BUILDING ENERGY MANAGEMENT AND REDESIGN RETROFIT (BEMARR)

AIR AND WATER ECONOMY CYCLES

CONTENTS PAGE

Figures

Tables

1. GENERAL

1.01 This section provides information on techniques and equipment available which will reduce energy requirements for mechanical cooling. Material used in this section has been extracted from the *Building Energy Management and Redesign Retrofit (BEMARR) Manual,* issued with GL 76-10-077 (EL-4857), dated October 7, 1976.

1.02 Whenever this section is reissued, the reason(s) for reissue will be listed in this paragraph.

2. SCOPE

2.01 In many buildings, it is necessary to operate mechanical refrigeration on a year-round basis to provide cooling for interior building spaces, computer equipment, and telephone switching equipment. This section provides information on techniques that can be employed to reduce mechanical cooling energy costs.

2.02 Economizer systems referred to as ***free cooling*** techniques are available which, when properly applied, can produce significant energy savings as compared to the continued use of energy intensive mechanical refrigeration. These techniques and their required supporting equipment have first costs as well as operating costs that make them far from free in their application. For example, equipment used for an evaporative cooling economy cycle which may consume 0.2 kilowatt per ton (kW/ton) of refrigeration is substituted for a refrigeration machine and cooling tower which might normally consume 1.0 kW/ton, thus reducing energy consumption by 80 percent.

2.03 The use of these techniques is dictated by the outdoor weather conditions and should be evaluated on the basis of the following parameters to determine their economic justification:

(a) Weather conditions—temperature/humidity

(b) Building occupancy requirements

(c) Compatibility with other building mechanical systems

(d) Systems control

(e) Installation costs

(f) Operation and maintenance costs.

3. OUTSIDE AIR ECONOMIZER CYCLES

3.01 The basic outside air economizer cycle has been in use for some time and is the most popular and widely used alternative to mechanical cooling. The concept of the outside air economy cycle is to use as much outside air as possible to meet the cooling requirements. In most cases where year-round cooling is required, mechanical refrigeration is operated in normal sequence until a lowering of the outside air dry-bulb temperature can assist mechanical cooling and, upon further lowering of the outside air dry-bulb temperature, replace the mechanical cooling altogether.

3.02 When this occurs, outside air is brought into the building, mixed with an appropriate amount of return air, and used as supply air to the space thereby allowing the chiller to be turned off.

3.03 The basic air economizer cycle is shown in Fig. 1. The return air fan may be required for control of building pressurization.

3.04 There are limitations to the use of the outside air economizer cycle that should be considered in the economic evaluation of such a system:

(a) If the outside air temperature which activates the system is too high, the humidity load may require refrigeration, thereby canceling any savings or even increasing the energy costs over using mechanical cooling with minimum outside air.

(b) Cool, dry air introduced into the system may impose a humidification load on the system.

(c) It is not used with heat recovery systems using cooling loads to develop heat for the system.

3.05 There are several control strategies that have been developed for use with the outside air economizer cycle. These control strategies are explained using psychrometric charts.

A. Dry-Bulb Economy Control

3.06 Figure 2 illustrates the traditional operating conditions of dry-bulb temperature control.

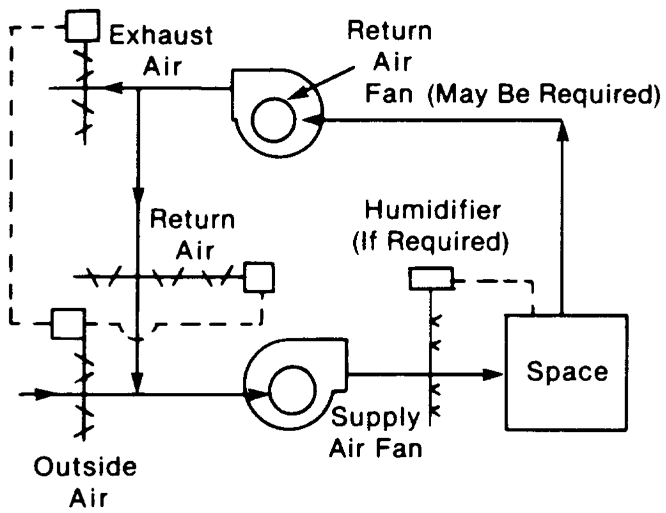

Fig. 1—Air Economizer System

Assuming that the supply air to the space requiring cooling is normally chilled to 62°F for 80°F space temperature, the mechanical refrigeration can be turned off and the outside air damper opened for cooling air whenever the outside air dry-bulb temperature is below 62°F. In this instance, the changeover control point is 62°F.

3.07 Additional energy savings are possible if the changeover control point is raised. For example, Fig. 3 illustrates the operating conditions for a 70°F changeover point. Total cooling with outside air (region A) occurs at 62°F outside dry-bulb temperature as in the traditional case. However, partial cooling with outdoor air is possible in the range of 62° and 70°F dry-bulb temperature or region B. Outdoor air, at conditions in region B, is easier to cool because its enthalpy is lower than that of the room return air.

3.08 It would appear that further raising of the changeover control point to a maximum of 80°F would provide further savings. There are two reasons why this is not necessarily true. First, this would increase the amount of time that outside air with a dew point above 62°F is used. An examination of Fig. 3 will indicate that such air above this dew point will cause the space to experience humidity problems with relative humidity greater than 55 percent. Secondly, using a higher changeover control point temperature could actually increase the energy required for cooling. If the enthalpy of the outside air is higher than that of the room air, using outside air could require more refrigeration energy than otherwise would be required if only the return air was cooled. Region C of Fig. 3 represents these conditions.

3.09 At a 70°F dry-bulb changeover control point temperature, the energy saved by using region B outweighs the small amount of extra energy required by region C. As the changeover point becomes higher, region C becomes dominant and will cause a net loss of energy. Evaluation of the desired changeover point requires consideration of these factors.

B. Enthalpy Economy Control

3.10 This method of outside air economizer control is based on the use of outside air to save mechanical refrigeration energy whenever the outside air enthalpy (or energy) is less than the enthalpy of the conditioned space.

3.11 Figure 4 illustrates the region of outside air utilization for enthalpy control. It is noted that whenever the outside air dry-bulb temperature is below normal supply air temperature (62°F in these examples), this type of control acts like the dry-bulb economy control by using outside air for 100 percent of the required cooling. Between this point and 80°F, the enthalpy control will measure the enthalpies of both the outside air and room air. If the outside air enthalpy is less than the room air, outdoor air will be used.

3.12 This is an advantage over the dry-bulb control in that under the conditions represented by

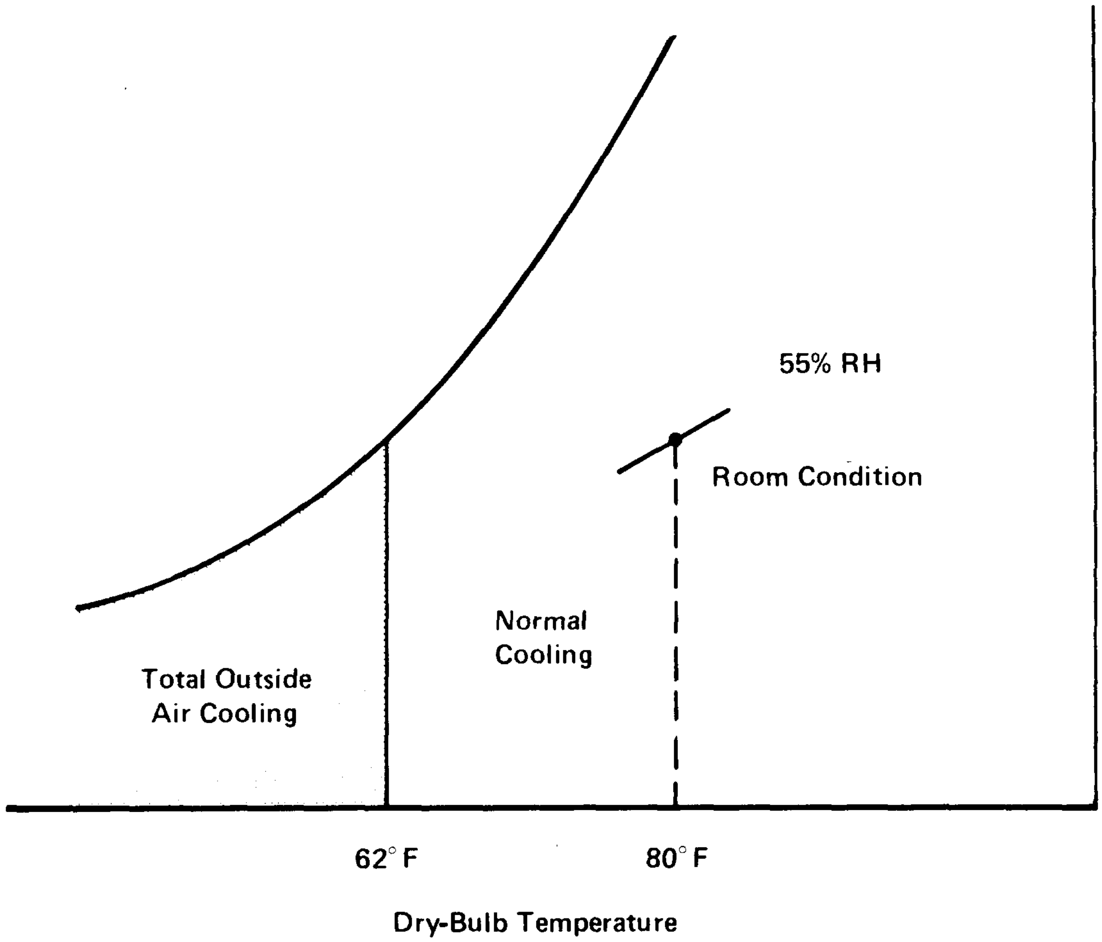

Fig. 2—Air Economizer System—Dry-Bulb Economy Control Operation With 62°F Changeover Point

region C, Fig. 3, outside air would not be used. Therefore, enthalpy will save more energy than the dry-bulb control which cannot distinguish between the attractive conditions in region B and the unattractive conditions in region C.

C. Dew Point Economy Control

3.13 This control will use outside air for cooling whenever the dew point and dry-bulb temperatures of the outside air are less than those of the space requiring the cooling.

3.14 The typical area of operation for the dew point economy control is illustrated by Fig. 5.

3.15 The dew point control prohibits the use of air with high dew point temperatures, thus eliminating the potential humidity problems associated with other economy controls.

3.16 The dew point control also makes considerable use of the outside air in the range of 62° to 80°F while not allowing outside air with unattractive enthalpies to be utilized.

3.17 The dew point economy control method appears to offer significant advantages. However, the dew point controller is expensive when compared to other control devices, and, in addition, it is difficult to maintain calibration. Therefore, this type of economy control is not recommended.

4. POTENTIAL SAVINGS OF OUTSIDE AIR ECONOMIZER CYCLES

4.01 Comparison studies of outside air economy cycles and their respective energy savings potential have been conducted using a computer model and the relatively high-internal heat loads and environmental constraints unique to telephone buildings. The studies were conducted for various outside air

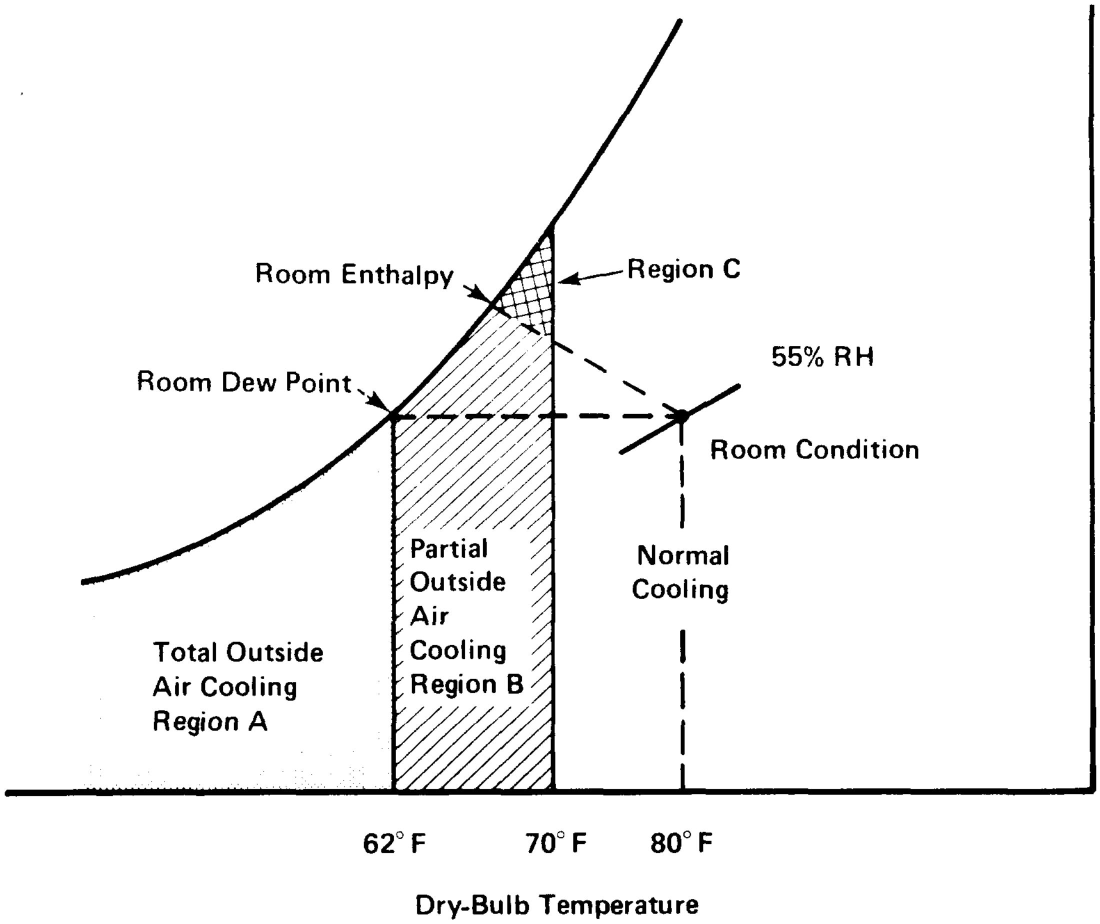

Fig. 3—Air Economizer System—Dry-Bulb Economy Control Operation With 70°F Changeover Point

dry-bulb changeover temperatures, varying internal heat loads, and enthalpy control, using weather data of cities representing different climatic areas of the country. The results of these studies are summarized in Table A.

4.02 The following conclusions can be drawn from these results:

(a) If compared to a dry-bulb economy cycle control which is set to operate (as many present systems are) at 60°F or below, the enthalpy economy control saves significantly more energy.

(b) At building heat loads of 5 watts per square foot (W/ft^2) or higher, if the same dry-bulb control is modified to operate at 70° or 75°F (depending on the geographic location), it will save almost the same energy as is saved by an enthalpy control.

(c) At low heat loads of $2W/ft^2$, the results vary significantly. In cities such as Denver, Billings, Seattle, Los Angeles, and Phoenix, the dry-bulb economy control can be set to save almost the same energy as the enthalpy control. In Miami, New Orleans, Atlanta, Dallas, and Kansas City, neither the dry-bulb nor enthalpy economy cycle will save a significant portion of the cooling energy. In New York, Chicago, Washington, Boston, and Minneapolis, the enthalpy control saves more energy than any dry-bulb control; however, even then, this extra savings is never more than about 10 percent of the total cooling load.

TABLE A

POTENTIAL COOLING SAVINGS
(PERCENT)

CITY	INTERNAL HEAT LOAD (w/ft²)	DRY-BULB CONTROL SWITCH POINT 60°F	65°F	70°F	75°F	ENTHALPY CONTROL
New York	2	0	0	8.5	-2.4	17.7
	5	11.7	23.5	35.7	31.6	41.2
	10	34.9	46.0	54.0	51.5	57.2
	15	43.4	53.4	59.9	57.9	62.5
Chicago	2	0	0	7.9	6.4	16.6
	5	10.4	21.8	34.2	35.0	40.1
	10	32.8	44.0	52.4	52.8	55.9
	15	43.6	53.5	60.0	60.4	62.8
Denver	2	0	0	10.7	22.4	24.2
	5	13.9	28.6	43.3	50.8	51.5
	10	38.8	52.4	61.7	65.8	66.2
	15	48.7	60.7	67.8	71.0	71.3
Miami	2	0	0	1.4	-8.2	4.6
	5	1.2	3.6	6.9	0.4	9.7
	10	2.7	6.0	9.2	3.6	11.7
	15	3.2	6.8	9.8	4.5	12.2
Washington, DC	2	0	0	5.7	0.3	13.5
	5	9.4	20.0	30.3	28.7	35.5
	10	29.7	40.6	47.9	46.9	51.2
	15	38.8	48.9	54.8	53.9	57.4

CITY	INTERNAL HEAT LOAD (w/ft²)	DRY-BULB CONTROL SWITCH POINT 60°F	65°F	70°F	75°F	ENTHALPY CONTROL
New Orleans	2	0	0	0.7	-19.7	4.5
	5	4.6	10.1	13.6	0.7	16.5
	10	13.8	20.6	23.7	13.8	25.9
	15	17.1	24.1	26.9	17.9	29.0
Billings	2	0	0	9.7	20.9	22.6
	5	14.4	27.6	42.1	49.3	50.1
	10	41.3	53.3	62.2	66.1	66.5
	15	51.4	61.9	68.7	71.7	72.0
Atlanta	2	0	0	3.6	-9.3	9.8
	5	7.7	16.3	24.5	18.4	29.0
	10	23.9	33.5	39.8	35.8	42.9
	15	30.2	39.6	45.1	41.6	47.6
Dallas	2	0	0	3.0	-0.1	6.9
	5	4.5	10.0	17.3	16.8	21.3
	10	16.8	23.9	30.4	30.0	33.2
	15	22.0	29.3	35.0	34.7	37.5
Seattle	2	0	0.1	24.0	42.1	43.6
	5	37.0	59.2	73.6	78.3	78.6
	10	63.5	78.3	85.1	87.2	87.4
	15	68.1	81.1	86.6	88.4	88.5

CITY	INTERNAL HEAT LOAD (w/ft²)	DRY-BULB CONTROL SWITCH POINT 60°F	65°F	70°F	75°F	ENTHALPY CONTROL
Los Angeles	2	0	0	32.1	49.9	52.8
	5	22.0	49.3	70.5	75.8	76.6
	10	34.0	60.5	74.9	78.3	78.7
	15	36.6	61.9	74.7	77.7	78.1
Phoenix	2	0	0	2.0	4.9	6.4
	5	3.8	8.3	14.6	18.8	19.8
	10	14.7	21.2	27.6	30.6	31.4
	15	18.9	25.8	31.7	34.3	35.0
Kansas City	2	0	0	4.0	-4.4	10.0
	5	6.7	15.3	24.4	21.4	29.3
	10	24.4	34.1	41.2	39.2	44.3
	15	33.8	43.0	48.7	47.2	51.2
Boston	2	0	0	11.1	10.8	21.7
	5	14.0	28.5	43.1	44.3	49.0
	10	38.8	51.8	60.8	61.5	64.2
	15	47.8	59.3	66.4	67.0	69.0
Minneapolis	2	0	0	9.5	11.7	19.7
	5	11.3	23.7	37.6	40.2	44.1
	10	32.1	44.5	53.9	55.5	57.9
	15	42.8	53.6	61.0	62.3	64.1

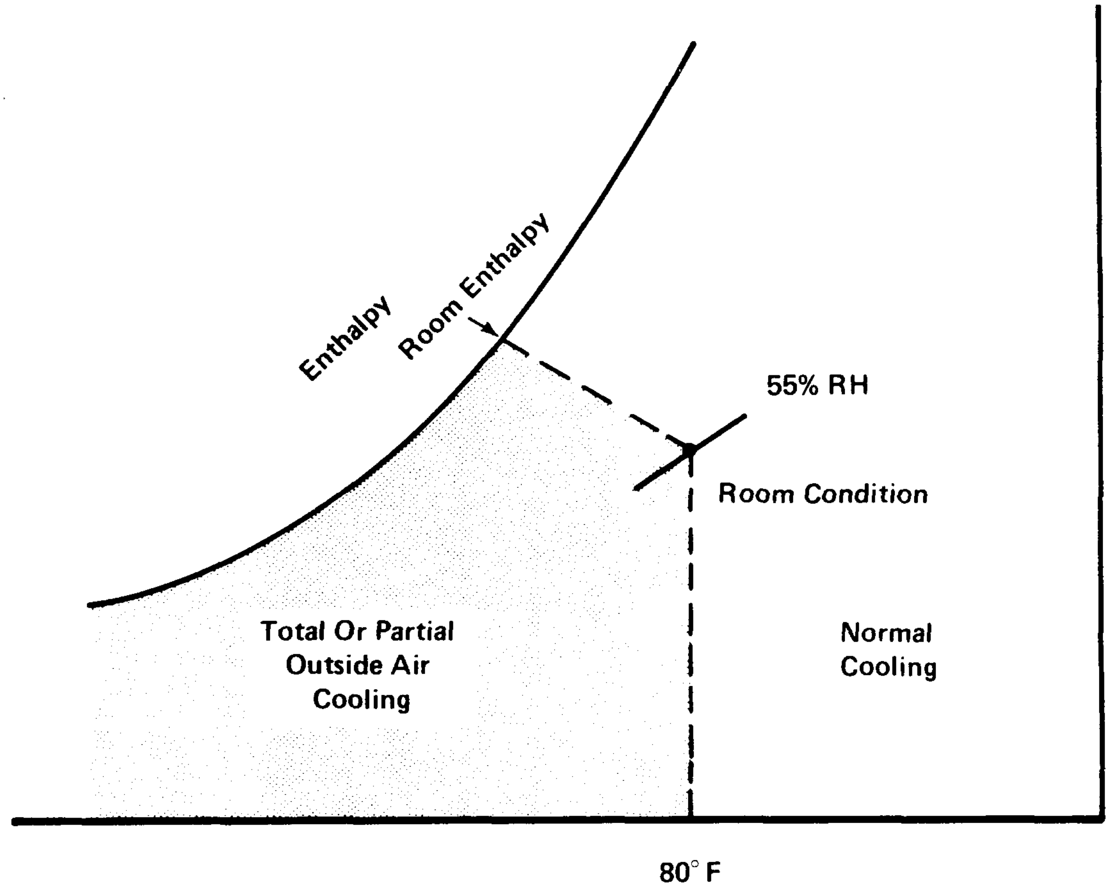

Fig. 4—Air Economizer System—Enthalpy Economy Control Operation

5. OPERATIONAL CONSIDERATIONS OF OUTSIDE AIR ECONOMIZER CONTROL

5.01 There are two operational problems that must be considered in connection with outside air economy cycles—calibration and humidity.

5.02 Enthalpy controls have a definite tendency to go out of calibration. The problem appears to be with the outdoor humidity sensor because it experiences large annual variations in temperature and humidity. Energy savings will be reduced whenever the sensor and controls are out of calibration.

5.03 The humidity problem may occur with either the dry-bulb or enthalpy controller. The potential for humidity problems arises whenever the outside air at a dew point greater than 62°F is used for economizer cooling of an office or area maintained at 80°F dry-bulb temperature and 55 percent relative humidity.

5.04 Consider, for example, an air-conditioning system supplying 62°F to an 80°F room (temperature differential = 18°F [ΔT = 18°F]) and is equipped with an enthalpy controller. Now, assume that the outside air is 66°F and has 100 percent relative humidity (ie, a 66°F dew point) (Fig. 6). It can be seen that the enthalpy of the outside air (point A) is slightly lower than the enthalpy of the room air (point C).

5.05 Under these conditions, the enthalpy controller will use the outside air to reduce cooling energy. Since the room requires 62°F supply air, the

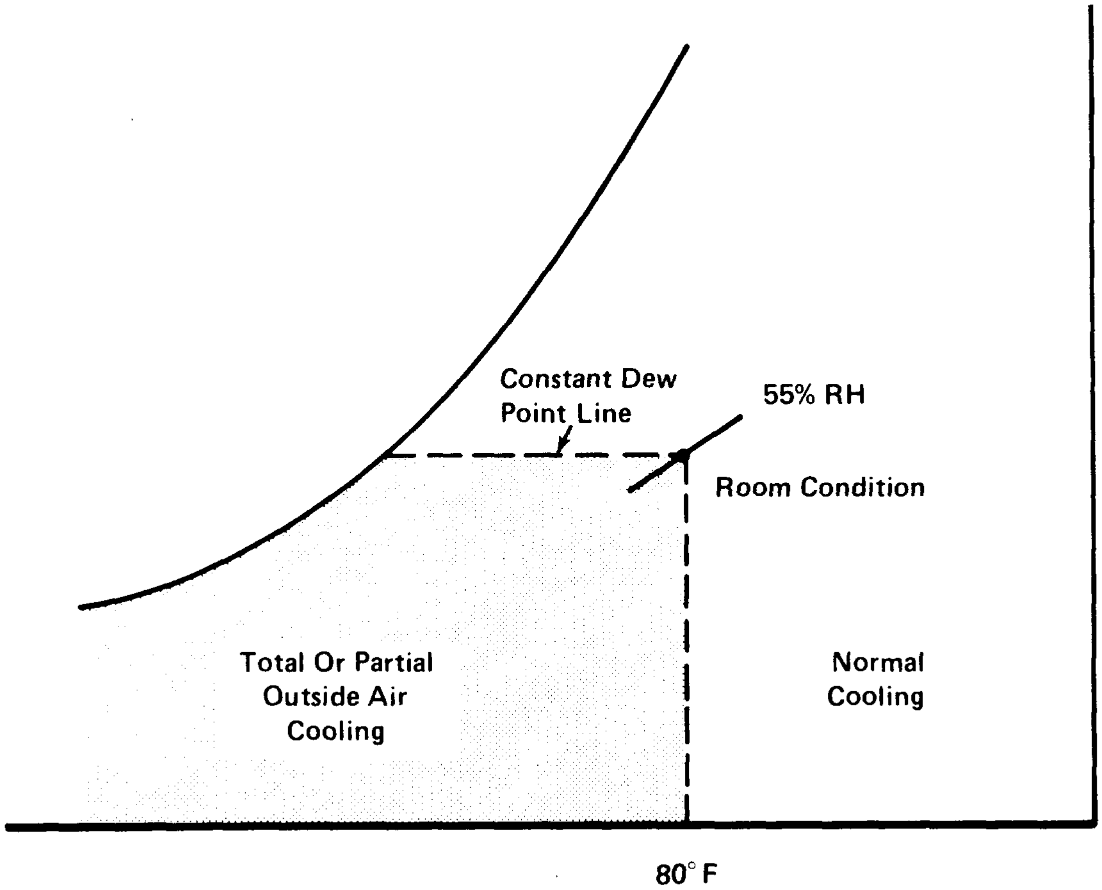

Fig. 5—Air Economizer System—Dew Point Economy Control Operation

chiller will have to operate in order to reduce the temperature of the outside air, which is now used as supply air, to 62°F (point B). Thus, the dew point of the air actually entering the conditioned space is acceptable.

5.06 Now, for example, consider a telephone equipment office only partially filled with equipment. Few offices are rebalanced for initial cooling loads; therefore, the actual load is less than design, and the system may be able to maintain space dry-bulb temperature with 66°F supply air rather than the design 62°F supply air. As before, assume outside air at 66°F and 100 percent relative humidity (Fig. 7, point A). The enthalpy control will determine that the enthalpy of the outside air is lower than the enthalpy of the space and will bring in outside air for cooling.

5.07 In this situation, the dry-bulb temperature of the outside air is sufficient to provide all of the sensible cooling required by the office. As a result, the chiller is turned off and outside air with 66° F dew point is brought into the office. Office relative humidity can, in time, exceed 65 percent humidity (Fig. 7).

5.08 There are probably other sets of circumstances where dry-bulb or enthalpy controllers could cause humidity problems. Therefore, it is recommended that any dry-bulb or enthalpy control system which could possibly use outside air with a dew point greater than 66°F should be equipped with a high limit humidistat. The humidistat will close the outside air intake dampers and halt further economy cycle operation if the relative humidity of the room exceeds 55 percent.

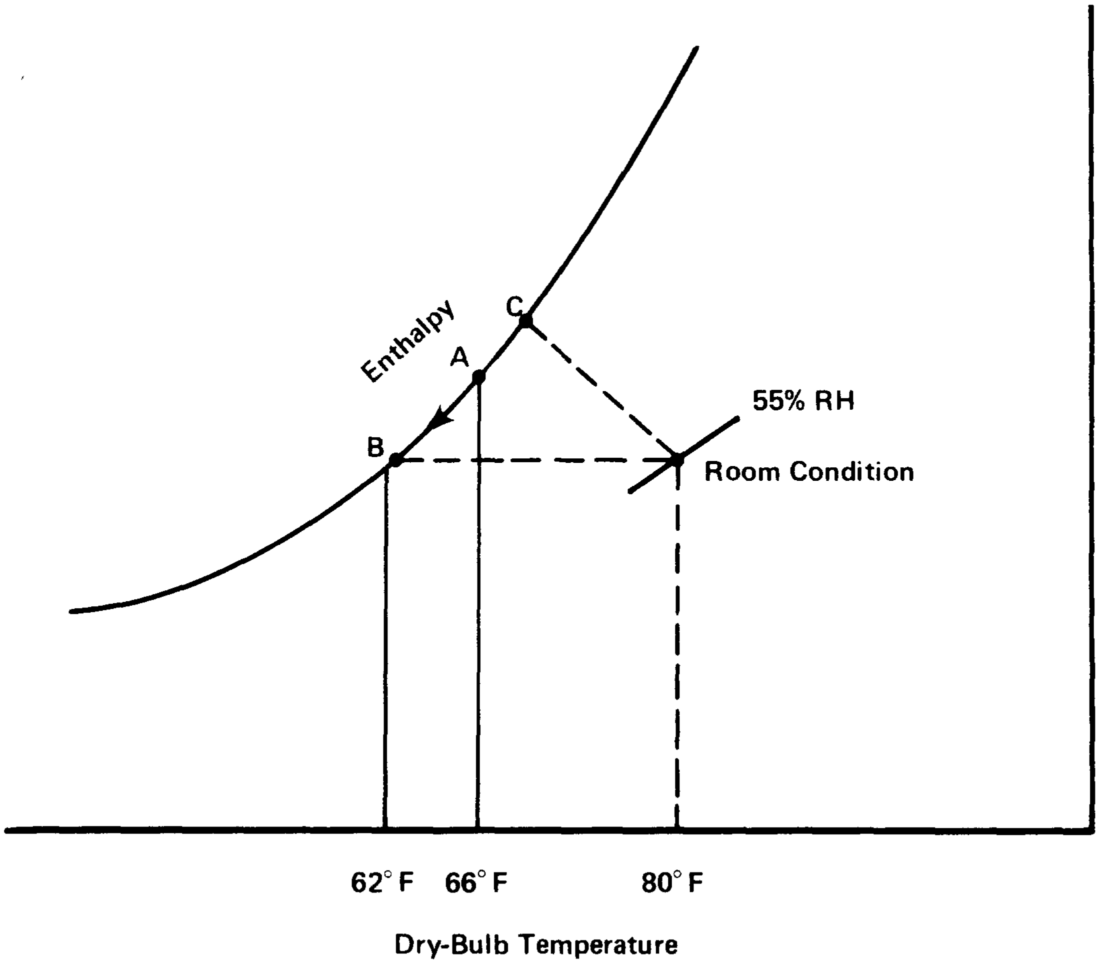

Fig. 6—Air Economizer System—Economy Cycle Operation Under Humid Outside Conditions

6. RECOMMENDATIONS FOR OUTSIDE AIR ECONOMIZER CYCLE

6.01 Table A, of the previously discussed study, indicates that the dry-bulb temperature control yields nearly the same energy savings as does the enthalpy control.

6.02 In a new installation, the cost for a dry-bulb economizer control system, including sensors, dampers, motors, etc, is nearly the cost for an enthalpy system ($1350 versus $1500).

6.03 It may appear to be cost-effective to convert an existing dry-bulb control system to enthalpy control. However, experience indicates that the extra enthalpy control hardware is generally not compatible with existing controls and dampers. As a result, significant and expensive changes are required.

6.04 It has been reported that the enthalpy controllers (the outside air humidity sensors) frequently lose calibration. This reduces the energy savings and may even save less energy than the dry-bulb temperature controls.

6.05 Therefore, it appears that the dry-bulb temperature control is the more attractive choice. To use the dry-bulb temperature control to maximum advantage requires that its changeover control be set at 70°F or higher depending upon the climatic conditions.

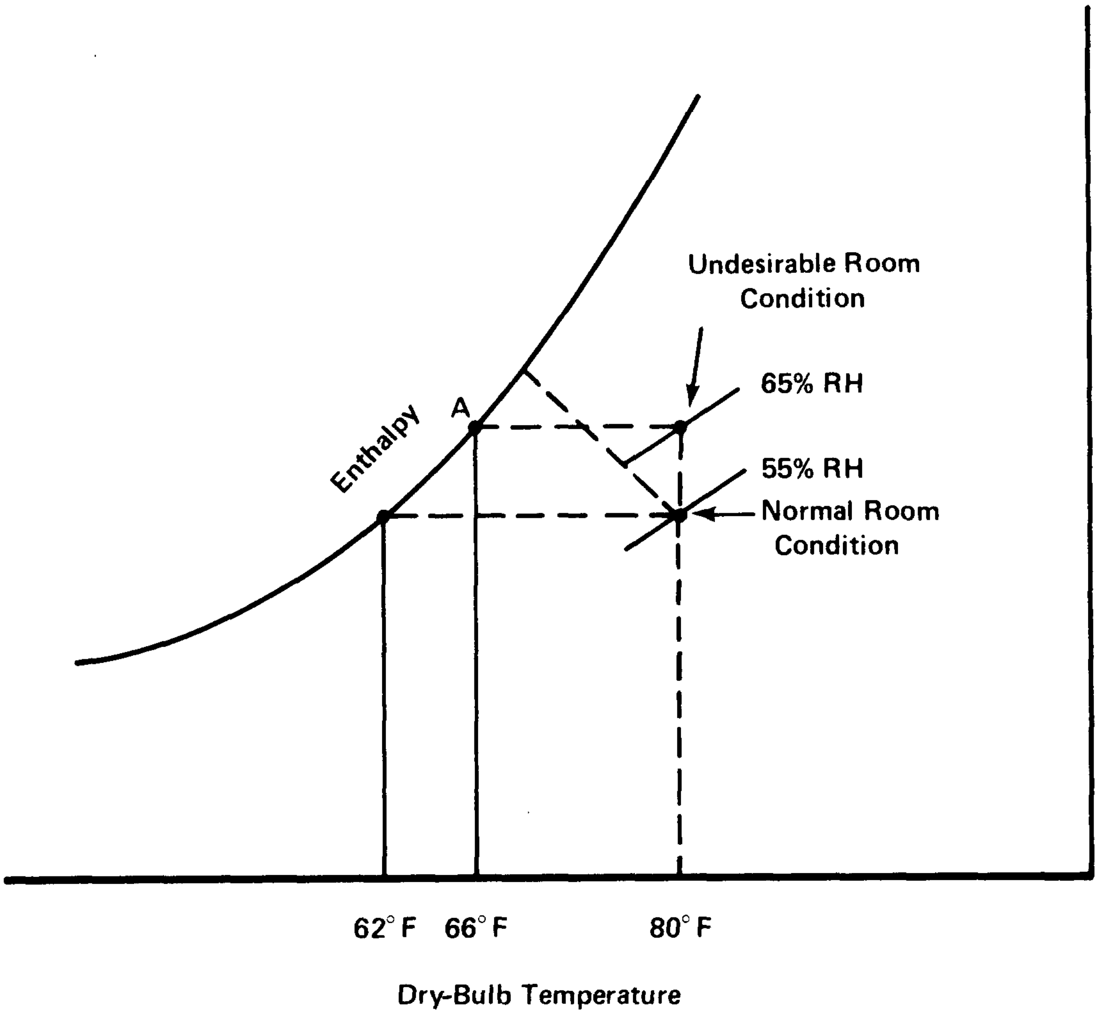

Fig. 7—Air Economizer System—Potential Humidity Problem Associated With Economy Cycle Operation

6.06 To protect against humidity problems with either type control will require a high limit room humidistat to override the economy cycle and close outside dampers in the event the room relative humidity exceeds 55 percent.

7. OTHER TYPES OF ECONOMY CYCLES

7.01 Evaporative chilling is a method of cooling water for refrigeration and air conditioning without using a chiller. During fall, winter, and spring, if the outside wet-bulb temperature drops to a sufficiently low value, enough cold water (condenser water) can be produced in the cooling tower to eliminate the need to run the refrigeration compressors. There are three basic types of evaporative chilling systems employed:

(a) Chiller vapor cycle

(b) Filtration system

(c) Cooling tower and heat exchanger.

7.02 Each of these systems can be classified as either ***open*** or ***closed*** to the atmosphere. In an open system, chilled water is passed directly through the open cooling tower and is exposed to the atmosphere. In a closed system, the chilled water circulates in a clean, closed circuit which is protected from atmospheric contaminants.

A. Chiller Vapor Cycle

7.03 This is the cycle that is most commonly known as free cooling. The chiller vapor cycle can provide cooling without operating the centrifugal-refrigeration compressor any time the condenser water temperature is below the desired chilled-water temperature. The chiller transfers heat through the refrigerant evaporation and condensing cycles without compression.

7.04 During the summer, this system operates as a conventional cooling tower and centrifugal chiller system. During the winter, the compressor is stopped, and an economizer valve between the condenser and evaporator is opened. (See Fig. 8.) Chilled and condenser water pumps operate as in the normal cooling cycle.

7.05 As the refrigerant removes heat from the chilled water in the evaporator, it is evaporated and migrates to the cooler condenser. The cooling tower water cools the tube surfaces of the condenser causing the refrigerant gas to condense on the tubes. The condensed liquid refrigerant then drains back to the evaporator to repeat the cycle.

7.06 The chiller vapor cycle is economically feasible when:

(a) Cooling load during the intermediate seasons is below 35 percent of the machine capacity.

(b) Low outdoor temperature/humidity prevails.

(c) Available low temperature condenser water is 6° to 10°F colder than the required chilled-water temperature.

(d) Centrifugal chillers with capacities of 200 tons or greater can be retrofitted. The chiller vapor cycle is also available as a factory option.

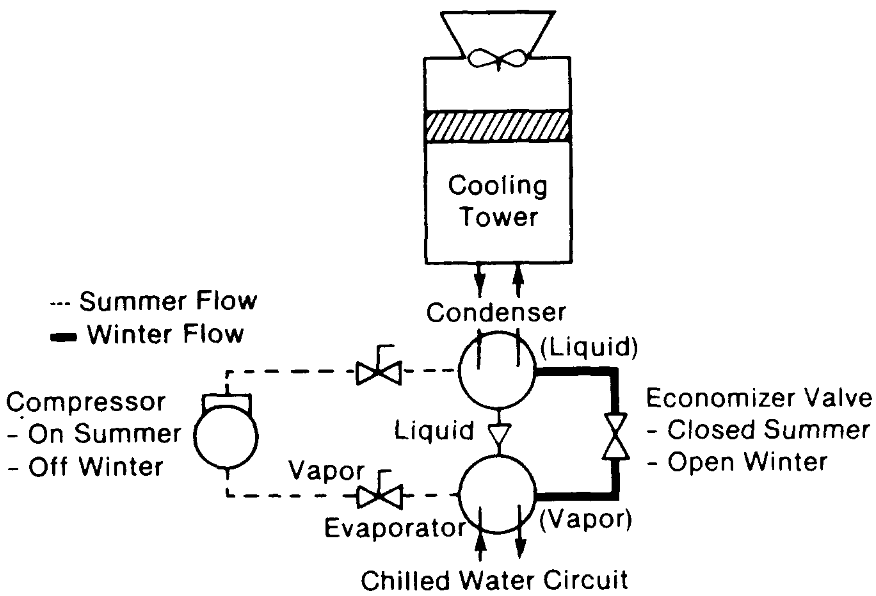

Fig. 8—Chiller Vapor Cycle

7.07 To obtain a maximum cooling effect, it is recommended that full design condenser water flow be circulated through the condenser.

7.08 To effectively operate the chiller vapor cycle for maximum heat transfer from the chilled water to the refrigerant, the entire interior tube surface in the evaporator must be wetted by the refrigerant.

7.09 It may be necessary to add a small refrigerant pump to produce a sprayed refrigerant surface. This is usually referred to as a pumped chiller vapor cycle. A patented system is available called Thermocycle*. (See Fig. 9.)

B. Filtration System

7.10 During the summer, this open-circuit system operates as a conventional cooling tower and chiller system. During the winter with the addition of strainers or water filters in the condenser water line, the cold water from the cooling tower is circulated directly through the chilled-water circuit to provide cooling. (See Fig. 10.)

7.11 The filtration or strainer system is intended to minimize corrosion or fouling in the chilled-water circuit by filtration of the undissolved contaminants from the water and air. These contaminants are concentrated by the evaporative cooling process, and particle sizes of 125 microns and larger are removed by the filter.

* Patented by Thermocycle International, Inc.

7.12 The filter is cleansed by an automatic backwash cycle. Approximately 150 gallons per minute (gpm) is used per backwash and, depending upon the water quality, could be repeated every 8 minutes which results in high water consumption and chemical treatment costs.

7.13 A backwash reclaim system can be used with the filter which is effective in reclaiming 90 percent of the water and chemicals used in each backwash cycle. The reclaim system uses a centrifugal separator to collect and discharge the contaminants to waste.

7.14 A patented filtration system is available called Strainercycle†.

C. Cooling Tower and Heat Exchanger System

7.15 During the summer, this system operates as a conventional cooling tower and chiller system. During the winter, the system provides for closed, chilled-water circuit in that the chiller is bypassed by the cold tower water and cools the chilled water through a heat exchanger. (See Fig. 11.)

7.16 This type of system has operated in northern regions for many years and can be economical in warmer regions if the heat exchangers are designed for a close temperature differential (2° to 5°F) between the condenser water and chilled water.

† Patented by Thermocycle International, Inc.

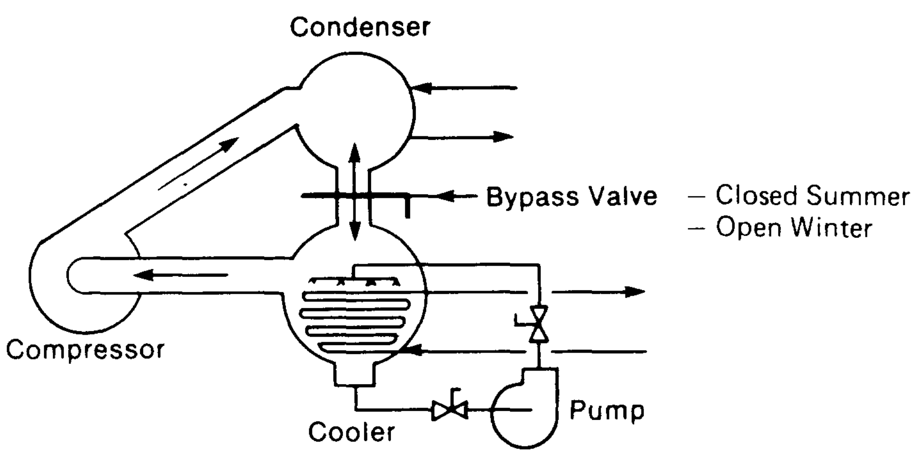

Fig. 9—Pumped Vapor Cycle

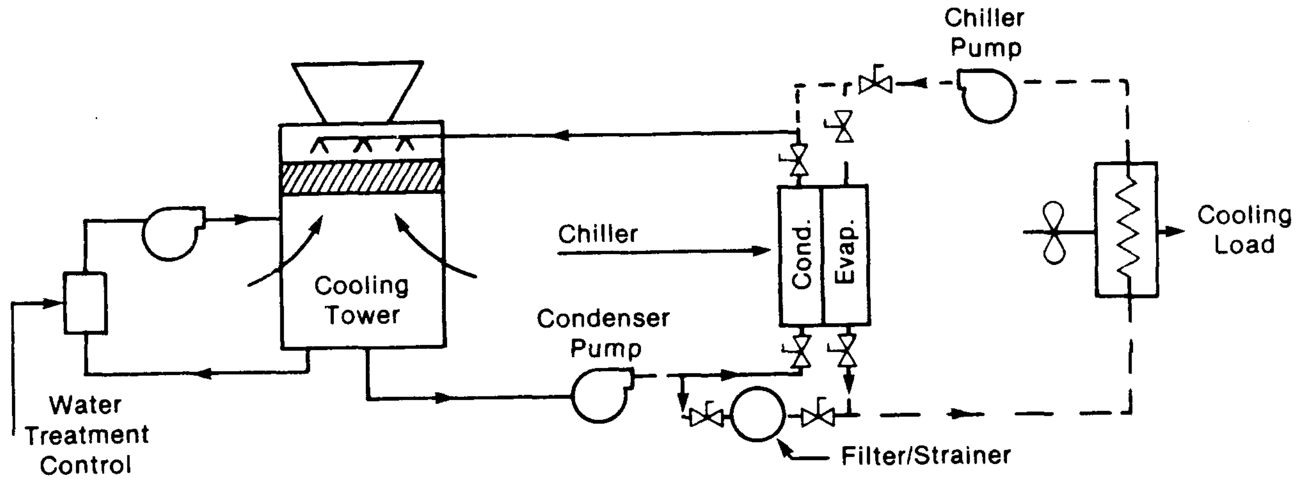

Fig. 10—Filtration or Strainer System

7.17 A fluid cooler, either an evaporative or dry type, may replace the conventional open cooling tower. This type of system is used in computer room winter cooling where the fluid (a mixture of glycol and water) is circulated through auxiliary coils located in the fan coil units.

8. ENERGY LOAD PROFILE

8.01 When considering the adaptability of an economy cycle, the heat gained or lost by a building or space compared to outdoor air dry- or wet-bulb temperatures must first be determined. The outdoor air economizer cycle with dry-bulb temperature control will be used as an example.

8.02 The first step is to graph heat gained or lost by the space versus outside air temperature (Fig. 12). The thermal conductance of the building must be determined at the outside design conditions for the building. Refer to Section 760-550-208* for design parameters. If, for example, the thermal conductance for a building is 90 MBH at 90°F outside

* Check Divisional Index 760 for availability.

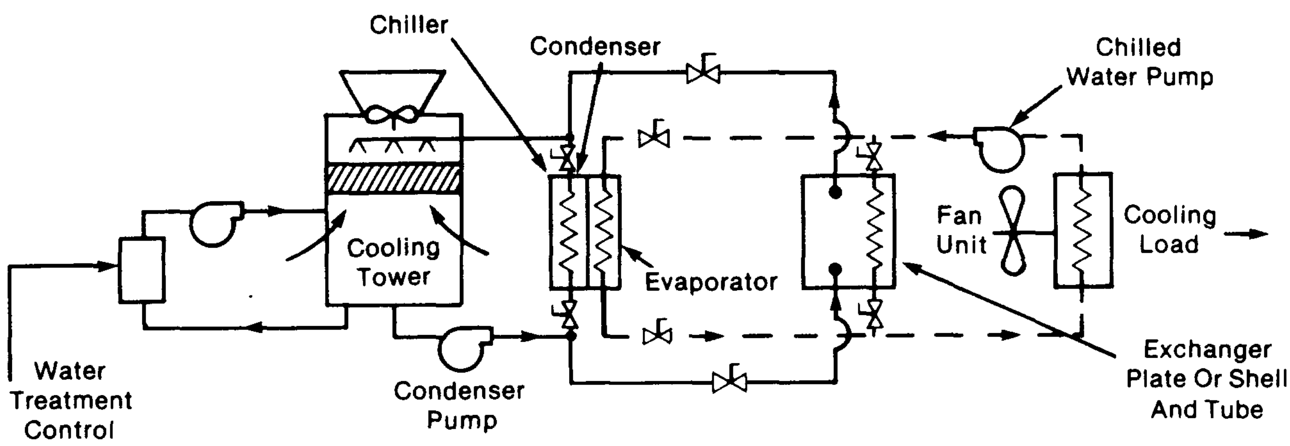

Fig. 11—Cooling Tower and Heat Exchanger

(point B, Fig. 12) and 80°F room temperature, the conductance of the building will be zero (point A, Fig. 12) when the outside temperature is equal to room temperature. The plot of these two points on the graph sets the slope for the building's energy.

8.03 The second step is to add the solar loading*, interior heat gain (people, lights, motors, etc), existing telephone equipment heat release, and the 5- and 10-year projected telephone equipment heat release.

8.04 Line 1 represents the total existing building heat load as it varies with outside air temperatures. Point C, where line 1 crosses the 0 line, represents the changeover from the cooling to the heating mode. Therefore, when outside air temperature is above 20°F, the structure must be cooled; if below 20° F, the structure must be heated.

* Solar loading should include time lag. During the heating period, it may be neglected. The ASHRAE Total Equivalent Temperature Differential Method determines conductance, solar loading, and time lag as one equation.

8.05 Lines 2 and 3 on the graph represent future loads as telephone equipment growth occurs within the building. Note that as point C moves to the left on the graph, cooling will be required for longer periods of time.

9. ANALYSIS

9.01 To determine whether an air economizer cycle can work in the example building, an additional item is required, ie, to plot the heat-removing ability of outside air at various outside temperatures (Fig. 13).

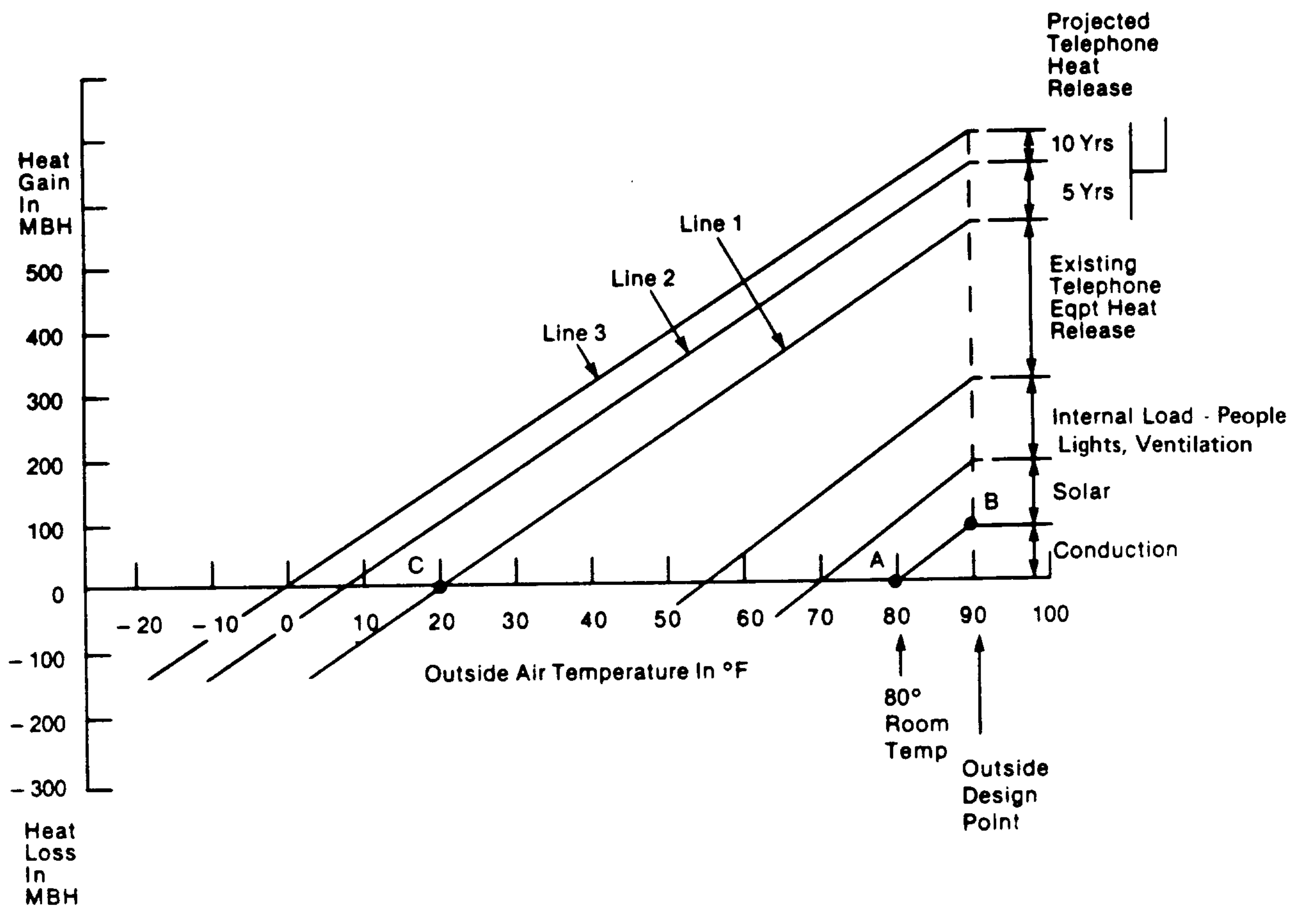

Fig. 12—Energy Load Profile Based on 80°F Room Temperature

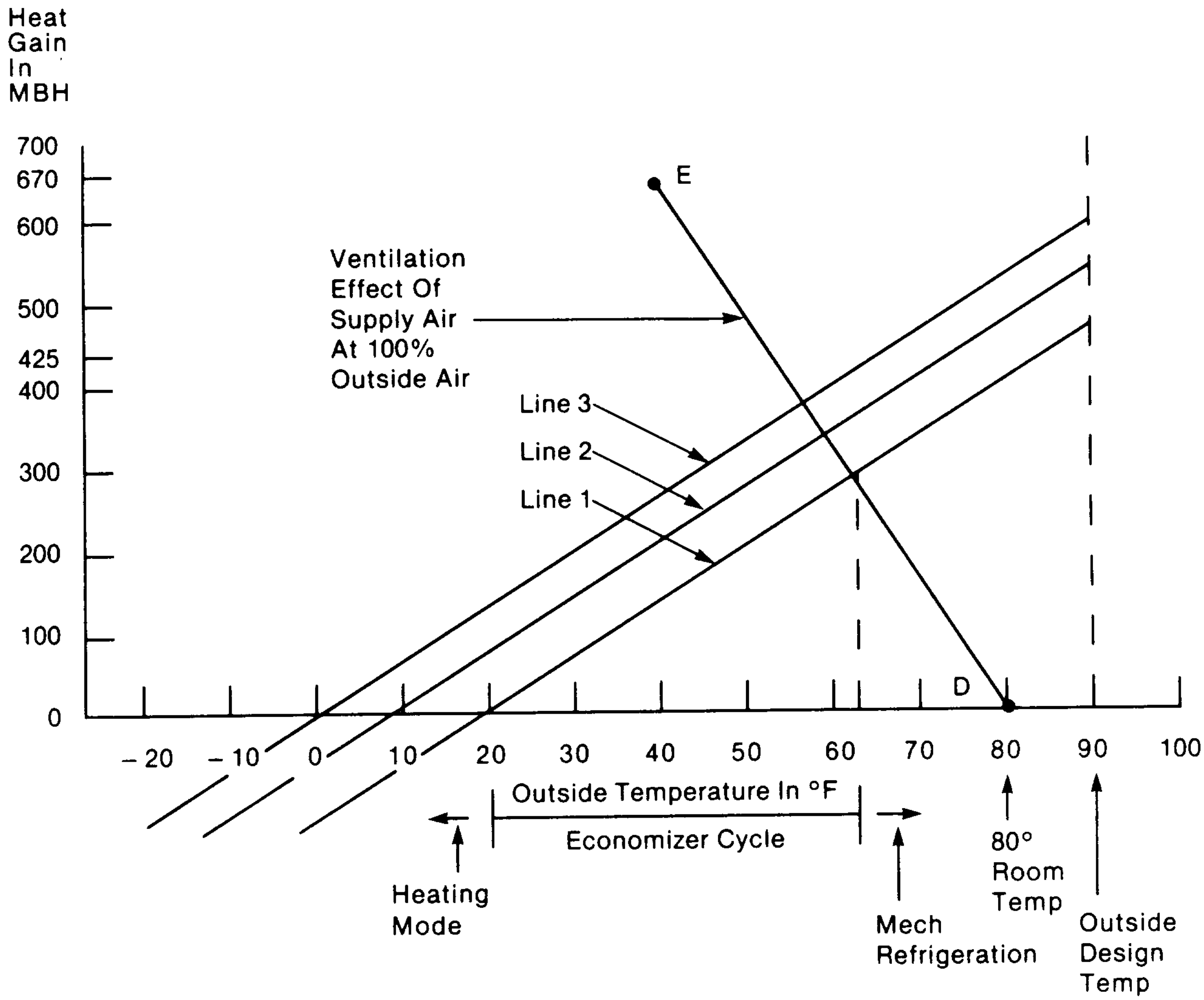

Fig. 13—Ventilation Effect of Supply Air

9.02 The first point on the graph (Fig. 13, point D) is the 0 heat-removal point that indicates the outside air temperature is the same as the room temperature. The second point is found through a simple calculation. The example building has a fan system operating at 15,500 cubic feet per minute (CFM). Using the following energy formula:

Heat quantity (BTUH) = CFM × ΔT°F × 1.08

and using any outside air temperature (ie, 40°F)

Heat quantity = 15,500 CFM × (80°F room temperature − 40) × 1.08

= 669,600 (British Thermal Units/Hour [BTUH]).

9.03 Now plot this point on the graph (Fig. 13, point E). The line D-E represents the heat removing ability of the ventilation system when using 100 percent outside air at any outdoor air temperature.

9.04 The total cooling requirements for this building can be met with outside air until the outside air dry-bulb temperature rises to 63°F (intersection of line D-E with line 1, Fig. 13). The outside air economizer would therefore operate 100 percent of the time between 20° and 63°F.

9.05 It is noted, however, that mechanical cooling may be required for the latent load (humidity). Cooling may then be handled by a high humidity override system that would close the outside air intake dampers and, upon further rise in relative humidity, start the mechanical refrigeration system to provide dehumidification.

9.06 Using an economizer system saves the cost of running the refrigeration system between 20° and 63°F ***assuming outside humidity is not a major problem.***

9.07 The total number of cooling hours required can be determined by using local weather data and tabulating the hours in temperature BINs. Table B shows a typical breakdown of 5°F BINs with corresponding annual hours of occurrence.

9.08 From Table B, the total annual hours that temperatures occur between 20° and 63°F (BINs D through L) are:

260 + 426 + 697 + 784 + 713 + 674 + 676 + 701 + 761 = 5692 hours.

TABLE B

HOURS WITHIN 5°F TEMPERATURE BINs

BIN	BIN RANGE OF	HOURS IN BIN
A	9°F	35
B	10/14	73
C	15/19	165
D	20/24	260
E	25/29	426
F	30/34	697
G	35/39	784
H	40/44	713
I	45/49	674
J	50/54	676
K	55/59	701
L	60/64	761
M	65/69	808
N	70/74	806
O	75/79	564
P	80/84	366
Q	85/89	190
R	90	51

9.09 The cooling load per temperature BIN can be calculated from Fig. 14 which shows an enlarged plot of line 1 from Fig. 13.

9.10 By knowing the total hours and heat gain per BIN, the total energy requirements for the temperature span can be determined. Table C shows the tabulated results for determining the grand total British Thermal Units × 1000 (MBTU).

9.11 Given the total cooling load for the temperature span (20° to 63°F), the cost of required mechanical refrigeration can be determined if the economizer system is not used. A refrigeration machine consumes energy at rates varying from 0.8 to 1.4 kW/ton. Assuming 1.2 kW/ton, then:

$$\frac{721{,}505\ \text{MBTU (Table C)}}{\frac{12\ \text{MBH}}{\text{ton}}} \times \frac{1.2\ \text{kW}}{\text{ton}} = 72{,}150\ \text{kWh}$$

If the electrical rate is 5 cents per kilowatt hour (kWh), then the maximum total savings are:

$$72{,}150\ \text{kWh} \times \$.05/\text{kWh} = \$3{,}607.50$$

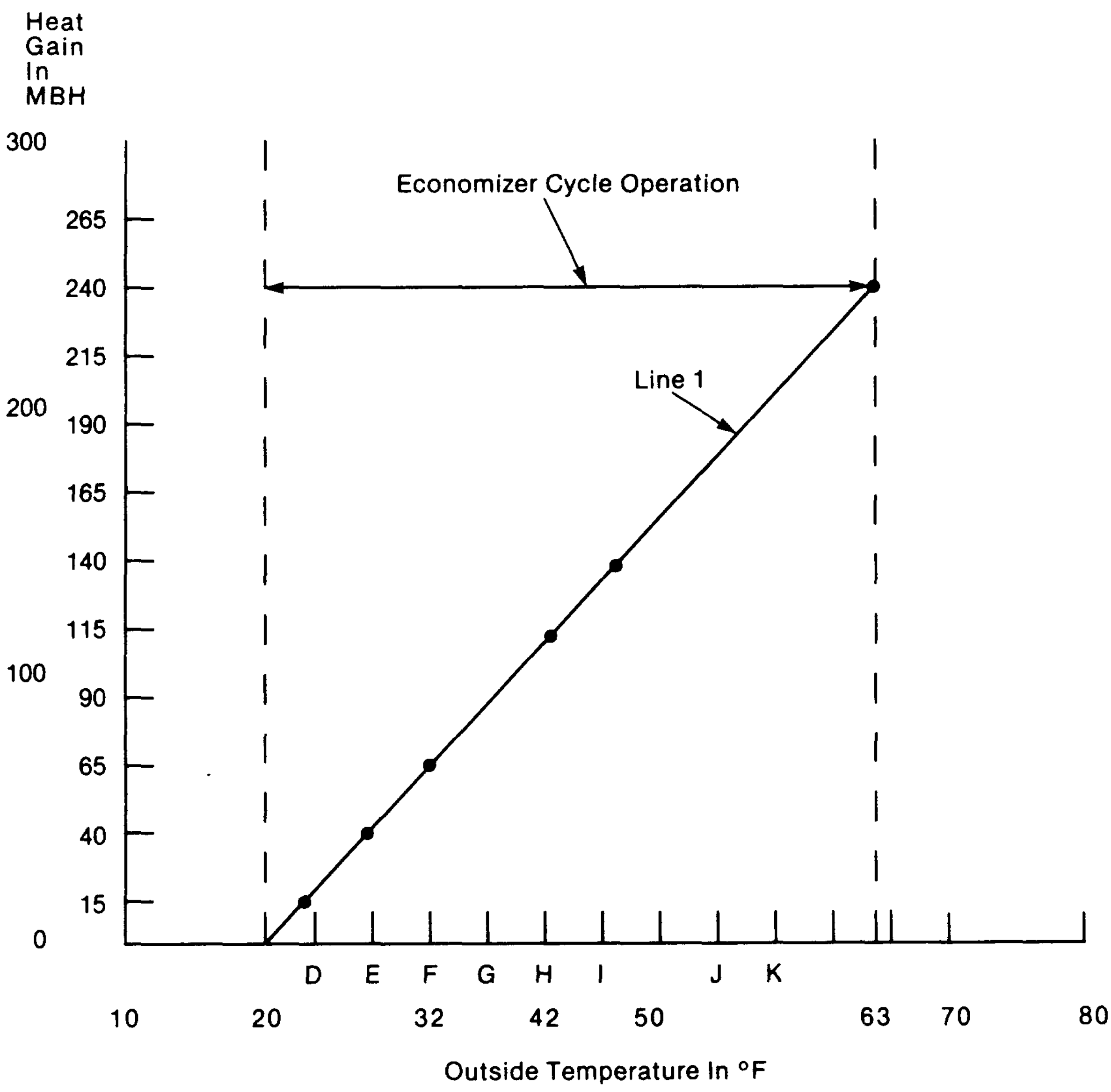

Fig. 14—Economizer Cycle—Energy Profile—MBH Heat Gain Per BIN

TABLE C

TABULATED RESULTS FOR DETERMINING TOTAL MBTU (BINs D THROUGH L)

I	II	III	IV	V	VI
TEMP RANGE	BIN MIDPOINT TEMP °F	BIN	HEAT GAIN (MBH) FROM FIG. 14	HOURS IN BIN	TOTAL MBTUS III × IV
20-24	22	D	15	260	3,900
25-29	27	E	40	426	17,040
30-34	32	F	65	697	45,305
35-39	37	G	90	784	70,560
40-44	42	H	115	713	81,995
45-49	47	I	140	674	94,360
50-54	52	J	165	676	111,540
55-59	57	K	190	701	133,190
60-64	62	L	215	761	163,615
			Totals	5,692	721,505

Comparing this with the total load for the entire cooling season without the economizer cycle (Fig. 15 and Table D) yields the following:

Total MBTU = 1,720,330

$$\frac{1{,}720{,}330 \text{ MBTU}}{\frac{12 \text{ MBH}}{\text{ton}}} \times \frac{1.2 \text{ kW}}{\text{ton}} = 172{,}033 \text{ kWh}$$

172,033 kWh × \$0.05/kWh = \$8,601.65

9.12 The use of an outside air economy cycle for this example results in an annual savings of:

$$41.9\% \text{ savings} = \frac{3{,}607.50}{8{,}601.65} \times 100$$

9.13 If it is determined that the outside air change-over control point for cooling is 57°F instead of 63°F because of humidity problems, the savings will be reduced to 32 percent.

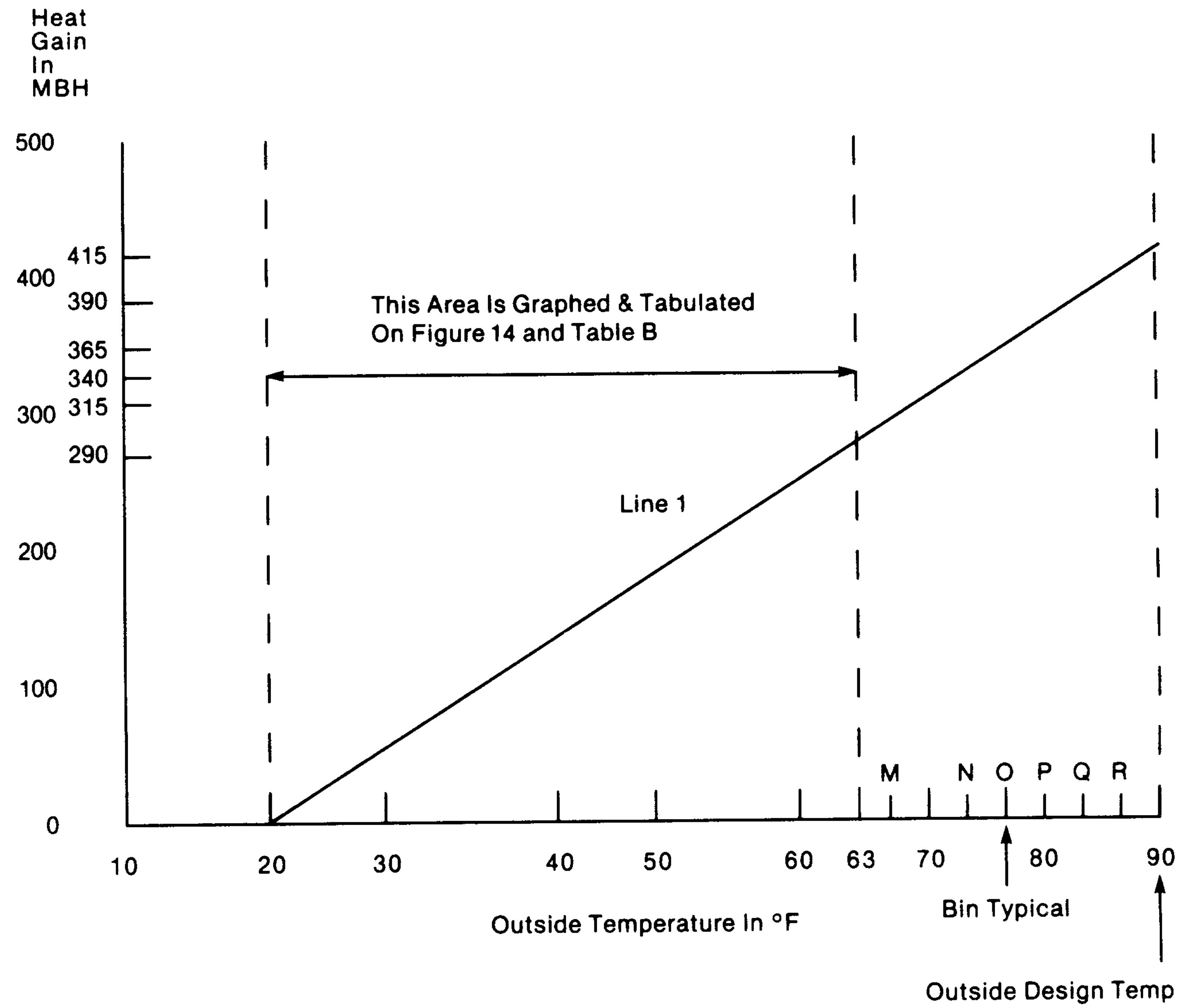

Fig. 15—Total Cooling Season—Energy Profile—MBH Heat Gain Per BIN

TABLE D

TABULATED RESULTS FOR DETERMINING GRAND TOTAL MBTUs

I	II	III	IV	V	VI
TEMP RANGE	BIN MIDPOINT TEMP °F	BIN	HEAT GAIN (MBH) FROM FIG. 9	HOURS IN BIN	TOTAL MBTUS III × IV
65-69	62	M	290	808	234,320
70-74	72	N	315	806	353,890
75-79	77	O	340	564	191,760
80-84	82	P	365	366	133,590
85-89	87	Q	390	190	74,100
90-	92	R	415	51	21,165
			Subtotals	2,785	1,008,825
			Subtotals from Table C	5,692	721,505
			Grand Totals	8,477	1,720,330