BUILDING ENERGY MANAGEMENT AND REDESIGN RETROFIT (BEMARR)

INTERNAL HEAT SOURCE HEAT PUMP

CONTENTS

1. GENERAL

1.01 This section provides building systems guidelines for the application of the internal heat source heat pump. 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 Technically, any refrigeration machine is a heat pump because it transfers heat from one location to another. And unquestionably, the heat produced by the heat pump is a useful product. In buildings that have high and low internal heat sources, it is possible to run the refrigeration machine to remove heat from one area, transport it, and reject it to another by means of an internal heat source heat pump.

1.04 By using piping and heat transfer coil changes, a refrigeration system can be converted to an internal heat source heat pump to save energy and fuel costs.

2. METHODOLOGY

2.01 For an example case study of an internal heat source heat pump, the following assumptions may be made:

(a) Outside winter design temperature is 0°F.

(b) Refrigeration system Coefficient of Performance (COP) is 2.5 (air handler with cooling coil, chilled water pump, water cooled chilled water machine, condenser water pump, and cooling tower).

(c) The system COP is defined as the ratio of the cooling effect divided by the power input of all the operating equipment expressed in consistent units:

$$\text{COP} = \frac{\text{Output}}{\text{Input}}$$

(d) For every 12,000 British Thermal Units per hour (BTU/hr) cooling performed, approximately 15,000 BTU/hr heat is rejected through the condenser water system.

(e) The building floor is 100 by 100 feet, the core area is 40 by 40 feet, and floors are occupied above and below this floor. (See Fig. 1.)

(f) The high internal heat area has a constant output of 7 watts per square foot per hour ($w/ft^2/hr$). The low internal heat area is 3 watts per square foot per hour occupied and 0 watts per square foot per hour unoccupied.

(g) The exterior building surface heat loss, at winter design, is 80,000 BTU/hr.

(h) The building load profile for the low internal heat area determines the changeover from cooling to heating to be at 55°F outside temperature.

(i) The office temperature is maintained at 65°F whenever heating is required. Heating is required when the outside temperature is below 55°F. The core area is maintained at 78°F year-around.

(j) Neglect conductance of heat through the walls from high to low heat areas.

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(k) The high heat area shall be considered isolated, and all heat must be removed by the air-handling system.

(l) Heat is supplied by a system whose efficiency is 50 percent at a gas cost of 45 cents per therm, and an electric rate of 5 cents per kilowatt-hour (kWh).

(m) Heating system efficiency is defined as the heat supplied to the space, divided by the fuel input to the heating unit, plus the power input to the heat delivery system (fans, pumps, etc).

2.02 The first item to consider is the amount of heat that the core area produces per hour. Since the loading is 7 w/ft²/hr and the area is 40 by 40 feet, the heat gain is:

$$7 \text{ w/ft}^2\text{/hr} \times 40 \text{ ft} \times 40 \text{ ft} = 11{,}200 \text{ watts/hr}$$

$$11{,}200 \text{ watts/hr} \times 3.41 \text{ BTUs/watt} = 38{,}192 \text{ BTU/hr}$$

2.03 Assume that for every 12,000 BTU/hr mechanically cooled, 15,000 BTU/hr is rejected. Rejected heat, in this case, is:

$$38{,}192 \text{ BTU/hr} \times \frac{15{,}000}{12{,}000} = 47{,}740 \text{ BTU/hr}$$

or 47.7 thousands of BTU per hour (MBH) of condenser water heat potentially available to offset the building heat loss in the office space.

2.04 The next step is to graph the heat loss of the building surfaces against the outside temperature. Using a BIN temperature format, tabulate the MBH loss per BIN, hours in BIN, and total MBTU during BIN period. (See Fig. 2 and Table A.)

2.05 Since the high heat source is constant, 47.7 MBH is always available for heating. Now tabulate the total and usable energy the heat pump can supply. Usable shall be the amount that can be used but not to exceed the heating loss for that BIN. (See Table B.)

2.06 The heat rejected by the heat pump can supply:

$$\frac{145{,}053.5 \text{ MBTU (Table B)}}{153{,}819.5 \text{ MBTU (Table A)}} \times 100$$

$$= 94\%$$

of the total annual heating requirements. It is noted that 8766 MBTUs must be supplied from the conventional heating system. But what about the fuel cost?

TABLE A

BUILDING ENVELOPE HEAT LOSS

COL 1	COL 2	COL 3	COL 4
BIN TEMP	HOURS IN BIN	MBH LOSS	TOTAL MBTU LOSS (COL 2 × COL 3)
50/54	583	4.5	2,623.5
45/49	579	11.5	6,658.5
40/44	592	18.0	10,656.0
35/39	796	26.5	21,094.0
30/34	935	33.5	31,322.5
25/29	633	41.0	25,953.0
20/24	453	48.5	21,970.5
15/19	290	55.5	16,095.0
10/14	163	62.5	10,187.5
5/9	74	70.0	5,180.0
0/4	27	77.0	2,079.0
		Total	153,819.5

2.07 For our heating system, the gas cost of supplying 145,053.5 MBTU is calculated as follows:

$$\text{Fuel Cost} = \frac{\text{MBTU heat loss}}{\text{EFFICIENCY of heating system}} \times \frac{\text{Cost of fuel}}{\text{therm}} \times \frac{\text{1 therm}}{\text{100 MBTU}}$$

$$= \frac{145{,}053.5 \text{ MBTU}}{.5} \times \frac{\$.45}{\text{therm}} \times \frac{\text{1 therm}}{\text{100 MBTU}}$$

$$= \$1305.47$$

TABLE B

HEAT REMOVED FROM HIGH INTERNAL HEAT AREA

COL 1	COL 2	COL 3	COL 4	COL 5
BIN TEMP	HOURS IN BIN	MBH OF REJECTED HEAT	TOTAL HEAT AVAILABLE (COL 2 × COL 3)	TOTAL USABLE HEAT — MBTU (NOTE)
50/54	583	47.7	27,809.1	2,623.5
45/49	579	47.7	27,618.3	6,658.5
40/44	592	47.7	28,238.4	10,656.0
35/39	796	47.7	37,969.2	21,094.0
30/34	935	47.7	44,599.5	31,322.5
25/29	633	47.7	30,194.1	25,953.0
20/24	453	47.7	21,608.1	21,608.1
15/19	290	47.7	13,833.0	13,833.0
10/14	163	47.7	7,775.1	7,775.1
5/9	74	47.7	3,529.8	3,539.8
0/4	27	47.7	1,287.9	1,287.9
		Totals	244,462.5	145,053.5

Note: COL 5 = The amount of heat in COL 4 which can be used to satisfy the envelope heat loss shown in COL 4 of Table A.

2.08 The heat pump electrical cost for 145,053.5 MBTU is calculated as follows:

$$\text{Fuel Cost} = \frac{\text{MBTU heat rejected}}{\text{COP}} \times \frac{1\ \text{kWh}}{3.41\ \text{MBTU}} \times \frac{12{,}000\ \text{BTU/hr}}{15{,}000\ \text{BTU/hr}} \times \frac{\text{Cost of fuel}}{\text{kWh}}$$

$$= \frac{145{,}054.5\ \text{MBTU}}{2.5} \times \frac{1\ \text{kWh}}{3.41\ \text{MBTU}} \times \frac{12{,}000\ \text{BTU/hr}}{15{,}000\ \text{BTU/hr}} \times \frac{\$.05}{\text{kWh}}$$

$$= \$680.60$$

or a savings of: $1305.47 – 680.60 = $624.87 per year.

2.09 This savings can be increased through thermal water storage. If the refrigeration machine were piped as shown in Fig. 3, on days when the rejected heat is greater than the usable heat (above 25°F outdoor temperature), heat could be stored for those times when the rejected heat requires supplemental heat to handle the building heating requirement (temperatures below 24°F outside).

2.10 The system would operate in the following manner. The primary condenser water pump (constant flow) would be designed to handle the 47.7 MBH of heat rejected from the condensing section with a 20°F temperature differential (ΔT). (Check with the manufacturer to determine if chiller can operate with 75°F incoming condenser water.)

2.11 In the heating mode, the water from the condensing section goes either into the tank on side 1 or to the secondary pump. The secondary pumping system is designed for peak flow at 0°F outside air temperature with a water temperature differential of 20°F. In our case, this is 77.0 MBH of required heating. This pump gallons per minute (gpm) flow will be about twice that of the primary pump, or in actual numbers:

$$\text{Primary pump gpm} = \frac{\text{BTU/hr}}{\Delta T \times 400^*}$$

$$= \frac{47{,}700\ \text{BTU/hr}}{20°\text{F} \times 400}$$

$$= 6\ \text{gpm}$$

$$\text{Secondary pump gpm} = \frac{\text{BTU/hr}}{\Delta T \times 400^*}$$

$$= \frac{77{,}00\ \text{BTU/hr}}{20°\text{F} \times 400}$$

$$= 10\ \text{gpm}$$

* Factor of 400 is based on a winterized system using 50 percent glycol rather than plain water.

2.12 Since the primary pump is only supplying 6 gpm (constant), the secondary pump, at 10 gpm during full demand, will require an additional 4 gpm to be drawn from the storage tank. At flows of less than 6 gpm from the secondary pump, the primary pump's excess water enters side 1 of the tank and recharges it.

2.13 If the tank is discharging for a prolonged time and can no longer supply the required temperatures for heating, the supplemental heater, sensing the return water temperature into side 2 of the tank, heats the supply heating water to make up the difference between required and usable heat.

2.14 While all this is happening, the 3-way valve is monitoring the return water to the chiller to maintain it at a constant 75°F.

2.15 Storage tank B in the chilled water loop could be used if the internal heat source of the high heat area were variable rather than constant.

2.16 Storage tank A can be sized for the maximum length of time between recharging or for the average variation of a typical heating day. See Section 760-550-215.

3. ECONOMIC ANALYSIS

3.01 In our example, we illustrated a savings of $624.87 per year with the internal heat source heat pump. The storage tank savings can be determined by following the same procedure as discussed in Section 760-550-215*.

3.02 Life-cycle costing can determine the economical feasibility of such an installation if the cost of modifying the piping system and the cost of the heating devices to use 75°F heating hot water with a 20°F temperature differential is given.

* Check Divisional Index for availability.

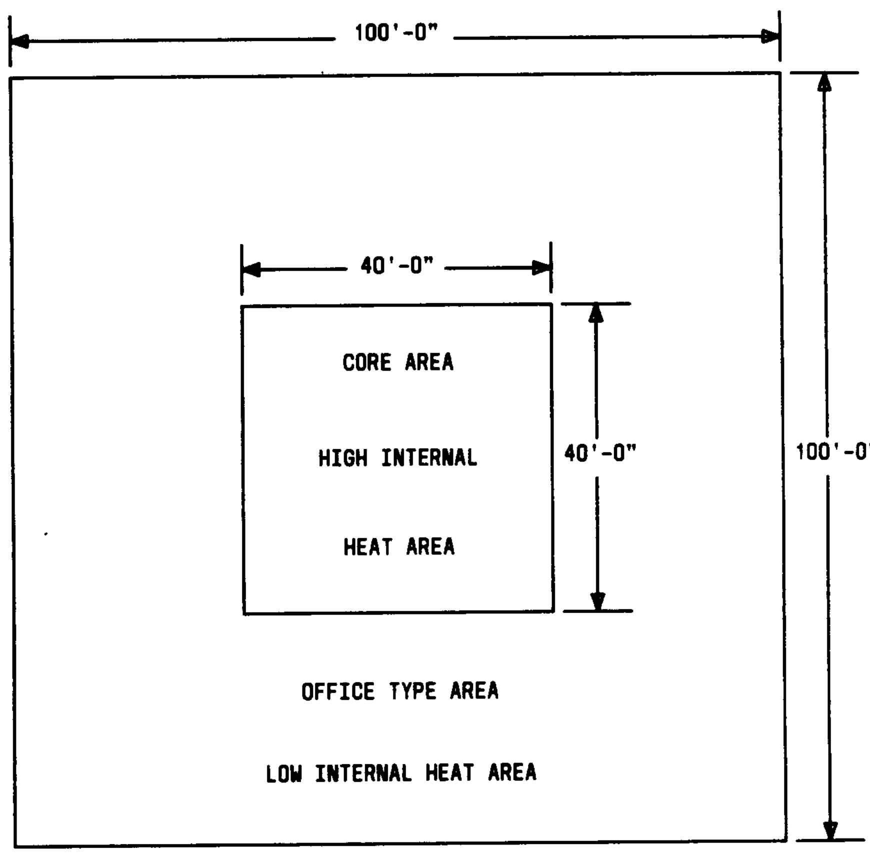

Fig. 1—Floor Plan of Case Study Building

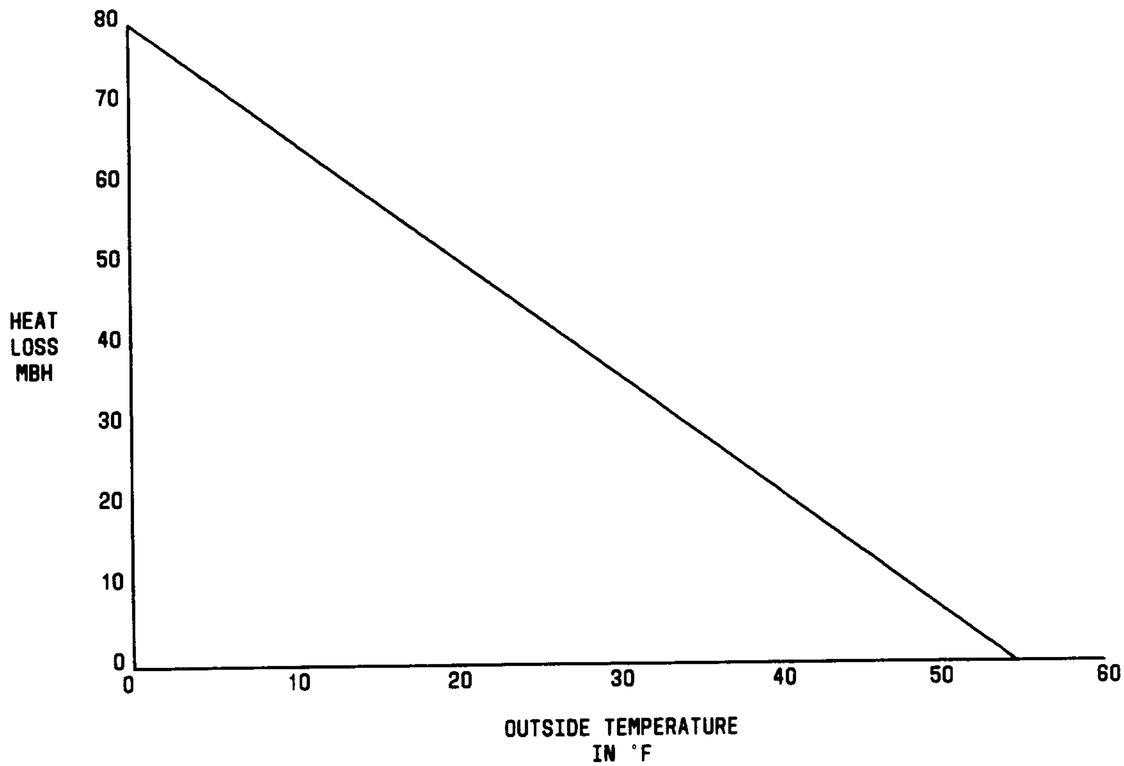

Fig. 2—Building Heat Loss

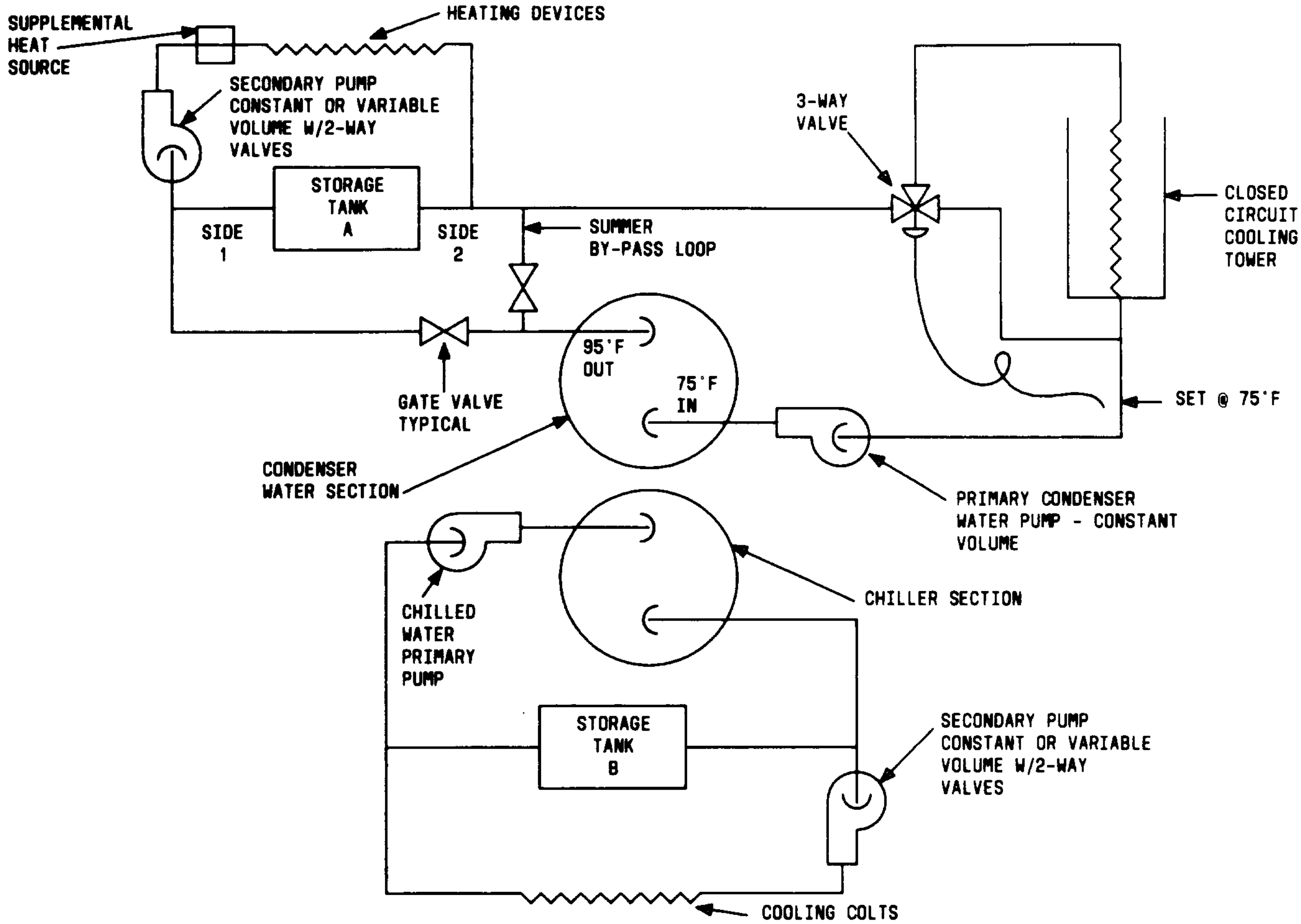

Fig. 3—Internal Heat Source Heat Pump With Thermal Storage