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EXERGY ANALYSIS OF HEAT PUMPS

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ABSTRACT

The HPSIM computer program for modeling heat pumps (Domanski and Didion 1983) has been augmented to include exergy analysis. Exergy transports between the components, to the loads, and to the environment are evaluated. The exergy consumptions in each component are also determined, as well as each type of consumption in the heat exchangers. Typical results are presented for the operation of a nominal 3.5 ton residential air-to-air heat pump, in both the cooling and heating modes, at four loads. Among the conclusions drawn for this particular heat pump, it is predicted that overall economy would be improved by increasing the heat transfer surface of the indoor coil, say at the expense of outdoor coil surface.

INTRODUCTION

A general heat pump computer simulation model called HPSIM (Domanski and Didion 1983) has been enhanced with two subroutines called HP1 and HP2. HPSIM simulates the performance of an electric air-to-air heat pump at given outdoor and indoor air conditions. With the results of HPSIM, HP1 and HP2 calculate exergy rates and consumptions and second law efficiences at key locations throughout the system. Through these calculations, the true inefficiences in the system can be evaluated and analyzed.

HPSIM models the main components of a heat pump, which are:

1. compressor
2. indoor coil
3. outdoor coil
4. expansion device
5. four-way valve
6. accumulator
7. piping connecting main components

Configurations of such a system are illustrated in Figure 1 for the cooling and heating modes, respectively. Figure 1 displays the relevant locations in the system, which are:

1 – evaporator exit, suction pipe inlet
2 – suction pipe outlet, low-pressure four-way valve inlet
3 – low-pressure four-way valve exit, compressor can inlet
4 – inside compressor can
5 – compressor cylinder at suction
6 – compressor cylinder at discharge
7 – discharge manifold
8 – compressor can exit, high-pressure four-way valve inlet
9 – high-pressure four-way valve exit, discharge pipe inlet
10 – discharge pipe outlet, condenser inlet
11 – condenser outlet, liquid line inlet
12 – liquid line outlet, expansion device inlet
13 – expansion device outlet, evaporator inlet
A – indoor coil unit air inlet
B – indoor coil unit air outlet
C – outdoor coil unit air inlet
D – outdoor coil unit air outlet

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CALCULATION PROCEDURES

Exergy Analysis

The exergy rates of the refrigerant at locations 1 through 13 in Figure 1 are calculated by

$$x = m_r[h - T_o s - (h_o - T_o s_o)] \tag{1}$$

The dead state of the refrigerant is taken at ambient temperature, T_o, and at a pressure, P_o, equal to the saturation pressure of the refrigerant at T_o (Wepfer and Gaggioli 1980).

For calculating the exergies at locations A through D in Figure 1, the following equation for the exergy of a moist air mixture is used (Wepfer et al. 1979):

$$x = m_a C_{p,d} T_o \{[1.0 + 1.852W][T/T_o - 1.0 - \ln(T/T_o)] \tag{2}$$

$$+ 0.2857[1.0 + 1.6078W]\ln(P/P_o)$$

$$+ 0.2857\ln[(\frac{1.0 + 1.6078W_o}{1.0 + 1.6078W})^{1.0+1.6078W}(W/W_o)^{1.6078W}]\}$$

The dead state for the air is at ambient temperature, T_o, ambient pressure, P_o, and ambient humidity ratio, W_o.

Exergy consumptions were determined through steady-state exergy balances on each particular piece of hardware or process. For the compression process, an exergy balance yields

$$C_{proc} = W_{compressor} + x_5 - x_6 \tag{3}$$

For each combination of fan and coil, an exergy balance gives

$$C_{coil} = W_{fan} + x_{r,in} + x_{a,in} - x_{r,out} - x_{a,out} \tag{4}$$

The consumptions by the expansion device, accumulator, four-way valve, and the piping connecting the equipment are calculated by a simple exergy balance upon each:

$$C = x_{r,in} - x_{r,out} \tag{5}$$

The computer model HP2 calculates the above quantities. The flow chart for HP2 is given in Figure 2.

Computer model HP1 was developed to perform an extensive exergy analysis on the heat exchangers on a tube-by-tube basis. Among HP1's results used in this paper are the consumptions due to the heat transfer resistance and the refrigerant-side pressure drop.

The equation used to determine the consumption due to the heat transfer resistance is

$$C_{ht,t} = x_{a,t} - x_{r,t} \tag{6}$$

where

$x_{a,t}$ is the net rate of thermal exergy from the air around a tube;

$x_{r,t}$ is the net rate at which it is delivered to the refrigerant inside the tube.

Both exergy rates are calculated by the following relation of exergy rate to heat transfer:

$$x_{j,t} = [1 - T_o/T_{j,t}]q_{j,t} \tag{7}$$

where

j = a or r.

The net heat transfer rate per tube, $q_{j,t}$, is calculated by an energy balance on the refrigerant, yielding

$$q_{a,t} = m_{r,t}[h_{out,t} - h_{in,t}] \quad (8)$$

The temperature of the refrigerant, $T_{r,t}$, is calculated by taking the average of the inlet and outlet refrigerant temperatures for each tube. The temperature of the air around each tube is calculated with

$$T_{a,t} = R_t q_{a,t} + T_{r,t} \quad (9)$$

where

R_t is the overall heat transfer resistance of each tube.

The exergy consumption due to refrigerant-side pressure drop is calculated by

$$c_{r,dp,t} = c_t - c_{ht,t} \quad (10)$$

where

c_t is the total exergy consumed per tube.

This consumption is obtained from an exergy balance on each tube, yielding

$$c_t = m_{r,t}[h_{r,in} - T_o s_{r,in} - h_{r,out} + T_o s_{r,out}] + x_{a,t} \quad (11)$$

where

the refrigerant enthalpies, $h_{r,in}$ and $h_{r,out}$, and entropies, $s_{r,in}$ and $s_{r,out}$, are for each tube.

Equations 6 through 11 give results on a tube-by-tube basis. The tube-by-tube results are then summed for each entire coil.

The amount of exergy consumed at the evaporator for mass transfer, to extract H_2O from the air, is estimated by HP2 with

$$c_{mt} = c_{coil} - c_{total} - W_{fan} \quad (12)$$

where

c_{coil} is given by Equation 4 and

c_{total} is given by Equation 11 but summed for the entire evaporator.

The consumption associated with air-side pressure drop is evaluated by calculating the friction rate of the coil, f, and multiplying by (T_o/T_a) to get the rate of exergy consumption (Obert and Gaggioli 1963):

$$c_{a,dp} = \frac{T_o}{T_a} f = \frac{T_o}{T_a} v_a \Delta P_a \quad (13)$$

where the air pressure drop across the coil, ΔP_a, is obtained from manufacturers' specifications or from laboratory experiments.

Finally, the difference between the fan work and $c_{a,dp}$ is the rate of exergy consumption in the fan and its motor.

Second Law Efficiencies

The second law efficiences of the system are calculated by the following expressions in HP2:

1. Overall

$$\eta_{II\,overall} = \frac{x_B - x_A}{W_{fan,ind} + W_{fan,outd} + W_{compressor}} \quad (14)$$

2. Overall (excluding consumptions by fans)

$$\eta_{II_{\text{overall, w/o fans}}} = \frac{x_B - x_A}{c_{a,dp,ind} + c_{a,dp,outd} + w_{compressor}} \tag{15}$$

3a. Indoor coil (heating mode)

$$\eta_{II_{\text{icoil}}} = \frac{x_B - x_A}{x_{10} - x_{12} + c_{a,dp,ind}} \tag{16a}$$

3b. Indoor coil (cooling mode)

$$\eta_{II_{\text{icoil}}} = \frac{x_B - x_A}{x_{13} - x_1 + c_{a,dp,ind}} \tag{16b}$$

4a. Compressor, associated hardware, four-way valve, piping to coils, outdoor coil, and expansion device (heating mode)

$$\eta_{II_{\text{comp, 4-way, ocoil, device, pipe}}} = \frac{x_{10} - x_{11}}{w_{compressor} + c_{a,dp,outd}} \tag{17a}$$

4b. Compressor, associated hardware, four-way valve, piping to coils, outdoor coil, and expansion device (cooling mode)

$$\eta_{II_{\text{comp, 4-way, ocoil, device, pipe}}} = \frac{x_{13} - x_1}{w_{compressor} + c_{a,dp,outd}} \tag{17b}$$

5a. Compressor, associated hardware, four-way valve, piping to coils, and outdoor coil (heating mode)

$$\eta_{II_{\text{comp, 4-way, ocoil, pipe}}} = \frac{x_{10} - x_{13}}{w_{compressor} + c_{a,dp,outd}} \tag{18a}$$

5b. Compressor, associated hardware, four-way valve, piping to coils, and outdoor coil (cooling mode)

$$\eta_{II_{\text{comp, 4-way, ocoil, pipe}}} = \frac{x_{11} - x_1}{w_{compressor} + c_{a,dp,outd}} \tag{18b}$$

6. Expansion device

$$\eta_{II_{\text{device}}} = \frac{\eta_{II_{\text{comp, 4-way, ocoil, device, pipe}}}}{\eta_{II_{\text{comp, 4-way, ocoil, pipe}}}} \tag{19}$$

7. Compressor, associated hardware, four-way valve, and piping to coils

$$\eta_{II_{\text{comp, 4-way, pipe}}} = \frac{x_{10} - x_1}{w_{compressor}} \tag{20}$$

8. Outdoor coil

$$\eta_{II_{\text{ocoil}}} = \frac{\eta_{II_{\text{comp, 4-way, ocoil, pipe}}}}{\eta_{II_{\text{comp, 4-way, pipe}}}} \tag{21}$$

9. Compressor and associated hardware

$$\eta_{II_{comp}} = \frac{x_8 - x_3}{w_{compressor}} \tag{22}$$

10. Four-way valve and piping to coils

$$\eta_{II_{4\text{-way, pipe}}} = \frac{\eta_{II_{comp,\ 4\text{-way, pipe}}}}{\eta_{II_{comp}}} \tag{23}$$

11. Fans

$$\eta_{II_{fan}} = \frac{f}{w_{fan}} \tag{24}$$

where

f is from Equation 13.

SUMMARY OF RESULTS

Table 1 shows the indoor and outdoor conditions for the four cases that have been analyzed. Exergy flow diagrams are given by Figures 4 through 7, for each of the cases, and Figures 8 and 9 give more detailed diagrams for one cooling case and one heating case. The principal exergy consumptions and component efficiencies are presented in Table 2, while Table 3 gives a further breakdown of the consumptions associated with the indoor and outdoor coils.

Consider first the relative exergy consumptions and efficiencies of the different devices. From Table 2, it is clear the the largest consumption is in the compressor—consuming about half the electric exergy supplied to it.

Except for the 17F outdoor case, the second largest consumption is associated with the heat transfer of the indoor coil, about one-sixth of the system's total.

The indoor and outdoor fans, with their motors, have the next largest exergy consumptions, about one-sixth and one-tenth of the total, although, during summer operation, the heat transfer consumption and losses of the outdoor coil are greater than the fan consumption.

The outdoor coil eliminates about a sixth of the total exergy consumption in summer, but only 4% to 7% in winter.

The throttling device consumes roughly 5% of the total, and the four-way valve about 2% or 3%. These two devices have the highest second law efficiencies.

For the coils, as shown by Table 3, by far the largest of the consumptions affiliated with heat transfer is that associated with the heat transfer resistance. The consumption from the refrigerant-side pressure drop and that from the air-side are negligible in comparison. During cooling operation, the consumption of exergy for mass transfer—for extracting H_2O from the air—is about one-third of that for heat transfer. In essentially every case, the consumption of the fan and motor exceeds that of heat transfer.

In general then, the hardware that consume the most exergy are the electro-mechanical devices, that is, the compressor and the fans.

Of the major components, the outdoor coil has the highest efficiency. Second is the compressor. Next is the indoor coil; its efficiency varies greatly with load and exceeds that of the compressor under some conditions. The fans, with values less than 5%, are the least efficient.

Next, consider the variation of efficiencies with outdoor conditions. The overall efficiency decreases as the difference between outdoor and indoor temperature decreases in magnitude. When that difference is low, so is the thermal exergy load on the system since $[1-T_o/T_{room}]$ is so small, while the total amount of exergy consumed by the hardware is about the same.

The columns of Table 2 are arranged so that $T_{cond}-T_{evap}$ is less for each succeeding column. The exergy consumption in the indoor coil becomes greater in each succeeding column because of the increased mass flow rate of refrigerant—leading to higher heat transfer resistance associated with the higher rate of heat transfer. For the 82F case, the indoor coil efficiency is very low, again because the exergy output of the evaporator is very low; since $[1-T_o/T_{room}]$ is so small, the sensible exergy output is practically nil. The principal output is humidity reduction for meeting the latent load. Furthermore, while the exergy load is lower, the system capacity is increased. Hence, the temperature difference between the air and the refrigerant is greater, causing greater exergy consumption from heat transfer resistance.

On the other hand, for the 17F case, the indoor coil efficiency is very high, as a result of the high value of $[1-T_o/T_{room}]$ and the low value of heat transfer rate. (That is, the heat flux [rate per unit of transfer area] is low.) Because of the low value, the temperature difference for heat transfer between refrigerant and indoor air is at its lowest value, leading to relatively low exergy consumption.

The outdoor coil has a higher efficiency than the indoor coil in every case, even while the outdoor coil is penalized not only for exergy consumption but also for the exergy loss in the discharge air.

The throttle consumption is more or less proportional to the product of flow rate and pressure differential; the former increases and the latter decreases for each succeeding column in Table 2.

DISCUSSION

Consider each device, in order of ascending exergy consumption.

Four-way Valve and Throttle

The four-way valve has what appears to be a negligible exergy consumption, accepted in order to gain convenience and flexibility in changing from the cooling mode to heating. The same comments hold for the throttling device.

Outdoor Coil and Fan/Motor

The total heat transfer resistance, with its large consumption, is primarily due to the low air-side heat transfer coefficient. The air-side pressure drop consumption, which is equal to the mechanical exergy of the air delivered to the coil by the fan, is much lower than the consumption from heat-transfer resistance. It appears, therefore, that the pressure-drop consumption should be increased, in order to lower the air-side heat-transfer consumption. For examples, the air velocity through the coil could be increased; fin shape and geometry could be changed to enhance turbulence; fin surface area could be increased.

However, such a change might be counterproductive unless the efficiency of the fan-and-motor combination, which is very low, were improved.

The conclusion then is that the prospect of improving overall system economics—by investing more capital in the outdoor unit for air-side heat transfer enhancement and for a better fan/motor—should be investigated. The same comments may be made for the indoor coil.

In any case, comparing the consumptions (and efficiencies) of the inside and outside coils, it is apparent that they are not optimal relative to each other. The indoor coil should be improved and/or the investment in the outdoor coil should be reduced. It is the indoor coil that transfers exergy from the system to the load. The remainder of the system, then, is subservient to the indoor coil; that is, its purpose is to deliver exergy to the indoor coil, in order to meet the load. For each unit of exergy consumed in the indoor coil, approximately four units are consumed in the rest of the system. Whereas, for every unit consumed or lost at the outdoor coil, about two units are consumed in the subsystem—compressor and four-way valve—supplying it. That is, exergy consumed in the indoor coil is more significant than that consumed in the outdoor coil. Yet, the outdoor coil is more efficient and has less consumption. Therefore, capital expenditure (say heat transfer surface) should be increased for the indoor coil and/or reduced for the outdoor coil. (Increasing the surface area of the indoor coil will cause the evaporator temperature to be higher. Therefore, the humidity of the air will be decreased less, although its temperature will be decreased more. It is likely that, with the higher humidity, occupants of the space would set the thermostat lower. The compressor might then run more. It is conceivable that overall economics would not improve.)

Of course, because of their large exergy consumptions, it will always be important to seek cost-effective ways of improving compressors. However, for the system at hand, the compressor efficiency is quite reasonable, except at the extreme operating conditions.

(Also, recall the very low second law efficiencies, overall and for the indoor coil, when the outdoor temperature is close to the room temperature. For cases when an air-conditioning system must run primarily under such conditions, say to meet latent and/or internal sensible loads, alternatives to the conventional single-speed vapor-compression system are worth developing--confirming the usefulness of ongoing R&D efforts devoted to such cases.)

CONCLUSION

A second law analysis has been presented for a typical residential heat pump, showing the breakdown of inefficiencies by subsystem and by device. It is believed that the results provide the engineer with a better, quantitative grasp of the inefficiencies and their relative magnitudes.

Furthermore, the results have motivated several recommendations for improving the overall system. Inasmuch as the ultimate test of prospective improvements is economic, the recommendations need to be followed up with analyses investigating the trade-offs between the decreases (or increases) in power costs versus the increases (or decreases) in fixed costs.

The results were developed via subroutines, HP1 and HP2, which augment the HPSIM simulation routine (which incorporates entropy calculations). Of course, other programs for simulating heat pumps (e.g., Jeter et al. 1986) could be enhanced in a similar manner.

NOMENCLATURE

- C = specific heat
- c = exergy consumption rate
- f = friction rate
- h = enthalpy
- m = mass flow rate
- P = pressure
- R = overall heat transfer resistance
- s = entropy
- T = temperature
- $\bar{T}$ = average temperature
- v = volumetric flow rate
- w = work input rate
- W = humidity ratio
- x = exergy rate

Subscripts

- a = air
- A-D = air key locations in the heat pump, as per Figure 1
- coil = coil
- comp = compressor and associated hardware
- compressor = compressor
- cond = condenser
- d = dry air basis
- device = expansion device
- dp = pressure drop
- evap = evaporator
- fan = fan and motor
- ht = heat transfer resistance
- icoil = indoor coil
- in = inlet
- ind = indoor
- j = component
- o = dead state
- ocoil = outdoor coil
- out = outlet
- outd = outdoor
- overall = overall
- mt = mass transfer
- p = constant pressure

pipe	=	piping from four-way valve to coils
proc	=	compression process
r	=	refrigerant
room	=	indoor
t	=	tube
w/o fans	=	excluding fan consumptions
II	=	second law
1-13	=	refrigerant key locations in the heat pump, as per Figure 1
4-way	=	four-way valve

Greek Letters

η = efficiency
Δ = change

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TABLE 1
Indoor and Outdoor Air Conditions for Loads Analyzed

(Indoor and outdoor air pressures are all 14.7 psia)

HEAT PUMP MODE	OUTDOOR AIR TEMPERATURE (F)	INDOOR AIR TEMPERATURE (F)	OUTDOOR AIR RELATIVE HUMIDITY	INDOOR AIR RELATIVE HUMIDITY
Heating	17	70	0.70	0.56
Heating	47	70	0.70	0.56
Cooling	82	80	0.40	0.51
Cooling	95	80	0.40	0.51

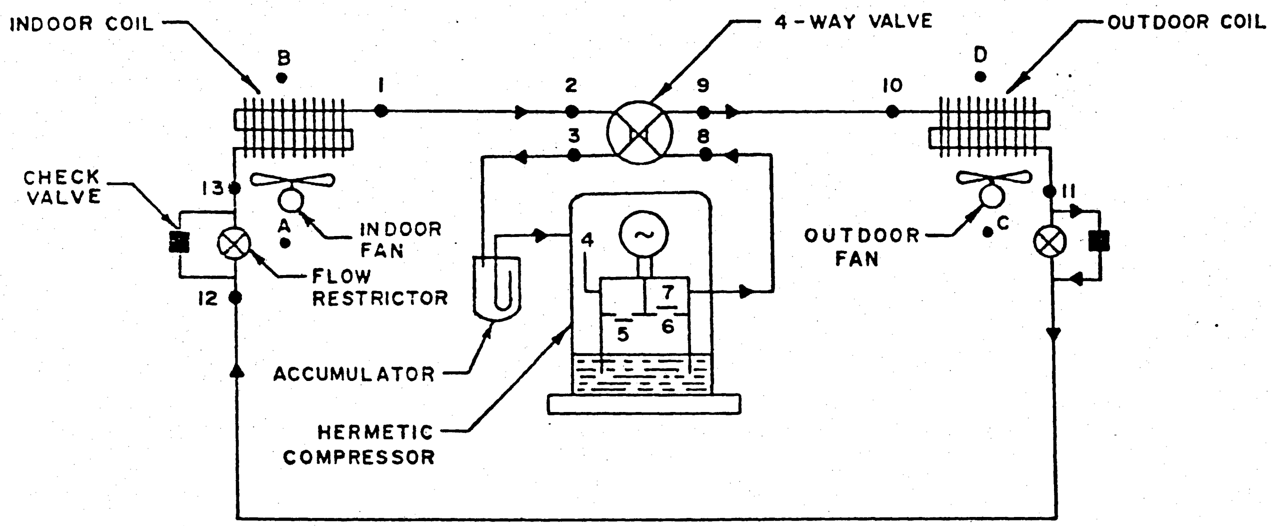
TABLE 2
Second Law Consumptions Plus Losses (and Efficiencies)

("Compressor" includes associated hardware;
"4-way Valve" includes piping to coils)

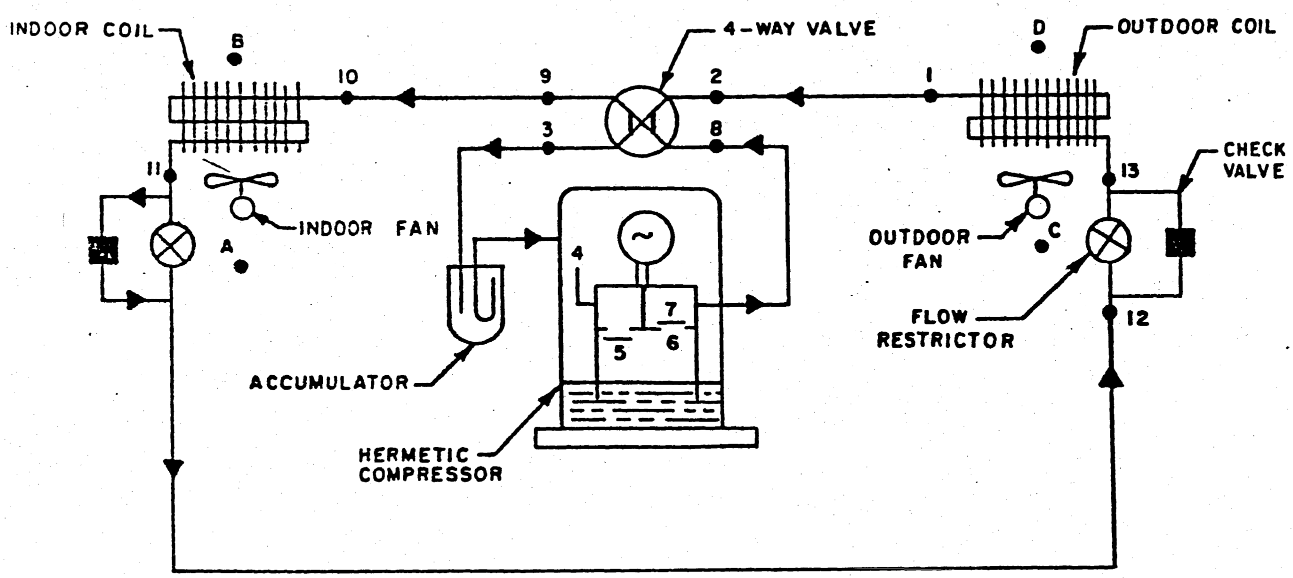
	17F	47F	95F	82F
Electricity Use	[5.28] hp	[6.51] hp	[6.81] hp	[6.49] hp
System Load	[1.15]	[1.17]	[0.58]	[0.22]
Overall	4.13+.01 (0.217)	5.26+.07 (0.181)	5.97+.26 (0.0843)	5.99+.28 (0.0340)
Overall w/o Fans	2.65 (0.301)	3.78 0.234)	4.51 (0.107)	4.52 (0.0439)
Indoor Coil	0.24 (0.826)	0.89 (0.570)	1.05 (0.353)	1.17 (0.158)
Compressor, 4-way Valve, Outdoor Coil, Throttle	2.40 (0.360)	2.89 (0.408)	3.46 (0.298)	3.35 (0.271)
Compressor, 4-way Valve, Outdoor Coil	2.10 (0.440)	2.55 (0.477)	3.17 (0.352)	3.17 (0.309)
Throttle	0.31 (0.816)	0.35 (0.855)	0.29 (0.845)	0.19 (0.877)
Compressor, 4-way Valve	1.95 (0.479)	2.24 (0.550)	2.44 (0.537)	2.42 (0.513)
Outdoor Coil	.15+.01 (0.920)	.31+.07 (0.869)	.73+.26 (0.656)	.75+.28 (0.602)
Compressor	1.85 (0.505)	2.06 (0.585)	2.20 (0.583)	2.19 (0.559)
4-way Valve	0.10 (0.947)	0.18 (0.939)	0.24 (0.921)	0.23 (0.918)
Outdoor Fan & Motor	0.65 (0.0441)	0.65 (0.0441)	0.65 (0.0441)	0.65 (0.0441)
Indoor Fan & Motor	0.83 (0.0349)	0.83 (0.0349)	0.81 (0.0581)	0.82 (0.0465)

TABLE 3
Exergy Consumptions in the Heat Exchangers

CONSUMPTION	17F OUTDOOR TEMPERATURE (HEATING)	47F OUTDOOR TEMPERATURE (HEATING)	95F OUTDOOR TEMPERATURE (COOLING)	82F OUTDOOR TEMPERATURE (COOLING)
INDOOR COIL:				
Heat Transfer Resistance	0.33 hp	0.92 hp	0.67 hp	0.77 hp
Refrigerant-Side Pressure Drop	~ 0	~ 0	0.07	0.08
Air-Side Pressure Drop	0.03	0.03	0.04	0.04
Mass Transfer	****	****	0.26	0.28
Fan and Motor	0.83	0.83	0.81	0.82
OUTDOOR COIL:				
Heat Transfer Resistance	0.06	0.17	0.64	0.66
Refrigerant-Side Pressure Drop	0.04	0.08	0.01	0.01
Air-Side Pressure Drop	0.03	0.04	0.04	0.04
Mass Transfer	0.01	0.02	****	****
Fan and Motor	0.65	0.65	0.65	0.65



a. cooling mode



b. heating mode

Figure 1. Schematic of heat pump

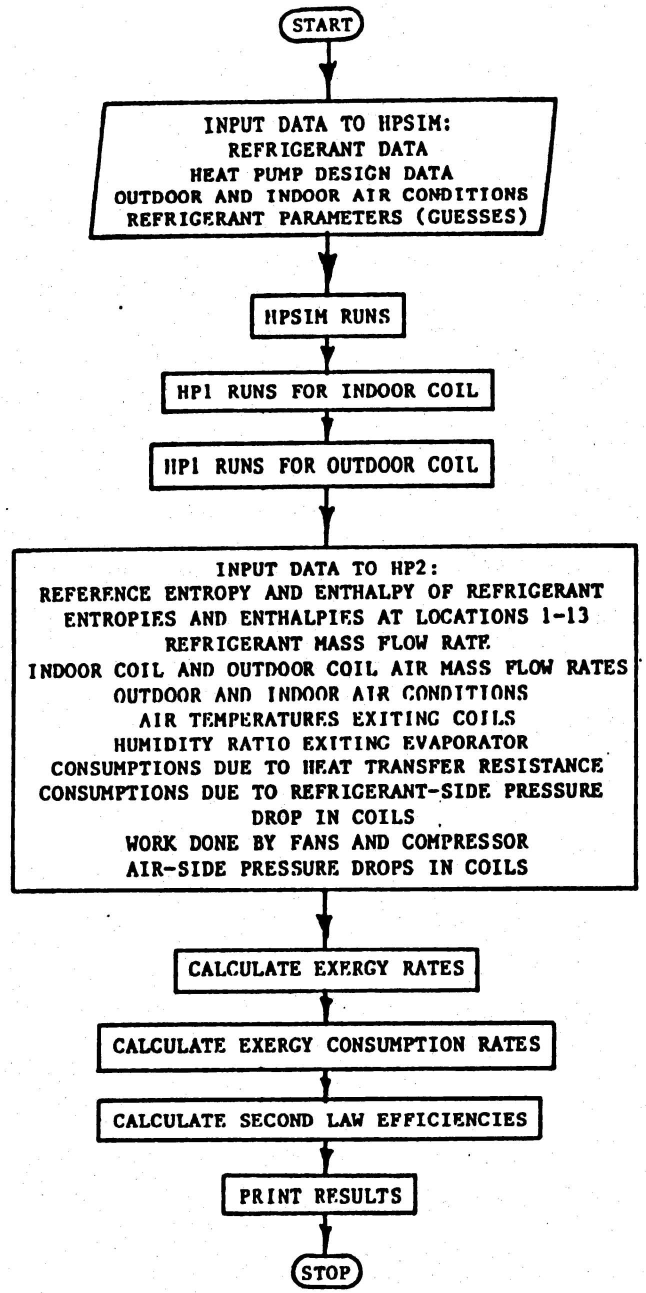


Figure 2. Flow chart for model HP2

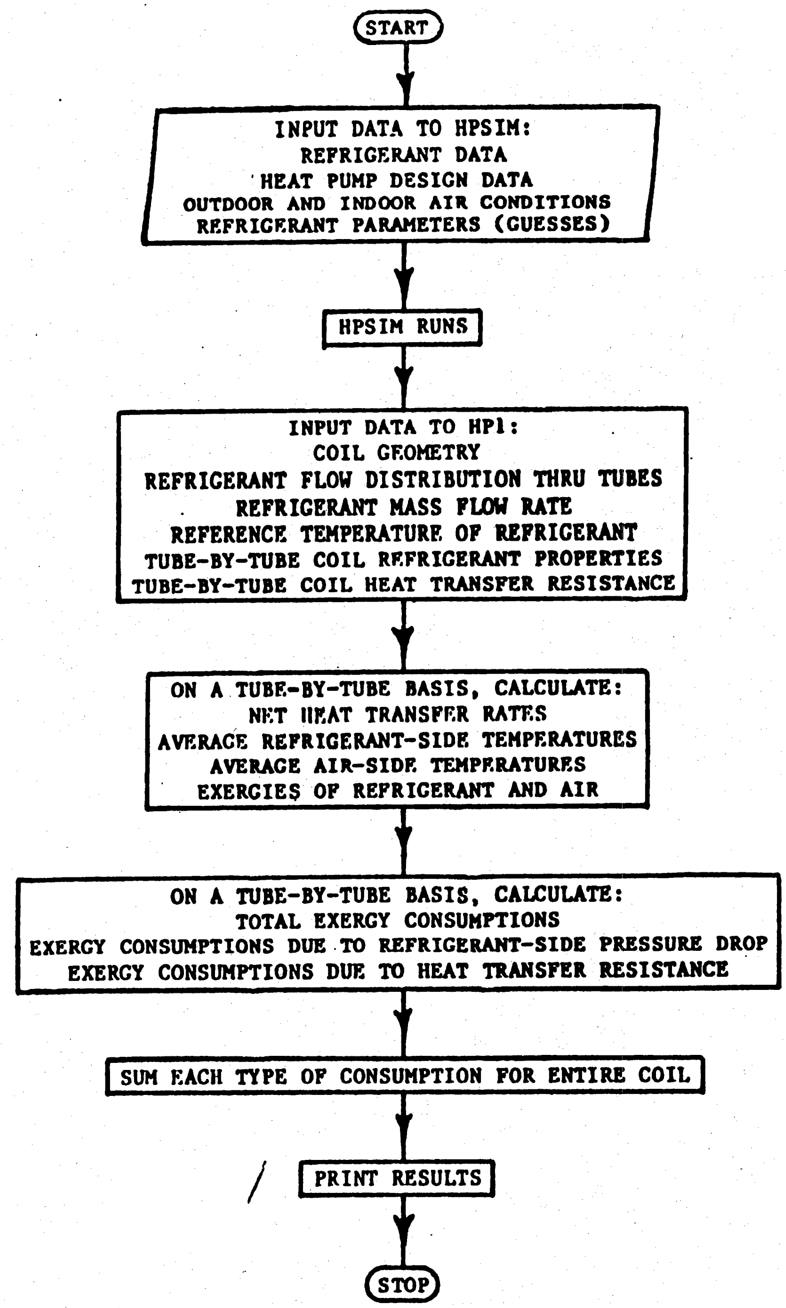


Figure 3. Flow chart for model HP1

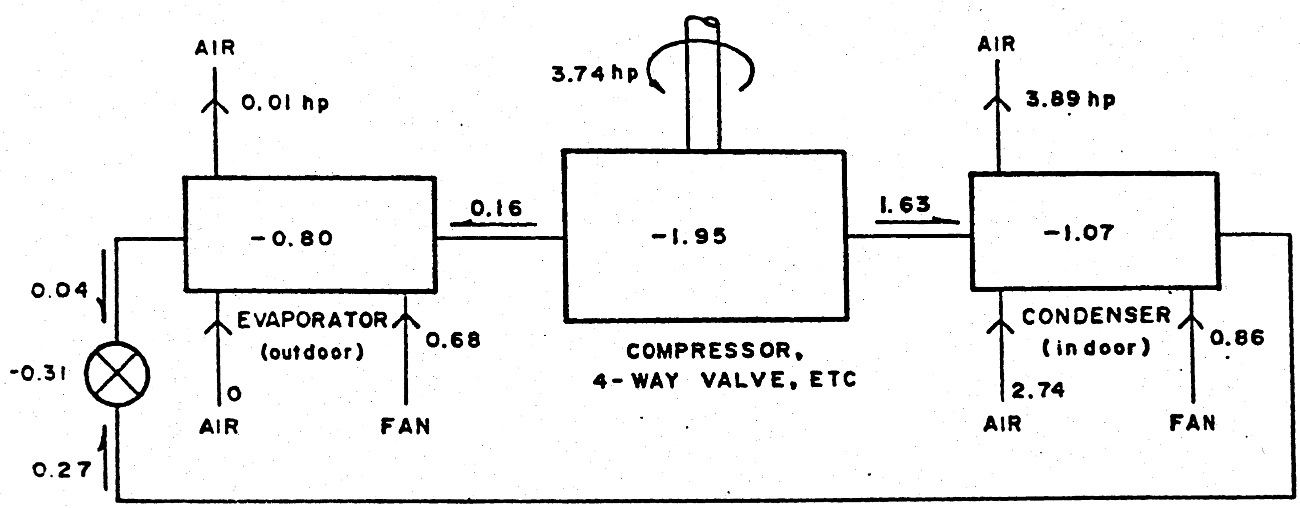


Figure 4. Exergy flow diagram for 17 F outdoor temperature, heating mode

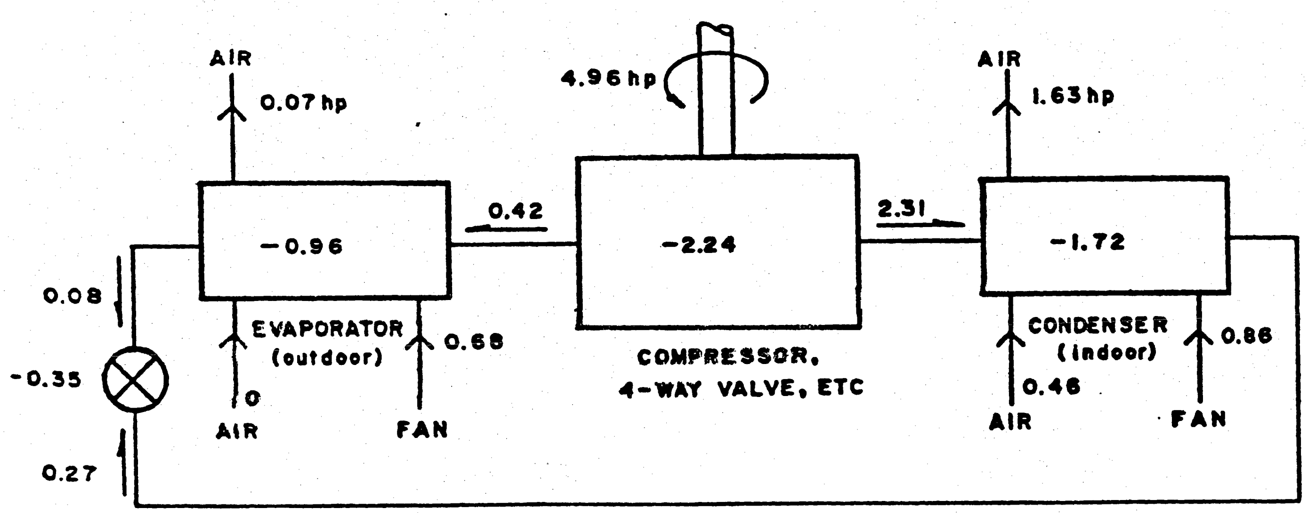


Figure 5. Exergy flow diagram for 47 F outdoor temperature, heating mode

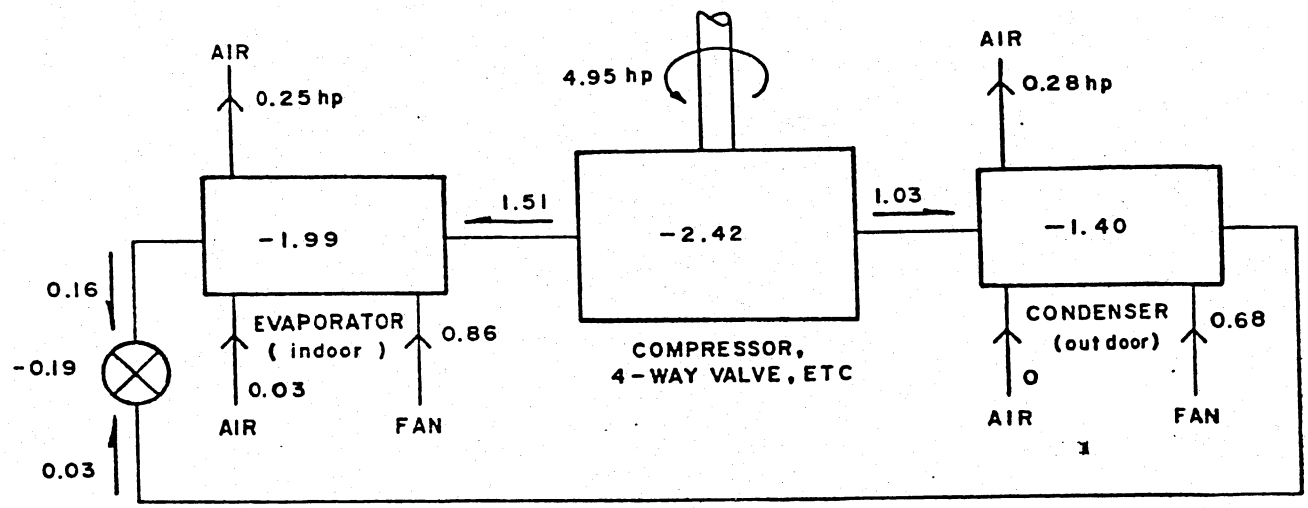


Figure 6. Exergy flow diagram for 82 F outdoor temperature, cooling mode

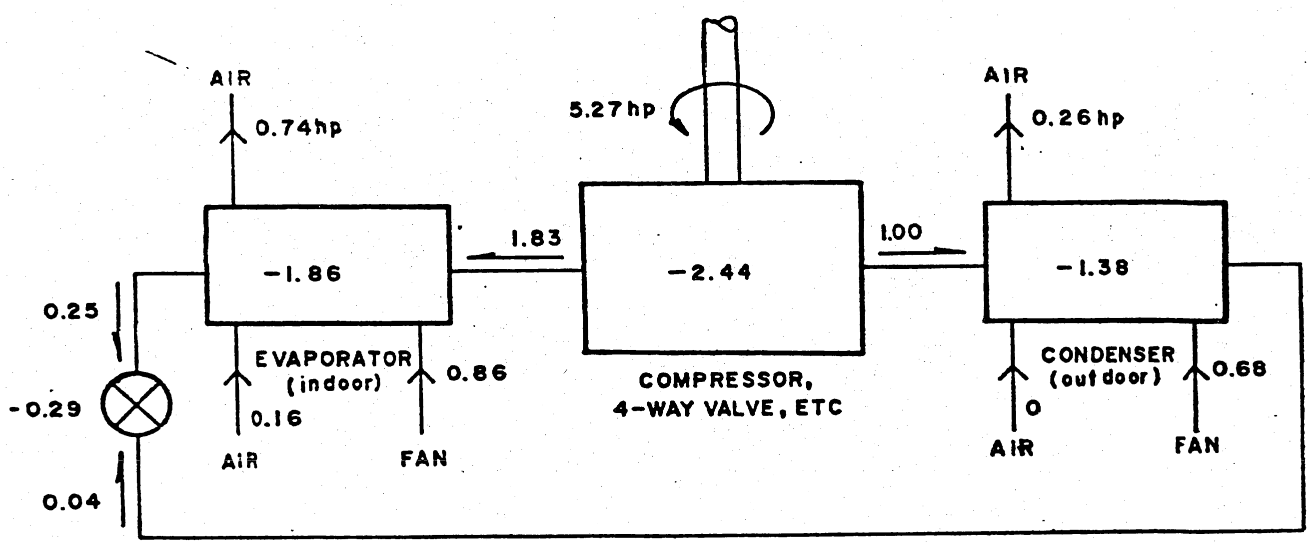


Figure 7. Exergy flow diagram for 95 F outdoor temperature, cooling mode

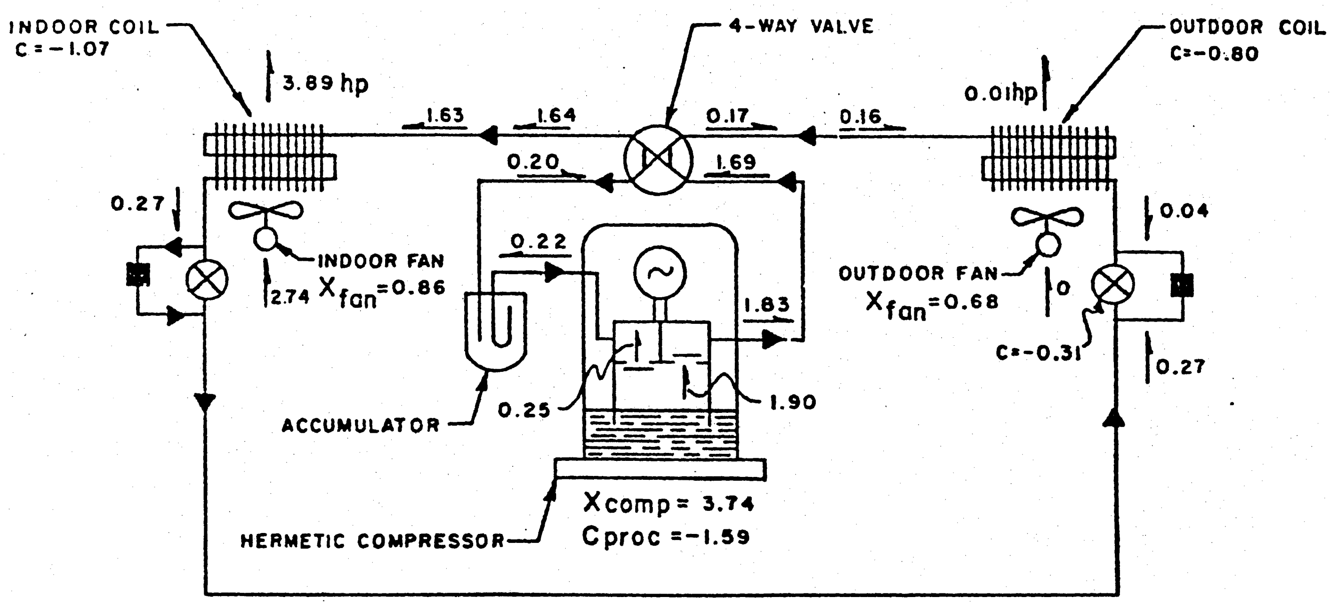


Figure 8. Detailed exergy flow diagram for 17 F outdoor temperature, heating mode

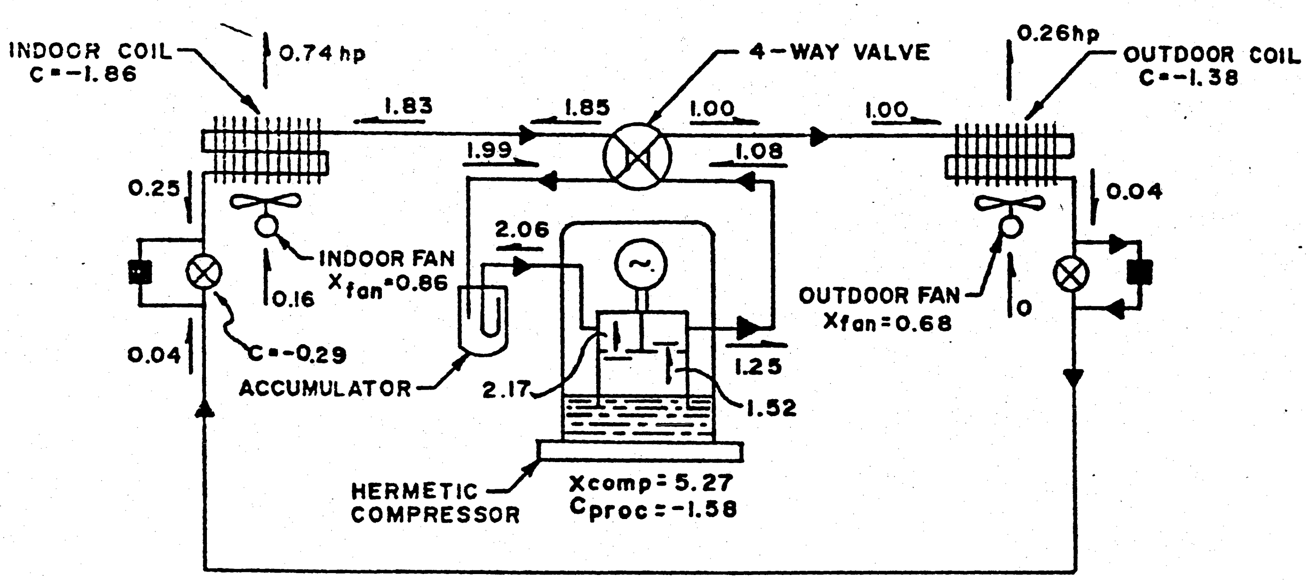


Figure 9. Detailed exergy flow diagram for 95 F outdoor temperature, cooling mode

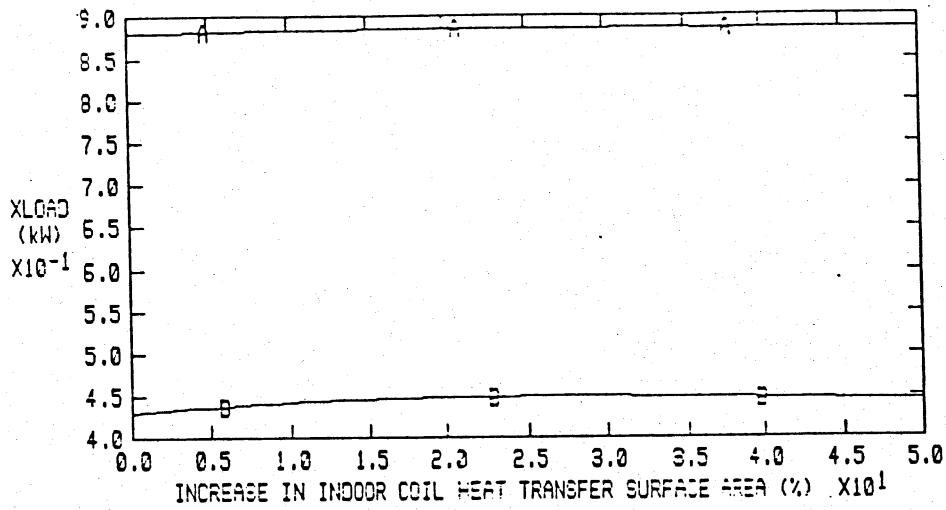


Figure A

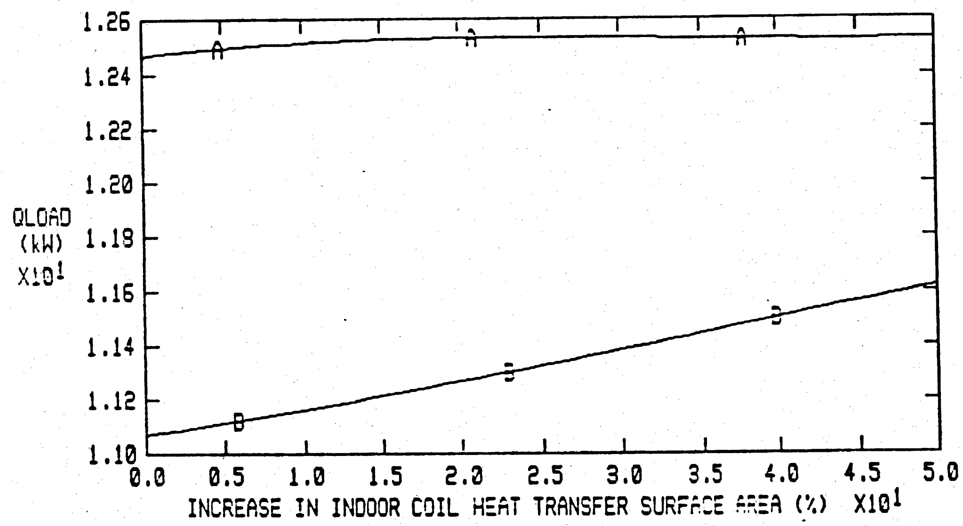


Figure B

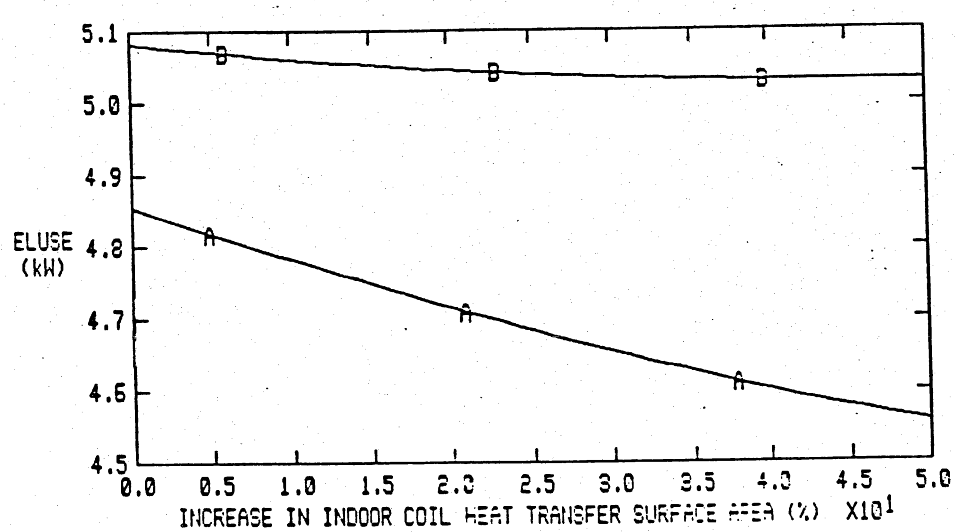


Figure C

Discussion

W. KOEHLER, University of Minnesota, Minneapolis: Which of the conclusions drawn could not have been drawn on a basis of the first law only?

GAGGIOLI: In theory, every conclusion could have been drawn without second law analysis. That is, the system could be modelled in the conventional fashion, using air, water and energy balances, property relations and performance characteristics of each piece of hardware--along with "boundary conditions" imposed by the load and the environment (satisfying the second law implicitly). By trial-and-error manipulation of the parameters of the model, the conclusions we have drawn could have been found. The more important question is not if they could but if they would have been found. The conclusions deduced via second law analyses could always be determined in another way. However, we believe that second law analyses lead to the conclusions more efficiently. For the case at hand, it is doubtful that the conclusions drawn from the second law analysis would have been found by trial-and-error modeling studies.

J.C. CHATO, University of Illinois, Urbana: Why does the exergy flow from the compressor to the evaporator?

GAGGIOLI: Although mass flows from the evaporator to the compressor, the specific exergy is negative, so the exergy flows in the opposite direction. Fundamentally, that is to be expected. Exergy is potential to cause change. The exergy input to the system is the electricity supplied to the compressor motor. The motor converts it from electrical to mechanical for; in turn, the compressor sends exergy to the evaporator (and the condenser), where a desired change is accomplished--the cooling of the air flowing over the evaporator coil (and the heating of the air flowing over the condenser).

ADDENDUM

Since the paper was written, various parametric studies were conducted with HPSIM to test the validity of the recommendations. One such study included removing heat transfer surface area from the outdoor coil and placing it in the indoor coil (an even swap). The outdoor coil air flow rate was decreased proportionately to the decrease in outdoor coil surface area--decreasing fan work, while the airflow through the indoor coil was held constant--decreasing fan work, also.

The results are shown in Figures A, B, and C. Figure A is a plot of exergy output, Figure B is a plot of capacity, and Figure C is a plot of electricity consumption. All curves are plotted with respect to the percent increase in indoor coil heat transfer surface area. The curves tagged "A" refer to a heating operation at 47 F, and the curves tagged "B" refer to a cooling operation at 95 F.

As can be seen, the results are positive. In both modes of operation, the capacity improves and the electricity consumption decreases. It is interesting to note that for the heating case, the capacity of the heat pump remains essentially constant while the electricity use decreases. Whereas for the cooling case, the capacity increases while the electricity consumption remains quite flat, decreasing slightly.

Regarding the cooling case (Curve B), it can be seen from Figures B and A that, while the capacity for heat removal increases steadily, the exergy output peaks out. Because, the increased cooling output consists of a rise in the sensible load carried, but a decrease in the latent load. The expression for exergy accounts for the relative value of these two changes, signaling the importance of the drop in system sensible heat ratio.

From the three figures presented, it may be concluded that the recommendations tested--drawn from exergy analysis--are valid.