph (3) below, for energy conversion values). Estimates shall be subdivided as follows:

1 For buildings incorporating submetering, separately identify building energy and metered process energy and the number of square feet associated with each type of energy usage. Include:

a Btu/year by types of energy.

b Total Btu/year.

c Btu/gross square foot/year.

2 For buildings without submetering, identify the energy usage type as either entirely building energy or entirely metered process energy and the number of square feet associated with the type of energy usage. Identification of energy usage types (building or metered process) and associated number of square feet must correspond to the actual metering and submetering features incorporated. Include:

a Btu/year by types of energy.

b Total Btu/year.

c Btu/gross square foot/year.

(g) Provide the following information with regard to the 45 percent energy-use reduction goal (see paragraphs 9 and 10, above):

1 Estimated baseline building energy use of a representative building of the appropriate category in FY 1975. Provide information corresponding to paragraph 14a(1)(f), above.

2 Estimated percentage of energy-use reduction (if less than the 45 percent goal, provide brief explanation).

3 Method used to estimate the baseline energy use of a representative building of the appropriate category in FY 1975 (see paragraph 9b, above).

4 Estimated additional construction costs to achieve the 45 percent goal, or to achieve the estimated percent reduction if either higher or lower than this goal (do not adjust these estimated costs to a 45 percent energy use reduction level).

5 Estimated energy cost savings over the projected life of the building (with respect to the additional investment costs in 4, above). Include the first year energy unit cost, the first year energy cost savings, and the present worth factor for each energy source included in the computation. The projected life of the building used in the computation shall also be included.

(h) Provide a comparison of estimated building energy use with applicable building energy performance standard for Federal buildings (see paragraph 11, above). This information is not required until such time as these standards are promulgated for required use.

(2) Reports for other energy-using facility projects (other than buildings/additions) shall include:

(a) Discussion of life cycle cost analyses made of energy conservation features and selections made, analyses of renewable energy source applications, evaluations made of nonrenewable energy supply alternatives and basis for selection(s), and conformance with the Department's fuels and energy use policy (DOE 4330.3).

(b) Estimates of energy use, and energy savings achieved, in Btu per year, by types of energy and total. See paragraph (3), below, for energy conversion values.

(3) Energy conversion values, from 10 CFR part 436, are listed below:

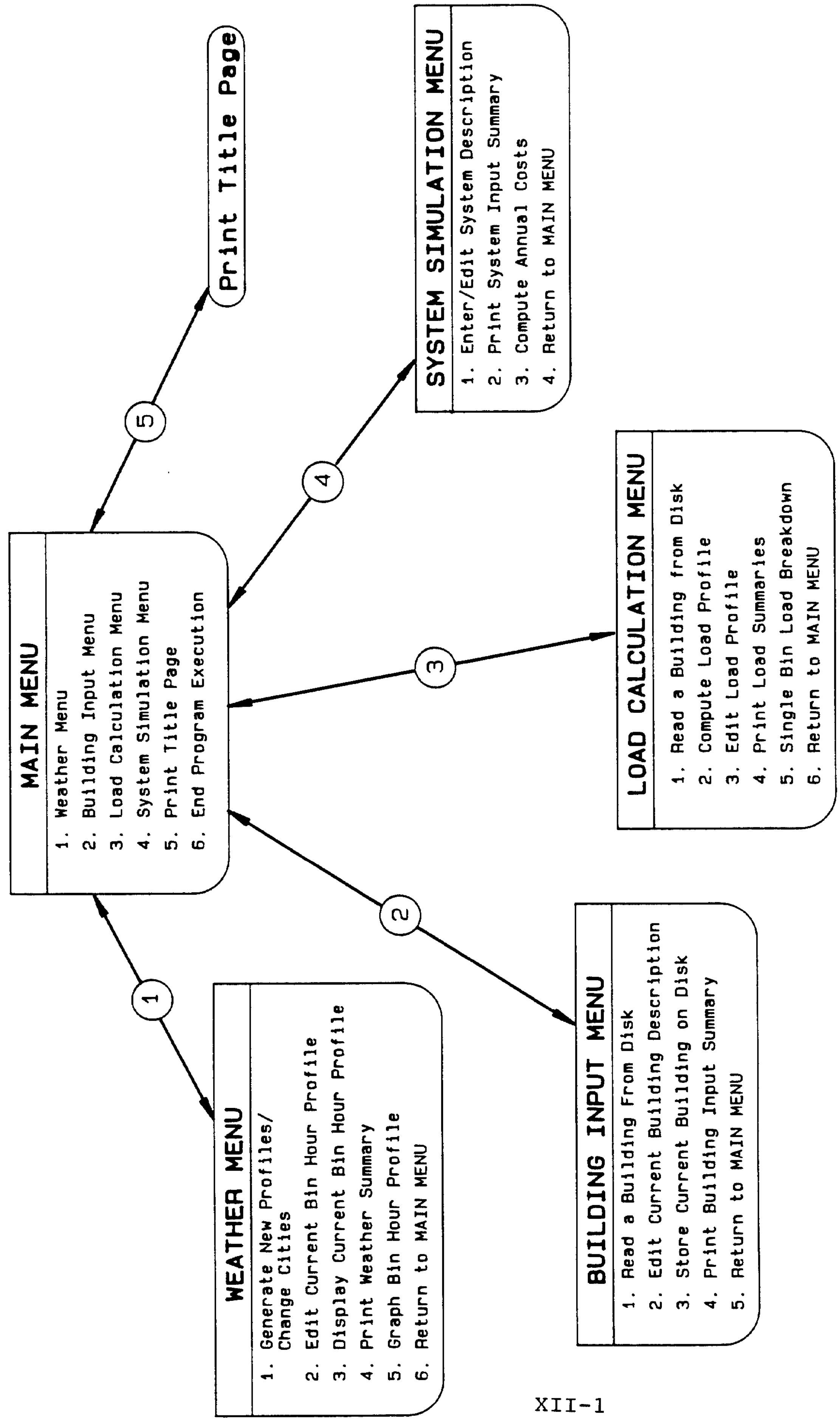
Electricity	11,600 and 3,412 Btu/kilowatt hour
Fuel Oil (distillate)	5,825,400 Btu/barrel
Residual Fuel	6,287,000 Btu/barrel
Natural Gas	1,030,000 Btu/1000 cubic feet
Liquified Petroleum Gas (LPG) including propane and butane	4,011,000 Btu/barrel
Coal	24,500,000 Btu/short ton
Steam (purchased)	1,390 and 1,000 Btu/pound
Energy sources not listed	Conversion factors from a standard engineering reference manual or other reliable reference.

The higher values for electricity and steam are only to be used in reporting energy use and energy savings. As stated in 10 CFR part 436, subpart 436.45, "---in calculating energy costs for life cycle costing purposes, only the conversion values of 3,412 Btu per kilowatt hour of electricity and 1000 Btu per pound of steam (purchased steam) shall be used."

(4) Energy metering provisions shall be discussed in the energy conservation report including types of permanent metering for energy inputs to the building, types of submetering for process energy use, compatibility with existing or projected energy monitoring and control systems (EMCS), and an estimate of the total costs for metering and submetering provisions. Include a narrative of the actions taken and the decisions and determinations made to assure compliance with the energy metering criteria in paragraphs 13c and 13d(2)(c). If metering provisions are not being made, provide brief explanation.

b. Distribution of Project Planning and Design Documents. In addition to other recipients of project planning and design documents (conceptual design reports, Title I design reports, Title II updates of information previously reported or not previously available, or other design documentation), DOE field organizations are requested to provide one copy of each document to the In-House Energy Management Branch, Office of Project and Facilities Management, at DOE Headquarters. The energy conservation reports and other directly-related information in these documents, is needed by the In-House Energy Management Branch for fulfillment of its energy management program responsibilities, including the reporting requirements under the provisions of 10 CFR part 436, "Federal Energy Management and Planning Programs."

FERMILAB ENERGY ANALYSIS
SOFTWARE DOCUMENTATION


MAIN MENU
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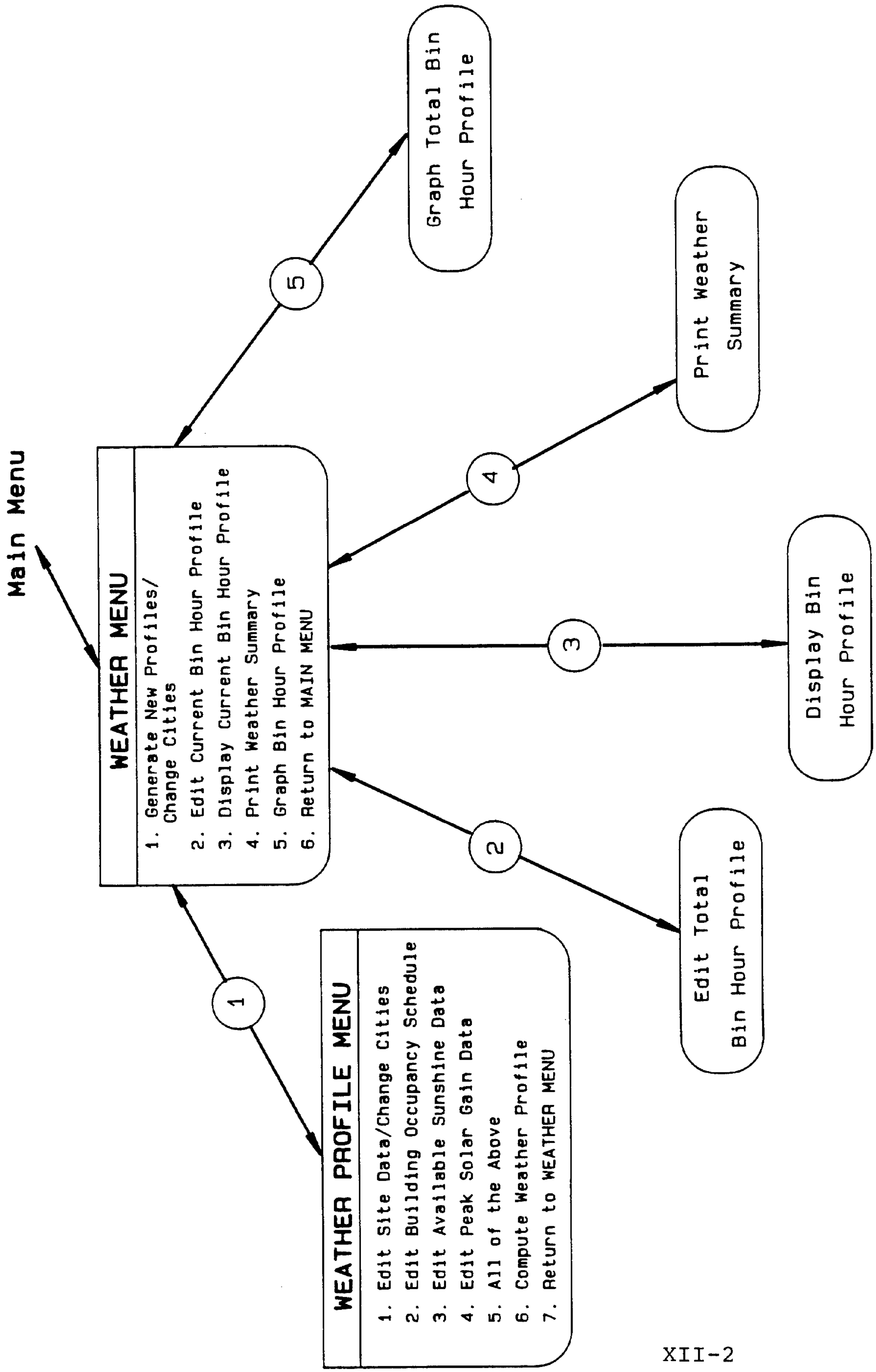
Main Menu
WEATHER MENU
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2. Edit Current Bin Hour Profile
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4. Print Weather Summary
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6. Return to MAIN MENU
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WEATHER PROFILE MENU
1. Edit Site Data/Change Cities
2. Edit Building Occupancy Schedule
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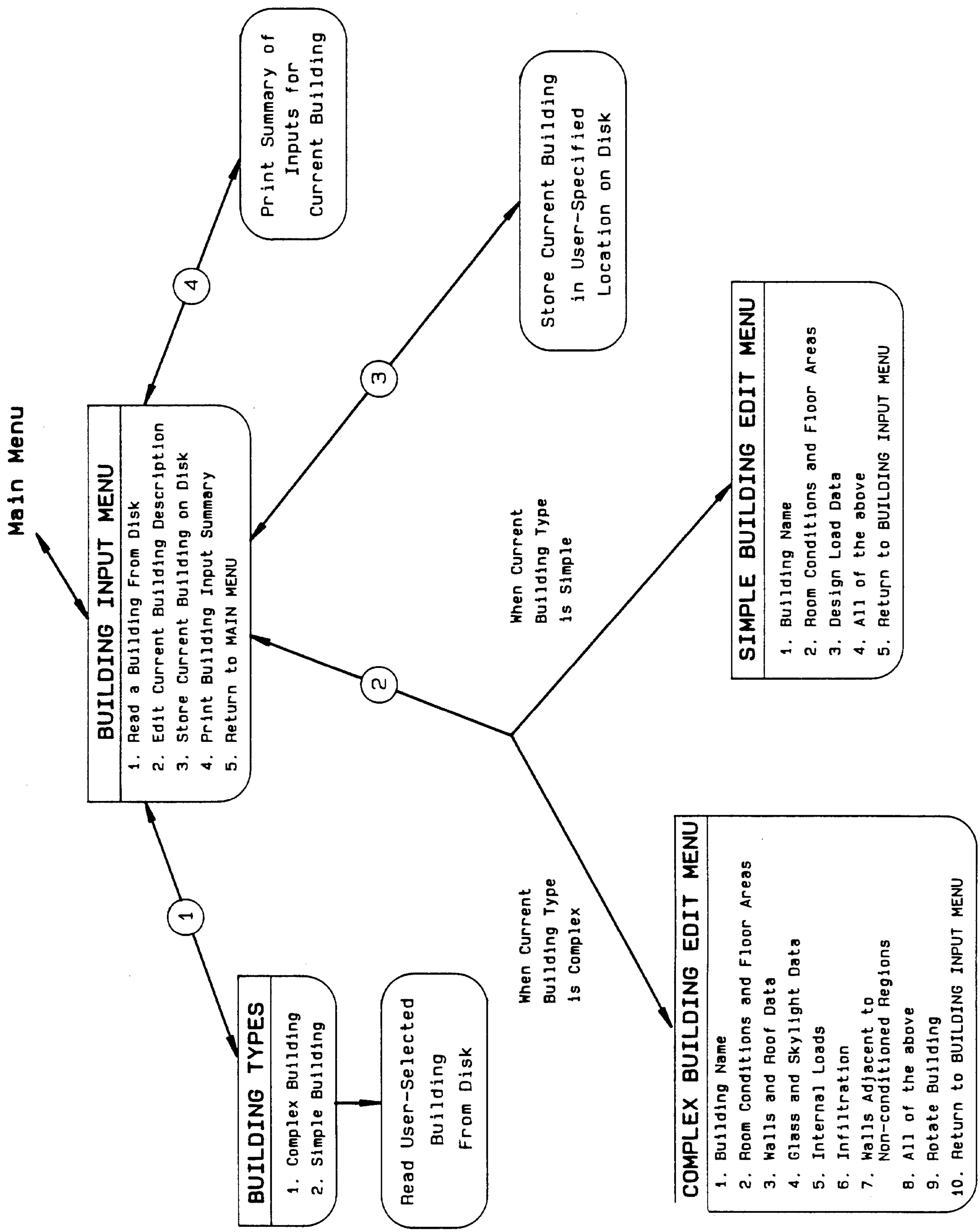
Main Menu
BUILDING INPUT MENU
1. Read a Building From Disk
2. Edit Current Building Description
3. Store Current Building on Disk
4. Print Building Input Summary
5. Return to MAIN MENU
Print Summary of Inputs for Current Building
Store Current Building in User-Specified Location on Disk
SIMPLE BUILDING EDIT MENU
1. Building Name
2. Room Conditions and Floor Areas
3. Design Load Data
4. All of the above
5. Return to BUILDING INPUT MENU
When Current Building Type is Simple
When Current Building Type is Complex
BUILDING TYPES
1. Complex Building
2. Simple Building
Read User-Selected Building From Disk
COMPLEX BUILDING EDIT MENU
1. Building Name
2. Room Conditions and Floor Areas
3. Walls and Roof Data
4. Glass and Skylight Data
5. Internal Loads
6. Infiltration
7. Walls Adjacent to Non-conditioned Regions
8. All of the above
9. Rotate Building
10. Return to BUILDING INPUT MENU
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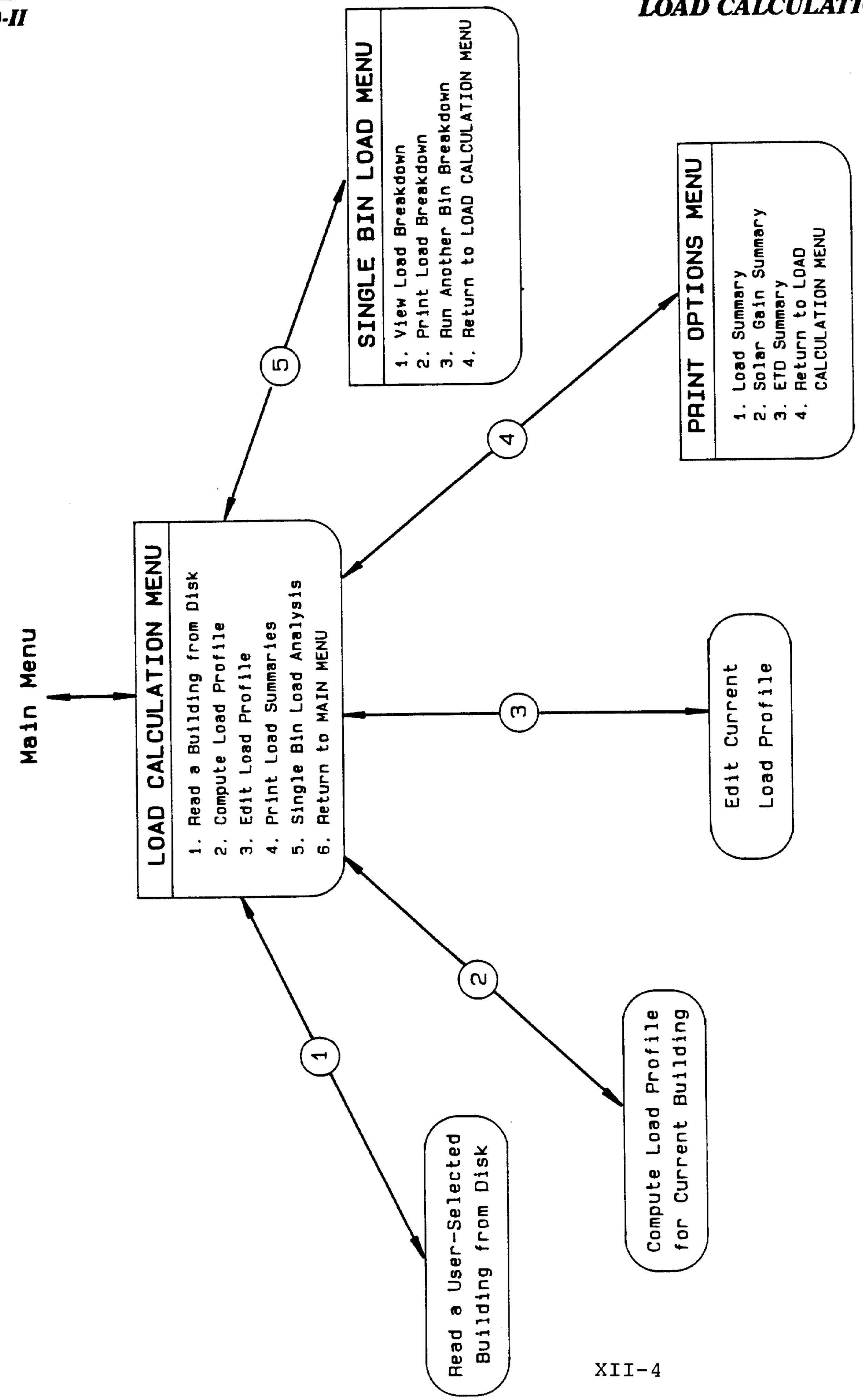
Main Menu
LOAD CALCULATION MENU
1. Read a Building from Disk
2. Compute Load Profile
3. Edit Load Profile
4. Print Load Summaries
5. Single Bin Load Analysis
6. Return to MAIN MENU
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Read a User-Selected Building from Disk
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Compute Load Profile for Current Building
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PRINT OPTIONS MENU
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2. Solar Gain Summary
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4. Return to LOAD CALCULATION MENU
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SINGLE BIN LOAD MENU
1. View Load Breakdown
2. Print Load Breakdown
3. Run Another Bin Breakdown
4. Return to LOAD CALCULATION MENU

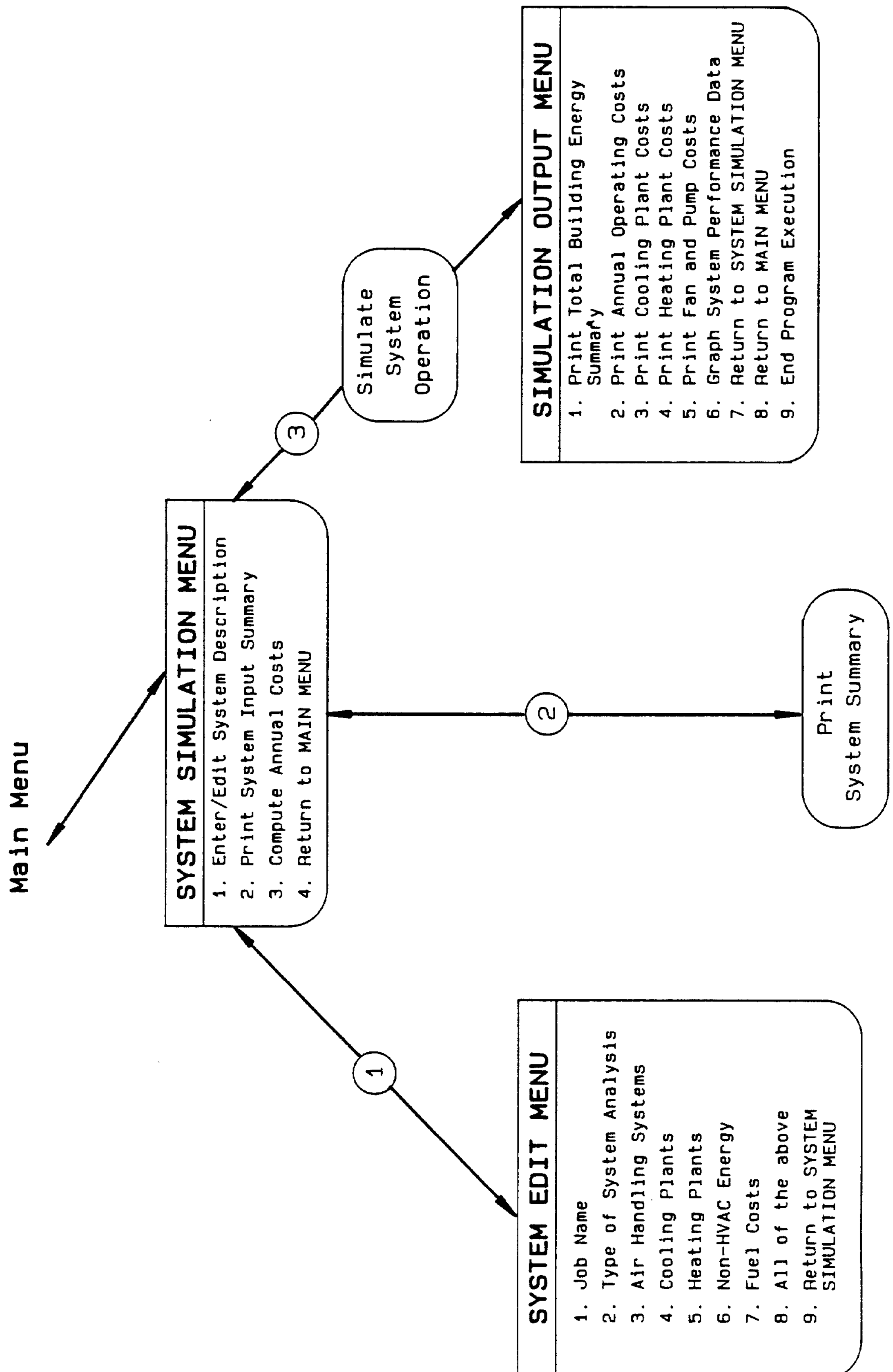
Main Menu
SYSTEM SIMULATION MENU
1. Enter/Edit System Description
2. Print System Input Summary
3. Compute Annual Costs
4. Return to MAIN MENU
1
SYSTEM EDIT MENU
1. Job Name
2. Type of System Analysis
3. Air Handling Systems
4. Cooling Plants
5. Heating Plants
6. Non-HVAC Energy
7. Fuel Costs
8. All of the above
9. Return to SYSTEM SIMULATION MENU
2
Print System Summary
3
Simulate System Operation
SIMULATION OUTPUT MENU
1. Print Total Building Energy Summary
2. Print Annual Operating Costs
3. Print Cooling Plant Costs
4. Print Heating Plant Costs
5. Print Fan and Pump Costs
6. Graph System Performance Data
7. Return to SYSTEM SIMULATION MENU
8. Return to MAIN MENU
9. End Program Execution

TABLE 1 SUMMARY OF CARRIER OPERATING COST ANALYSIS PROCEDURE

COMPUTE	GIVEN
A. WEATHER CALCULATIONS (see Chapter 1)	
1. Outdoor air temperature and humidity conditions 2. Average occupied and unoccupied hours by bin 3. Peak solar heat gains	1. Site latitude, elevation and design temperatures 2. Occupancy schedule 3. Peak solar heat gains
B. BUILDING THERMAL LOAD CALCULATIONS (see Chapters 2 and 3)	
1. Typical and design building sensible, latent and plenum thermal loads by bin 2. Building indoor dry bulb temperatures by bin	1. Outdoor temperature and humidity conditions 2. Peak solar heat gains 3. Building description
C. SECONDARY SYSTEM SIMULATION (see Chapter 4)	
1. Fan input KW by bin 2. System cooling coil loads by bin 3. System heating coil loads by bin 4. Ventilation reclaim device input kw by bin	1. Building thermal loads 2. Building indoor dry bulb temperatures 3. Outdoor temperature and humidity conditions 4. Secondary system description
D. PRIMARY SYSTEM SIMULATION (see Chapter 5)	
1. Cooling plant input power by bin 2. Primary heating plant input power by bin 3. Auxiliary heating plant input power by bin 4. Pump input power by bin	1. System cooling coil loads 2. System heating coil loads 3. Outdoor air temperature and humidity conditions 4. Primary system description
E. NON-HVAC SYSTEM ENERGY CALCULATIONS (see Chapter 6)	
1. Lighting, miscellaneous and other electrical input kw by bin 2. DHW plant input power by bin	1. Lighting, miscellaneous and other electrical power description 2. DHW system description
F. AVERAGE ANNUAL OPERATING COST CALCULATIONS (see Chapter 7)	
1. Fan costs 2. Pump costs 3. Cooling plant costs 4. Heating plant costs 5. Lighting, miscellaneous and other electrical costs 6. DHW costs 7. Grand total operating cost	1. Fan input kw by bin 2. Pump input kw by bin 3. Cooling plant input power by bin 4. Heating plant input power by bin 5. Lighting, misc. and other input kw by bin 6. DHW plant input power by bin 7. Occupied and unoccupied hours by bin 8. Fuel costs.

WEATHER CALCULATIONS

INTRODUCTION

The purposes of the weather calculations are to compute:

1. Bin dry bulb temperatures and corresponding average humidity conditions which are used in the estimation of bin hours, peak solar gains, building loads, and system performance levels.
2. Occupied and unoccupied bin hour profiles which are used with energy cost and system component input power data to determine component operating cost/year for each bin.
3. Peak solar gain values for use in the calculation of solar gain loads and equivalent temperature differences.

These three categories of computations are discussed in the following sections.

1.1 TEMPERATURE AND HUMIDITY CALCULATIONS

The purpose of this section is to discuss the calculation of outdoor dry bulb temperatures, wet bulb temperatures, specific humidity ratios and the design dewpoint temperature.

a. **Dry Bulb Temperatures**

Bin dry bulb temperatures are defined for the temperature range between the 1% summer and 99% winter design dry bulbs for a given site. Each dry bulb corresponds to one of a number of temperature bins.

A bin is a 5 F sub-division of the dry bulb range for which building load and system performance conditions are analyzed. The bin temperature 70 F, for instance, corresponds to the bin containing dry bulb temperatures T_a such that $70 =< T_a < 75$. A maximum of 24 average bins are permitted in the profile for any one site. This represents a range of 120 F between summer and winter design conditions and is sufficient for most sites. When it is not, bins adjacent to the winter design condition are omitted. Average bins contain temperatures at which average building load and system performance characteristics are studied. Two design bins are also included: one each at the 1% summer and 99% winter design conditions. These special "bins" each have a temperature range of exactly 1 F.

b. **Wet Bulb Temperatures**

A set of wet bulb temperatures corresponding to average humidity conditions at each bin dry bulb temperature is defined in order to facilitate the calculation of the bin specific humidity ratio, Wa.

The following assumptions are used in computing average bin wet bulbs:

i. Outdoor air is saturated at and below 50 F. Examination of actual bin weather data shows this to be a sound assumption.

ii. At the summer design condition, the average wet bulb temperature may be approximated as follows:

$$T_{dwb} = T_{swb} - .1\,(T_{swb} - 50)$$

where:

T_{dwb} = Average wet bulb temperature at 1% summer design condition, F

T_{swb} = 1% Summer design wet bulb temperature, F.

iii. The average wet bulb temperature varies linearly with bin dry bulb temperature between T_{dwb} and 50 F.

As a result of the approximating assumptions, wet bulb temperatures are computed as follows:

$T_{wb} = T_{swb}$ for summer design bin

$T_{wb} = (T_{dwb}-50)/(T_{sdb}-50)*(T_{db}-50)-50$ if $T_{db} > 50$ F

$T_{wb} = T_{db}$ if $T_{db} =< 50$ F

where:
T_{wb} = Wet bulb temperature corresponding to T_{db}, F.
T_{sdb} = 1% summer design dry bulb, F
T_{swb} = 1% summer design wet bulb, F
T_{db} = Bin dry bulb temperature, F

c. **Specific Humidity Ratio**

The bin specific humidity ratio is computed for use in a variety of subsequent calculations such as that of the latent infiltration and ventilation loads. The calculation described below utilizes psychrometric principles:

$$W_a = [(1093-.556T_{wb})W_s-.24(T_{db}-T_{wb})]/(1093+.444T_{db}-T_{wb})$$

where:
Wa = Specific humidity ratio, lb/lb
T_{db} = Bin dry bulb temperature, F.
T_{wb} = Bin wet bulb temperature, F.
W_s = Specific humidity ratio at T_{wb} for saturation condition, lb/lb.
= $.62198P_s/(14.696 - P_s)$
P_s = Atmospheric pressure at T_{wb} for saturation condition, psia.
= $4.993x10^8 exp(-11040/(T_{wb}+460))$ if $T_{wb}<32$
= $2.4074x10^7 exp(-9548/(T_{wb}+460))$ if $T_{wb}>=32$

d. **Design Dewpoint Temperature**

The design dewpoint temperature is used in correcting Carrier peak solar heat gain values and in the calculation of the bin hour profiles. It is calculated using psychrometric relations as follows:

$$T_{dp} = -11040/\ln(P_s/4.993x10^8)-460 \text{ if } P_s<.089788$$

$$T_{dp} = -9548/\ln(P_s/2.4074x10^7)-460 \text{ if } P_s>=.089788$$

where:
T_{dp} = Dewpoint temperature computed at 1% summer design condition, F.
P_s = Atmospheric pressure at T_{swb} for saturation condition, psia. See equations in 1.1c.
T_{swb} = 1% summer design wet bulb temperature, F.
ln = Natural logarithm function.

1.2 BIN HOUR PROFILE CALCULATIONS

The purpose of these calculations is to determine the number of hours per year during which the outdoor air temperature is within each bin. Separate profiles are computed for occupied and unoccupied periods. Bin hour profiles are intended to represent the distribution of hourly temperatures during the year for average weather conditions.

In the first step of the calculation procedure, bin hour profiles for the three daily observation periods are determined. Next, occupied and unoccupied period profiles are computed using this data and the building occupancy schedule. Each calculation step is discussed in a separate sub-section below.

a. **Observation Period Bin Hour Profiles**

Bin weather data are listed in the United States Air Force Weather Manual [1] both as total hours/year/bin and as hours/year/bin for the following three daily observation periods:

Observation Period (As listed in ref [1])	Period Begins	Period Ends
01—08	1:00 am	8:59 am
09—16	9:00 am	4:59 pm
17—24	5:00 pm	12:59 am

Occupied and unoccupied period bin hour profiles are computed by assigning hours from each of the observation periods to the occupied and unoccupied profiles according to the building occupancy schedule. This procedure is described in (b) below. First, however, bin hour profiles for the three observation periods must be determined.

Bin hour profiles for the three observation profiles are computed using a modified version of the Erbs Method [5]. The Erbs Method is a statistical calculation procedure which provides accurate bin hour profiles for sites in North America. The method was developed from a study of hourly weather data for a number of sites in North America. Certain simplifying assumptions were introduced into the method so that it could be used in the OPCOST program. A summary of the modified Erbs Method appears below.

The modified Erbs Method utilizes the following three input values:

T_{sdb} = 1% summer design dry bulb temperature, F.
T_{wdb} = 99% winter design dry bulb temperature, F.
DR = Daily temperature range, F.

Bin hour profiles are computed using the following five stage procedure:

> 1. Estimate mean monthly temperatures for each month with a correlation involving T_{sdb}, T_{wdb} and DR.

> 2. Compute the mean monthly temperature for each observation period using the equation:

$$T_{mp} = T_{mm} + C\,[.4632(\sin(t_2^* - 3.805) - \sin(t_1^* - 3.805)) + .0492(\sin(2t_2^* - .36) - \sin(2t_1^* - .36))]$$

where:
T_{mp} = Mean monthly temperature for observation period, C.
T_{mm} = Mean monthly temperature, C.
C = $12A/(p(t_2 - t_1))$
A = Amplitude of daily ambient temperature variation, C. Assumed to be equal to the daily range, DR.
p = Pi, 3.141593.
t_2^* = $2p(t_2 - 1)/24$
t_1^* = $2p(t_1 - 1)/24$
t_1 = Starting time for observation period, 1 for 1am, 9 for 9am, 17 for 5 pm.
t_2 = Ending time for observation period, 9 for 9am, 17 for 5pm, 25 for 1am.

The preceeding equation is derived by integrating the Fourier series approximation of the hourly temperature variation for a typical day in the month,

$$T_a = T_{mm} + A\,[0.4632\sin(t_2^* - 3.805) + 0.0984\sin((2t_1^* - .36)]$$

where:
T_a = Hourly ambient temperature, C.

between the starting and ending times for the observation period, and dividing by the length of the observation period.

> 3. Determine the mean annual temperature for each observation period from the mean monthly temperatures. Compute the standard deviation for the mean monthly temperatures from the mean annual temperature for each observation period.

> 4. Compute the standard deviation for the distribution of ambient temperatures about the mean monthly temperature for each month and observation period. Standard deviations are computed in this case using the following correlation:

$$D_{pm} = 1.45 - .0209T_{pm} + .0664D_{pa}$$

where:
D_{pm} = Standard deviation for ambient temperatures about the mean monthly temperature for an observation period.
T_{pm} = Mean monthly temperature for the observation period, C.
D_{pa} = Standard deviation for mean monthly temperatures about the mean annual temperature for each observation period.

> 5. Compute the monthly bin hour profiles for each observation period. Temperatures are assumed to have a normal distribution about the mean monthly temperature. Bin hours are computed as follows:

$H_b = 24N(Q(h_2) - Q(h_1))$

where:
H_b = Bin hours in one bin, hrs/yr/bin.
$Q(h)$ = Ambient temperature cumulative distribution function.
= $1/(1 + \exp(-3.396h))$
h = Non-dimensional temperature scale parameter.
= $(T-T_{pm})/(D_{pm}\sqrt{N})$
T = Ambient temperature, C.
h_2 = h value computed for the temperature at the upper limit of a temperature bin.
h_1 = h value computed for the temperature at the lower limit of a temperature bin.
N = Number of days in month.

Bin hour profiles are computed for each observation period, month by month in this way. Monthly profiles are then summed to obtain the annual profile. Annual observation profiles are summed to obtain the total bin hour profile.

More information concerning this method may be found in reference [5] or is available upon request.

b. **Occupied / Unoccupied Bin Hour Profiles**

Having determined bin hour profiles for each of the observation periods, the next task is to assign hours to the occupied and unoccupied period profiles according to the building occupancy schedule. In order to carry out this task, the average fractions of each observation period which occur during the occupied and unoccupied period are calculated. These fractions depend directly upon the user-specified building occupancy schedule. Occupied bin hours are then computed as follows:

$H_O = R_1 H_1 + R_2 H_2 + R_3 H_3$

where:
H_O = Occupied period bin hours for a sample bin, hr/yr.
R_1 = Average fraction of hours in first observation period which occur during occupied period.
H_1 = Bin hours in sample bin occuring during first observation period.
R_2 = Average fraction of hours in second observation period which occur during occupied period.
H_2 = Bin hours in sample bin occuring during second observation period.
R_3 = Average fraction of hours in third observation period which occur during occupied period.
H_3 = Bin hours in sample bin occuring during third observation period.

A similar equation is utilized to compute the unoccupied period bin hours.

The following example illustrates this final step in the procedure:

Day	Schedule: Occupied Begins:	Period Ends:	Occupied Period 1am - 9am	9am - 5pm	5pm - 1am	Unoccupied Period 1am - 9am	9am - 5pm	5pm - 1am
Sun	— Off —		0	0	0	8	8	8
Mon	6am	12am	3	8	7	5	0	1
Tue	6am	12am	3	8	7	5	0	1
Wed	6am	12am	3	8	7	5	0	1
Thu	6am	12am	3	8	7	5	0	1
Fri	6am	12am	3	8	7	5	0	1
Sat	— Off —		0	0	0	8	8	8
	Total Hrs/Wk ->		15	40	35	41	16	21

The average fractions of hours in each observation period which occur during the occupied and unoccupied period are then computed:

Period	1am to 9am	9am to 5pm	5pm to 1am
Occupied	15/56 = 0.27	40/56 = 0.71	35/56 = 0.63
Unoccupied	41/56 = 0.73	16/56 = 0.29	21/56 = 0.37

For a sample bin, suppose that:

Total hours = 600 hr/yr
1am - 9am hours = 100 hr/yr
9am - 5pm hours = 300 hr/yr
5pm - 1am hours = 200 hr/yr

Then, for this example:

Occupied hrs = (100)(.27)+(300)(.71)+(200)(.63) = 366 hr/yr
Unoccupied hrs = (100)(.73)+(300)(.29)+(200)(.37) = 234 hr/yr

A final adjustment is made to the occupied and unoccupied profiles when a summer shutdown period is specified. Certain types of buildings such as schools and factories are closed and not air conditioned for extended periods each year. By specifying the number of days the building is shut down and that the building is not cooled during the unoccupied period, the effect of summer shutdown on annual operating cost can be estimated. The estimation procedure is discussed below.

The first step in the procedure is to compute the occupied hours affected by the shutdown period:

$H_{sd} = H_{occ} * D_{sd}$

where:
H_{sd} = Total number of occupied hours affected by the shutdown period, hr/yr.
H_{occ} = Average occupied hours per day, computed using the building occupancy schedule.
D_{sd} = Number of shutdown days, days/yr.

Starting with the highest temperature bin and proceeding downward, hours are transferred from the occupied to the unoccupied profiles until a total of Hsd hours have been transferred. The rationale for this transfer is that bin hours in the highest temperature bins occur predominantly during the months of June, July and August, the time frame during which the shutdown period is assumed to occur. Building shutdown during months other than June, July and August cannot be accurately modeled with this procedure. As a result, the maximum shutdown period allowed is the length of time between June 1 and August 31: 92 days.

1.3 PEAK SOLAR GAIN CALCULATIONS

Peak solar heat gain (PSHG) values must be defined so that solar gain loads and equivalent temperature differences may be computed as described in sections 2.3a and 2.3b. The program user may elect to use PSHG data directly from Table 6 of the Carrier Design Manual [2]. These data are pre-stored in the program. If the user desires to utilize data from an alternate source, PSHG data may be manually entered. Only PSHG values for the months of January and July need be entered for the reasons described in section 2.2. The alternate methods of defining PSHG data are described below.

a. **Use of Pre-Stored Carrier PSHG Values**

For a given site latitude, PSHG values are computed using a straight-line interpolation method with data from Table 6 of the Carrier Design Manual [2]. Table 6 data are summarized in Table 1.1.

TABLE 1.1 CARRIER PEAK SOLAR HEAT GAINS

Lat. Deg N	Month	Exposure N	NE/NW	E/W	SE/SW	S	Horiz
0	Jan	10	52	152	153	67	233
	Jul	48	153	152	52	14	233
10	Jan	9	37	143	161	106	210
	Jul	30	148	158	66	14	247
20	Jan	8	26	128	164	141	180
	Jul	19	138	163	85	14	251
30	Jan	7	16	116	162	159	145
	Jul	16	131	164	100	30	246
40	Jan	5	12	100	156	166	103
	Jul	15	127	164	125	69	233
50	Jan	4	9	64	127	153	53
	Jul	14	117	163	143	106	211
60	Jan	2	4	29	81	104	15
	Jul	17	108	162	156	142	183

The PSHG values computed using Table 1.1 are automatically corrected for altitude and design dewpoint as follows:

Altitude correction : +0.7% for every 1000 ft above sea level
-0.7% for every 1000 ft below sea level

Dewpoint correction : -7% for every 10 degrees above 67 F
: +7% for every 10 degrees below 67 F

In addition, the sash correction factor of 1.17 is used since the majority of commercial buildings today have sashless or steel sash windows.

b. **Manually Entered PSHG Values**

When the program user wishes to utilize PSHG values from an alternate source such as the ASHRAE Fundamentals Manual or wishes to use Carrier PSHG values without a sash correction, PSHG values must be manually entered. All entered values must be fully corrected for site altitude and dewpoint conditions.

COMPLEX BUILDING THERMAL LOAD CALCULATIONS

2.0 INTRODUCTION

Design loads and average load profiles are computed for a complex building using Carrier E20 load calculation methods and certain load averaging assumptions. These loads are computed using detailed information concerning the building construction and the nature of load generating components. Building thermal loads are subsequently utilized in the simulation of the behavior of the air handling systems.

A complex building may be comprised of one or two zones. A zone is defined as a region of the building with a single thermostatic control. Two zone types exist: the perimeter zone which may have exterior wall, roof and glass exposures, and the interior zone which may only have an exterior roof exposure. Zones, in turn, are made up of a number of elements such as walls, roofs, glass, people, and lights, which influence the thermal load a zone experiences.

Sensible, latent and plenum thermal loads are computed for each zone. It should be noted that a zone thermal load does not include ventilation air or fan heat gain components. These load components are system-related and can be properly computed only during simulation of the air handling system operation. It should also be noted that zone latent thermal loads, comprised of latent heat gain from people, infiltration air and miscellaneous internal sources, may exist even at low ambient temperature conditions. When cold, dry ventilation air at these conditions is introduced into the zone during the air handling system operation, these latent thermal loads are in most cases eliminated. Because of these considerations, the user should not attempt to manually estimate system coil loads by adding the sensible and latent thermal load components.

Zone thermal loads are computed for cooling and heating design conditions and for average occupied and unoccupied conditions bin by bin. Table 2.1 shows a breakdown of the three types of thermal loads for each zone. Development of the load calculation is discussed in the following sections.

TABLE 2.1 ZONE THERMAL LOAD COMPONENTS

		Perimeter Zone			Interior Zone	
Load Component		S	L	P	S	L	P
Solar Gain:	Windows	x					
	Skylights	x			x		
Transmission:	Walls	x					
	Roof	x		x	x		x
	Windows	x					
	Skylights	x			x		
Transmission thru walls Adj. Non-Cond Region		x			x		
Lighting Heat Gain		x		x	x		x
Misc Electric Heat Gain		x			x		
People Heat Gain		x	x		x	x	
Misc Internal Heat Gain		x	x		x	x	
Infiltration		x	x				

Notes: a. S = Sensible thermal load
L = Latent thermal load
P = Plenum thermal load
b. A fixed portion of the roof transmission load and a fixed portion of the lighting heat gain comprise the plenum load when a plenum exists and when lights are recessed. See sections 2.3b and 2.3e for more information.

2.1 DESIGN LOAD CALCULATIONS

The purpose of design load calculations is to determine extreme cooling and heating thermal load conditions in each zone. Design loads are used for equipment sizing and the computation of design air flow rates and supply air temperatures.

The Carrier E20 load calculation method, described in reference [2], is used to compute design loads. Major assumptions used in the calculations are summarized in Table 2.2 below. As shown, the design cooling load is assumed to occur at 3pm in July, when the summer design temperature typically occurs. Maximum lighting and occupancy levels are also assumed for the cooling design calculation. The design heating load includes only transmission and infiltration components. Calculation of individual load components is described in section 2.3 .

TABLE 2.2 DESIGN LOAD CALCULATION ASSUMPTIONS

	Assumptions For:	
Load Component	Cooling Design	Heating Design
Solar Gains	Peak solar gains, 3pm July, no cloud cover	Not considered.
Wall, Roof Transmission	ETD values based on peak solar, July, 3pm	Use actual temperature differences instead of ETDs
Glass Transmission	Normal procedure	Normal procedure
Transmission Thru Walls Adj to Non-Cond Region	Normal procedure	Normal procedure
Lighting Heat Gain	Maximum wattage used	Not considered.
Miscellaneous Electric Heat Gain	Occupied load value	Not considered.
People Heat Gain	Maximum occupancy	Not considered.
Miscellaneous Internal Heat Gain	Occupied load values	Not considered.
Infiltration	Normal procedure	Normal procedure

2.2 TYPICAL LOAD CALCULATION METHOD

In addition to computing design zone thermal loads, typical loads corresponding to each bin must be computed. These load data are utilized in the simulation of average air handling system performance bin by bin. Loads are computed using principles from both the Carrier E20 Load Calculation Method [2] and the ASHRAE TC 4.7 Simplified Energy Analysis Method [3]. The development of the OPCOST load calculation method is discussed below. A discussion of individual load component calculations is found in section 2.3.

a. **Basic Principles**

As stated, the goal of this load calculation is to compute a thermal load which corresponds to average load condition existing when the outdoor air temperature is in a certain temperature bin. These loads will henceforth be referred to as "bin loads". Program users often have the misconception that a bin load is computed for a specific month and time of day. Bin loads are actually the average of many loads occuring for a variety of months and times of day when the ambient dry bulb is in a certain temperature bin.

The bin thermal load can be mathematically stated as follows:

$$\text{Load} = \frac{\text{(Sum of loads occuring when OA temperature is in bin)}}{\text{(Number of hours OA temperature is in bin)}}$$

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Further, the bin load is the sum of the average component loads. For example, the average sensible thermal load can be written as:

Bin Sensible Load = Bin solar gain
+ Bin wall transmission
+ Bin lighting heat gain
+ . . .etc. . . . . .

Finally, each component load can be computed as:

$$\text{Bin Component Load} = \frac{\text{(Sum of bin loads for one component)}}{\text{(Number of hours in bin)}}$$

To precisely compute bin loads, an hourly simulation would be run for the 8760 hours in the year and a listing of hourly temperatures and component loads would be printed. Loads would then be segregated by bin and period, and averaged. Because this program is not an hourly simulation, certain approximating assumptions must be made in estimating bin component loads. In defining these assumptions the precise analysis methodology above is kept in mind to make realistic assumptions.

b. **Load Dependency**

When computing average bin loads, we seek to define load components as a function of bin temperature. The assumptions necessary to accomplish this depend upon what other variables each load component is a function of. For example, wall transmission is a function of wall area, U-value and indoor-outdoor equivalent temperature difference (ETD). For a particular zone, wall area and U-value are constants, while the ETD varies with solar flux, indoor and outdoor temperature and time of day. Broadly, we can say that wall transmission is solar and temperature dependent. By similar analysis, each load component can be categorized by dependency as follows:

1. Period dependent loads
   People sensible and latent heat gain
   Lighting heat gain
   Miscellaneous electric heat gain
   Miscellaneous internal sensible and latent heat gain.
2. Temperature dependent loads
   Glass transmission
   Sensible infiltration
   Transmission for wall adjacent to non-conditioned region
3. Humidity dependent load
   Latent infiltration
4. Solar dependent loads
   Solar gains
5. Solar and temperature dependent loads
   Exterior wall transmission
   Roof transmission

Assumptions necessary for the estimation of each type of load component is discussed in a following sub-section.

c. **Period Dependent Loads**

Period dependent loads are independent of outdoor air temperature and are assumed constant during the occupied and the unoccupied period. By defining the average number of people occupying a zone, the average lighting wattage, the average miscellaneous electrical wattage and average miscellaneous internal heat gains for each period, loads in this category can be accurately estimated.

d. **Temperature Dependent Loads**

Temperature dependent loads are solely dependent upon the indoor-outdoor air temperature difference. The indoor air temperature is determined by a separate calculation described in section 2.4 . Thus, for our purposes, these loads are solely dependent upon the outdoor air dry bulb temperature and no averaging assumptions are necessary.

e. **Humidity Dependent Load**

This load is dependent upon the difference between indoor and outdoor specific humidity ratios. Indoor specific humidity is a function of indoor design relative humidity and dry bulb values, and with the stated assumptions from 2.3i is independently computed. Outdoor air specific humidity is a function of the outdoor dry bulb temperature, as described in 1.1c. Thus, this average load component is an indirect function of outdoor air temperature and no additional assumptions are neccessary.

f. **Solar Dependent / Solar and Temperature Dependent Loads**

These two load categories are discussed together because of the common solar dependency. Estimation of loads which are both solar and temperature dependent involve the calculation of an equivalent temperature difference (ETD). Each ETD is comprised of a solar- and a temperature-related component. The temperature-related component is a direct function of outdoor air temperature as discussed in (d) above. Estimation of the solar component is discussed below.

Solar dependent loads are the most challenging to estimate, because the relationship between time-delayed solar gains and outdoor air temperature is difficult to ascertain. As an example, consider the estimation of the average solar gain through a south facing window for the 70 F bin. The precise calculation of the average solar gain would utilize the equation:

$$\text{Avg solar gain} = \frac{\text{(Sum of south glass solar gains, 70 F bin)}}{\text{(Number of hours, 70 F bin)}}$$

For a sample site such as Chicago, 70 F bin temperatures could occur in any month from April to October and for any hour of the day. The distribution of hours in each bin by month and time of day depends upon a number of site- and climate-related factors such as latitude, elevation, proximity to bodies of water and to mountains, and prevailing wind patterns. As a result, the distribution is extremely difficult to compute without actual hourly weather data. Thus, unless some way can be found to correlate bin temperatures with a month and time of day, average solar gains cannot be satisfactorily estimated.

Fortunately, there exist two temperatures at which reasonable correlations with month and time can be made: the 1% summer and 99% winter design temperatures. Analysis of actual bin weather data has shown that in an average year, there are 30 hours at or above the 1% summer design temperature and 22 hours at or below the 99% winter design temperature [4]. For North American sites, it is reasonable to assume that hours at and above the summer design condition occur in June, July and August, with the preponderence of hours at or near July 3pm.

Likewise, hours at and below the winter design condition can be assumed to occur during December, January and February, with the preponderance of hours at or near January 3am. The choice of 3am is not critical since there is no instantaneous solar flux, and storage load factors and ETDs vary slightly from hour to hour here.

Having identified months and hours corresponding to two temperatures, solar gains and the solar portions of ETDs can be computed for each exposure at each condition. Carrier E20 methodology is used here, modified only by the use of the percentage of available sunshine factors to attenuate peak solar gains. Solar gains and the solar portion of ETDs are approximated as linear functions of outdoor air dry bulb temperature between the design points. The use of linear functions is judged to provide the best unbiased approximation of solar dependent load components possible under the terms of this simplified energy analysis.

It is very important that the reader understand exactly what the solar gain values in each bin represent. Bin solar gains are the average solar gains occuring when the outdoor air temperature is within the limits of each bin. Thus, for bins near the summer design temperature, the bin solar gains represent average values for daylight hours since this is when most high temperature bin hours occur. For bins near the winter design temperature, the bin solar gains represent average values for night hours since this is when most low temperature bin hours occur. Finally, for intermediate bins, the solar gains represent daily averages since bin hours at these temperatures tend to be more evenly split between daylight and night hours.

Because bin solar gains are average values, the same solar gains are used for both occupied and unoccupied periods. At first glance, this may seem to be an improper use of the solar gains. However, it is the best application of the solar gain data for reasons discussed in the following paragraph.

First, it is important to consider when bin hours occur. In high temperature bins, hours occur mainly during daylight hours. If the occupied period occurs during daylight hours, the use of the average solar gain in high temperature bins for unoccupied period load calculations may be a poor approximation. However, because there would be very few unoccupied hours in this bin, and many more occupied hours, the impact of this approximation on annual system energy totals and costs becomes negligible. The same is true of low temperature bins where the solar gain tends to be an average for night hours. Use of these solar gains for occupied period load calculations may also be a poor approximation, but has little effect on annual system costs because so few occupied period bin hours would exist. Finally, for intermediate bins the solar gains represent daily averages and are good approximations for both occupied and unoccupied period load calculations. Using the same type of logic, and a slightly more detailed analysis, we would find the same to be true regardless of the building schedule.

Finally, the reader should consider the overall value of the solar gain approximation. The approximation may only be flawed for the unoccupied period in high temperature bins, and occupied periods in low temperature bins. The numbers bin hours for the occupied period in high temperature bins and for the unoccupied period in low temperature bins are small enough that the impact of the poor approximation on annual system energy use and costs is made trivial. Thus, the application of solar gains using this method provides a good engineering estimate of building and system behavior.

2.3 LOAD COMPONENT CALCULATIONS

The purpose of this section is to provide documentation of the calculation of the various thermal load components. Where applicable, differences between design and average bin load calculations will be noted.

a. **Solar Gains**

Basic solar gain equation:

$$Q_{sg} = PSHG * F_s * SLF * SF * A_g$$

where:

Q_{sg} = Solar gain for one exposure, Btu/hr.
PSHG = Corrected peak solar heat gain for desired month and exposure, Btu/hr. Calculation of PSHG is discussed in 1.3.
F_s = Fraction of available sunshine for desired month.
SLF = Storage load factor for one exposure. Value is dependent upon building weight, operating schedule, use of internal shading and time of day as shown in Tables 2.3 and 2.4.
SF = Internal shade factor which defines the fraction of solar energy transmitted through the glass panes and internal shade covering. Values used are:
  0.56 for light colored shades
  0.65 for medium colored shades
  0.75 for dark colored shades
  1.00 for no internal shades
  or other value defined by the program user.
A_g = Glass area for the exposure, sqft.

For cooling design solar gains, use:

i. PSHG values for July
ii. SLF values for 3pm
iii. F_s =1.00 (i.e. assume clear sky)

For the heating design condition, solar gains are not considered.

For bin solar gains, the following equation is used:

$$Q_{sga} = (Q_{sgs}-Q_{sgw})*(T_a-T_{sdb})/(T_{sdb}-T_{wdb}) + Q_{sgs}$$

where:
Q_{sga} = Solar gain for one glass exposure corresponding to typical conditions at T_a, BTU/hr.
Q_{sgs} = Solar gain for one glass exposure corresponding to typical conditions at T_{sdb}, Btu/hr. The following values are used in the calculation of Q_{sgs}:
i. PSHG for July
ii. SLF for 3pm
iii. F_s for summer condition

Q_{sgw} = Solar gain for one glass exposure corresponding to typical conditions occuring at T_{wdb}, Btu/hr. The following values are used in the calculation of Q_{sgw}:
i. PSHG for January
ii. SLF for 3am
iii. F_s for winter conditions
T_a = Outdoor air bin dry bulb temperature, F.
T_{sdb} = 1% summer design dry bulb temperature, F.
T_{wdb} = 99% winter design dry bulb temperature, F.

Finally, the operating schedule used in choosing storage load factor values is determined as follows. Carrier SLF values are defined for the following 3 operating shedules:

24 hour : First hour = 6am, last hour = 5am
16 hour : First hour = 6am, last hour = 9pm
12 hour : First hour = 6am, last hour = 5pm

Because the program user is free to define any building operating schedule he desires, a procedure must be used to decide which storage load factors to utilize for any given user-entered schedule:

12 hour SLFs used if a. Cooling not provided during unoccupied period.
b. 5am $=< h_1 =<$ 8am
c. 5pm $=< h_2 =<$ 9pm

16 hour SLFs used if a. Cooling not provided during unoccupied period.
b. 5am $=< h_1 =<$ 8am
c. 9pm $< h_2 =<$ 12am

24 hour SLFs are used in all other cases.

h_1 = Average weekly starting time for building occupancy.
h_2 = Average weekly ending time for building occupancy.

b. **Wall and Roof Transmission Loads**

Basic wall and roof transmission load equation:

$$Q_w = U_w * A_w * ETD$$

where:
Q_w = Transmission load for wall or roof exposure, BTU/hr.
U_w = Wall or roof exposure U-factor, BTU/(hr-sqft-F).
A_w = Wall or roof exposure area, sqft.
ETD = Equivalent temperature difference for wall or roof exposure, F.

Basic equivalent temperature difference equation:

$$ETD = K_w(R_sF_s/R_m)(t_{em} - t_{es}) + t_{es} + X_d$$

where:
K_w = Wall or roof color correction factor:
1.00 for dark wall or roof surface
0.78 for medium wall or roof surface
0.55 for light wall or roof surface
R_s = Peak solar heat gain for wall or roof exposure, BTU/(hr-sqft). Computed as described in 1.3.
R_m = Peak solar heat gain for wall or roof exposure at 40 N latitude, month of July, at sea level with design dewpoint of 67 F and clear sky conditions, BTU/(hr-sqft).
F_s = Fraction of available sunshine.
t_{em} = Equivalent temperature difference for sunlit wall or roof exposure and a specific time of day, F. This value is taken from either Table 19 or 20 of the Carrier Design Manual [2] and is uncorrected for design conditions. Values are summarized in Table 2.5.

t_{es} = Equivalent temperature difference for shaded wall or roof exposure and a specific time of day, F. This value is taken from either Table 19 or 20 of the Carrier Design Manual [2] and is uncorrected for design conditions. Values are summarized in Table 2.5.

X_d = Design temperature correction factor, F
= $T_{ad} - T_i - .5 * DR - 5$
(This equation is derived from data in Table 20A of reference [2]

T_{ad} = Design dry bulb temperature for desired month at 3pm, F.
T_i = Indoor design dry bulb temperature, F.
DR = Daily temperature range, F.

Cooling design ETDs are computed using the following values:

i. R_s values for July
ii. t_{es} and t_{em} values for 3pm
iii. F_s = 1.00
iv. T_i = Indoor cooling dry bulb temperature.

At the heating design condition actual indoor-outdoor temperature differences are utilized instead of ETDs. The indoor design temperature used is the occupied heating temperature.

TABLE 2.3 Carrier Storage Load Factors, With Internal Shade

	3 PM Values									3 AM		
	24 hr			16 hr			12 hr			24 hr		
Ex	L	M	H	L	M	H	L	M	H	L	M	H
N	.98	.87	.84	.98	.87	.84	1.0	.98	.96	.01	.10	.12
NE	.13	.16	.17	.13	.16	.17	.13	.19	.21	0	.03	.04
E	.14	.18	.18	.14	.18	.18	.14	.21	.23	0	.03	.05
SE	.20	.24	.24	.20	.24	.24	.20	.28	.30	0	.04	.06
S	.50	.59	.59	.50	.59	.59	.50	.65	.67	0	.06	.08
SW	.86	.70	.66	.86	.70	.66	.87	.77	.74	.01	.07	.08
W	.65	.54	.52	.65	.54	.52	.67	.61	.61	.02	.07	.09
NW	.39	.34	.33	.39	.34	.33	.41	.40	.41	.02	.06	.08
H	.50	.59	.59	.50	.59	.59	.50	.65	.67	0	.06	.08

*From Tables 7, 9 and 11 of Carrier Design Manual [2]

TABLE 2.4 Carrier Storage Load Factors, Without Internal Shade

	3 PM Values									3 AM		
	24 hr			16 hr			12 hr			24 hr		
Ex	L	M	H	L	M	H	L	M	H	L	M	H
N	.97	.76	.69	.95	.80	.75	1.0	.95	.92	.02	.18	.23
NE	.16	.21	.22	.16	.21	.22	.16	.26	.29	0	.05	.07
E	.18	.25	.26	.18	.25	.26	.18	.31	.34	0	.06	.08
SE	.32	.36	.35	.32	.36	.35	.32	.44	.45	0	.08	.10
S	.75	.57	.51	.75	.57	.51	.76	.68	.65	.01	.11	.14
SW	.78	.50	.43	.78	.50	.47	.80	.62	.57	.02	.12	.14
W	.49	.30	.27	.49	.30	.27	.52	.43	.42	.03	.13	.15
NW	.27	.19	.17	.27	.19	.17	.30	.30	.30	.03	.11	.13
H	.75	.57	.51	.75	.57	.51	.76	.68	.65	.01	.11	.14

*From Tables 8, 10 and 11 of Carrier Design Manual [2].

TABLE 2.5 Carrier Equivalent Temperature Differences

	3 PM			3 AM		
Exposure	L	M	H	L	M	H
North (or Shade)	12	8	1	-1	0	2
Northeast (Sunlit)	13	11	16	-3	1	8
East (Sunlit)	13	13	19	-2	1	12
Southeast (Sunlit)	19	21	15	-1	3	10
South (Sunlit)	28	25	7	0	1	9
Southwest (Sunlit)	34	24	7	0	4	8
West (Sunlit)	32	19	10	0	3	16
Northwest (Sunlit)	19	10	6	-1	1	11
Horizontal (Sunlit)	36	33	31	6	11	16
Horizontal (Shaded)	13	10	6	-4	-3	0

*From Tables 19 and 20 of Carrier Design Manual [2].

Key for Tables 2.3-2.5:
Building Wts (for SLFs): L = 30; M = 100; H = 150 lb/sqft floor area
Wall Weights (for ETDs): L = 20; M = 60; H = 140 lb/sqft wall area
Roof Weights (for ETDs): L = 20; M = 40; H = 60 lb/sqft roof area

For bin wall or roof transmission loads, the following equation is used:

$Q_{wa} = U_w * A_w * ETD_a$

where:
Q_{wa} = Wall or roof exposure transmission load corresponding to typical conditions at T_a, BTU/hr.
ETD_a = Wall or roof exposure ETD corresponding to typical conditions at T_a, F.
T_a = Outdoor air bin dry bulb, F

The computation of ETD_a is developed below:

ETD_a =
$= K_w(R_sF_s/R_m)(t_{em}-t_{es}) + t_{es} + X_d$
$= K_w(R_sF_s/R_m)(t_{em}-t_{es}) + t_{es} + T_{ad} - T_i - .5*DR - 5$
$= [K_w(R_sF_s/R_m)(t_{em}-t_{es}) + t_{es} + T_{ad} - .5*DR - 5] - T_i$
$= T_{ae} - T_i$

where:
T_i = Indoor dry bulb temperature, determined as described in section 2.4, F.
T_{ae} = Outdoor air equivalent temperature, F. This temperature is computed considering both the outdoor air dry bulb and the effect of the transmission of solar energy through the wall or roof exposure.

Outdoor air equivalent temperature equation:

$T_{ae} = (T_{aes} - T_{aew})(T_a - T_{sdb})/(T_{sdb} - T_{wdb}) + T_{aes}$

where:
T_{aes} = Outdoor air equivalent temperature computed for typical conditions at T_{sdb}, F. Values used to compute T_{aes}:
i. R_s for July
ii. t_{es} and t_{em} for 3pm (see Table 2.5)
iii. F_s for summer conditions
iv. $T_{ad} = T_{sdb}$

T_{aew} = Outdoor air equivalent temperature computed for typical conditions at T_{wdb}, F. Values used to compute T_{aew}:
i. R_s for January
ii. t_{es} and t_{em} for 3am (see Table 2.5)
iii. F_s for winter conditions
iv. T_{ad} = estimated maximum temperature on winter design day.
$= T_{wdb} + DR$

T_a = Outdoor air bin dry bulb, F
T_{sdb} = 1% summer design dry bulb temperature, F.
T_{wdb} = 99% winter design dry bulb temperature, F.

Finally, if a plenum is used, it is assumed that 70% of the roof transmission heat is transferred to air in the plenum; 30% of the load reaches the zone.

c. **Glass and Skylight Transmission Load**

Glass and skylight transmission load equation:

$$Q_g = U_g * A_g * (T_a - T_i)$$

where:
Q_g = Transmission load for glass or skylight exposure, BTU/hr.
U_g = Glass or skylight exposure U-factor, BTU/(hr-sqft-F).
A_g = Glass or skylight exposure area, sqft.
T_a = Outdoor air bin dry bulb, F.
T_i = Indoor air dry bulb, F.

d. **Transmission Through Walls Adjacent to Non-Conditioned Region**

This load element allows the user to model transmission through walls adjacent to a region not conditioned by the HVAC system. With this element, the user can model adjacent regions such as an unconditioned warehouse.

Transmission load equation:

$$Q_{wu} = U_{wu} * A_{wu} * (T_u - T_i)$$

where:
Q_{wu} = Transmission loss through wall adjacent to non-conditioned region, BTU/hr.
U_{wu} = Wall U-factor, BTU/(hr-sqft-F).
A_{wu} = Wall area, sqft.
T_i = Indoor air dry bulb, F.
T_u = Air dry bulb temperature in non-conditioned region, F. T_u modeled in four ways:

i. Adjacent region is only heated:
$T_u = T_{uh}$ if $T_a =< T_{uh}$
$T_u = T_a$ if $T_a > T_{uh}$

ii. Adjacent region is only cooled:
$T_u = T_{uc}$ if $T_a >= T_{uc}$
$T_u = T_a$ if $T_a < T_{uc}$

iii. Adjacent region is cooled and heated:
$T_u = T_{uc}$ if $T_a >= T_{uc}$
$T_u = T_a$ if $T_{uh} < T_a < T_{uc}$
$T_u = T_{uh}$ if $T_a =< T_{uh}$

iv. Adjacent region is neither cooled nor heated:
$T_u = T_a$

T_{uc} = Adjacent region indoor dry bulb for cooling, F.
T_{uh} = Adjacent region indoor dry bulb for heating, F.
T_a = Outdoor air bin dry bulb, F.

e. **Lighting Heat Gain**

Heat gain from lights is assumed to be instantaneous. No lighting storage load factors are used in estimating lighting heat gain.

Lighting heat gain equation:

$$Q_l = E_l * H_f * D_l * A_f * 3.413$$

where:
Q_l = Total lighting heat gain, BTU/hr
E_l = (Lighting watts)/(sqft zone floor area).
H_f = Heating factor for lighting ballast:
1.25 if ballast is used in lights.
1.00 otherwise.

D_l = Lighting diversity factor: the typical fraction of total lighting watts in use for the occupied or unoccupied period.

Design cooling load : $D_l = 1.00$
Typical occupied period : D_l = User defined.
Typical unoccupied period : D_l = User defined.
Design heating load : $D_l = 0.00$

A_f = Zone floor area, sqft.
3.413 = (Btu/hr)/Watt

If a plenum is used, the lighting heat gain may be split between the plenum and the zone as follows:

$Q_{lp} = R_l * Q_l$ = Portion of lighting load to plenum
$Q_{lz} = (1 - R_l) * Q_l$ = Portion of lighting load to zone

where:
R_l = Fraction of lighting load to plenum.
= 0.30 if a plenum exists and lights are recessed.
= 0.00 otherwise.

f. **Miscellaneous Electrical Heat Gain**

The purpose of this load element is to model heat generation in the zone by electrical machinery and appliances. Separate miscellaneous electrical wattages are defined by the user for the occupied and unoccupied periods.

Heat gain equation:

$$Q_e = E_m * A_f * 3.413$$

where:
Q_e = Miscellaneous electrical heat gain, BTU/hr.
E_m = (Miscellaneous electrical watts)/(sqft floor area).
A_f = Zone floor area, sqft.
3.413 = (BTU/hr)/Watt

g. **People Sensible and Latent Heat Gain**

Heat gain equations:

$$Q_{ps} = P_s D_p A_f / P_a \qquad Q_{pl} = P_l D_p A_f / P_a$$

where:
Q_{ps} = People sensible heat gain, BTU/hr.
Q_{pl} = People latent heat gain, BTU/hr.
P_s = (BTU/hr sensible heat gain)/person
P_l = (BTU/hr latent heat gain)/person
P_a = (sqft floor area)/person
D_p = People diversity factor: the typical fraction of the maximum occupancy level for either the occupied or unoccupied period.
i. Design cooling load : $D_p = 1.00$
ii. Typical occupied load : D_p = user defined.
iii. Typical unoccupied load : D_p = user defined.
iv. Design heating load : $D_p = 0.00$

A_f = Zone floor area, sqft.

P_s and P_l may be user defined or may be chosen from one of the five pre-stored activity levels in Table 2.6.

TABLE 2.6 People Sensible and Latent Heat Gain Factors

Activity Level	Sensible Heat Gain (BTU/hr)/person	Latent Heat Gain (BTU/hr)/person
1. Seated at rest	230	120
2. Office or retail	245	205
3. Light work	280	270
4. Medium work	295	455
5. Heavy work	525	925

h. **Miscellaneous Sensible and Latent Heat Gains**

Miscellaneous heat gain equations:

$$Q_{msz} = Q_{ms}A_{fz}/(A_{fp} + A_{fi})$$

$$Q_{mlz} = Q_{ml}A_{fz}/(A_{fp} + A_{fi})$$

where:
Q_{msz} = Miscellaneous sensible heat gain for zone z, BTU/hr.
Q_{mlz} = Miscellaneous latent heat gain for zone z, BTU/hr.
Q_{ms} = Total miscellaneous sensible heat gain for occupied or unoccupied period, BTU/hr.
Q_{ml} = Total miscellaneous latent heat gain for occupied or unoccupied period, BTU/hr.
A_{fz} = Zone floor area, sqft.
A_{fp} = Perimeter zone floor area, sqft.
A_{fi} = Interior zone floor area, sqft.

i. **Infiltration Sensible and Latent Loads**

Infiltration is assumed to occur only in the perimeter zone. Separate infiltration air flow rates are entered by the program user for the occupied and unoccupied periods.

Infiltration load equations:

$$Q_{is} = 1.1 * V_i * (T_a - T_i)$$

$$Q_{il} = 4840 * V_i * (W_a - W_i)$$

where:
Q_{is} = Sensible infiltration load, BTU/hr.
Q_{il} = Latent infiltration load, BTU/hr.
V_i = Infiltration air flow rate, cfm.
T_a = Outdoor air bin dry bulb, F.
T_i = Indoor air dry bulb, F.
W_a = Outdoor air bin specific humidity ratio, lb/lb.
Computed as described in 1.2c.
W_i = Indoor design air specific humidity ratio, lb/lb.
This value is computed assuming the indoor relative humidity is fixed at the user-specified value. The computation is only an approximation since OPCOST does not simulate humidity control and thus, the zone relative humidity may actually float on either side of the design value. Wi is computed using the specific humidity ratio equations described in Chapter 1 of the Documentation Guide.
R_h = Indoor design relative humidity, decimal.

j. **Warm-Up Cycle Loads**

When the thermostat is set back during unoccupied heating periods, a warm-up cycle will occur at the end of these periods as the equipment warms the building from the unoccupied to occupied thermostat settings. The size of the warm-up load depends upon a number of factors including the magnitude of the setback. To account for warm-up cycles, unoccupied heating loads are increased by a user-specified percentage. In this way, the effect of the warm-up load is modeled by spreading the load evenly over the unoccupied period. The program user must estimate the warm-up percentage by taking into consideration the size of the setback, the length of the unoccupied period, and the thermal mass of the building.

k. **Pulldown Cycle Loads**

When cooling is not provided during the unoccupied period, a pulldown cycle will occur at the beginning of each occupied cooling period as the equipment cools the building to the thermostat setting. It is assumed that equipment is turned on one hour before an occupied period begins to pull the building temperature down. This is typically the case. To account for pulldown cycles, occupied period cooling loads are increased by a fixed percentage. In this way, the effect of the pulldown load is modeled by spreading the load evenly over the occupied period. The pulldown factor is computed as follows:

$$F_{pd} = (H_o + H_o/h_o)/H_o = 1 + 1/h_o$$

where:
F_{pd} = Pulldown factor. Occupied period cooling loads are multiplied by this factor to account for the pulldown load.
H_o = Number of hours in an occupied temperature bin, hr/yr.
h_o = Length of occupied period, hr/period.

2.4 ZONE DRY BULB TEMPERATURE CALCULATION

The purpose of this section is to describe the simple procedure used to determine the representative indoor dry bulb temperature for each bin load condition. The procedure utilizes the following thermostat data provided by the user:

T_c = Indoor cooling thermostat setting, F.
T_h = Indoor occupied heating thermostat setting, F.
T_{sb} = Unoccupied heating setback degrees, F.

It is assumed the air conditioning system maintains zone air at the thermostat setting temperature under all load conditions, except in the dead band region where the temperature floats between the cooling and heating settings. Determination of design and average bin zone temperatures, as well as dead band modelling are discussed in the following sub-sections.

a. **Cooling Design Bin**

At the cooling design condition, the zone air is assumed to be maintained at the cooling thermostat setting, T_c.

b. **Heating Design Bin**

At the heating design condition, the zone air temperature is assumed to be maintained at the occupied heating thermostat setting, T_h. The purpose of the heating design load calculation is to estimate the extreme heating load condition. Use of T_h will yield a worst case load while use of the unoccupied thermostat setting may not.

c. **Typical Bins**

A simple trial and error procedure is used determine the representative zone air temperature for each bin.

For the first trial, loads are computed using the cooling thermostat setting, T_c, as the indoor air temperature. If the resulting sensible thermal load is a cooling load, the indoor air temperature assumed is correct. If the load computed is a heating load, we proceed to the second trial.

For the second trial, loads are computed using the heating thermostat setting as the indoor air temperature. For the occupied period, this setting is Th; for the unoccupied period, it is (T_h - T_{sb}). If the resulting load is a heating load, the correct indoor air temperature value was assumed. If the computed load is a cooling load, then the thermostat must be in the dead band region.

d. **Dead Band Condition**

When dead band conditions occur, the indoor dry bulb temperature floats between the cooling and heating settings so that transmission loads are balanced by internal gains and the zone sensible thermal load is zero.

a. **Derivation of Basic Fan Power Equation**

First, we define:

$P_f = .746 * BHP/.9$

where:
P_f = Fan input power, kw.
BHP = Fan brake horsepower, hp.
.746 = kw/hp conversion factor.
.9 = Assumed value of fan motor efficiency.

Next, we can define:

$BHP = 0.000157\ V_f\ T_s\ /\ n_f$

where:
V_f = Fan air flow rate, cfm.
T_s = Total static pressure across fan, inches of water, gage.
n_f = Fan drive and mechanical efficiency.
.000157 = Conversion factor:
= (62.3 lbm/ft³ water)(ft/12 in)(hp-min/33000 ft-lb)

Combining these equations, we obtain:

$P_f = (.746)(.000157)V_fT_s/(.9n_f)$ (Eqn 4.1)

Finally, we can also define:

$Q_f = 3.413 * P_f$

where:
Q_f = Fan heat gain, BTU/hr.
3.413 = MBH/kw

Varied and unvaried versions of equation 4.1 are used for the five categories of fans as described below.

b. **Central Single Speed Constant Volume Fan Model**

Equation:

$P_f = (.746)(.000157)V_fT_s/(.9n_f)$

where values used for nf are defined in Table 4.1.

c. **Central 2-Speed Constant Volume Fan Model**

Equation:

$P_f = (.746)(.000157)V_fT_sR_f^3/(.9n_f)$

where:
R_f = (Low speed rpm)/(High speed rpm), if low speed setting used.
= 1.00 if high speed setting used.
n_f values are shown in Table 4.1.

d. **Central Single Speed Variable Volume Fan Model**

Equation:

$P_f = (.746)(.000157)V_fT_sM_{fe}/(.9n_f)$

where:
M_{fe} = Fan energy multiplier for part load fan operation
= $A + B*R + C*R^2$, such that $M_{fe} > E$
R = (Part load fan air flow rate)/(Design fan air flow rate), cfm/cfm.
A,B,C,E and n_f values are defined in Table 4.1.

e. **Central 2-Speed Variable Volume Fan Model**

Equation:

$P_f = (.746)(.000157)V_f T_s M_{fe} R_f^3/(.9n_f)$

where:
M_{fe} = Fan energy multiplier. See (e) above.
R_f = Part load fan air flow ratio. See (d) above.
A, B, C, E, and n_f values are defined in Table 4.1

f. **Terminal Single Speed Constant Volume Fan**

Equation:

$P_f = (.746)(.000157)V_f T_s/(.9n_f)$

where:
n_f is a user entered average fan efficiency.

TABLE 4.1 Central Fan Data

Fan Type	n_f	A	B	C	E
1. FC	.600	.375	.625	.000	.40
2. FC with dampers	.560	.300	.300	.400	.45
3. FC with variable freq. drive	.490	.073	.246	.681	.10
4. FC with variable speed drive	.530	.158	-.565	1.407	.10
5. FC with 2-speed motor	.600	.375	.625	.000	0.0
6. BI/AF	.600	.092	2.325	-1.417	.45
7. BI/AF with inlet vanes	.550	.448	-.173	.725	.37
8. BI/AF w/ var. freq. drive	.470	.073	.246	.681	.10
9. BI/AF w/ var. speed drive	.475	.158	-.565	1.407	.10
10. Controlled pitch axial	.600	.158	-.565	1.407	.10

Key: FC = Foward curved
BI/AF = Backward inclined or air foil.

4.16 OUTDOOR AIR ECONOMIZERS

Outdoor air economizers control the outdoor ventilation damper position. By modulating the flow of outdoor air to the system, the proper mixture of outdoor and return air can be achieved to minimize or eliminate the cooling coil load. Four basic economizer control strategies are offered in the OPCOST program, and are discussed below. In the discussion, "economizer cooling" refers to the use of outdoor air to reduce the coil load. Mechanical cooling refers to the use of a refrigeration plant to provide cooling.

a. **Integrated Enthalpy Control**

This type of control involves the comparison of outdoor and return air enthalpies and dry bulb temperatures to determine outdoor air damper position. Mechanical and economizer cooling may occur simultaneously. The control strategy is shown in Figure 4.23, and is subsequently discussed.

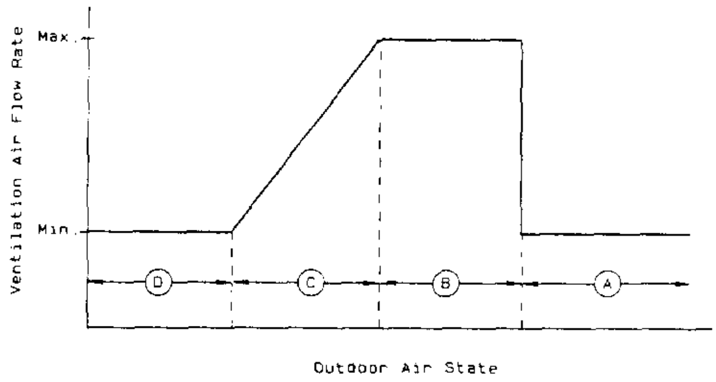

FIGURE 4.23
Integrated Enthalpy Economizer Operation

Operating Mode	Discussion
A	$h_r < h_a$. Ventilation dampers are held at minimum flow position. Use of additional outdoor air will not reduce the cooling coil load.
B	$h_r >= h_a$ and $T_a > T_s$. Ventilation dampers are full open. Supply air is 100% outdoor air. The cooling coil load is reduced, but mechanical cooling is still required.
C	$h_r > h_a$ and $T_a =< T_s$. The ventilation damper position is varied between maximum and minimum flow positions. Outdoor and return air streams are mixed so that no mechanical cooling is needed.
D	Ventilation dampers are held at the minimum flow position. Use of the minimum quantity of outdoor air eliminates the need for mechanical cooling, but may create the need for heating of the supply air stream.

Key:
h_r = Return air enthalpy, BTU/lbm.
h_a = Outdoor air enthalpy, BTU/lbm.
T_s = Cooling coil outlet dry bulb temperature, F.
T_a = Outdoor air dry bulb temperature, F.

b. **Non-Integrated Enthalpy Control**

This type of control involves the comparison of outdoor and cooling coil outlet air enthalpy with the C-Scale threshold of the Honeywell enthalpy sensor. The C-Scale threshold closely tracks the coil outlet dry bulb temperature. Therefore, a sufficient approximation of this control strategy is to use outdoor and cooling coil outlet air dry bulb temperatures to control ventilation damper position. See the description of Non-Integrated Dry Bulb Control for more information.

c. **Integrated Dry Bulb Control**

Integrated dry bulb control involves a comparison of the outdoor and return air dry bulb temperatures to determine the ventilation air damper position. With the integrated control strategy, mechanical and economizer cooling may occur simultaneously. The control scheme is illustrated in Figure 4.24, and is discussed below.

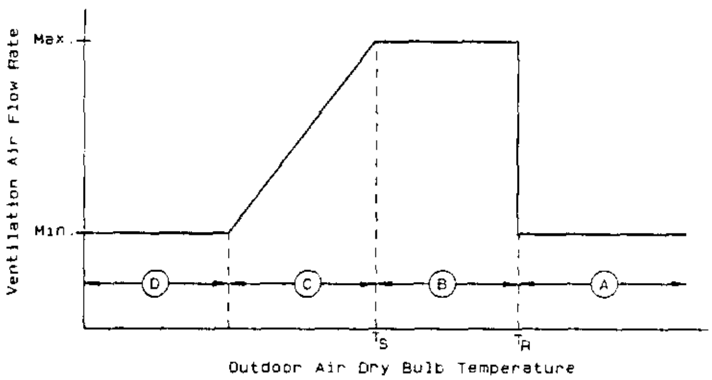

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FIGURE 4.24
Integrated Dry Bulb Economizer Operation

Operating Mode	Discussion
A	$T_a > T_r$. Ventilation dampers are held at the minimum flow position. An increase in ventilation air flow will not reduce the coil load.
B	$T_s < T_a = < T_r$. Ventilation dampers are opened to the maximum flow position. The use of 100% ventilation air reduces the cooling coil load, but some amount of mechanical cooling is still required.
C	$T_a = < T_s$. Ventilation damper position varies between maximum and minimum flow positions. Proper amounts of outdoor and return air are mixed to result in a supply air stream with temperature T_s. No mechanical cooling is therefore required.
D	Ventilation dampers are held at the minimum flow position. When the minimum quantity of ventilation air is used, no mechanical cooling is required, but heating of the supply air stream may be required.

Key:
T_a = Outdoor air dry bulb temperature, F.
T_r = Return air dry bulb temperature, F.
T_s = Required cooling coil outlet dry bulb temperature, F.

d. **Non-Integrated Dry Bulb Control**

Non-integrated dry bulb control involves a comparison of outdoor and cooling coil outlet air dry bulb temperatures to determine the ventilation air damper position. With non-integrated control, mechanical and economizer cooling do not occur simultaneously. The control scheme is illustrated in Figure 4.25 and is discussed below.

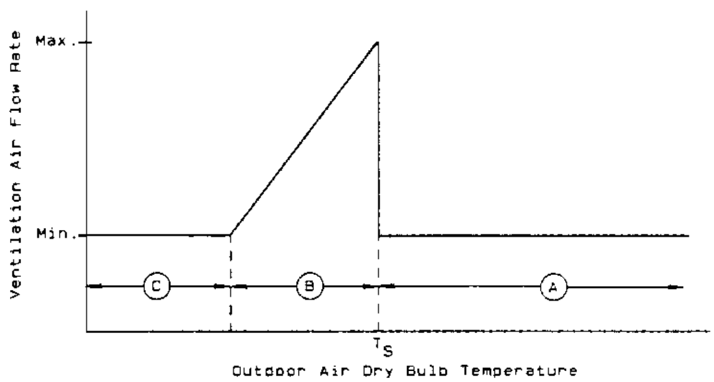

FIGURE 4.25
Non-Integrated Dry Bulb Economizer Operation

Operating Mode	Discussion
A	$T_a > T_s$. Ventilation dampers are held at the minimum flow position.
B	$T_a = < T_s$. Ventilation dampers modulate between maximum and minimum flow positions so that outdoor air and return air streams mix to create a supply air stream with temperature T_s. At this condition, no mechanical cooling is required.
C	Ventilation dampers are held at the minimum flow position. Use of the minimum quantity of outdoor air elimates the need for mechanical cooling, but may create the need for heating of the supply air stream.

Key:
T_a = Outdoor air dry bulb temperature, F.
T_s = Required cooling coil outlet air dry bulb temperature, F.

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4.17 VENTILATION AIR RECLAIM

A ventilation reclaim device is used to transfer heat between exhaust and ventilation air in order to reduce central coil loads. To reduce cooling coil loads, heat is transferred from hot outdoor ventilation air to cooler exhaust air to reduce the coil inlet temperature. To reduce heating coil loads, heat is transferred from hot exhaust air to cooler ventilation air to increase the coil inlet temperature. The heat transfer mechanism may be a pump around system, a heat wheel, a heat pipe or some similar device. The system is illustrated schematically in Figure 4.26. The two modes of system operation are discussed below.

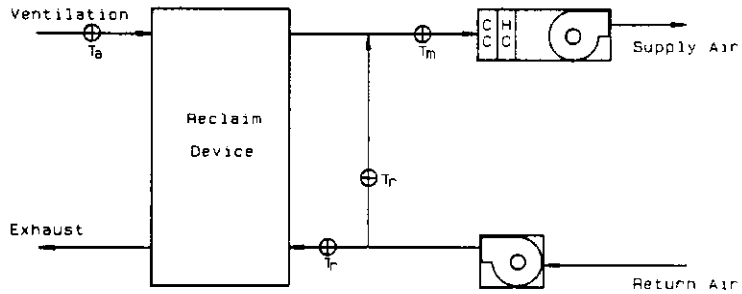

FIGURE 4.26
Ventilation Air Reclaim

a. **Cooling Mode**

For cooling reclaim, the temperatures T_a, T_r, and T_m, shown in Figure 4.26, are sensed by thermostats located in the appropriate air ducts. When $T_a > T_r$, it is possible to transfer heat from hot outdoor air to cooler exhaust air, and the reclaim device is turned on. As heat is transferred, the outdoor air is cooled, and the mixed air temperature, Tm is lowered. This in turn reduces the cooling coil load. The amount of heat transferred is estimated using the following equation:

$$Q_{vr} = 1.1\ V_v\ e_c\ (T_a - T_r)$$

where:
Q_{vr} = Cooling ventilation reclaim heat transfer, BTU/hr
V_v = Ventilation air flow rate, cfm.
e_c = Reclaim device efficiency. This value is the estimated average fraction of total heat transfer possible using the device
T_a = Outdoor air dry bulb temperature, F.
T_r = Return air dry bulb temperature, F.

Knowing Q_{vr}, T_r, the supply air flow rate and the ventilation air flow rate, the mixed air temperature, T_m can be computed. This temperature is subsequently used in the calculation of the cooling coil load.

b. **Heating Mode**

For heat reclaim operation, the temperatures T_a, T_r, and T_m are again measured by thermostats in the appropriate air ducts. When the condition $T_a < T_r$ occurs, heat can be transferred from the hot exhaust air stream to the cooler ventilation air stream, and the reclaim device is turned on. Heat is transferred from the exhaust to the ventilation air in order to elevate the mixed air temperature, T_m. The amount of heat transferred between air streams is estimated using the following equation:

$$Q_{vr} = 1.1\ V_v\ e_c\ (T_r - T_a)$$

where:
Q_{vr} = Heating ventilation reclaim heat transfer, BTU/hr.
The remaining variables were defined in (a) above.

Knowing Q_{vr}, T_r, and the ventilation and supply air flow rates, the mixed air temperature, Tm can easily be determined. This temperature is subsequently used in the calculation of the heating coil load.

4.18 CONTROL OF SUPPLY AIR TEMPERATURE

In the descriptions of air handling system operation (sections 4.1 - 4.14) the control of supply air temperature to maintain a constant temperature or to vary this temperature is often alluded to. The purpose of this section is to briefly describe the various methods of controlling supply air temperature. All the methods involve controlling the heat transfer at a cooling coil. The major methods used are listed in the table below.

Cooling Plant Type	Temperature Controlled By:
1. Direct Expansion	Cycling Cooling Plant As the cooling plant is cycled on and off, the supply air temperature fluctuates. The end result of this control is that the time-averaged supply temperature is at the desired level.
2. Direct Expansion	Unloading Compressors The unloading of compressors reduces the capacity of the plant so that the proper amount of cooling is provided to obtain the desired supply temperature.
3. Liquid Chilling	Varying Chilled Water Temperature By varying the temperature of chilled water entering the cooling coil, the heat transfer at the coil and ultimately, the supply temperature are controlled.
4. Liquid Chilling	Varying Chilled Water Flow Rate to Coil By varying the flow of constant temperature water to the cooling coil, heat transfer at the coil and ultimately, the supply temperature are controlled.
5. Any	Bypassing Air Around Coil By varying the flow of air over the coil, the coil outlet temperature may be controlled.

T_{ec} = Effective condenser entering air temperature
= T_a + E/1180
Use when condenser performance is not adjusted for altitude. See 5.1b for discussion.
= T_a
Use otherwise.
T_a = Outdoor air dry bulb, F.
N_c = Nominal plant capacity at 95 F outdoor air dry bulb temperature, Tons.
N_k = Nominal plant KW/Ton at 95 F outdoor air dry bulb temperature.
T_{lw} = Leaving chilled water temperature, F.
= T_{dw}
If chilled water reset control is not utilized.
= $T_{dw} + 10(1 - R_{pl})$
and $T_{lw} = < T_{dw} + 10$, and $T_{lw} = < 60$ F
If chilled water reset control used.
R_{pl} = Part load ratio.
= $Q_c/(12N_c)$
T_{dw} = Design leaving chilled water temperature, F.
Q_c = Total cooling coil load, MBH.
12 = MBH/Ton
MBH = 1000 BTU/hr

5.3 WATER COOLED RECIPROCATING COOLING PLANT

This cooling plant is a vapor compression machine which utilizes one or more reciprocating compressors and a water cooled condenser. The unit may be direct expansion or liquid chilling. It is assumed that the compressor motors are electric.

Energy costs considered for this plant are classified as follows:

i. Cooling Plant Costs:
Compressor energy cost
Cooling tower fan energy cost (if applicable)
ii. Pump Costs:
Condenser water pump energy costs
Chilled water pump energy costs (if applicable)

The calculation of pump input power is described in section 5.16, while that for the cooling tower fan power is summarized in section 5.15. The compressor input power computation is developed below.

a. **Plant Part Load Performance Equation**

The part load performance of this cooling plant is determined using the following equation:

$P_c = R_{kw} N_{kw}$

where:
P_c = Compressor input power, KW.
R_{kw} = Fraction of nominal compressor KW for plant operating condition.
N_{kw} = Nominal compressor input KW at 85 F condenser entering water temperature.

The fraction R_{kw} is dependent upon the part load ratio, entering water temperature, capacity control mechanism, and the use of suction temperature control. The relationship between Rkw and these variables was quantified via a statistical analysis of the part load performance characteristics of a number of water cooled reciprocating units. The calculation of R_{kw} and N_{kw} are summarized below.

$R_{kw} = (R_{ew}\, C_{st}\, C_{cc}\, Q)/(12\, N_c)$

$N_{kw} = N_k\, N_c$

where:
R_{ew} = Condenser entering water temperature factor.
= $(0.133 + .01T_{ew})/(1.645 - .0076T_{ew})$

C_{st} = Suction temperature control factor. See section 5.1b for discussion.
= 1.0
Use if suction temperature is fixed.
= $.92+.08Q/(12N_c)$, C_{st} = <1
Use if suction temperature is scheduled.
C_{cc} = Capacity control factor. See section 5.1b for discussion.
= $.4(Q/(12N_c))^2 - .1(Q/(12N_c)) + .7$, C_{cc} = <1
Use for multiple compressors per condenser circuit arrangement.
= 1.0
Use for one compressor per condenser circuit arrangement, with no compressor cycling.
= $1.15 - .15(Q/(12N_c))$, C_{cc}> = 1
Use for one compressor per condenser circuit arrangement, with compressor cycling.
Q = Cooling plant load, MBH
= $12\ R_u\ N_c$
Use when hot gas bypass control is utilized and the last unloading step has been reached. See section 5.1b for discussion.
= Qc
Use for all other cases.
N_c = Nominal plant capacity at 85 F condenser entering water temperature, Tons.
N_k = Nominal compressor KW/Ton at 85 F condenser entering water temperature.
T_{ew} = Condenser entering water temperature, F. See section 5.15 for discussion of computation.
R_u = Part load fraction at final unloading step.
Q_c = Cooling coil load, MBH.
12 = MBH/Ton.
MBH = 1000 BTU/hr.

5.4 AIR COOLED CENTRIFUGAL CHILLER

This cooling plant is a vapor compression machine which utilizes a centrifugal compressor and an air cooled condenser. The unit is liquid chilling. The compressor motor is assumed to be electric.

Energy costs considered for this plant are classified as follows:

i. Cooling Plant Costs:
   Compressor energy cost
   Condenser fan energy cost
ii. Pump Costs:
   Chilled water pump energy cost

This type of cooling plant is currently marketed by only one major air conditioning manufacturer from whom it was not possible to obtain part load performance data. In lieu of such data, the part load performance of this machine is approximated using the model for a water cooled centrifugal chiller with a closed circuit cooling tower. The only departure from the standard water cooled model is that no minimum entering water temperature restriction is imposed. Input power calculations are discussed for the compressor in section 5.5, for the cooling tower fans in section 5.15 and for the chilled water pump in section 5.16.

5.5 WATER COOLED CENTRIFUGAL CHILLER

This cooling plant is a vapor compression chiller which utilizes a centrifugal compressor and a water cooled condenser. The unit is liquid chilling. The compressor motor is assumed to be electric.

Energy costs considered for this plant are classified as follows:

i. Cooling Plant Costs:
   Compressor energy cost
   Cooling tower fan energy cost (if applicable)
ii. Pump Costs:
   Condenser water pump energy cost
   Chilled water pump energy cost

The calculation of pump input power is discussed in section 5.16 while that for the cooling tower fan input power is described in section 5.15. The computation of compressor input power is developed below.

a. **Chiller Part Load Performance Equation**

The part load performance of this cooling plant is determined using the following equation:

$P_c = R_{kw} N_{kw}$

where:
P_c = Compressor input power, KW.
R_{kw} = Fraction of nominal compressor input power at plant operating condition.
N_{kw} = Nominal compressor input power at 85 F condenser entering water temperature, KW.

The fraction R_{kw} is dependent upon the part load ratio, the entering water temperature, suction temperature control and the use of sequenced chillers. The relationship between R_{kw} and these variables was determined from a statistical analysis of the part load performance characteristics of a number of water cooled centrifugal chillers. The calculation of R_{kw} and Nkw described below applies to the use of a single chiller. Modification of the calculations for use of sequenced chillers is described in sub-section (b).

$R_{kw} = C_{st}\, C_{pl}\, C_w$

$N_{kw} = N_k\, N_c$

where:
C_{st} = Suction temperature control factor. See section 5.1b for discussion.
= 1.0
Use if suction temperature is fixed.
= $.92+.08Q_c/(12N_c)$
Use if suction temperature is scheduled.
C_{pl} = Part load factor.
= $.375R_{pl}^2 + .4R_{pl} + .225$
C_w = Condenser entering water temperature factor.
= $.00015T_{ew}^2 - .013T_{ew} + 1.021$
N_c = Nominal plant capacity at 85 F condenser entering water temperature, Tons.
N_k = Nominal plant KW/Ton at 85 F condenser entering water temperature.
R_{pl} = Part load ratio.
= $Q_c/(12N_c)$
T_{ew} = Condenser entering water temperature, F. See section 5.15 for discussion.
Q_c = Cooling coil load, MBH.
12 = MBH/Ton
MBH = 1000 BTU/hr

b. **Part Load Perforamance Analysis for Sequenced Chillers**

In some applications, a number of centrifugal chillers are connected to the chilled water loop to enchance part load performance for the system. It is assumed that each chiller connected to the loop has the same nominal capacity. The performance of each chiller in the system is analyzed assuming the chillers are controlled as described in the following example:

Two 200-ton centrifugal chillers are connected to the chilled water loop. For the sake of simplicity in this example, the condenser entering water temperature and the plant capacities remain constant as the coil load varies. It is assumed that the system is controlled so that the part load fraction experienced by each chiller corresponds to the sample schedule shown below:

It is assumed that the same heating plant which provides heat for the zone or building provides steam for the absorption chiller. Therefore, the generator input, Pgi, is added to the heating coil loads to determine the total heating plant load. The heating plant input power is computed using the method described for the chosen plant (see sections 5.9 – 5.12).

5.8 WATER SOURCE HEAT PUMP

This dual cooling and heating system utilizes reversible water cooled reciprocating heat pumps. The heat pumps are connected to a common water loop as shown in Figure 5.2. The system has three distinct operating modes as described below.

i. Cooling Mode
Both zones require cooling. The units operate as direct expansion cooling plants. Heat is rejected through the water loop to a cooling tower or heat sink.

ii. Reclaim Mode
One zone requires cooling, the other, heating. Accordingly, heat pumps in the cooling zone provide cooling and reject heat to the water loop. Units in the heating zone operate using the water loop as a heat source. As a result, heat rejected by the cooling units is reclaimed and used by the heating units.

iii. Heating Mode
Both zones require heating. The heat pumps operate using the water loop as a heat source. An auxiliary heating plant is used to maintain the loop water at a specified minimum temperature.

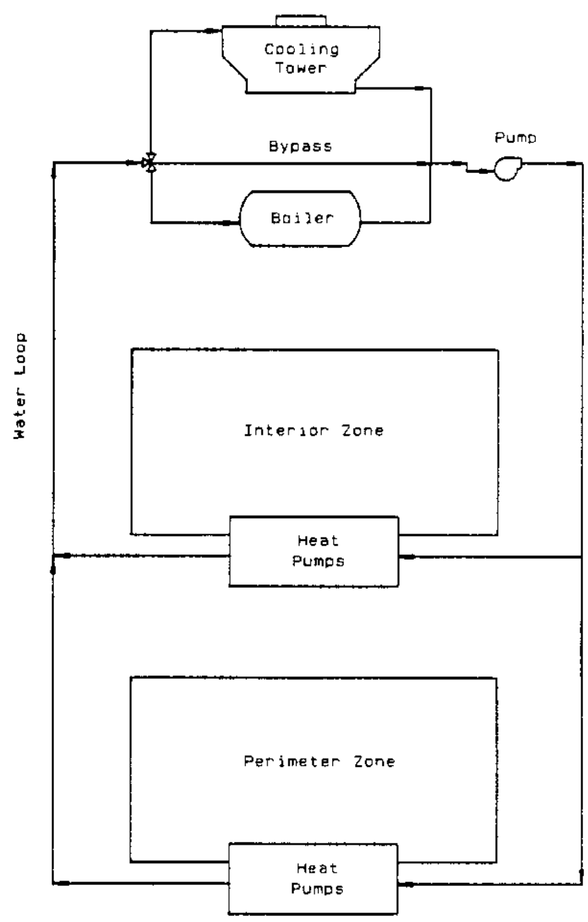

FIGURE 5.2
Water Source Heat Pump System Diagram

The energy costs considered for this system are classified as follows:

i. Cooling Plant Costs:
   Compressor input energy cost
   Cooling tower fan energy cost (if applicable)
ii. Heating Plant Costs:
   Auxiliary heating plant energy cost
iii. Pump Costs:
   Condenser (loop) water pump energy cost

The calculation of pump power is described in section 5.16. Computation of cooling tower fan power is summarized in section 5.15. The analyses of heat pump and auxiliary heating plant performance are developed below. In separate sections, the heat pump part-load performance equations for cooling and heating duty are described, and the modeling of each of the three system operating modes is discussed.

a. **Heat Pump Part-Load Performance Equation for Cooling Duty**

For cooling duty, the heat pump performance is analyzed using a modified version of the water cooled reciprocating cooling plant model. The derivation of this model is discussed in section 5.3. The model and its modifications are summarized below.

$P_c = R_{kw}N_{kw}$

$R_{kw} = (R_{ew}\, C_{st}\, C_{cc}\, Q)/(12\, N_c)$

$N_{kw} = N_k\, N_c$

where:
P_c = Compressor input power, KW. It is assumed that the compressor motor is electric.
R_{kw} = Fraction of nominal compressor KW for plant operating condition.
N_{kw} = Nominal compressor input KW at 85 F condenser entering water temperature.
R_{ew} = Condenser entering water temperature factor.
= $(0.133 + .01T_{ew})/(1.645 - .0076T_{ew})$
C_{st} = Suction temperature control factor. See section 5.1b for discussion.
= 1.0
The heat pumps considered are direct expansion units in which the suction temperature is fixed.
C_{cc} = Capacity control factor. See section 5.1b for discussion.
= $1.15 - .15(Q/(12N_c))$, $C_{cc} >= 1$
The heat pumps considered have one compressor per condenser circuit. Compressors are cycled to regulate capacity.
Q = Cooling plant load, MBH. Hot gas bypass control is not used with these heat pumps. As a result no false loading occurs and the plant cooling load is always equal to the coil load.
N_c = Nominal plant capacity at 85 F condenser entering water temperature, Tons.
N_k = Nominal compressor KW/Ton at 85 F condenser entering water temperature.
T_{ew} = Condenser entering water temperature, F. See sub-sections (c), (d) and (e) below for discussion.
12 = MBH/Ton
MBH = 1000 BTU/hr

b. **Heat Pump Part Load Performance Equation for Heating Duty**

For heating duty, the part load performance of the heat pump is analyzed using a modified version of the ground source heat pump model. The derivation of this model is described in section 5.14. The model and its modifications are summarized below.

$P_c = R_{kw}\, N_{kw}$

$R_{kw} = Q_p\, R_{ew} / Q_{cap}$

where:
P_c = Compressor input power, KW.
N_{kw} = Nominal compressor input power at 70 F evaporator entering water temperature, KW.
R_{kw} = Fraction of nominal compressor input power at plant operating condition.
R_{ew} = Evaporator entering water temperature factor.
$.00022T_{ew}^2 - .0216T_{ew} + 1.42$

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Q_p = Plant heating load. In this system, the heat pump must be sized so that under all conditions, the plant capacity is sufficient to meet the heating load. As a result, the plant heating load is equal to the heating coil load.
Q_{cap} = Plant capacity, MBH.
= $(.00018T_{ew}^2 - .0124T_{ew} + 1)H_c$
T_{ew} = Evaporator entering water temperature, F. See sub-sections (c), (d), and (e) for discussion.
H_c = Nominal plant capacity at 70 F evaporator entering water temperature, MBH.
MBH = 1000 BTU/hr.

c. **System Operation in Cooling Mode**

When both zones require cooling, the heat pumps operate as cooling plants, rejecting heat to the water loop. Compressor input power for units in each zone is computed using the equation described in sub-section (a). The condenser entering water temperature is computed using the appropriate heat sink model, described in section 5.15. This temperature is always held below the specified maximum entering water temperature and above the minimum temperature. The auxiliary heater is not used in this mode, and therefore consumes no energy.

d. **System Operation in Heat Reclaim Mode**

In this mode, heat pumps in the zone requiring cooling operate as cooling plants and reject heat to the water loop. The units in the zone requiring heating operate as heating plants and extract heat from the water loop. In this way heat is reclaimed.

To analyze the performance of the system, a basic heat balance is written for a control volume including the water loop and the heat pumps:

(Heat rejected to loop) - (Heat extracted from loop) = 0

$(Q_c + Q_{cc}) - (Q_h - Q_{ch}) = 0$

where:
Q_c = Cooling zone total coil load, MBH.
Q_h = Heating zone total coil load, MBH.
Q_{cc} = Compressor heat for cooling zone heat pumps, MBH.
= 3.413*(Compressor input KW)
Q_{ch} = Compressor heat for heating zone heat pumps, MBH.
= 3.413*(Compressor input KW)

In order to compute compressor input powers, the steady state entering water temperature must first be determined. By substituting the heat pump performance equations from sub-sections (a) and (b) into the heat balance above, an equation is obtained in which zone loads and the entering water temperature are the unknowns. For given zone loads, the steady state entering water temperature can be calculated using this equation. Compressor input power is then computed using the performance equations from sub-sections (a) and (b).

In the event that the computed entering water temperature is below the specified minimum temperature, the auxiliary heater must be used. Compressor power and heat are computed for entering water at the minimum temperature. The load on the auxiliary heater is computed using the heat balance below:

$Q_{aux} + (Q_c + Q_{cc}) - (Q_h - Q_{ch}) = 0$

where:
Q_{aux} = Auxiliary heater load, MBH.

The auxiliary heating plant input power is computed using the model for the chosen plant (see sections 5.9 - 5.12).

e. **System Operation in Heating Mode**

When both zones require heating, the units provide heat using the water loop as a heat source. The auxiliary heater is used to maintain the entering water at the specified minimum temperature. Compressor power and heat are computed using this entering water temperature value. The auxiliary heating plant load is then computed using the following heat balance for a control volume including the water loop and heat pump units:

Q_{aux} - (Heat removed from loop) = 0

$Q_{aux} - (Q_h - Q_{ch}) = 0$

where:
Q_{aux} = Auxiliary heating plant load, MBH.
Q_h = Sum of heating coil loads for interior and perimeter zones, MBH.
Q_{ch} = Sum of compressor heat gains for units in interior and perimeter zones, MBH.

The auxiliary heating plant input power is computed using the model for the chosen heating plant (see sections 5.9 - 5.12).

5.9 ELECTRICAL RESISTANCE HEATING PLANT

Heat is provided by direct heating elements or by electrically heated water with this system. It is assumed that the plant or coil element is properly sized to meet all heating loads encountered. Further, 100% conversion efficiency is assumed.

Energy costs considered for this means of heating are classified as follows:

i. Heating Plant Costs:
   Coil element or plant input energy cost
ii. Pump Costs:
   Hot water pump energy cost (only for hydronic HVAC systems)

Pump input power calculations are described in section 5.16. Total heating element or plant input power is computed using the following equation.

$P_h = Q_h/3.413$

where:
P_h = Heating input power, KW.
Q_h = Total heating load, MBH
3.413 = MBH/KW
MBH = 1000 BTU/hr

5.10 COMBUSTION HEATING PLANTS

With this group of plants, a fuel is burned either to heat air directly or to heat water. The fuel used may be natural gas, fuel oil or propane.

Energy costs considered for these plants are classified as follows:

i. Heating Plant Costs:
   Heating plant fuel cost
ii. Pump Costs:
   Hot water pump energy cost (only for hydronic HVAC systems)

Pump input power calculations are described in section 5.16. Heating plant input power is computed using the following equation.

$P_h = Q_h F / n_p$

where:
P_h = Plant input power in therm/hr for natural gas, US or Imperial gallons/hr for fuel oil, or lb/hr for propane.
Q_h = Heating load, MBH.
F = Conversion factor or heating value factor:
= (1 therm natural gas/hr)/(100 MBH)
= (1 US gallon fuel oil/hr)/(138.7 MBH)
= (1 Imperial gallon fuel oil/hr)/(168 MBH)
= (1 lb propane/hr)/(21.68 MBH)

n_p = Plant efficiency.
= Average seasonal efficiency if value defined by user.
= $.45+.669(Q_h/H_c)-.361(Q_h/H_c)^2$
If user chooses to have computer generate part load plant efficiency. The efficiency is limited to the range $.45 = < n_p = < .758$. This polynomial was derived from data generated for a study of combustion plant part load efficiency by the California Energy Commission.
H_c = Heating plant rated capacity, MBH.
MBH = 1000 BTU/hr.

5.11 AIR COOLED CONDENSER

With this system, waste heat is provided to the zone at no cost. Typically, the source of the waste heat is the air cooled condenser of a refrigeration unit. This heating plant option may also be used, however, to model any situation in which free heat is provided.

It is assumed that sufficient heat is available at all times to meet the heating load. No heating plant or pumping costs are incurred with this system.

5.12 REMOTE SOURCE HEATING

With this system, hot water is provided to the building from a remote source. Typically, such a system is used for a college campus or complex of buildings. Hot water is generated at a central source and is provided to the buildings. A pump in the building is then used to distribute the water throughout the building. It is assumed that sufficient heat is available at all times to meet all heating loads.

Energy costs considered for this system are classified as follows:

i. Heating Plant Costs:
   Cost of purchased hot water
ii. Pump Costs:
   Hot water pump energy cost

Hot water pump calculations are described in section 5.16. Assuming no pipe losses, the heat purchased is equal to the zone or building heating load.

5.13 AIR SOURCE HEAT PUMP HEATING PLANT

This heating plant is a vapor compression heat pump which utilizes one or more reciprocating compressors and air as the heat source. The heat pump provides some or all of the required heat as the plant capacity permits. Any additional heat required is provided by an auxiliary heating plant. It is assumed that the compressor motor is electric.

Energy costs considered for this plant are classified as follows:

i. Heating Plant Costs
   Compressor and evaporator fan energy cost
   Auxiliary heating plant energy cost
ii. Pump Costs
   Hot water pump energy cost (if used with auxiliary plant)

Hot water pump calculations are described in section 5.16. The input power calculations for the heat pump and auxiliary plant are described below.

a. **Heat Pump Input Power**

The part load performance of the heat pump is determined using the following equation:

$P_h = R_{kw} N_{kw}$

5.15 COOLING TOWERS AND HEAT SINKS

The water cooled cooling plants modeled in the OPCOST program utilize one of three heat sink types listed below:

i. Open cooling tower
ii. Closed circuit cooling tower
iii. Constant temperature heat sink

The purpose of this section of documentation is to describe the models used to compute the condenser entering water temperature for a given set of atmospheric and load conditions, and the cooling tower fan input power (if applicable) for each heat sink. Heat sink models are discussed in sub-sections below.

a. **Open Cooling Tower**

In an open cooling tower, condenser water is sprayed over baffles or fill in the tower as shown in Figure 5.3. The tower fan draws ambient air through the tower. Condenser water is cooled by contact with the air, and then collected in the sump before being pumped back to the condenser.

A study of the performance of open cooling towers yielded the following correlation for condenser entering water temperature as a function of ambient and part load conditions.

$$T_{ew} = A_0 + A_2R_{pl} + A_2R_{pl2}$$

where:

A_0 = $3.065065 + 2.72273h_a - .0217201h_a^2$,
$A_0 = < T_a - 3$
A_1 = $61.1089 - .900838A_0 + 2.96571 \times 10^{-3}A_0^2$
A_2 = $-22.4969 + .337364A_0 - 9.42044 \times 10^{-4}A_0^2$
T_a = Outdoor air dry bulb temperature, F.
h_a = Outdoor air enthalpy, BTU/lbm.
= $.24T_a + W_a(1061 + .444T_a)$
W_a = Outdoor air specific humidity ratio, lb/lb. W_a is computed using the equations described in section 1.1.
R_{pl} = Cooling tower part load ratio.
Q_c = Plant cooling load, MBH
N_c = Cooling plant rated capacity, Tons
12 = MBH/Ton
MBH = 1000 BTU/hr

The following restrictions are imposed, and adjustments made to the computed value of T_{ew}:

i. $T_{mw} <= T_{ew} = < 130$
It is assumed that the cooling tower is controlled so that T_{ew} is always less than or equal to 130 F and greater than or equal to a user-defined minimum entering water temperature, T_{mw}.
ii. When hot gas bypass control is used with a water cooled reciprocating chiller, it is assumed that the cooling tower is controlled so that $T_{ew} >= 80$ F.
iii. If the tower performance has not been adjusted for altitude, T_{ew} is increased by the quantity (Elevation)/1180 to account for altered tower performance due to the change in air density with elevation. See sub-section 5.1b for discussion.

The preceeding equation for T_{ew} is used in an iterative solution method to determine the steady state system operating condition. This method is necessary because of the inter-relation of cooling plant and cooling tower performance. The entering water temperature is affected by the quantity of heat rejected by the cooling plant. In turn, the plant heat rejection is affected by compressor heat gain, which is dependent upon the entering water temperature. A small number of iterative cycles is usually required to determine the steady state value of T_{ew}.

Finally, the cooling tower fan input power is estimated as follows:

$$P_f = F_f Q_{cap}$$

where:

P_f = Cooling tower fan input power, KW.
F_f = Fan power factor. The value of this factor was determined from a study of the performance of open cooling towers.
= 1.177×10^{-2} KW/Ton

Q_{cap} = Nominal heat rejection capacity, Tons. It is assumed that the tower is sized to handle the plant heat rejection at the design condition.
= $N_c + .2844 * N_k * N_c$
N_c = Nominal cooling plant capacity at 85 F condenser entering water temperature, Tons.
N_k = Nominal cooling plant KW/Ton at 85 F condenser entering water temperature.
.2844 = [3.413 MBH/KW][Ton/(12 MBH)]

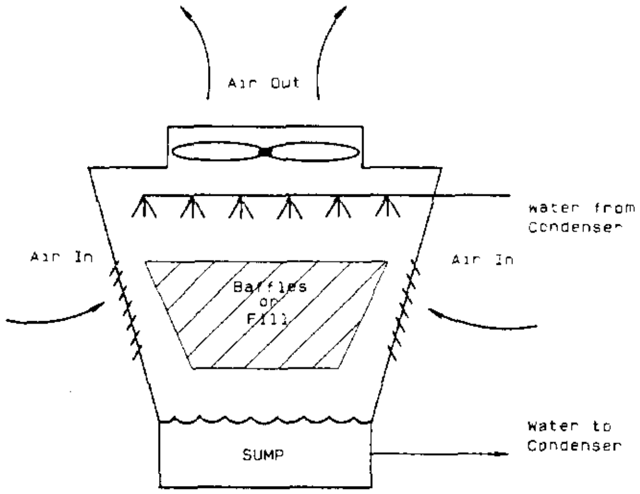

FIGURE 5.3
Open Cooling Tower

b. **Closed Circuit Cooling Tower**

In a closed circuit cooling tower, condenser water flows through coils in the tower as shown in Figure 5.4. Tower water is pumped from the sump and sprayed over the coils while a tower fan draws air through the tower. Spray water is cooled by contact with ambient air. The spray water in turn cools the tower coils causing the transfer of heat from the condenser water to the spray water.

To compute the condenser entering water temperature, a modified version of the open cooling tower model is used:

$$T_{ew} = A_0 + A_1 R_{pl} + A_2 R_{pl}^2 + 5$$

where all equation components are the same as defined in sub-section (a) above.

The cooling tower fan input power is computed using the equation:

$$P_f = F_f Q_{cap}$$

where:
P_f = Cooling tower fan input power, KW.
F_f = Fan power factor. The value of this factor was determined from a study of the performance of closed circuit cooling towers.
= 5.884×10^{-3} KW/Ton.
Q_{cap} = Nominal heat rejection capacity, Tons. It is assumed the cooling tower is sized to handle the heat rejection of the cooling plant at the design condition.
= $N_c + .2844 * N_k * N_c$
N_c = Nominal cooling plant capacity at 85 F entering water temperature, Tons.
N_k = Nominal cooling plant KW/Ton at 85 F entering water temperature.
.2844 = [3.413 MBH/KW][Ton/(12 MBH)]

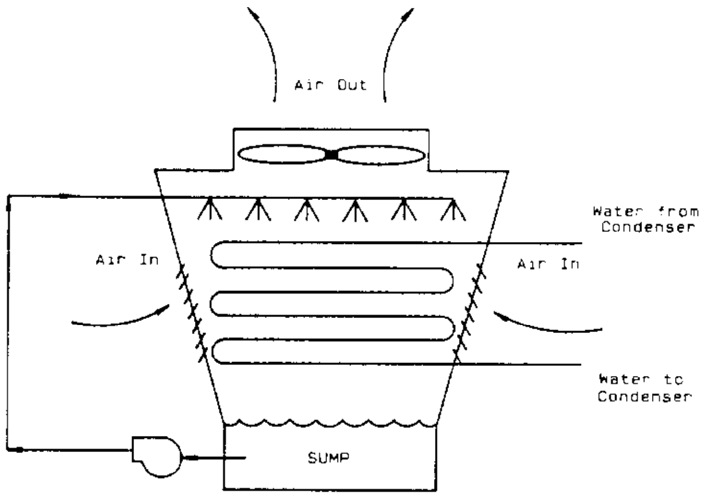

FIGURE 5.4
Closed Circuit Cooling Tower

c. **Constant Temperature Heat Sink**

With this heat sink, constant temperature water is provided to the condenser throughout the year. This model may be used for such heat sinks as cooling ponds and ground water. The condenser entering water temperature is set equal to a user-defined typical entering water temperature. This temperature represents average entering water conditions during the year. No fans are associated with this heat sink.

5.16 PUMP INPUT POWER CALCULATIONS

The energy consumption of pumps associated with heating and cooling systems is considered in computing the system operating cost. The following four pumping systems are examined:

i. Chilled water pumping system which transports water between the chiller and the cooling coils.
ii. Hot water pumping system which transports water between the water heater and the heating coils.
iii. Condenser water pumping system which transports water between the cooling plant condenser and the heat sink.
iv. Ground water pumping system which transports ground or well water to the evaporator of a ground source heat pump.

The purpose of this section is to describe the calculation of pump input power for these four pumping systems. In separate sub-sections below modeling assumptions are described, and the pump power equation is derived.

a. **Assumptions**

A simple pump system model is used in which all pumps are assumed to be of the constant speed, constant flow, constant head variety. The use of variable speed pumps, pump cycling, and the operation of pumps in various series and parallel configurations are not considered. Additional assumptions used in the pump model are that:

i. Pump motors are electric.
ii. The pump mechanical efficiency is 85%.
iii. The pump motor efficiency it 90%.
iv. Piping systems are well insulated so that thermal piping losses may be neglected.
v. Each pumping system contains one pump.

b. **Derivation of Basic Pump Power Equation**

Pump input power is computed using the equation:

$$P_p = (V_p\ Dp)/n_p$$

where:
P_p = Pump input power.
V_p = Volumetric flow rate.
D_P = Pump head.
n_p = Pump mechanical and electrical efficiency.

Because each pumping system is used to transfer heat in some way, the total heat transfer may be used to compute the volumetric flow rate. The calculation follows:

$V_p = Q_p/(p_w C_p D_T)$

where:
Q_p = Total heat transferred in system loop, BTU/hr. This may be a cooling coil load, condenser heat rejection, heating coil load or evaporator heat gain.
p_w = Density of water: 62.3 lbm/ft3.
C_p = Heat capacity of water: 1 BTU/(lbm-F).
D_T = System delta-T: the change in water temperature in the system loop, F.

The preceeding two equations are combined to obtain pump input power as a function of pump head, D_p, pump system temperature difference, D_T, and pump system heat transfer, Q_p:

$P_p = (Q_p D_P F)/(p_w C_p D_T n_p)$

where:
P_p = Pump input power, KW.
Q_p = System heat transfer. BTU/hr.
= Cooling plant nominal capacity for chilled water system.
= Heating plant rated capacity for hot water system.
= $12000(N_c + .2844 * N_k * N_c)$ for condenser water system. This the cooling plant heat rejection at the design condition.
= The heat pump nominal capacity at 70 F evaporator entering water temperature for the ground water system.
D_P = Pump head, feet of water, gage.
D_T = Pump system delta-T: the change in water temperature in the system loop, F.
p_w = Density of water: 62.3 lbm/ft3.
C_p = Specific heat of water, 1 BTU/(lbm-F).
n_p = Pump mechanical and electrical efficiency: (.85)(.90).
F = Units conversion factor.
= [(hp-min)/(33000 ft-lb)]*[(hr/(60 min)]*
[kw/(1.341 hp)]*[(62.3 lbm water)/ft^3]
N_c = Plant nominal capacity at 85 F entering water temperature, tons.
N_k = Plant nominal KW/ton at 85 F entering water temperature.
.2844 = [3.413 MBH)/KW][on/(12 MBH)]
12000 = (BTU/hr)/Ton

5.17 HYDRONIC ECONOMIZERS

A hydronic economizer is an accessory used with a water cooled cooling plant to reduce the mechanical refrigeration load. The economizer consists of a pre-cooling coil and associated controls which will use cooling tower water to pre-cool air before it passes through the main cooling coil. Cooling tower water is only allowed to flow through the pre-cooling coil when the water temperature is less than the coil entering air temperature. A schematic of the economizer is shown in Figure 5.5. The operation of the economizer is integrated so that the economizer and the mechanical refrigeration plant can operate simultaneously.

The amount of cooling occuring at the pre-cooling coil is estimated using the following equation:

$Q_e = 1.1 V_s e_e (T_{ce} - T_{ew})$

where:
Q_e = Economizer cooling, BTU/hr.
V_s = Volumetric air flow rate through pre-cooling coil, cfm.
T_{ce} = Dry bulb temperature of air entering pre-cooling coil, F.
T_{ew} = Condenser entering water temperature, F. See section 5.16 for discussion of calculation of T_{ew}.
e_e = Average economizer efficiency. This value is an average fraction of the total possible heat transfer which occurs at the coil. Calculation of this value is discussed below.

a. **Calculation of Economizer Efficiency**

If a specific economizer is being modeled, the economizer efficiency may be approximated using the product data. For several entering water temperature conditions and, if possible, several entering air conditions, the efficiency should be computed:

$e_e = TC/(1.1\ V_s\ (T_{ce} - T_{ew}))$

where:
TC = Total cooling capacity, BTU/h.

The efficiency values computed should then be averaged and this average value should be entered as the economizer efficiency.

If a specific product is not being modeled, some representative efficiency value should be used. A typical value is 70%.

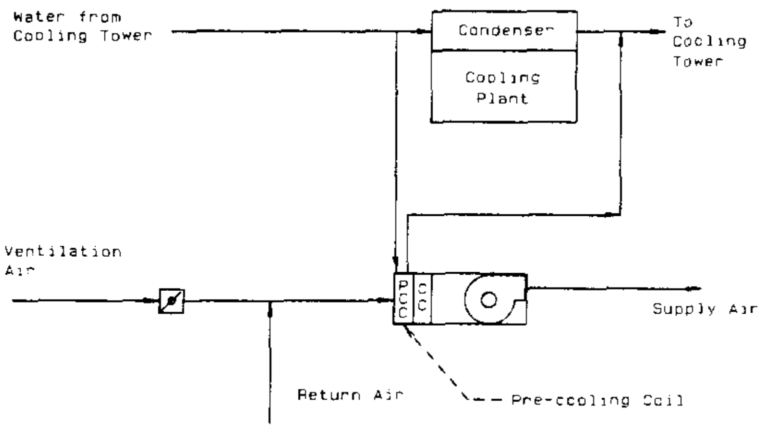

FIGURE 5.5
Hydronic Economizer

AVERAGE ANNUAL OPERATING COST CALCULATIONS

a. **Introduction**

In this final stage of the analysis, annual operating costs are computed using input power and energy consumption data generated in previous stages. Simple summation calculations are carried out to derive component operating costs from bin by bin power consumption data. These calculations are briefly discussed in the following sections.

b. **Summary**

Average annual operating costs are compiled from component costs as shown below.

Fan Energy Costs	
Pump Energy Costs	
Cooling Plant Energy Costs	Non-HVAC Electrical Costs
+ Heating Plant Energy Costs	+ DHW Energy Costs
Total HVAC Energy Cost	Total Non-HVAC Energy Cost

Total HVAC Energy Cost
\+ Total Non-HVAC Energy Cost

Average Annual Operating Cost

c. **HVAC Component Energy Cost Calculation**

HVAC component energy costs are computed by summing input power data over all bins, for each zone and period:

Fan Energy Costs = Sum of [$P_f * H_b * C_e$]
Pump Energy Costs = Sum of [$P_p * H_b * C_e$]
Cooling Plant Energy Cost = Sum of [$P_c * H_b * C_e$]
Heating Plant Energy Cost = Sum of [$P_h * H_b * C_e$]

where:
P_f = Fan input power for conditions in one bin, KW.
P_p = Pump input power for conditions in one bin, KW.
P_c = Cooling plant input power for conditions in one bin. Units vary with cooling plant type. See documentation of the cooling plants for an explanation of how plant input power is accounted for.
P_h = Heating plant input power for conditions in one bin. Units vary with plant type.
H_b = Bin hours in one temperature bin, hr/yr.
C_e = Energy cost, $/fuel unit. A number of fuel costs are defined by the program user. Each cost is applied in the summation calculation with the appropriate plant, fan or pump. Electrical energy costs are sometimes broken into three rate categories: compressive energy-for compressor operation; inductive energy-for fan and pump operation; resistive energy-for electrical resistance heating.

d. **Non-HVAC Energy Cost Calculations**

Non-HVAC energy costs are computed using the following simple equations:

$$\text{Non-HVAC Electrical Cost} = (E_{lo} + E_{mo} + E_{oo})C_{eo} + (E_{lu} + E_{mu} + E_{ou})C_{eu}$$

where:
E_{lo}, E_{lu} = Annual energy use for lighting in occupied and unoccupied periods, KWH/yr. See section 6.1a.
E_{mo}, E_{mu} = Annual energy use for miscellaneous electrical appliances and machinery, KWH/yr. See section 6.1b.
E_{oo}, E_{ou} = Annual energy use for other electrical devices and systems, KWH/yr. See section 6.1c.

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C_{eo}, C_{eu} = Electrical energy rates. $/KWH. Rates may vary for occupied and unoccupied time periods. and for type of use. If these multiple costs are used. the resistive costs are utilized in these calculations.

DHW Energy Cost = $E_{wo} * C_{eo} + E_{wu} * C_{eu}$

where:

E_{wo}, E_{wu} = Annual energy use for domestic water heating during the occupied and unoccupied periods, respectively. Units of energy vary with the heating plant used.

C_{eo}, C_{eu} = Fuel or energy cost, $/fuel unit. Units depend upon the heating plant used. When an electrical resistance heating plant is used, it is possible to define separate electricity costs, C_{eo} for the occupied period, and C_{eu} for the unoccupied period. For all other types of DHW plants, C_{eo} and $C_{e}u$ are equal.

REFERENCES

1. Air Force Manual 88-29, *Facility Design and Planning Engineering Weather Data*, Government Printing Office, Washington D.C., 1978.

2. *Carrier System Design Manual, Part 1: Load Estimating*, Carrier Air Conditioning Company, Syracuse, New York, 1960.

3. Knebel, David E., *Simplified Energy Analysis Using the Modified Bin Method*, American Society of Heating, Refrigerating and Air Conditioning Engineers, 1983.

4. Knebel, David E., *Simplified Energy Analysis Using the Modified Bin Method*, American Society of Heating, Refrigerating and Air Conditioning Engineers, 1983, pg 2-1.

5. Erbs, Daryl G., *Models and Applications for Weather Statistics Related to Building Heating and Cooling Loads*, Doctor of Philosophy Thesis in Mechanical Engineering, University of Wisconsin-Madison (1984), pp 39-74.