3. U. Bonne, A. Patani, R. Jacobson and D. A. Mueller, "Electric-Driven Heat Pump Systems: Simulations and Controls II", ASHRAE Trans. 86I, 687 (1980). Since both Figs. 5 and 6 in this memo were computed for the heating mode, we selected the "normal operation" evaporator size in the cooling mode from Fig. 6 to be 0.4 rather than 0.6 units as marked, in order to allow for the generally larger capacity of the outdoor coil.

4. Dave Murphy, Jr., Air Conditioning Coil Heat Transfer Tests, Report No. 3800, Tests 1-5, Air Filter Testing Laboratories, Crestwood, Kentucky, Nov. 1985.

APPENDIX

1. Calculation of Power Demand Changes

To determine the input power change as a function of the changing capacity of the indoor coil when used either as a condensor or as an evaporator, we continued to use the curve in Fig. 3 representing the average dust load, i.e. doubling the pressure drop in 7.38 years, which is equivalent to reducing the COP to 0.8 of its initial value during that period or to 0.7 in a period of 11 years.

From Fig. 4, we determined that a COP degradation to 0.7 corresponds to a coil capacity reduction to .44 of its initial value. The H_5-curves of Figs. 5 and 6 provide input power values at 0.44 of the "normal operating" coil capacity of $.25 \times 10^6$ cal/(hC) (heating) and a capacity of $.4 \times 10^6$ cal/(hC) in the cooling mode. Fig. 6 represents an analysis of outdoor coil capacity influence on heat pump parameters; therefore a smaller capacity of 0.4×10^6 was selected to better represent the indoor coil performance. The reductions then correspond to values of $.11 . 10^6$ and $.176 . 10^6$ cal/(hC) coil capacity, respectively. The corresponding compressor power inputs normalized to "normal operation" are, from Figs. 5 and 6, again for heating and cooling: 2.48/2.19 = 1.1324 and 1.97/2.02 = .876, respectively. However, while the indoor fan power input is reduced in both heating and cooling (it unloads as the mass flow is reduced), this power opposes the trend of the compressor input power during the heating mode. We therefore corrected the heating mode input power change (after 11 years) by determining the mean power demand change:

$$dp_H = \underbrace{[(1.1324)8.787/2.6}_{\text{COMPRESSOR FAN}} + \underbrace{.35}_{\text{OUTD. FAN}} + \underbrace{(.8786)0.35]}_{\text{INDOOR FAN}}/\underbrace{(3.827+.7)}_{\text{INPUT POWER}} = 1.0946$$

where 2.6 = the instantaneous COP at the same outdoor temperature (47F) as the heat pump capacity of 2.5 x 12,000/3414 = 8.787 kW and

.35 kW = the assumed input power for each of the indoor and outdoor fans.

We could then compare and tabulate the change in input power (under continuous operation) in the heating and cooling modes, P_H and P_C, both without and with an EAC:

$P_O = 1$ and $P_{O,EAC} = 1.023\ P_O$, under clean coil conditions

$P_H = 1 - (1-1.0946)FZ_L$ and $P_{H,EAC} = 1-(1-1.0946)FRZ_L\ 1.023$

$P_C = 1 - (1-.876)FZ_L$ and $P_{C,EAC} = 1-(1-.876)FRZ_L\ 1.023$

Columns 5 and 6 of Table 1 list the values of $P_{H,EAC}/P_H$ and $P_{C,EAC}/P_C$, respectively.

Note that these changes in peak power demand occur under outdoor temperature conditions above the design cooling load and at the balance point. Below the balance point the change in demand is increasingly determined by the required resistance heat, the consumption of which is dependent on the status of the distribution system: A clean coil will most likely get the heat to the proper place to satisfy the thermostat sooner than an air stream slowed down by a partially clogged coil.

2. Calculation of Cooling Degree Days

The cooling load, given as Z_C = 900 compressor hours, was used to compute total annual operating hours, which determine the amount of coil soiling. In order to add the cooling load cost to that of the heating load, we converted Z_C to degree days, L_C, and with the specified building load and average cooling COP, into input kWh and cost:

$$L_C = Z_C H_{2,C}/(B_{9,C}\ 24) = 1200 \text{ degree F days}$$

where $H_{2,C}$ = heat pump cooling capacity $\simeq$ (assumption!) $H_{2,47}$ e_C/e_H = 30,000 2.64/2.2

$B_{9,C}$ = B_9 x correction factor for latent load

$= B_9(H_{2,\text{sensible}} + H_{2,\text{latent}})/H_{2,\text{sensible}}$

$\simeq B_9$ 1.65; 24 converts Z_C from hours into days.

This is consistent with a cooling design temperature of about 105F since

$$L_C = Z_C\ (105-78)H_{2,105}/H_{2,C}/24$$

$$= 37.5(27)(H_{2,47} - 20S_1)/H_{2,47} = 37.5(27)1.2 = 1215 \text{ Fd}$$