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# AIR-CONDITIONING AND HEAT PUMP OPERATING COST SAVINGS BY MAINTAINING COIL CLEANLINESS

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ABSTRACT

Previous analyses of the cumulative influence of collected dust, tobacco smoke/tars and particulates generated in kitchen activities on the coefficient of performance (COP) of heat pumps indicated that the particulate-retention efficiency of finned tube coils ranges from 40 to 78%. This retention was experimentally verified, and can lead to significant heat exchange deterioration over the useful service life of a heat pump or air conditioner. This means that significant operating cost savings can be achieved by the use of high efficiency air cleaners (HEAC) which retain 90 ± 5% of residential particulates to prevent such deterioration.

We expanded the analyses and complementary experimental work to quantify the operating cost savings due to avoidance of cleaning costs and maintenance of high operating COP. Field surveys show that users seldom have their coils cleaned and, therefore, could benefit from a program to maintain coil cleanliness via application of efficient air cleaners. Our results show that this would save from 10 to 25% of averaged operating costs (for each of the assumed 15 years of service life), or from 25 to 55% during the 15th year of operation, for the range of considered climatic and dust-loading conditions. Lower efficiency air cleaners or filters, retaining e.g. only 20% of the particulates, would only realize about 1/8 of the above savings under otherwise equal conditions. The magnitude of these benefits suggests that means should be developed to credit the seasonal energy performance figures of merit of air conditioners and heat pumps when they are equipped with appropriate air cleaners.

## INTRODUCTION AND SUMMARY

This paper presents an analysis, backed by experimental evidence, of the benefits provided to heat pump systems by using high efficiency air cleaners (HEAC), in place of (or in addition to) commonly used dust stop filters, in terms of: 1. Annual cost savings and 2. Electric demand changes. The results are presented in both graphical and tabular form for these conditions:

- o A medium residential dust load, ranging from .05 to .2 mg/m$^3$.
- o An annual heat pump operating time of 3190 hours (vs. 3900 hours in Pittsburgh) to represent the southeastern region of the US, where heat pumps enjoy a large market penetration.
- o A dust-retention rate causing the indoor coil pressure to double in 7.38 years (vs. 6 in Pittsburgh), and
- o No coil cleaning during the assumed service life.
- o The reference system is equipped with the filters commonly used today with heat pumps and furnaces. They are flat panel filters composed of coarse fibers. These filters have a high porosity and are sometimes coated with a viscous substance, such as oil, to enhance particle retention. They are of low pressure drop, low cost, but their arrestance efficiency is usually less than 20-30%.

While we had used the climate of Pittsburgh in our earlier calculations(1), for easier comparison to furnace efficiency calculations often made for this average U.S. heating season climate, we used a southeastern U.S. region because of the prevalence of heat pump applications there.

We determined <u>annual energy consumption savings</u> data for up to 20 years of service. For example, after 15 years of service, the operating cost savings ranged from 25 to 55%, with a typical value of 35%. On the 20th year of operation (without any interim coil cleaning) that value had increased to 49%, while the electric peak heating power demand was found to be reduced by a factor of 0.894; the demand would increase by a factor of 1.172 <u>without an HEAC</u> but only by a factor of 1.048 <u>with</u> an efficient HEAC. The peak cooling power demand would increase by a factor of 1.278 (.774 vs. .990) as a result of using an HEAC. This assumed that no significant deterioration in comfort occurred during that period, even under peak cooling conditions. Had we included the increased use of resistance auxiliary heat during low outdoor temperatures, as the balance temperature increased over the years of service without an HEAC, the savings would be greater still.

<u>Heating power demand</u> increases with operational life both with and without an air cleaner, but it increases data 7x slower rate if a high efficiency air cleaner (HEAC) is used. While the compressor discharge pressure and power requirements increase as the condensor coil plugs up, the partially unloaded circulating fan requires less power and the balance temperature moves up. However, this last effect has not been included in the analysis to date.

<u>During cooling power demand</u> , the air cleaner reduces the following effects as the coils soil: It slows down the trend of lowering evaporator temperature and reducing air-flow, which cause a reduction in the peak power demand during continuous operation. In other words, the design power cooling demand with HEAC drops at a slower rate than without an HEAC. During both heating and cooling operation, the air cleaner reduces the energy consumption (kWh) during cycling (part-load operation) because of the improved heat pump efficiency.

We computed the above results by modifying our earlier quadratic efficiency degradation approach to a linear one. The latter technique provides a better representation of the approximate constant particulate-retention efficiency and the computed linear COP degradation. We stressed the term "efficient air cleaner" several times above, for a good reason: We define high efficiency air cleaners as those that remove 90±5% of residential particulates, as determined according to ASHRAE Std. 52-76, "Initial Atmosphere Dust Spot Test". Savings achievable with <u>low efficiency</u> air cleaners or filters, e.g., those with a removal efficiency of only 20%, would be about (100-90)/(100-20) = <u>1/8</u> of those near an efficiency of 90%.

## ANALYSIS

To determine the influence of air cleaners on the achievable savings/year for each year and any associated changes in the peak demand for electric power, we assumed the following heat pump system parameters for a typical southeatern region of U.S.:

- o Heating season, $L_H$ = 3500 degree F days (vs. 5200 Fdays for Pittsburgh)
- o Cooling compressor hours, $Z_C$ = 900 compressor hours (vs. 700h)
- o Balance point, $T_B$ = 28F (vs. 26F)
- o Heating seasonal COP, $e_H$ = 2.2 (vs. 2.0)
- o Cooling seasonal COP, $e_C$ = 2.64 (vs. 3.0)
- o Building load approx. $B_9$ = 730 BTU/(hF) or 214 W/F (unchanged)
- o Electric rate, $C_e$ = 0.048\$/kWh or $4.8\times10^{-5}$ \$/Wh (vs. $6.92\times10^{-5}$ \$/Wh)

In addition we derived the following parameters: building load, annual operating hours and demand changes when operating at steady state during heating (i.e. at the balance point) and cooling. The first two were obtained by inserting appropriate values into the known relation for building load:

$$B_9 = H_{2,B}/(T_R - T_B) \text{ in W/F} \qquad (1)$$

where $H_{2,B}$ = heat pump output at the balance (outdoor) temperature, $T_B$, given in F; and $T_R$ represents the reference (outdoor) temperature at which the need for heating starts. Since the heat pump output generally is only given at 47F, $H_{2,47}$, we converted that output to the value at $T_B$ with

$$H_{2,B} = H_{2,47} - S_1(47 - T_B) = 25{,}336 \text{ BTU/h} \quad (2)$$

where the slope, $S_1$, was set equal to the system COP vs. outdoor temperature slope derived earlier(3) as part of our linear heat pump model development:

$$S_1 = H_{2,47}(e_{47} - e_{17})/(e_{47}(47 - 17)) = 245.5 \text{ BTU/(hF)} \quad (3)$$

where $e_{47}$ = 2.2 or COP at 47F and
$e_{17}$ = 1.6 or COP at 17F.

For heat pumps of 2.5- and 3-ton capacity at 47F, the above calculations resulted in building loads of 200 and 240 W/F, respectively. We selected the former, since it was closest to the above average home size of 214 W/F.

From the seasonal "equivalent heat output", $H_{2,E} = H_{2,B}$ = 25,336 BTU/h and the building load, we derived the seasonal operating hours, $Z_{op}$:

$$Z_{op} = B_9 L_H\, 24/H_{2,E} + Z_C = 3167\text{h} \quad (4)$$

(vs. 3900h for Pittsburgh)

where $H_{2,E}$ = heat pump output at the equivalent outdoor temperature, which for this region is about 28F, i.e. the steady- state COP and output at that temperature are equal to the seasonal average(2), see Fig. 1*. Having picked a convenient value, $T_B = T_E$ and $H_{2,47}$ = 30,000 BTU/h, we had obtained $H_{2,E}$ = 25,336 BTU/h or 7,420 W with eq. (2).

The resulting annual operating cost is now simply approximated by:

$$C = (Z_H\, H_{2,E}/e_H + Z_C H_{2,E}(e_C/e_H)/e_c)C_e$$

$$C = (Z_{op}\, H_{2,E}/e_H)C_E = 512.74\$ \quad (5)$$

(680.92$ for Pittsburgh)

assuming that the cooling output and COP show similar temperature dependencies. For homes requiring 3-ton heat pumps, the annual operating costs obtained were 615.48 and 816.90$, respectively.

The influence of dust/particulate buildup on heat pump COP degradation and on peak power demand changes was determined by using results computed earlier(1), and presented in Figs. 2 and 4. Fig. 2 was obtained for Pittsburgh's average of 3900h of annual operation. This was converted to sputheastern regional conditions by decreasing the rate of degradation in inverse proportion to the shortened hours of annual operation of 3167. This resulted in the COP vs. service life plots of Fig. 3, with and without an air cleaner. Fig. 4 also shows how COP degrades as a function of particulate buildup and how this buildup influences coil capacity, which in turn influences compressor and fan power demand.

This latter influence was taken from the HFROST (= heat pump performance simulation program) simulation results(3) of heat pump compressor input power, $H_5$, vs. coil capacity presented in Figs. 5 and 6. These results were normalized and led to the following data on compressor and heat pump peak power demand changes (see Appendix for details):

o During the heating season, an increasingly soiled coil causes the discharge and condensing pressure to rise, while the fan unloads and its

*The curves of Fig. 1 were obtained by computing the <u>seasonal</u> average COPs vs. balance temperature, and substituting the COPs for the temperature at which the same steady-state COP would be obtained.

power requirement gradually decreases. The net result is a 9.46% increase in the power consumption at any (= assumption!) heating season condition, after 11 years of operation. This value was proportioned linearly from 0 to 20 years of operation without the air cleaner. So, the 6th and 5th columns in Table 1 represent peak power demand ratios with ($P_{H,EAC}$) and without ($P_H$) an air cleaner, respectively, relative to the initial demand without an EAC, $P_{H,O}$ and again linearizing the effect of the EAC by increasing the length of time needed to achieve a given demand change by a factor of R=7.

- o During the cooling season, the increase in coil soiling decreases evaporating temperature and pressure and tends to unload <u>both</u> the compressor and fan. We obtained a decrease to 87.6% after 11 years. With the EAC, this change in demand would not happen before 77 years of service. The corresponding ratio of demands without an EAC ($P_C/P_{C,O}$) over the one with an EAC ($P_{C,EAC}/P_{C,O}$) is listed in columns 7 and 8 of Table 1.

Other benefits or changes resulting from the use of an efficient air cleaner are shown in columns 2-4 of Table 1:

- o Column 2: The ratio of actual (e) to initial ($e_o$) heat pump COP after the number of service years listed in the first column, <u>without</u> an EAC,
- o Column 3: The ratio of actual ($e_{EAC}$, with an EAC) to initial ($e_o$, without an EAC) COP; the COP degradation to .9775 at the first year is caused by the 50 W of power needed by the EAC; and
- o Column 4: The operating cost savings for the listed years, resulting from the use of an EAC, obtained by computing

$$dC/C = 100(eo/e - e_o/e_{EAC})/(e_o/e)$$

where: $e/e_o = 1 - (1-d_1)FZ_L$,

$e_{EAC}/e_o = 1 - (1-d_1)FRZ_L/R_P$,

$R_P$ = heat pump power ratio with/without an EAC, $\simeq$ 1.023 for a 2.5 ton heat pump at 47F, because of the 50 W needed by the EAC,

$d_1 = 0.8$,

R = 7, and

$Z_p$ = 7.38 years

By letting $Z_p$ adopt additional values of 4.92 and 9.85 years, which represent residential environments with low dust and high dust concentrations, we also provided these kWh savings in graphical form in Fig. 7. The corresponding cumulative savings (including a 100$ cleaning bill every 15 years) are shown in Fig. 8 for a service life of up to 30 years.

## EXPERIMENTAL

As part of an ongoing project, preliminary measurements were made with three air-conditioning "A" coils, which had been removed from existing residences. They are labeled #1, #2, and #3 in Table 2, which lists the measured pressure drop vs. airflow and were sent to an independent lab[4] that performed the following tests:

1. Determined the heat transfer characteristics of these A-coils, as found, by measuring the heat transferred to air while hot water was circulated through the refrigerant tubes to simulate refrigerant flow.
2. Cleaned the coils and repeated test #1.

3. Artificially loaded up the coils with particulates in a simulated and accelerated life test, and repeated test #1 after completing each dust-loading increment.

Unfortunately, the consistency of the obtained heat transfer data (heat rate added to the air stream heat rate lost from the water stream) was poor, so that no quantitative evaluation was possible. However, a trend is observable in the plotted data shown in Fig. 9, in spite of the poorly satisfied consistency condition, $H_a = H_w$. The data show energy into air vs. energy lost by water, before (full dots) and after cleaning (empty dots) for air flows of 600 and 1200 $ft^3/min$ (1019 and 2039 $m^3/h$) in the form of small and large dots, respectively. A very similar graph was obtained for the respective U-values (BTU/(min F)) by simply dividing by the average differences between water and air temperatures. The trend of moderate to significant increases in energy transferred as a result of cleaning the A-coils is visible in Fig. 9. Unfortunately, none of the history of these A-coils is known to us.

Test #3 was conducted with ground cotton lint particulates (=ASHRAE test dust), fed in 20 gram increments until a total of 400 grams was reached.[4] The air and water flow rates were maintained at 600 $ft^3/min$ and 6.59 lb/min, respectively. The corresponding changes in pressure drop ranged from 0.08 to 0.55 inches of water column. The A-coil retention of the particulates was high, amounting to 75.7, 92.4 and 67.7%. These high values should be qualified by the (unknown) size distribution of the used ground lint particulates and the retention efficiency of the arrestance filter downstream of the A-coil. However, this retention efficiency, measured by a professional air filter testing lab[4], verifies at least qualitatively the high values computed and reported earlier[1].

As the lint particulates increased, the amount of exchanged energy decreased only moderately within a range from 100% down to 70%. This set of experiments should be repeated, since 1. The temperatures of the air and water flows overlapped, 2. The results lacked consistency, as did those discussed earlier in Fig. 9; and 3. The air flow was artificially kept constant, rather than allowed to decrease as provided by the fan characteristic in a field installation.

To obtain information of the air flow effect, we measured its influence on the COP of a heat pump installed in one of our test chambers. The result is shown in Fig. 10. It shows that the COP can drop from 3.12 to 2.76 or 11.5% when only the air flow drops from 1000 to 500 $ft^3/min$ (1529 to 850 $m^3/h$), or by 13.2% if the insulating effect of the dust layer is taken into account. This soiling was obtained for a 3-ton heat pump by retaining 600 g of a 1000 g dust load.

## CONCLUSIONS

The use of high efficiency air cleaners (HEAC) upstream of finned tube coils of air conditioners and heat pumps is recommended to maintain efficient and trouble-free operation. HEACSs are defined here as having a 90 ± 5% removal efficiency for residential particulates. Such air cleaners can maintain operation throughout the service life of an air conditioner (A-coil) or heat pump, while low efficiency air cleaners, defined here as having a removal efficiency of 20-40%, cannot. The operating cost savings achievable with a HEAC air cleaner in a typical southeastern U.S. region characterized by 3500 heating degree F days (1940 °Cd) amounted to average values from 10 to 25% for a heat pump service life of 15 years (or peak savings of 25 to 55% during the 15th year of service) without coil cleaning. These values are the result of considering a range of typical residential dust loads.[1]

The particulate accumulation also influences the corresponding peak electricity demand rates (kW), see Table 1, by unloading the indoor fan and compressor in the cooling mode, while unloading the indoor fan and loading the compressor in the heating mode. The use of an air cleaner sharply reduces these effects.

These values are supported by experimental evidence obtained by testing old coils in the lab before and after cleaning and by dust-loaded and air-starved heat pump performance data obtained under controlled test chamber conditions. The former doubled the heat transfer rate (-average of 6 tests), while the latter showed a 10-13% decrease in COP under dirty coil conditions. Air-conditioning or heat pump systems featuring such means to maintain efficient operation via air cleaners over long periods of service life should be credited with appropriately modified seasonal efficiencies. Appendix 2 presents the rationale for converting compressor cooling hours into cooling degree days.

## NOMENCLATURE

- $B_g$ Building load, in BTU/(hF), W/°C or W/F
- C Cost of operation, in $
- e Heat pump COP
- F Frequency or inverse of operating time, in $years^{-1}$
- d Degradation factor (of COP), dimensionless
- D Dust layer thickness in mm
- H Heat output of heat pump
- K Thermal conductivity in BTU/(hf ft)
- L Heating or cooling load in Fd (degree-days)
- P Pressure
- R Ratio of operating times, with and without an air cleaner to achieve equal pressure drops
- S Coil fin spacing, in mm
- T Temperature in F or °C
- $T_B$ Balance (outdoor temperature) point, at which the steady state heat pump output equals the heating or cooling load of the building. Typically, $T_B$ is about 20 to 35F for heating
- $T_R$ Reference (outdoor) temperature (start of heating demand), taken here as 65F in agreement with older practices; more realistic values today range from 55 to 62F for residential housing.
- $Z_p$ Time of operation needed for event to occur (like doubling of coil pressure drop), in years.

## SUBSCRIPTS

- B Balance point
- C Cooling
- E Equivalent temperature, at which $e = e_{SEASONAL}$, for a U.S. heat pump system with electric resistance auxiliary heat.
- e Electric
- H Heating
- L Life
- O Clean coil
- P Pressure
- R Reference
- 1,3 Outdoor
- 2,4 Indoor
- 5 Compressor

## REFERENCES


1. Krafthefer, B., and Bonne, U., 1986 "Energy Use Implications of Methods for Maintaining Heat Exchanger Coil Cleanliness", ASHRAE 1986 Seminannual Meeting, Symposium on "Case Histories and Energy Use Implications of Operation with Soiled Finned-Tube Heat Exchangers", San Francisco, CA 19-22 Jan. 1986, ASHRAE Trans. 92, Pt. 1,420 (1986).

2. U. Bonne and A. Patani, "Modeling the Influence of Heat Pump Sizing, Climate and Test Conditions on Seasonal Efficiency", Conf. on "Efficiency and Performance of HVAC Equipment and Systems III, Purdue Univ., W. Lafayette, Ind., 23-25 Oct. 1978.

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 $0.4 \times 10^6$ 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:

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

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

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

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. <u>Calculation of Cooling Degree Days</u>

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

$$B_{9,C} = B_9 \times \text{correction factor for latent load}$$

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

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

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

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

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

TABLE 1

Heat Pump Energy Performance vs. Years of Service, With and Without a High-Efficiency Air Cleaner to Maintain Indoor Heat Exchanger Cleanliness

| SERVICE TIME $Z_L$ YEARS | COP DEGRADATION: HP ALONE $e/e_o$ | COP DEGRADATION: HP + EAC $e_{EAC}/e_o$ | OP. COST SAVINGS dC/C \$/yr | PEAK POWER DEMAND RATIOS: $P_H/P_{H,O}$ | $P_{H,EAC}/P_{H,O}$ | $P_C/P_{C,O}$ | $P_{C,EAC}/P_{C,O}$ |
|---|---|---|---|---|---|---|---|
| 0 | 1.0000 | .9775 | -2.300 | 1.0000 | 1.0230 | 1.0000 | 1.0230 |
| 1 | 0.9729 | .9737 | 0.086 | 1.0086 | 1.0243 | 0.9887 | 1.0214 |
| 2 | 0.9458 | .9699 | 2.490 | 1.0172 | 1.0255 | 0.9775 | 1.0197 |
| 3 | 0.9187 | .9662 | 4.913 | 1.0258 | 1.0268 | 0.9662 | 1.0181 |
| 4 | 0.8916 | .9624 | 7.355 | 1.0344 | 1.0280 | 0.9549 | 1.0164 |
| 5 | 0.8645 | .9586 | 9.816 | 1.0430 | 1.0293 | 0.9436 | 1.0148 |
| 6 | 0.8374 | .9548 | 12.297 | 1.0516 | 1.0305 | 0.9324 | 1.0131 |
| 7 | 0.8103 | .9510 | 14.798 | 1.0602 | 1.0318 | 0.9211 | 1.0115 |
| 8 | 0.7832 | .9472 | 17.318 | 1.0688 | 1.0331 | 0.9098 | 1.0098 |
| 9 | 0.7561 | .9435 | 19.859 | 1.0774 | 1.0343 | 0.8985 | 1.0082 |
| 10 | 0.7290 | .9397 | 22.420 | 1.0860 | 1.0356 | 0.8873 | 1.0065 |
| 11 | 0.7019 | .9359 | 25.002 | 1.0946 | 1.0368 | 0.8760 | 1.0049 |
| 12 | 0.6748 | .9321 | 27.605 | 1.1032 | 1.0381 | 0.8647 | 1.0032 |
| 13 | 0.6477 | .9283 | 30.229 | 1.1118 | 1.0393 | 0.8535 | 1.0016 |
| 14 | 0.6206 | .9245 | 32.875 | 1.1204 | 1.0406 | 0.8422 | 0.9999 |
| 15 | 0.5935 | .9208 | 35.542 | 1.1290 | 1.0419 | 0.8309 | 0.9983 |
| 16 | 0.5664 | .9170 | 38.232 | 1.1376 | 1.0431 | 0.8196 | 0.9966 |
| 17 | 0.5393 | .9132 | 40.943 | 1.1462 | 1.0444 | 0.8084 | 0.9950 |
| 18 | 0.5122 | .9094 | 43.678 | 1.1548 | 1.0456 | 0.7971 | 0.9933 |
| 19 | 0.4851 | .9056 | 46.435 | 1.1634 | 1.0469 | 0.7858 | 0.9917 |
| 20 | 0.4580 | .9018 | 49.215 | 1.1720 | 1.0481 | 0.7745 | 0.9901 |
| | | | Peak Demand Change: | .8943 | | 1.2784 | |



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* Under SE region conditions of:

- o Heating season, $L_H$ = 3500 degree F days
- o Cooling compressor hours, $Z_c$ = 900 compressor hours
- o Balance point, $T_B$ = 28F
- o Heating seasonal COP, $e_H$ = 2.2
- o Cooling seasonal COP, $e_c$ = 2.64
- o Building load $B_g$ = 684 BTU/(hF) or 200 W/F
- o Electric rate, $C_e$ = 0.048\$/kWh or $4.8 \times 10^{-5}$ \$/Wh
- o Annual Oper. Time, $Z_{op}$ = 3167 hours
- o Time to double coil dp, 1/F = 7.38 years, or 7x7.38 years with EAC
- o COP degradation at that point, $d_1$ = 0.8

TABLE 2

Airflow Resistance and Heat Transfer Tests on Three Used Air-Conditioning Coils(4)

**AIR CONDITIONING COILS AS RECEIVED**

REPORT NO. 3800
TEST NO. 1

**AIR FLOW VS. RESISTANCE:**

| AIR FLOW RATE CFM | RESISTANCE IN. W.G. COIL MARKED #1 | COIL MARKED #2 | COIL MARKED #3 |
|---|---|---|---|
| 400 | .03 | .045 | .05 |
| 600 | .05 | .085 | .095 |
| 800 | .08 | .145 | .16 |
| 1000 | .12 | .22 | .24 |
| 1200 | .16 | .30 | .34 |

**HEAT TRANSFER DATA**

| COIL MARKED | AIR FLOW RATE CFM | WATER FLOW RATE LB/MIN. | TEMP. AIR IN $T_1$ °F | TEMP. AIR OUT $T_2$ °F | TEMP. WATER IN $T_3$ °F | TEMP. WATER OUT $T_4$ °F | AIR | | WATER | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | $T_2 - T_1$ | BTU/min | $T_3 - T_4$ | BTU/min |
| #1 | 600 | 6.59 | 77.6 | 80.2 | 90.4 | 81.6 | 2.6 | 28.08 | 8.8 | 57.99 |
| | 1200 | 6.59 | 78.6 | 79.4 | 89.4 | 80.6 | 0.8 | 17.28 | 8.8 | 57.99 |
| #2 | 600 | 4.94 | 85.2 | 89.4 | 103.2 | 88.2 | 4.2 | 45.36 | 15.0 | 74.10 |
| | 1200 | 4.94 | 86.2 | 87.2 | 101.4 | 86.8 | 1.0 | 21.6 | 14.6 | 72.12 |
| #3 | 600 | 6.42 | 85.6 | 92.2 | 98.6 | 89.4 | 6.6 | 71.28 | 9.2 | 59.06 |
| | 1200 | 6.42 | 85.6 | 90.0 | 96.8 | 87.4 | 4.4 | 95.04 | 9.4 | 60.35 |

**AIR CONDITIONING COIL CLEANED**

TEST NO. 2

| COIL MARKED | AIR FLOW RATE CFM | WATER FLOW RATE LB/MIN. | TEMP. AIR IN $T_1$ °F | TEMP. AIR OUT $T_2$ °F | TEMP. WATER IN $T_3$ °F | TEMP. WATER OUT $T_4$ °F | AIR | | WATER | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | $T_2 - T_1$ | BTU/min | $T_3 - T_4$ | BTU/min |
| #1 | 600 | 6.59 | 78.2 | 90.2 | 100.8 | 85.0 | 12.0 | 129.60 | 15.8 | 104.12 |
| | 1200 | 6.59 | 79.4 | 85.0 | 95.4 | 82.0 | 5.6 | 120.96 | 13.4 | 88.31 |
| #2 | 600 | 4.94 | 78.6 | 83.8 | 97.8 | 81.8 | 5.2 | 56.16 | 16.0 | 79.04 |
| | 1200 | 4.94 | 78.2 | 80.4 | 96.2 | 79.6 | 2.2 | 47.52 | 16.6 | 82.00 |
| #3 | 600 | 6.42 | 77.6 | 84.4 | 91.8 | 82.4 | 6.8 | 73.44 | 9.4 | 60.35 |
| | 1200 | 6.42 | 77.6 | 82.6 | 90.6 | 80.4 | 5.0 | 108.00 | 10.2 | 65.48 |

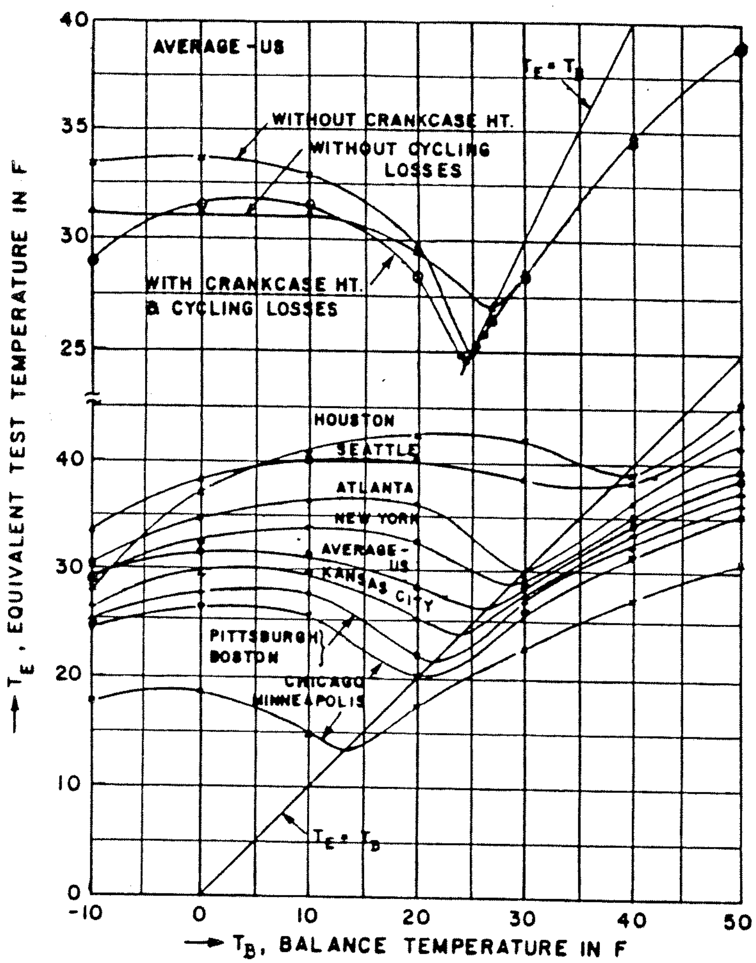


*Figure 1.* *Equivalent heat pump test temperature vs. balance temperature for various climates*

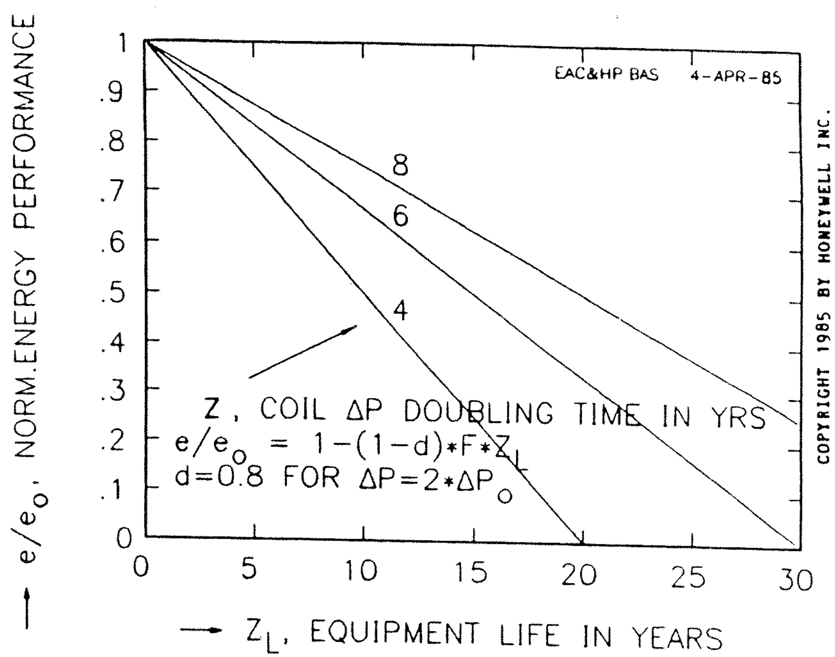


*Figure 2.* *Heat pump efficiency degradation vs. time due to particulate buildup on its indoor heat exchanger fins. Climate: Pittsburgh, PA, with about 3900 hours of annual heat pump operating time*

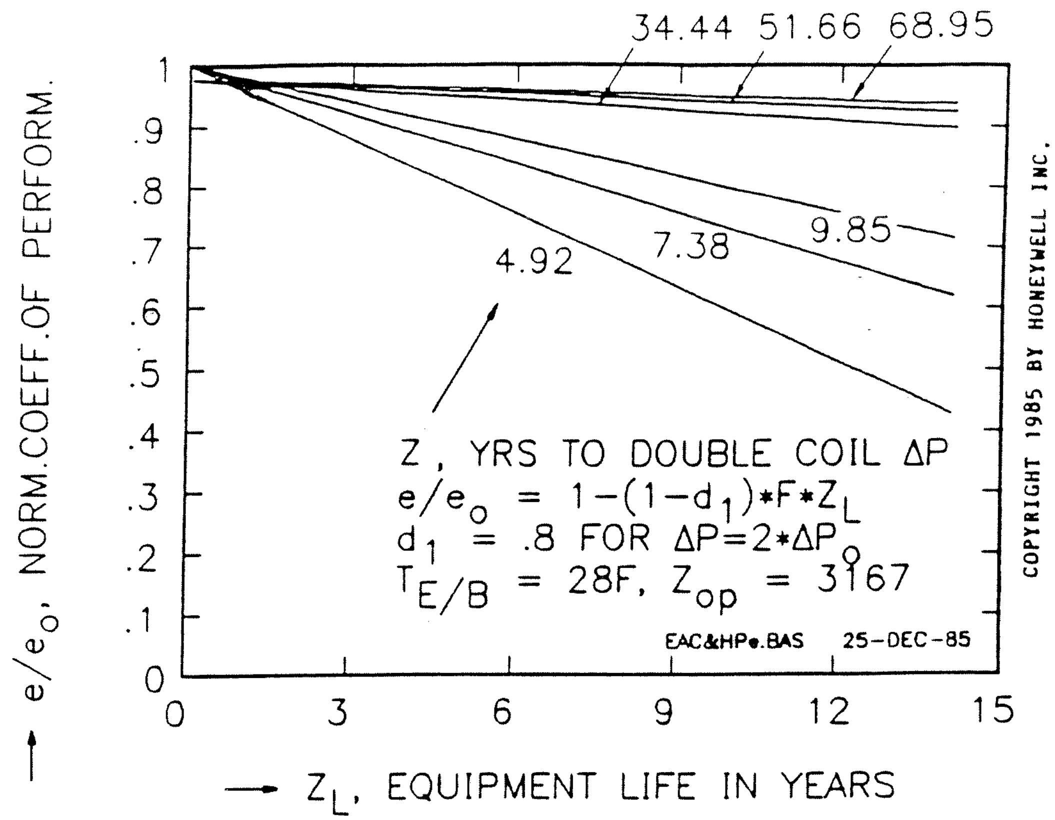


*Figure 3. Heat pump efficiency degradation vs. time due to particulate buildup. $B_g$ = 200 W/F, $L_H$ = 3500 Fd, $Z_C$ = 900 h, R = 7, $e_h$ = 2.2, $e_c$ = 2.64, $C_e$ = 4.8 c/kWh, $P_{EAC}/P_{HP}$ = .023, climate: southeastern United States*

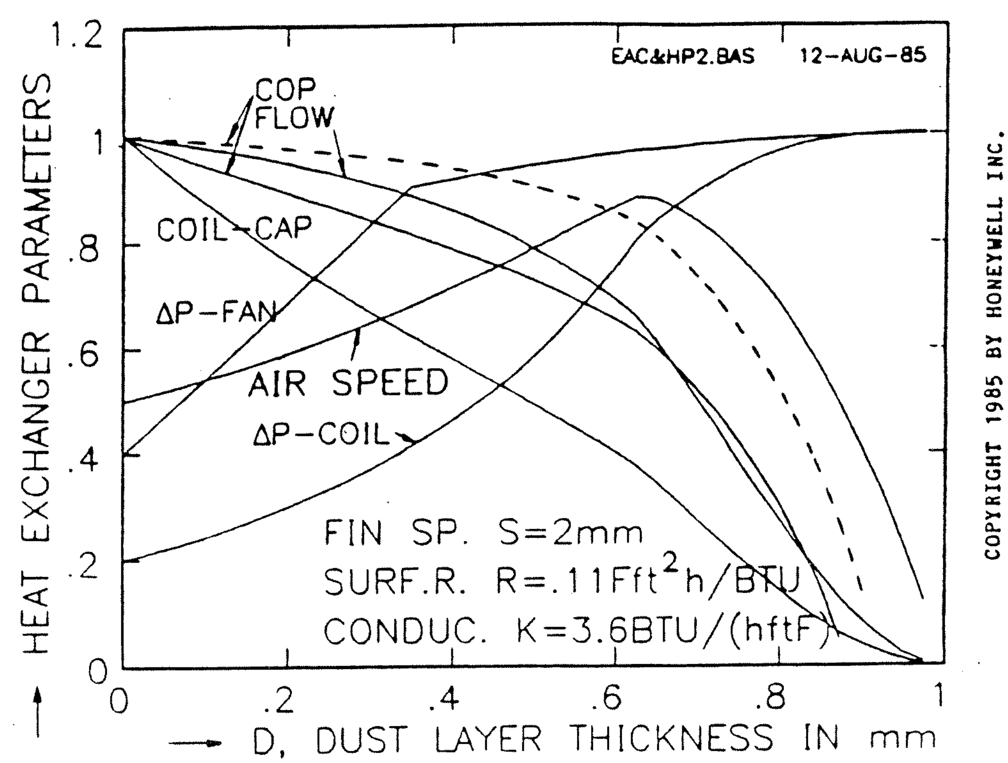


*Figure 4. Performance of a dust-loaded, finned, forced-air heat exchanger*

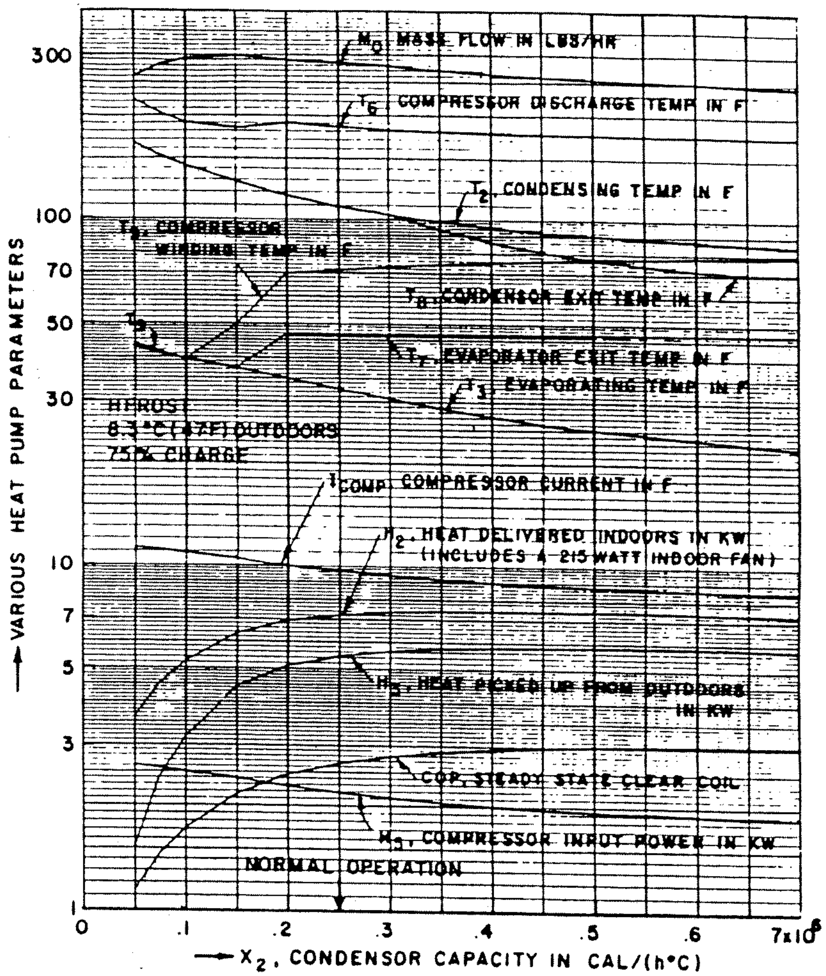


*Figure 5. Heat pump performance vs. condensor capacity (inverse of coil resistance) in the heating mode; capillary tube system*

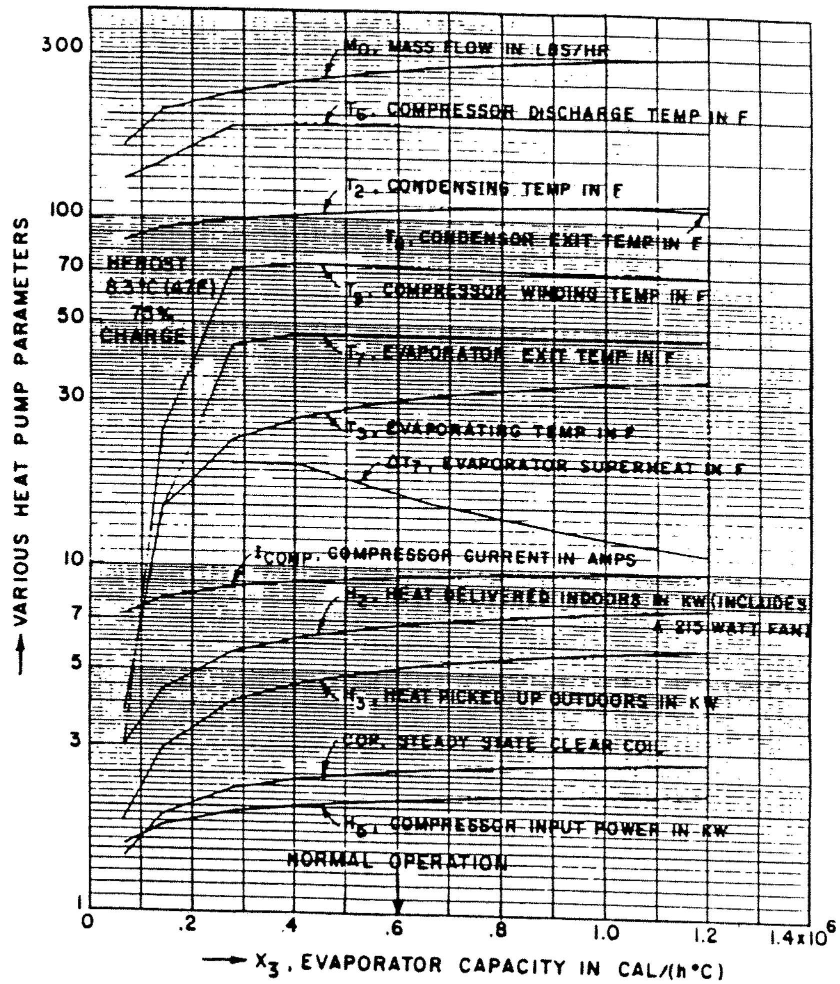


*Figure 6. Heat pump performance vs. evaporator capacity in the heating mode; capillary tube system*

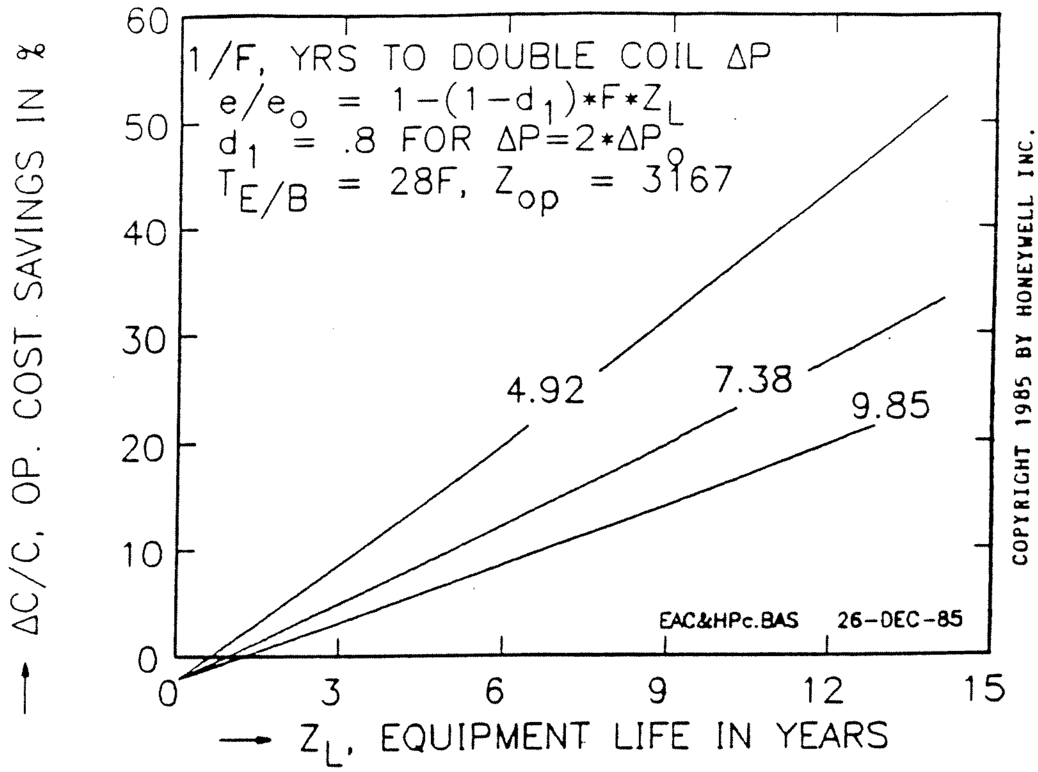


*Figure 7.* *Heat pump operating cost savings due to using a high-efficiency air cleaner. $B_g$ = 200 W/F, $L_H$ = 3500 Fd, $Z_C$ = 900 h, R = 7, $e_h$ = 2.2, $e_c$ = 2.64, $C_e$ = 4.8 c/kWh, $P_{EAC}/P_{HP}$ = .023*

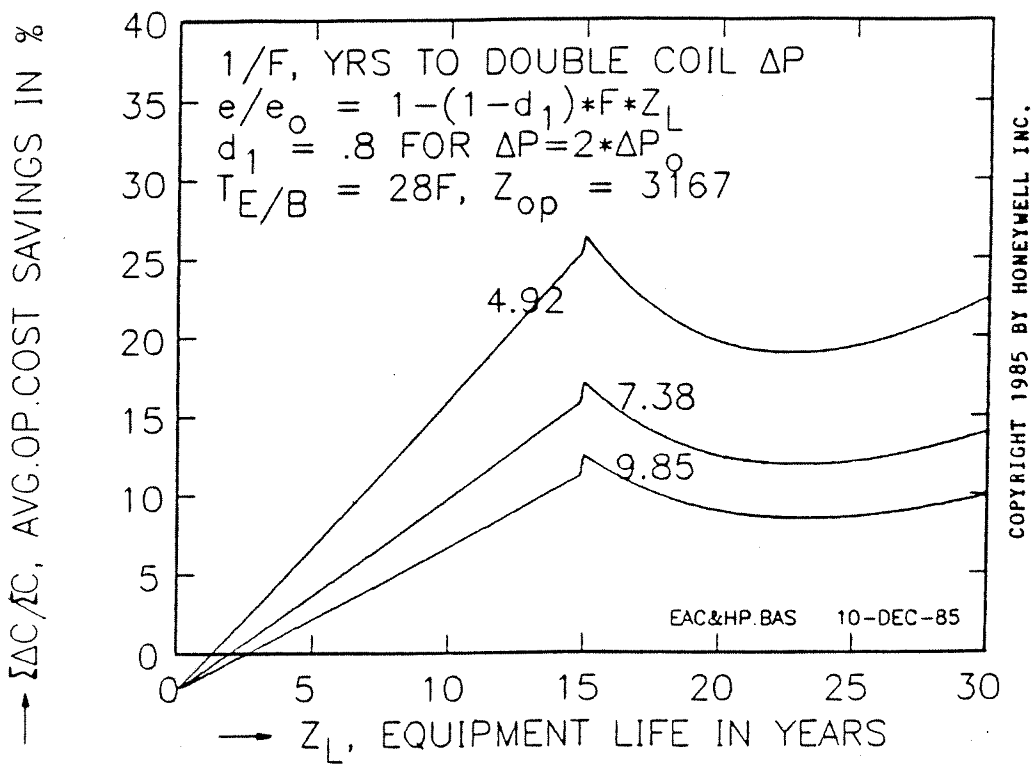


*Figure 8.* *Cumulative cost savings resulting from use of a high-efficiency EAC on a 2.5-ton heat pump. $B_g$ = 200 W/F, $L_H$ = 3500 Fd, $Z_C$ = 900 h, R = 7, $e_h$ = 2.2, $e_c$ = 2.64, $C_e$ = 4.8 c/kWh, $P_{EAC}/P_{HP}$ = .023*

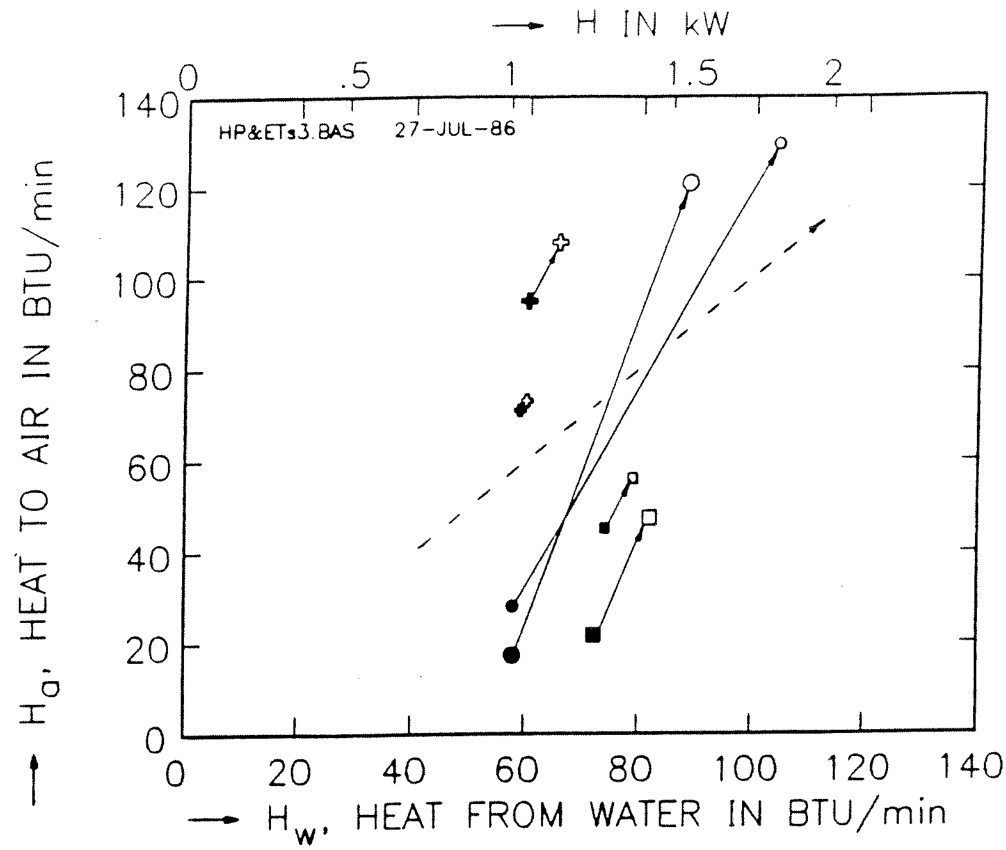


*Figure 9. A-coil heat transfer to airflow from hot water flow in the coil tubes. Airflow: large points at 1200 $ft^3$/min (2039 $m^3$/h), small points at 600 $ft^3$/min (1019 $m^3$/h); water flow: 4.94-6.59 lb/min (2.24-2.99 kg/min)*

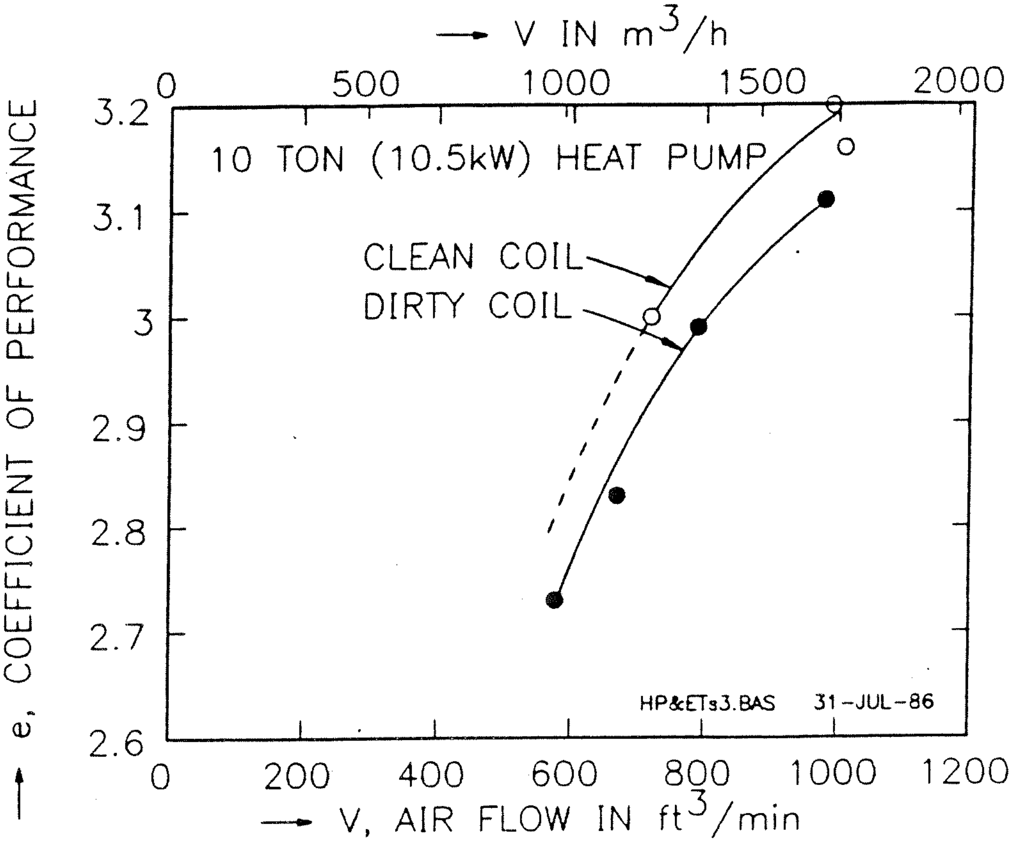


*Figure 10. Heat pump COP as a function of airflow and coil cleanliness*