BUILDING ENERGY MANAGEMENT AND REDESIGN RETROFIT (BEMARR)

HEAT RECOVERY

CONTENTS PAGE

1. GENERAL

1.01 Recovering some of the heat from buildings that would otherwise be rejected into the exhaust air from a building helps reduce energy costs. Heat recovery devices and equipment generally require higher capital investment than conventional Heating, Ventilating, and Air-Conditioning (HVAC) equipment; therefore, they should be carefully analyzed for cost effectiveness. This section provides information which will aid in the analysis of heat recovery systems. The 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.

1.03 ***Abbreviations and Acronyms:*** Refer to Table A for a list of abbreviations and acronyms used in this section.

TABLE A

ABBREVIATIONS AND ACRONYMS

ABBREVIATION	TERM
AHU	Air-Handling Unit
Avg	Average
BOC	Bell Operating Companies
BTU	British Thermal Unit
BTU/hr	British Thermal Unit Per Hour
C	Pressure Contactor
CFM	Cubic Feet Per Minute
Cu ft	Cubic Feet
dB	Decibel
Equip	Equipment
F	Fahrenheit
fpm	Feet Per Minute
gal	Gallon
gpm	Gallons Per Minute
hr	Hour
HVAC	Heating, Ventilating, Air Conditioning
lb	Pound
MBH	BTU Per Hour × 1000
MBTU	BTU × 1000
Min	Minute
NASA	National Aeronautics and Space Administration
NC	Normally Closed
NO	Normally Open
Req	Required
Supp	Supplemental
T	Temperature
Tel	Telephone
Typ	Typical

2. SCOPE

2.01 The possible energy savings that may be achieved through the use of heat recovery devices and systems are discussed in this section. Heat recovery can be accomplished between any two fluids such as air and liquid. The transfer of heat can take place from one fluid to another by convection or conduction.

2.02 Heat recovery systems can be as simple as supplying air from a hot space to a space requiring heat (ie, exhausting treated building air through mechanical equipment or elevator machine rooms). The more complex heat recovery systems use refrigeration equipment to elevate the temperature of the recovered heat before distribution.

2.03 It should be noted that even the most efficient heat recovery systems and equipment cannot recover all of the waste heat that may be available. Existing equipment should be fine-tuned before investments in recovery equipment are made. Further, large amounts of waste heat (ie, high boiler exhaust stack temperatures) can sometimes be an indicator of serious defects or problems in existing systems or equipment.

2.04 The basic categories of heat recovery devices are systems which transfer heat from:

(a) ***Air-to-Air Exchangers:*** These devices include ventilation air recovery systems and recuperators. Recuperators contain an array of tubes in which the waste heat of boiler exhaust gases is used to preheat incoming combustion air. These devices are not economical for systems that produce less than 50,000 pounds of steam per hour. Ventilation air recovery systems work by circulating warm outgoing air through an exchanger that transfers heat to the incoming supply air. Typical of these devices are heat wheels and heat pipes.

(b) ***Air-to-Liquid Exchangers:*** These devices are used where the source of heat is in a gaseous form, and the heat to be used is desired in the liquid form; included are economizers and heat recovery boilers. Economizers are similar to recuperators in that high temperature, boiler flue gases are utilized in the case of an economizer to preheat boiler makeup water.

(c) ***Liquid-to-Liquid Exchangers:*** These devices are used where the source of heat is available in liquid form and is required to be transferred to another liquid. Included in this category are those devices which operate directly through temperature differentials of the liquids and those that operate on the heat pump principle.

3. HEAT RECOVERY ANALYSIS

3.01 In a multifunction building with low- and high-internal heat areas, it is possible by means of heat recovery to balance the heating requirements in low internal heat areas with the reclaimed heat from high-internal heat areas.

3.02 The first step in the analysis to determine feasibility of heat recovery is to develop an energy profile for the building, a floor, or a section of the space. Refer to Section 760-550-210* for details on developing an energy profile.

* Check Divisional Index 760 for availability.

3.03 Figure 1 represents the energy profile for high- and low-heat gain areas. On the graph, the area above the datum line represents heat that must be removed from the space, and the area below the datum line represents heat that must be added. Note that in this example the changeover from heating to cooling for the telephone equipment space occurs at −10°F, while the changeover from heating to cooling for the office space occurs at 40°F.

3.04 The equipment space (Fig. 1) is maintained at 80°F inside temperature until the outside temperature reaches 0°F. Then, changeover is made to maintain 65°F inside, and heating starts to be required at −10°F outside.

3.05 The office space is maintained at 78°F until the outside temperature reaches 50°F. Heating is required at 40°F outside temperature to maintain an inside temperature of 65°F. Area A-B-C of Fig. 1 represents the amount of energy that could be

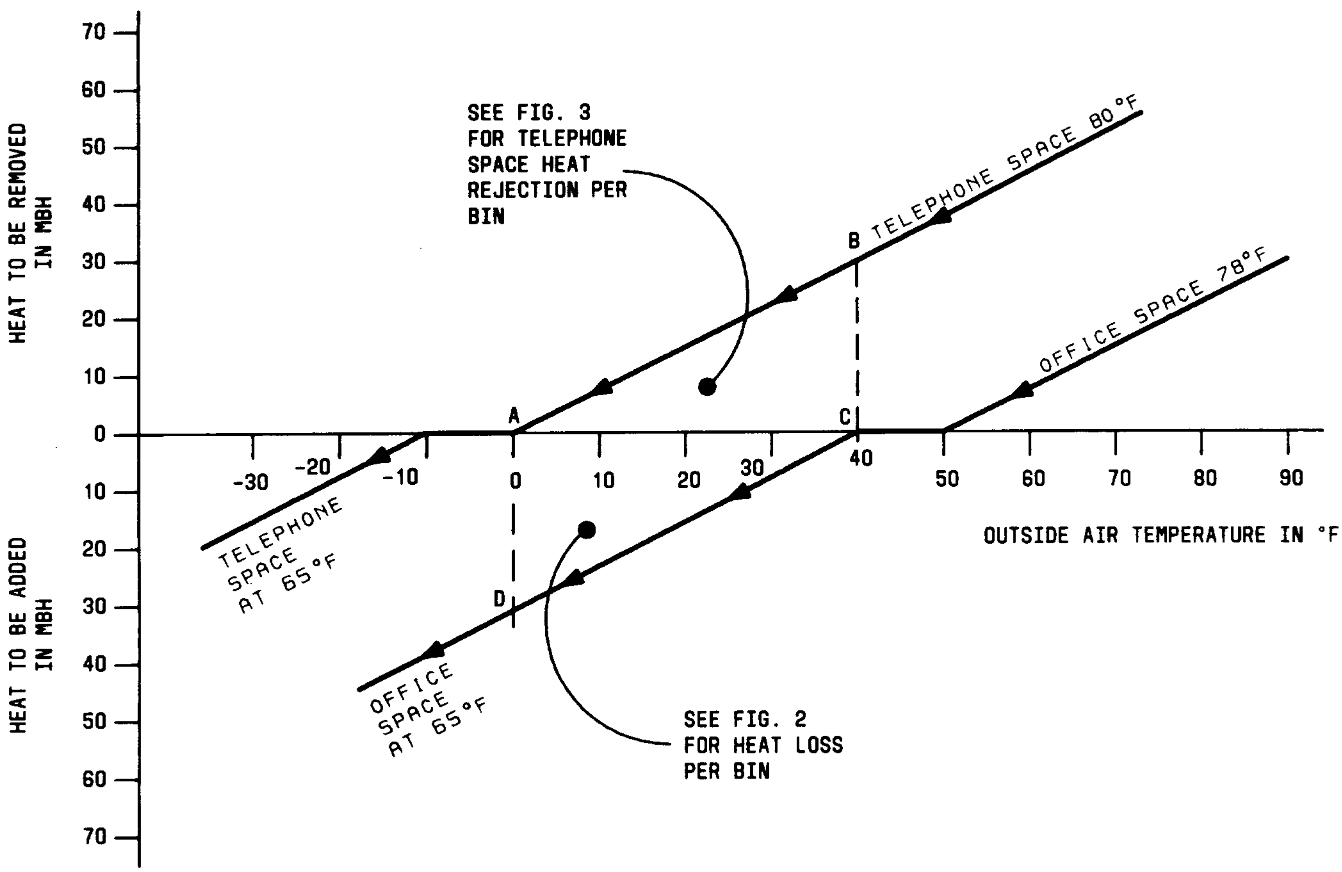

Fig. 1—Energy Load Profiles of Telephone and Office Spaces

reclaimed from the equipment space. Area A-D-C represents the amount of energy required to heat the office area.

3.06 To find the actual British Thermal Units (BTU) at 100 percent recovery:

(a) Determine the number of hours per temperature BIN. These hours vary according to geographic locations and are calculated and charted by the National Weather Bureau. Table B shows typical 4°F temperature BINs.

TABLE B

TYPICAL 4°F TEMPERATURE BINs

BIN	BIN RANGE	HOURS IN BIN
A	0/9	70
B	10/13	67
C	14/17	214
D	18/21	316
E	22/25	305
F	26/29	430
G	30/33	565
H	34/37	577
I	38/41	530

(b) From the graph (Fig. 2), determine HEAT LOSS in thousands of BTUs per hour (MBH) per each BIN range for heat to be added to the office space.

(c) From the graph (Fig. 3), determine HEAT GAIN in MBH available for recovery from the telephone equipment space in each temperature BIN range.

(d) Table C tabulates the total heating requirements for the office space and the heat rejec-

3.07 Once a recovery system has been chosen and its thermal efficiency established through design calculations or manufacturer's data, the total energy savings can be determined. For example, three different efficiencies were chosen arbitrarily: 20, 45, and 70 percent. The information taken from Table C was arranged into a new table as shown in Table D. For example, if the heat reclaim equipment is 20 percent efficient and 20 percent of the heat rejected (usable heat) from the telephone equipment is recovered, 9,600 MBH can be used to offset the 26,007 MBH grand total office heating loss between 0° and 40°F outside temperature. The resulting savings would equal:

$$\frac{\text{9,600 MBH Reclaimed}}{\text{26,007 MBH Grand Total Heat Loss}} \times 100 = 37\% \text{ Energy Savings}$$

> ***Note:*** Usable heat is heat from the equipment space which can be used to offset the heat loss from the office space. It cannot exceed 100 percent of the heat loss per temperature BIN.

3.08 For comparison, if it were possible to reclaim and transfer 100 percent of the usable heat rejected from the telephone equipment space to the office space to offset the grand total heat loss (Table C), the theoretical energy savings would equal:

$$\frac{\text{21,712 MBH Usable Reclaimed Heat}}{\text{26,007 MBH Grand Total Heat Loss}} \times 100 = 84\% \text{ Energy Savings}$$

With the 20 percent efficient reclaim system, the 9600 MBH reclaimed figure (Table D) yields:

$$\frac{\text{9,600 MBH Reclaimed Heat}}{\text{21,712 MBH Usable Heat Rejected}} \times 100 = 44\% \text{ of the available heat energy is reclaimed and used.}$$

3.09 Note that even though the system reclaims only 20 percent of the usable heat that is rejected, it will offset the energy consumption by 37 percent in utilizing 44 percent of the heat rejected in the 0° to 40°F temperature range. The same procedure can be applied to the 45- and 70-percent systems

Fig. 1—Energy Load Profiles of Telephone and Office Spaces

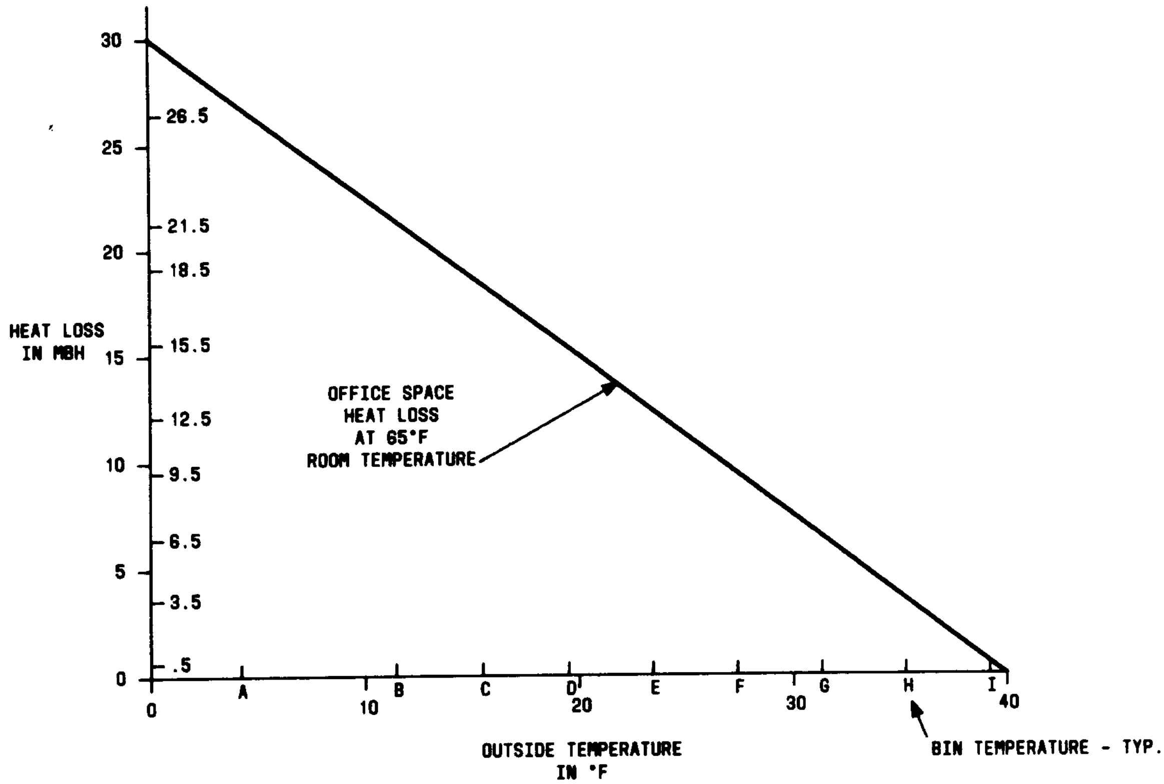

Fig. 2—Office Space Heat Loss Graph

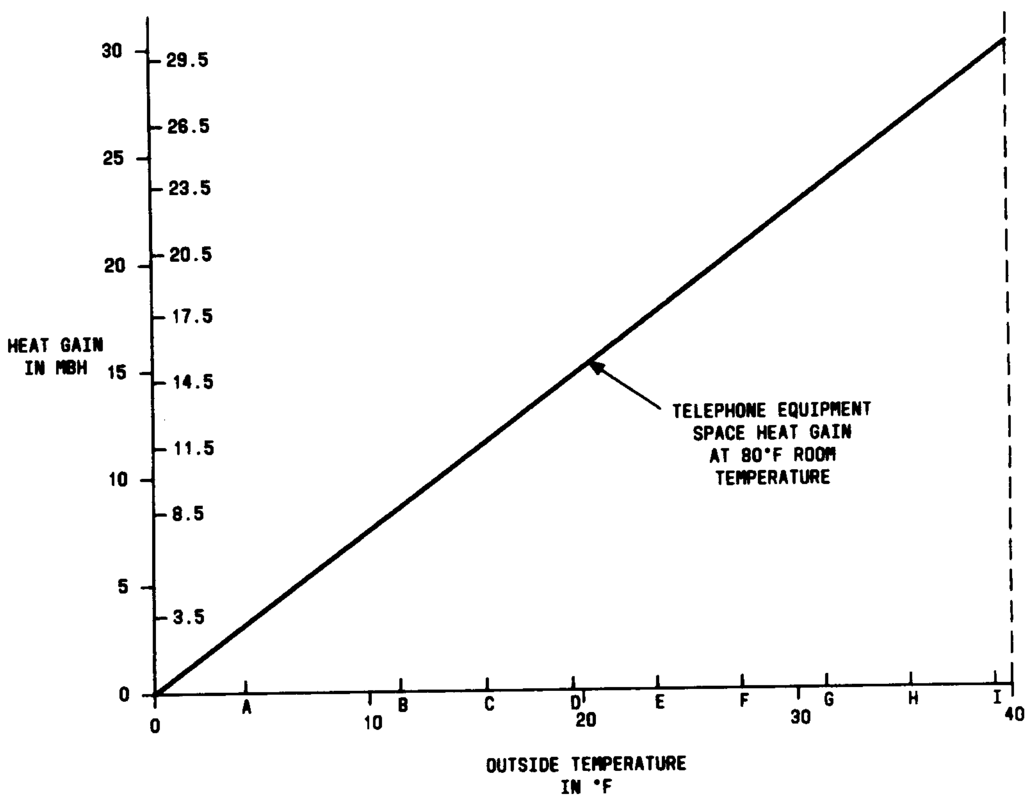

Fig. 3—Telephone Equipment Space Heat Rejection Graph

TABLE C

GRAND TOTAL HEAT REQUIREMENTS AND HEAT REJECTION (NOTE 1)

I	II	III	IV	V	VI	VII	VIII
BIN	HOURS IN BIN	OFFICE HEAT LOSS MBH (FIG. 2)	TELEPHONE HEAT REJECTION MBH FIG. 3)	TOTAL OFFICE LOSS MBTU (II × III)	TOTAL TELEPHONE REJECTION MBTU (II × IV)	TOTAL SUPPLEMENTAL HEAT REQUIRED MBTU (NOTE 2) (V - VI)	TOTAL SURPLUS HEAT MBTU (NOTE 2) (VI - V)
A	70	26.5	3.5	1,855.0	245.0	1,610	—
B	67	21.5	8.5	1,440.5	569.5	871	—
C	214	18.5	11.5	3,959.0	2,461.0	1,498	—
D	316	15.5	14.5	4,898.0	4,582.0	316	—
E	305	12.5	17.5	3,812.5	5,337.5	—	1,525
F	430	9.5	20.5	4,085.0	10,105.0	—	6,020
G	565	6.5	23.5	3,672.5	13,277.5	—	9,605
H	577	3.5	26.5	2,019.5	15,290.5	—	13,271
I	530	0.5	29.5	265.0	15,635.0	—	15,370

NOTES:

1: Grand Total Heat Loss in Office Space — 26,007 MBTU
Grand Total Heat Rejection in Telephone Space — 67,503 MBTU
Grand Total Usable Heat Rejection — 21,712 MBTU
Grand Total Supplemental Heat — 4,295 MBTU.

2: Based on 100% Reclaim.

TABLE D

HEAT RECLAIM EFFICIENCY

BIN	OFFICE HEAT LOSS MBTU	TEL EQUIP HEAT REJECT MBTU	20% RECLAIM (USABLE) MBTU	SUPP HEAT MBTU	45% RECLAIM (USABLE) MBTU	SUPP HEAT MBTU	70% RECLAIM (USABLE) MBTU	SUPP HEAT MBTU
A	1,855.0	245.0	49.0	1,806.6	110.3	1,744.8	171.5	1,683.5
B	1,440.5	569.5	113.9	1,326.6	256.3	1,184.2	398.7	1,041.9
C	3,959.0	2,461.0	492.2	3,466.8	1,107.5	2,851.5	1,722.7	2,236.3
D	4,898.0	4,582.0	916.4	3,981.6	2,061.9	2,836.1	3,207.4	1,690.6
E	3,812.5	5,337.4	1,067.5	2,745.0	2,401.9	1,410.6	3,736.3	76.3
F	4,085.0	10,105.0	2,021.0	2,064.0	4,085.0	—	4,085.0	—
G	3,672.5	13,277.5	265.5	1,017.0	3,672.5	—	3,672.5	—
H	2,019.5	15,290.5	2,019.5	—	2,019.5	—	2,019.5	—
I	265.0	15,635.0	265.0	—	265.0	—	265.0	—
Totals	26,007.0	67,503.0	9,600.0	16,407.0	15,979.9	10,027.3	19,278.6	6,728.6

TABLE E

HEAT REJECTION AND ENERGY CONSUMPTION

RECOVERY SYSTEM EFFICIENCY (%)	TOTAL HEAT LOSS OFFSET (%)	AVAILABLE HEAT ENERGY USED (%)
20	37	44
45	61	74
70	74	89

3.10 The calculations for percent of total heat loss offset and of available energy used depend on the amount of heat rejected by telephone equipment versus the heat required for the office space. In addition, consideration should also be given to the removal of the heat from the high-heat area that is in excess of the low-heat area requirements. The process provides the input data necessary for Life Cycle Analysis by showing accurate heat requirements and heat available for recovery and reuse. This procedure can be applied to each article in this section as an aid in economic analysis for each type of recovery system.

4. HEAT RECOVERY AND TRANSFER THROUGH THE USE OF THE CHILLED-WATER CIRCULATING SYSTEM

4.01 Figure 4 represents a typical chilled-water circuit handling one high- and one low-internal heat source area. The basic principle of using the chilled-water system to transfer heat necessitates a temperature difference between the spaces. If all spaces have the same room temperature, no heat transfer will take place. The load profile, as described in Part 3 of this section, indicates that the temperature in the telephone equipment space will be 80°F while the office space will be 65°F during the office space heating cycle when outside temperatures are greater than 0°F but less than 40°F.

4.02 The basic operation of the heat recovery and transfer system is to operate the fan coil units and pumps to continuously circulate the chilled water through the cooling coil. The heat from the high-internal heat area heats the chilled water as it flows through the cooling coil. This heat is then released to the area by the water flowing through the associated coil.

4.03 The system can be modified to include a chiller bypass which increases the system efficiency during the heating cycle by reducing the heat loss to the chiller mass and reducing the head on the chilled-water pump. It should be noted that the cooling coils in both the low- and high-heat gain areas should be 100 percent open.

4.04 New and existing coils should be checked for capacities if they are to be used in this manner. All factors affecting their capacities should be considered (ie, the water flow rates, water temperatures, and airflow rates). The coil manufacturer may have a computer program to determine or verify local calculations of the actual amount of transferable heat available in the systems involved.

4.05 The temperature controls for the system may require modification to accomplish the functions described. To modulate water into coils which require heat, other modifications will be needed. The control system should be reviewed to ensure proper operation for the spaces involved.

5. THE USE OF EXCHANGED RETURN AIR AS AN ENERGY SAVER

5.01 In many telephone equipment buildings, it is possible to swap a portion of the return air from a high, internally heated space to help heat a space with a low-internal heat source. This technique is very simple, and the savings will depend on the amount of air that can be exchanged.

5.02 When feasible, the use of direct heat exchange is more efficient than using an intermediate medium such as water.

5.03 The tables and graphs in Part 3 of this section are used for a case study. An idealized setup is shown in Fig. 5. When air swapping is possible, damper 4 closes and damper 3 opens so that all the air is taken from the "A" side rather than the "B" side. Conversely, damper 2 opens to allow airflow equal to the amount borrowed by air-handling unit (AHU) 2 and damper 1 closes to decrease the return air from "A" space by the amount borrowed.

5.04 In Fig. 5, AHU 2 can use up to 1930 cubic feet per minute (CFM) of air at 80°F from the "A" side, while AHU 1 uses 1930 CFM of air at 65°F from the "B" side. The system can recover, per hour, the following amount of heat:

$$\text{Recoverable Heat} = \text{CFM} \times (\text{"A" Room Temp} - \text{"B" Room Temp}) \times 1.08$$

$$= 1{,}930\ \text{CFM} \times (80°\text{F} - 65°\text{F}) \times 1.08\ \frac{\text{BTU/Min}}{\text{hr cu ft °F}}$$

$$= 31{,}266\ \text{BTU/hr or } 31.3\ \text{MBH}$$

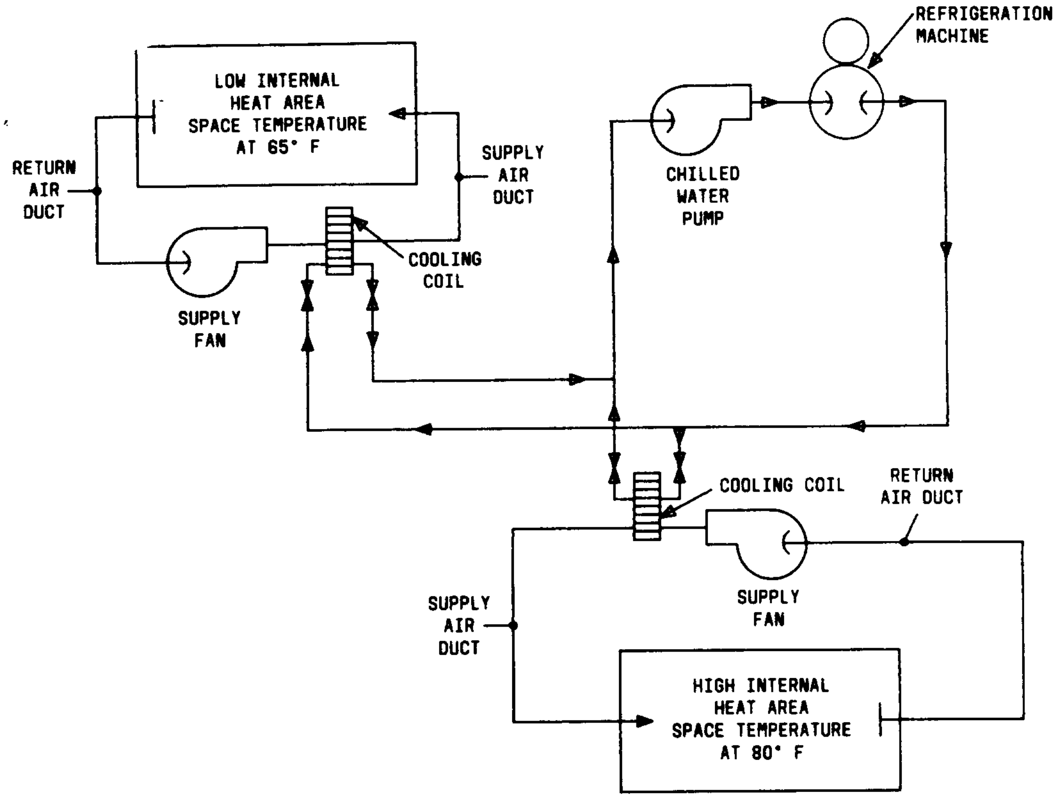

Fig. 4—Typical Chilled-Water Loop

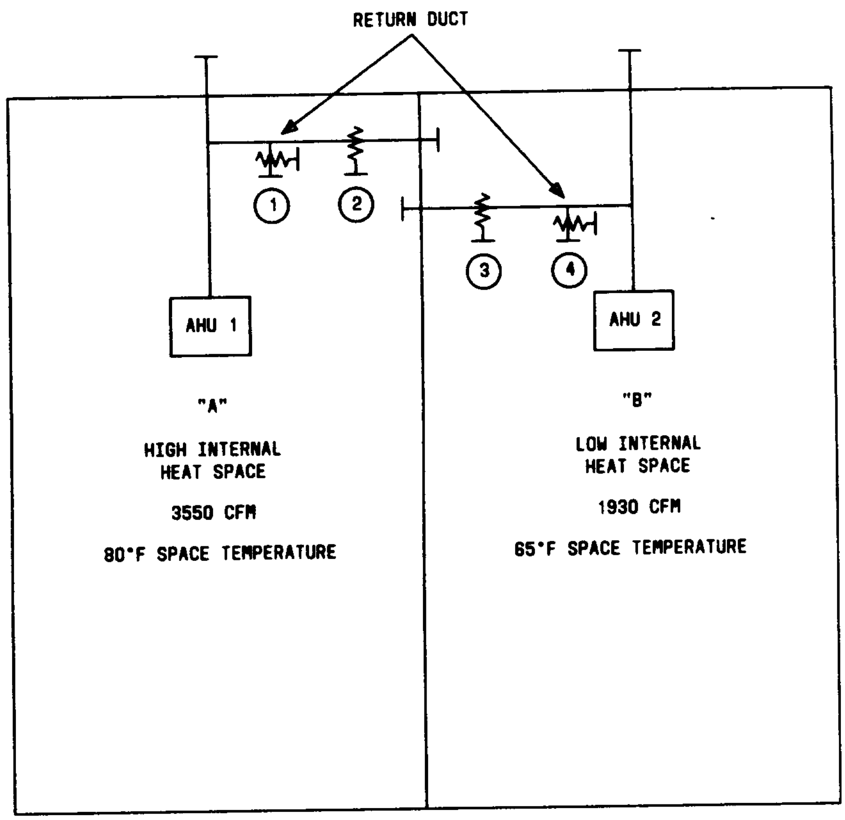

Fig. 5—Return Air Swapping for Energy Conservation

5.05 Using the temperature BIN method, Table F shows:

(a) The return air can transfer all the heat rejected by the telephone equipment (31.3 MBH capacity versus 29.5 MBH rejected in BIN I, column IV).

(b) The total heat loss of the office space (or low-heat release space) is 26,007 MBTU (column V), and the total usable reclaimed heat from the high-heat release area is 21,712 MBTU (column VII) or:

$$\frac{\text{Reclaimable Heat}}{\text{Heat Loss}} = \text{Portion of Reclaimed Heat}$$

$$\frac{21{,}712\ \text{MBTU}}{26{,}007\ \text{MBTU}} = $$ 0.83 or 83 percent of the total heat required can be reclaimed by swapping all or a portion of that amount.

TABLE F

HEAT AVAILABLE AND HEAT REQUIRED

I	II	III	IV	V	VI	VII	VIII
BIN	HOURS IN BIN	OFFICE HEAT LOSS MBH (FIG. 3)	TELEPHONE HEAT REJECTION MBH (FIG. 4)	TOTAL OFFICE LOSS MBTU (II × III)	TOTAL TELEPHONE REJECTION MBTU (II × IV)	TOTAL RECOVERED HEAT MBTU	TOTAL SUPP HEAT REQD MBTU (V - VII)
A	70	26.5	3.5	1,855.0	245.0	245.0	1,610
B	67	21.5	8.5	1,440.5	569.5	569.5	871
C	214	18.5	11.5	3,959.0	2,461.0	2,461.0	1,498
D	316	15.5	14.5	4,898.0	4,582.0	4,582.0	316
E	305	12.5	17.5	3,812.5	5,337.5	*3,812.5	—
F	430	9.5	20.5	4,085.0	10,105.0	*4,085.0	—
G	565	6.5	23.5	3,672.5	13,277.5	*3,672.5	—
H	577	3.5	26.5	2,019.5	15,290.5	*2,019.5	—
I	530	0.5	29.5	265.0	15,635.0	265	—
Totals		—	—	26,007.0	67,503.0	21,712.0	4,295

* That portion of the heat available from the telephone equipment that can be utilized in the office space.

5.06 Table G shows the amount of return air to be taken from the high-heat release area for each BIN. The 3.5 MBH of heat is available in BIN A. To transfer this heat, 216 CFM must be taken from side "A":

$$\text{CFM} = \frac{\text{Heat to be Transferred}}{(80°\text{F} - 65°\text{F}) \times 1.08}$$

$$216\ \text{CFM} = \frac{3500\ \text{BTU}}{15°\text{F} \times 1.08}$$

In BIN G, while there is 23.5 MBTU available, the office area can only use 6.5 MBTUs. The required quantity of air is now:

$$\text{CFM} = \frac{6500}{(80°\text{F} - 65°\text{F}) \times 1.08}$$

Quantity = 401 CFM

TABLE G

RETURN AIR QUANTITY REQUIRED FOR HEAT TRANSFER

BIN	CFM
A	216
B	525
C	710
D	895
E	772
F	586
G	401
H	216
I	31

5.07 The control of the system may be an outside temperature reset on the four dampers concerned. As the temperature of the outside air decreases from 40° (BIN I) to 20°F (BIN D), the amount of swapped return air increases. As the outside temperature decreases from 20° to 0°F (BIN A), the amount of swapped air decreases.

5.08 An outside thermostat can be arranged to index the dampers so that space "B" uses the tabled CFM at each BIN temperature range. Below BIN A, both areas require supplemental heating; above BIN I, both areas require cooling.

6. HEAT RECOVERY FROM EXHAUST AIR

A. General

6.01 Many buildings use a large proportion of outside air for ventilation. This is to satisfy codes. Waste energy from exhausted air may be recovered and used to heat or cool the makeup air or to preheat water. In an office building where the amount of outside air required is high, recovery of energy may be economically practical using such systems as described herein.

B. Heat Recovery Wheel

6.02 Heat recovery wheels (Fig. 6) are rotary air-to-air heat exchangers of which there are two types:

(a) The metallic wheel constructed of knitted aluminum or stainless steel heat transfer media which transfers sensible heat from one air stream to another with virtually no moisture transfer

(b) The desiccant wheel that has corrugated air passages composed of silicate-backed asbestos impregnated with lithium chloride. The desiccant wheel can transfer both sensible and latent heat between air streams. The wheel is constructed to minimize carryover or cross-contamination between air streams. Although particle carryover is minimal, the use of the desiccant-type wheel is ***not recommended*** due to the use of asbestos material.

6.03 The heat recovery wheel rotates slowly in a casing between the outside air intake duct and the exhaust air duct. As it rotates through the counterflowing air streams, the wheel absorbs energy from the higher energy air stream and transfers it to the lower energy one.

6.04 The efficiency of heat wheels varies from 60 to 90 percent depending upon the face velocity through the unit. Some units have 90 percent efficiency at 200 feet per minute (fpm), but only 70 percent at 800 fpm. The lower the velocity, the larger the size of the unit. The pressure drop across these units is approximately 1 inch of water.

6.05 Retrofit applications of heat recovery wheels are limited because of their typically large size and because supply and exhaust duct work must be in close proximity to one another.

C. Coil Loop Runaround

6.06 Coil loop runaround is a device used to recover waste energy from the exhaust air and supply it to the makeup or incoming fresh air. In a typical layout (Fig. 7), two finned tube coils are linked together by a loop of pipe. One coil is placed in the exhaust air stream, and the other coil is placed in the makeup air stream. A pump continuously circulates a fluid such as a mixture of glycol and water (to prevent freezing) between the two air streams.

6.07 In winter, the exhaust air releases its heat at the exhaust coil. The fluid transfers this energy to heat the incoming air.

6.08 The system is seasonally reversible, thereby helping to preheat air when the outdoor air is cooler than the exhaust air and helping to precool air when the outdoor air is warmer than the exhaust air.

6.09 Coils and pumps are selected to achieve sensible heat recovery efficiencies of 40 to 60 percent. Greater efficiencies may be achieved by adding additional coils in the heat exchangers to increase their capacity. This increases the pressure drop across the coils which increases the fan horsepower required. The additional coils will increase the pump energy required.

6.10 The runaround coil has flexibility for retrofit application since the supply and exhaust air ducts can be remote from each other.

D. Efficiency of Air-to-Air Heat Exchanger

6.11 The efficiency of an air-to-air heat exchanger is:

$$\text{Efficiency} = \frac{\text{Emergency Recovered From Exhaust}}{\text{Maximum Possible Heat Recovery}}$$

$$\text{Percent} = \frac{T_3 - T_4}{T_3 - T_1}$$

Where:

T_1 = Temperature of supply air entering

T_3 = Temperature of exhaust air entering

T_4 = Temperature of exhaust air leaving.

E. Heat Pipe

6.12 The next system is the heat pipe (Fig. 8). The basic principles of the heat pipe were developed and patented about 30 years ago. This recovery unit has no moving parts. It appears to be a dehumidification coil with a partition separating the face into two sections. The unit is made of an array of extended surface tubes sealed at both ends. These tubes are the actual heat pipes. Each heat pipe consists of a tube, wick, and a working fluid which undergoes a phase change between the temperature range of the two airstreams. Refrigerant R-12 is commonly used.

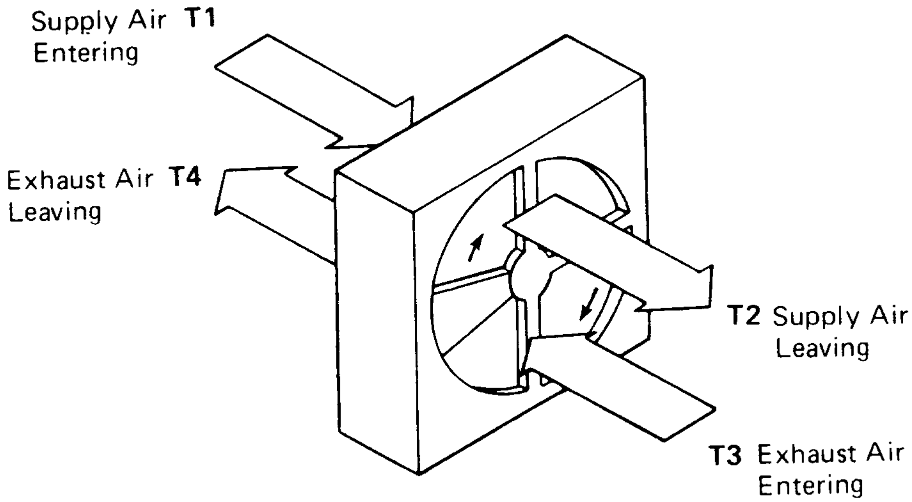

Fig. 6—Heat Recovery Wheel

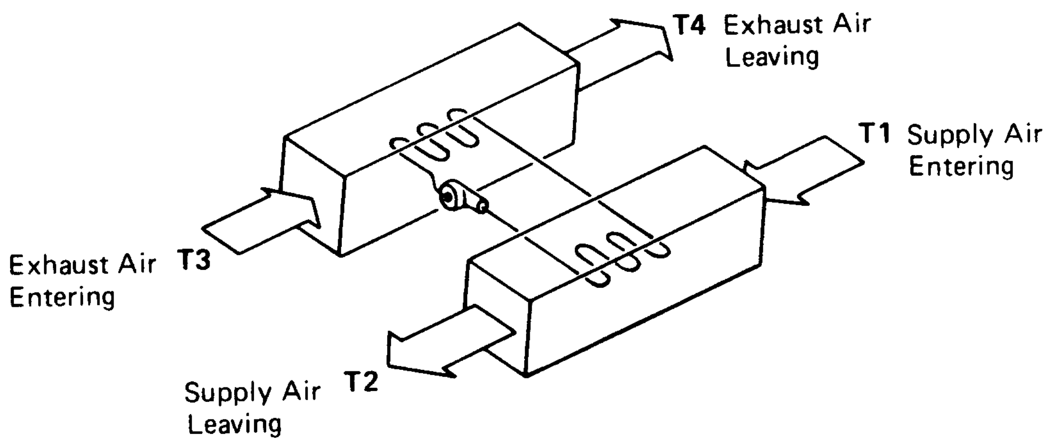

Fig. 7—Coil Loop Runaround

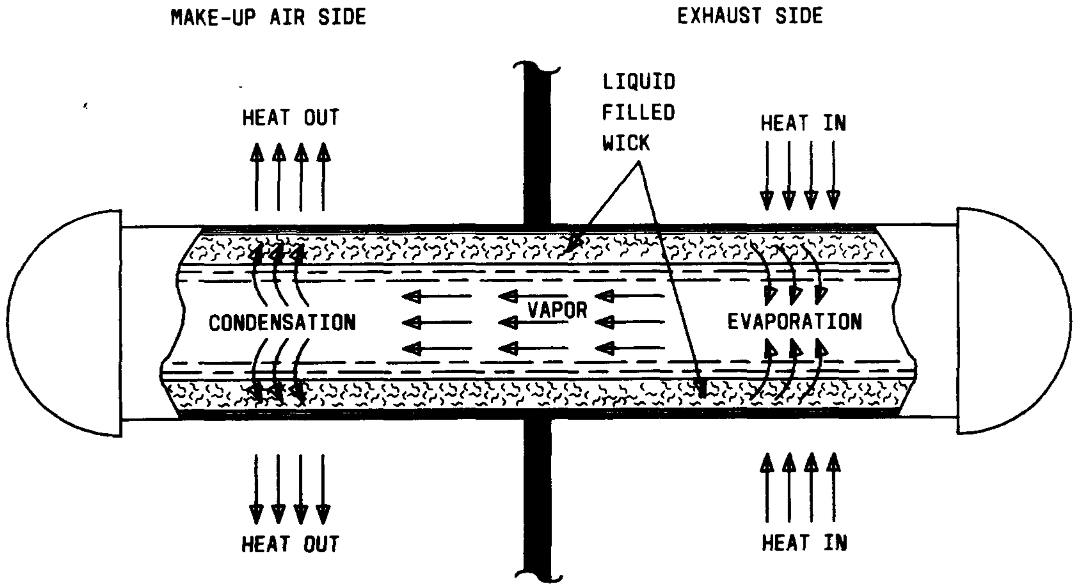

Fig. 8—Heat Pipe Air-to-Air Heat Exchanger

6.13 When heat is applied at one end (hot end), the refrigerant is evaporated and migrates as a gas to the other end (cold end) where it is condensed. The condensed refrigerant fluid is then collected at the bottom of the heat pipe where it returns to the hot end to complete the cycle.

6.14 The heat pipe system recovers sensible heat only. Since the air streams are separated by a partition, there is no problem with cross-contamination, and no pumps or drives are necessary.

6.15 The heat pipe is a completely reversible isothermal device. Air streams must be counterflow for maximum efficiency. Typical efficiency varies from 45 percent (4-row coil at 700 fpm coil face velocity) to 70 percent (8-row coil at 300 fpm coil face velocity).

6.16 The application of the heat pipe for retrofit in existing building systems is limited because duct work is frequently required in order to bring the hot and cold air ducts together.

F. Counterflow Heat Exchanger

6.17 The counterflow heat exchanger is sometimes called a static heat exchanger (Fig. 9). This recovery unit has no moving parts. It resembles an open-ended steel box with a rectangular cross section that is divided into a number of narrow layered passages. Every other passage carries exhaust air, alternating with those carrying supply air. The flow is counterflow, and efficiency varies with the number of passages. A reasonably attainable efficiency is 60 to 70 percent. Heat is transferred through the passage walls by conduction. Cross-contamination is not a problem due to the heat exchanger design.

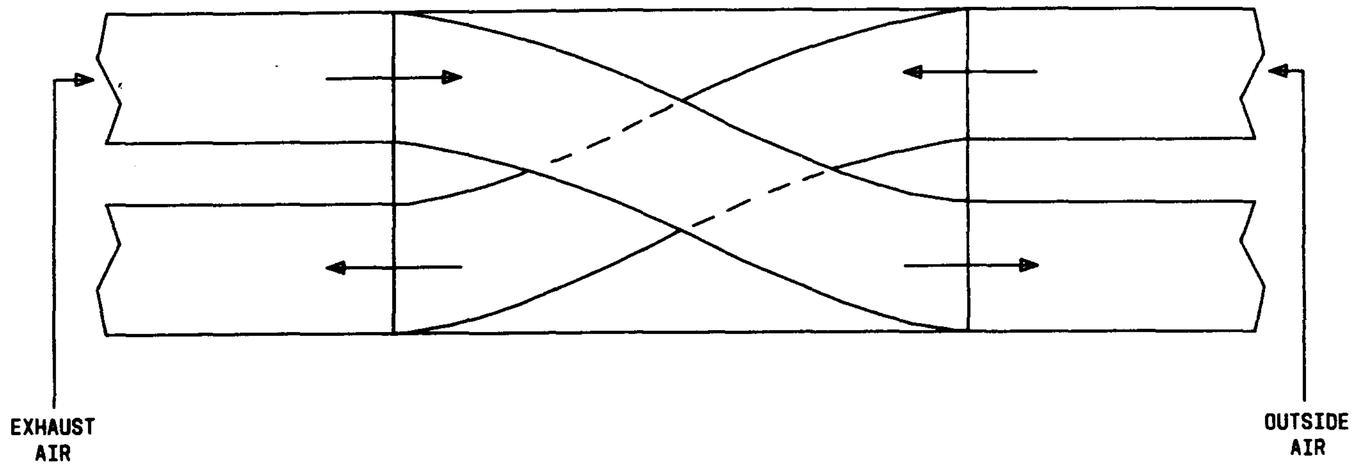

Fig. 9—Static Heat Exchanger

G. Evaluating Exhaust Air Systems

6.18 Heat recovery systems as described are not always feasible or cost-effective for a given condition. Several variables should be considered when evaluating exhaust air heat recovery systems:

(a) Amount of air being exhausted

(b) Temperature of exhaust

(c) Cost of energy

(d) Outside temperature range

(e) Internal heat gain from lights, equipment, and people.

7. HEAT RECOVERY CHILLERS

7.01 Heat recovery chillers are designed to utilize the heat energy that is normally rejected to the atmosphere. By reclaiming this heat, substantial savings in heating energy and costs can be realized. The recovered heat can be used for perimeter heating, preheating ventilation air, and preheating and heating domestic hot water. Water temperatures are available in the range of 100° to 130°F.

7.02 The heat recovery chiller is similar to the standard water chiller of either the reciprocating or centrifugal compressor type. An additional condenser is generally mounted on the equipment, or a double-bundle condenser is constructed with two separate condensing water circuits enclosed within the same shell. The basic configuration of a double-bundle condenser machine is shown in Fig. 10.

7.03 In the operation of a heat recovery chiller, hot refrigerant gas from the compressor is discharged into the condenser shell where the heat is then absorbed by one of the water circuits or both circuits simultaneously, depending on the system requirements at the time.

7.04 The tower water condensers are capable of 100 percent of the total heat rejection of the equipment when there is no building heat load demand.

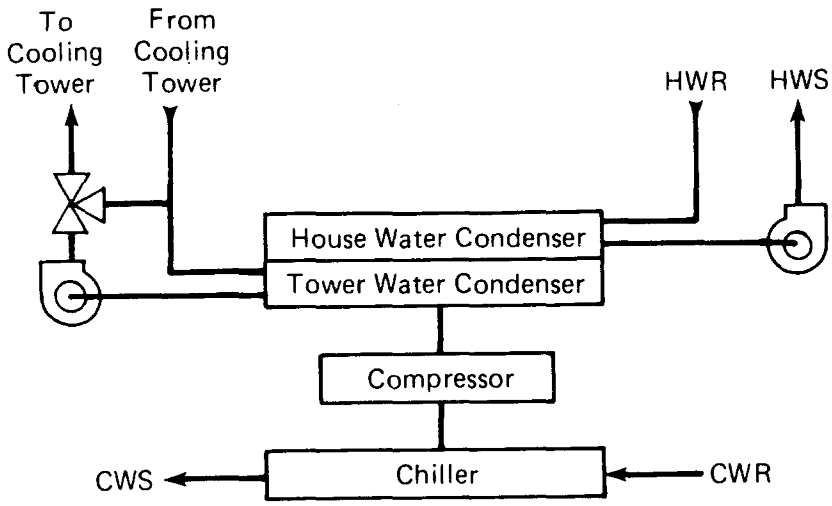

Fig. 10—Double-Bundle Heat Recovery Chiller

7.05 The heat recovery condensers are also capable of 100 percent of the total heat rejection of the units supplying the maximum available heat to the building. In the event of moderate heating demands, only partial condensing will occur in the heat recovery condenser.

7.06 In certain applications, the amount of heat reclaimed during the occupied hours may exceed daytime heating requirements. When this occurs, storage tanks can be added to the system allowing heat to be stored in the water in the tank for later release when the building is unoccupied.

7.07 Heat recovery equipment requires higher condensing pressures and temperatures than the standard water chiller. This is to elevate the temperature of the reclaim water circuit. The result is a higher power consumption per ton of refrigeration as well as a reduction in the refrigeration cooling capacity.

7.08 A refrigeration system can supply a heat recovery or reclaim system only when the cooling system is in operation. A heat recovery system is not economical unless there is a use for the reclaimed heat at the same time there are requirements for cooling. For efficient operation:

(a) Condensing pressure must not be so high that too much refrigeration capacity is lost or that the operating (energy) costs become too high.

(b) Condensing pressure must not go below the minimum level required for adequate flow of refrigerant.

(c) The refrigeration system should not be designed to operate for cooling unless heating is required.

8. HEAT PUMPS

8.01 Heat recovery from refrigeration systems is easily accomplished. However, the heat recovered is often at a very low level and not of much use unless the temperature is raised.

8.02 Heat pumps are heat transfer machines that can elevate the temperature of a low-temperature heat source. For example, heat can be extracted from a cold water source and used for heating domestic hot water or water for heating the building perimeter areas. See Section 760-550-214 for additional information on the internal heat source heat pump.

8.03 Heat pumps recover heat from a low-temperature source and raise its temperature to a usable level. For example, a heat pump can absorb heat from 95°F condenser water and raise its temperature to the 100° to 130°F range where it is usable to heat domestic hot water. They also have been used to reclaim heat from computer or equipment areas to be used for perimeter heating of other areas or even separate buildings in close proximity.