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SUBJECT: Energy and Cost Analysis of Room Air Conditioners

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ABSTRACT

A production-possibilities curve for room air conditioners in the 8000-Btu/hr-capacity range was constructed by upgrading a unit from an energy-efficiency ratio (EER) of 6.3 to 9.8 Btu/Whr through energy-saving improvements in heat exchangers, compressor, and fan. An energy consumption and price-estimation model was developed. An initial increase of 30% in EER is possible using four-pole compressor and permanent split capacitor fan motors at an 8% price increase. Increasing condenser frontal area by 40%, doubling the number of coil rows, and replacing cooling fins with spines decrease energy consumption by 35% with a 40% increase in price.

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1. SUMMARY

The energy consumption and initial cost of a variety of energy-saving design alternatives for room air conditioners (RACs) were evaluated. An 8000-Btu/hr-baseline model with an energy-efficiency ratio (EER) of 6.3 Btu/Whr was chosen. An energy-estimation model was constructed to provide a simple and flexible method for predicting energy consumption and cooling capacity over a range of ambient temperatures.

Energy-saving improvements included a 50% increase in condenser frontal area, the replacement of cooling fins by spines, an increase of spine density by 30%, an increase in the number of coil rows from 3 to 6, and a decrease of condenser air flow rate by 30%. Use of more efficient four-pole compressor motors and permanent split capacitor fan motors were also evaluated. These improvements yielded a 9.8 EER unit, close to the maximum EER reported for this capacity.

The cost of these improvements was based on major component cost increases. Increasing fan and compressor motor efficiencies would improve EER by 30% for an additional cost of 8%. Further combinations of these improvements with increases in condenser and evaporator dimensions can bring EER up by 53% for a cost increase of 40%.

A production possibilities curve was constructed for the range of technological alternatives. The payback period for a 9.8 EER unit with respect to the baseline unit was calculated to be 3.6 yr for New York City and 2.6 yr for Miami based on electricity prices of 7.3 and 3.5¢/kWh, respectively.

2. INTRODUCTION

2.1 Motivation

The growing public interest in energy conservation has encouraged the study of energy-efficient technological improvements for household appliances. Energy-policy makers need information to evaluate the feasibility and cost of energy-efficient targets. An engineering-economic model was developed at ORNL to evaluate residential energy consumption (6). This model uses a production-possibilities curve for each household appliance as input. The major objective of this project was to construct such a correlation between energy consumption and initial price for room air conditioners.

2.2 Background

Room air conditioners (RACs) perform four major functions: (1) cool room air by transporting room heat to the ambient (outside) air, (2) dehumidify room air to enhance comfort, (3) circulate room air to provide uniform cooling, and (4) filter room air to remove solid impurities, such as pollen and dust. Additional features may include: (1) ventilation, i.e., exchanging inside for outside air, (2) multiple indoor fan speeds, (3) continuous fan operation for room air circulation and purification during non-cooling periods, and (4) thermostatic temperature control.

RACs are presently rated according to their cooling capacities (Q) and their energy-efficiency ratios (EER). Cooling capacity is the maximum rate (Btu/hr) at which a unit can remove room heat under standard conditions of 95°F ambient temperature, 80°C room temperature, and 51% room relative humidity (1). Capacity increases with decreasing ambient temperature and increasing relative humidity. Capacities range from 5000 to 36,000 Btu/hr with 34% of the units sold in 1976 rated below 7000 Btu/hr, 28% rated between 7000 and 11,000 Btu/hr, and 10% of the units above 20,000 Btu/hr (2, 13).

The EER is defined as the ratio of a unit's cooling capacity to the electric power it consumes (W, watts) under identical standard conditions:

$$\text{EER} = \frac{Q}{W} \text{ Btu/Whr}$$

EERs range from 4.9 to 11.6 (2), the average for 1975 being 6.5 (11). EER increases as capacity increases for 115 V (up to 1700 W) units. For units larger than 1700 W, EERs are greater for 115-V units than for 230-V units of the same capacity.

Due to the variations in climatic conditions, a more accurate measure of a unit's efficiency would be the seasonal efficiency ratio (SER). This is the ratio of a unit's total cooling load over its total seasonal energy use, based on temperature, humidity, and cooling load information for a particular geographic area and the standard room (1).

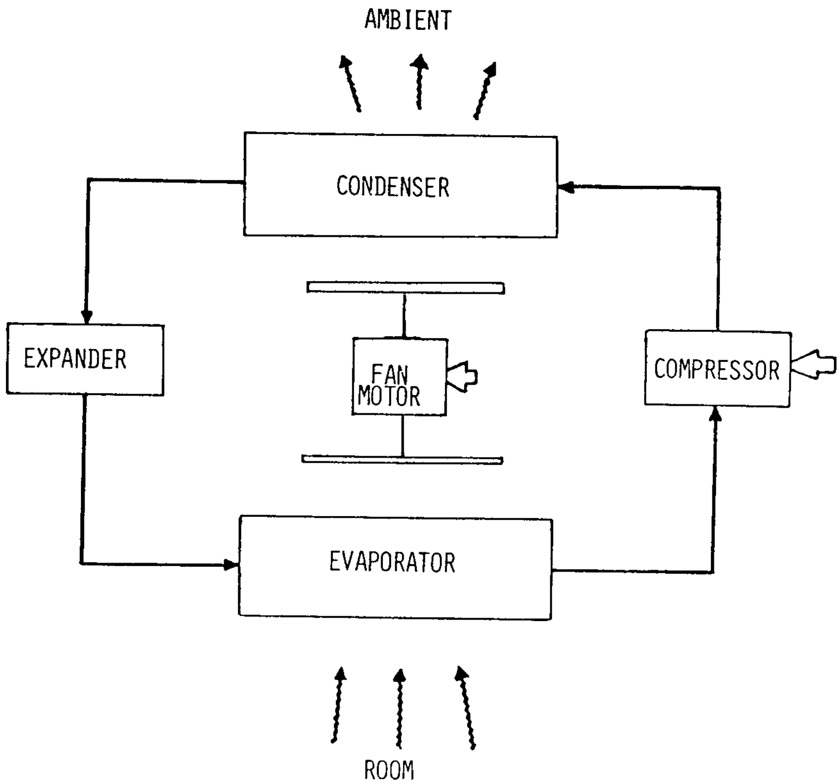
3. DESCRIPTION

3.1 Components

Components in a typical RAC are shown in Fig. 1.

a) Heat Exchangers. The two heat exchangers (evaporator and condenser) are characterized by their frontal area, fin or spine density, and number of coil rows. Material used is predominantly aluminum.

b) Fan and Fan Motors. Each heat exchanger is equipped with a fan, both driven by the same motor. Motors are either of the shaded pole (SP) type with an efficiency of 35% or the more costly permanent split capacitor (PSC) type with an efficiency up to 70%. Fans are characterized by the air flow rate they generate and the frontal area across which they move air.



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TYPICAL ROOM AIR CONDITIONER

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Fan work is a strong function of volumetric air flow and contributes 20% of the total electricity consumption. Fans are also a major contributor to noise as related to air flow rate and pressure drop. Maintaining a low noise level requires a balancing of air flow rate and frontal area.

c) Compressor. As the major mechanical component, it pumps a superheated (low pressure and temperature) vapor to a higher pressure and temperature. Compressor motors are either two-pole, 3600 rpm, with a 68% efficiency or the more costly four-pole with an 80% efficiency.

d) Expansion Device. A capillary tube or thermostatic expansion valve is used to expand a high pressure refrigerant exiting the condenser. The latter is superior since it permits the control of compressor superheat independent of ambient temperature. Capillary tubes produce a greater level of superheat at lower ambient temperatures and decrease efficiency, but they are preferred by manufacturers due to their reliability and lower cost.

3.2 Cooling Cycle

The cooling cycle is a typical Rankine cycle. In the evaporator, heat is transferred from the room air ($\sim$80°F) to the refrigerant ($\sim$45°F) which evaporates and superheats. The refrigerant is then compressed to a higher temperature and pressure and enters the condenser ($\sim$130°F) where it liquefies by transferring heat to the ambient air ($\sim$95°F). The cycle is completed by expanding the high pressure fluid to low pressure.

3.3 Constraints

There are several cycle and component constraints affecting RAC performance:

a) Large air temperature differences are required across the heat exchangers to maintain the desired heat flow rates.

b) Evaporator temperature should be cool enough to permit dehumidification without frosting the coils. The upper and lower limits are thus set by room dew point temperature and the freezing point of water. Most units operate between 40 and 50°F.

c) Condenser temperature should be high enough so that after expansion, the evaporator fluid temperature is not below 32°F. This may happen at cooler ambient temperatures. Therefore many units incorporate heaters (low ambient kits) to maintain minimum condenser temperature ($\sim$112°F) (5).

3.4 Design Improvements

The following energy-saving improvements to increase EER, some of which already have been incorporated in the more energy-efficient models, have been proposed for RACs:

a) Heat Exchanger Efficiency: Changing from fins to spines and increasing spine density, frontal area, number of rows of coils, and air flow rates will increase heat transfer capability.

b) Compressor Efficiency: Using high efficiency compressor motors (four-pole), minimizing heat losses, and improving valve efficiencies will decrease compressor energy use.

c) Fan Efficiency: Using more efficient (PSC) motors, decreasing air pressure drop by decreasing number of coil rows, decreasing air flow rates, and improving the air ducts will decrease fan energy use.

The seasonal efficiency ratio can be improved by:

a) Having an autofan in which the fan operation is synchronized with the compressor. Disadvantages would be poor air circulation and stratification during non-cooling periods. Also, on-off noise may be annoying.

b) Using multiple or variable speed compressors that will adjust the air conditioner capacity to the room cooling load providing for continuous operation. This would decrease the refrigerant mass flow rate at lower ambient temperatures. Therefore, lighter cooling loads would be supplied easily. Evaporator and condenser temperature constraints can be met without the need for heater kits (4).

3.5 Constraints on Improvements

Constraints that limit the degree of change in RACs are:

a) RACs must be small enough to fit in a window.

b) Inside fan noise level is limited by the user's comfort, and outside noise is limited by municipal ordinance.

c) Compressor variations are limited to those supplied by manufacturers. Two-speed compressors are now available in RACs. A RAC with a variable-speed compressor has not yet been manufactured due to an expected high cost (13).

4. ANALYSIS AND RESULTS

4.1 Baseline Unit

A baseline unit was chosen that was representative of a large percentage of the units sold in the United States in capacity and EER. Design and rating specifications of a Gibson AL08C4EEBA unit are given in Table 1.

Table 1. Design Specifications of Gibson Model AL08C4EEBA

Cost	$250 (1975 dollars)
Capacity	8000 Btu/hr
Voltage	115 V/AC
Current	12 amp
Power	1275 W
Unit EER	6.3 Btu/Whr
Fan Power	192 W
Fan Type	single pole
Compressor Power	1083 W
Compressor EER	8.5
Compressor Type	two pole
Frontal Area, Evaporator	0.625 ft^2
Frontal Area, Condenser	0.866 ft^2
Fin Density, Evaporator	13/in.
Fin Density, Compressor	13/in.
Equivalent Spine Density	433/in.
Evaporator Depth	3 tubes
Condenser Depth	3 tubes
Air Rate, Evaporator	200 ft^3/min
Air Rate, Condenser	355 ft^2/min

4.2 Energy-Estimation Model

A mathematical model was developed to predict power requirement and EER for a variety of design options and ambient conditions. The model was simple to use and yet flexible enough to study multiple design variations. Appendix 7.1 presents a detailed description of the model. The following assumptions were made:

a) Heat generated by the mechanical parts is not transferred to the refrigerant circuit. Since mechanical parts are located on the exterior side and are isolated from the heat transfer surfaces, they should transfer heat to the ambient air.

b) Evaporator temperature is an increasing linear function of the outdoor temperature and is constrained between 40 and 50°F.

c) Compressor efficiency is proportional to the Carnot efficiency between the condenser and evaporator temperatures. Moyers (7) has correlated performance data for 31 RAC compressors. Values of the proportionality constant were obtained from evaporator and condenser temperature assumptions.

d) Minimum condenser temperature is set at 113°F due to expander limitations and evaporator temperature constraints (see Sect. 3.3).

The model was used to estimate compressor and fan energy consumption, cooling capacity, and EER for the rating conditions of 95°F ambient temperature and room conditions of 80°F and 51% relative humidity. Results were also obtained for ambient temperatures of 85 and 105°F.

4.3 Energy-Estimation Results

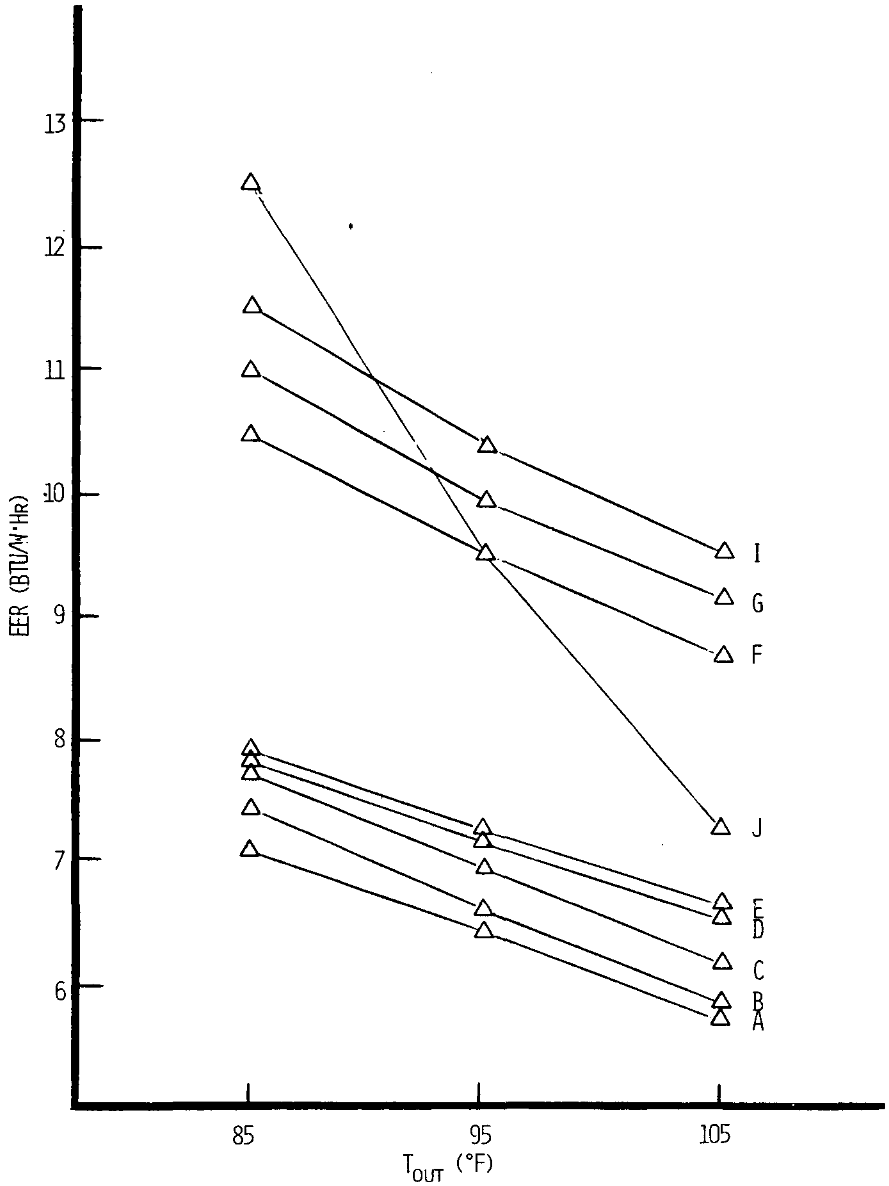
Design improvements and the corresponding energy consumption and EER are listed in Table 2. Figure 2 presents EER versus outdoor temperature for several design improvements.

A variable speed compressor is represented by line J. It was modeled by matching its capcity to the cooling load of a typical room and relaxing the constraint on condenser temperature. At rated conditions its EER is the same as that of the constant speed unit F. At higher temperatures, efficiency drops more drastically since the variable speed compressor is meeting increased capacity, whereas the one speed unit would not meet the demand. The effectiveness of a variable speed compressor is most pronounced at lower temperatures.

Decreasing the condenser air flow rate is found to be energy efficient since fan work increases with increased flow rate to the 2.68 power, whereas condenser heat rejection increases only to the 0.37 power. An optimum air flow rate can then be found balancing the two effects. The air flow rate of the base model was higher than the optimum flow rate.

Table 2. Improvements Represented in Fig. 2

Unit	% Increase in Condenser A_c	S_c	V_c	N_c	Evaporator A_E	S_E	Motors Efficiency K	η_F	Power (W) 85°	95°	105°	EER (Btu/Whr) 85°	95°	105°	(1975 $) Price Increase ($) ΔP_{he}	ΔP_c	ΔP_f	ΔP_{cont}	ΔP_{tot}	Price P	Comments
A									1172	1262	1381	7.00	6.39	5.63						250	base unit
B								10	1071	1101	1281	7.68	6.89	6.07			8		8	258	
C					10	10			1120	1198	1269	7.39	6.66	6.12	10				10	260	
D	25	15	-20	33					1055	1130	1210	7.79	7.05	6.42	15			17	32	282	
E							-10		1032	1130	1194	7.97	7.08	6.51		7			7	257	
F	40	20	-30	100			-15	10	787	852	916	10.45	9.38	8.48	37	12	8	30	87	337	
G	40	30	-30	100	10	10	-15	10	747	818	881	11.00	9.78	8.81	51	12	8	30	101	351	
I	40	30	-30	100	10	10	-15	10	714	780	848	11.51	10.25	9.15	81	12	8	30	124	374	copper condenser spines
J	40	20	-30	100			-15	10	498	853	1338	12.50	9.38	7.30							variable speed compressor
Other Single Component Improvements																					
	50	15	-30	67					1013	1094	1169	8.12	7.31	6.64	30			38	68	318	
	15	15	20	33					1243	1292	1361	6.61	6.19	5.67	13			10	23	373	
							-15		1004	1021	1131	8.89	7.47	6.70		12			12	262	
	15	15	20	33				10	1032	1081	1158	7.96	7.40	6.70	13		8	10	31	281	
Other Combined Improvements																					
							-15	10	904	971	1031	9.10	8.29	7.35		12	8		20	270	
	25	20	-30	33			-15	10	841	910	970	9.71	8.79	8.00	16	12	8	17	53	303	
	25	15	-20	33			-10	10	878	950	1016	9.37	8.42	7.65	15	7	8	17	47	297	
	50	15	-30	67			-10	10	848	920	987	9.70	8.64	7.87	30	7	8	38	83	333	
	15	15	20	33			-10	10	909	985	1053	9.05	8.13	7.38	13	7	8	10	38	288	
	25	20	-30	66			-15	10	808	877	939	10.17	9.12	8.23	29	12	8	17	61	311	
	25	20	-30	100			-15	10	788	858	921	10.43	9.32	8.43	32	12	8	17	68	318	
	40	20	-30	66			-15	10	810	870	937	10.28	9.20	8.33	29	12	8	30	79	329	
	50	20	-30	67			-15	10	795	865	928	10.42	9.25	8.37	32	12	8	38	90	340	
	50	20	-30	33			-15	10	813	893	954	10.12	8.93	8.14	23	12	8	38	81	331	



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ENERGY EFFICIENCY RATIO AS A FUNCTION OF AMBIENT TEMPERATURE FOR 8000 BTU/HR ROOM AIR CONDITIONERS

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4.4 Price Model

A price estimation model, details of which are given in Appendix 7.2, was developed based on published information (7, 12) and discussion with the Carrier Corp. (16). The following was considered in developing the model:

a) The new retail price (1975 dollars) is the sum of the baseline price and price increases of the major components: heat exchangers, compressor fans, fan motor, and the container.

b) Container and heat exchanger prices are correlated to physical dimensions using experimental scaleup factors.

c) Fan and compressor improvements result in discrete price increases.

4.5 Discussion

Table 3 is a summary of the most promising design alternatives. They represent the production-possibilities curve shown in Fig. 3 by lines A, B, and the lower portion of C. The PPC developed here is in close agreement with Moyers (7), with his data inflated to 1975 prices, and with the model developed by SAI (15) based on recent market information. Energy estimates presented here tend to be more optimistic than the other studies, which can be attributed to the inefficient unit used as a baseline. Although this study is based on an 8000-Btu/hr-capacity unit, the PPC developed here reflects the trade-off between energy savings and capital cost for a wider range of capacities.

5. CONCLUSIONS

1. Production-possibilities curves for RACs of 8000 Btu/hr capacity show a trade-off between energy use and initial cost.

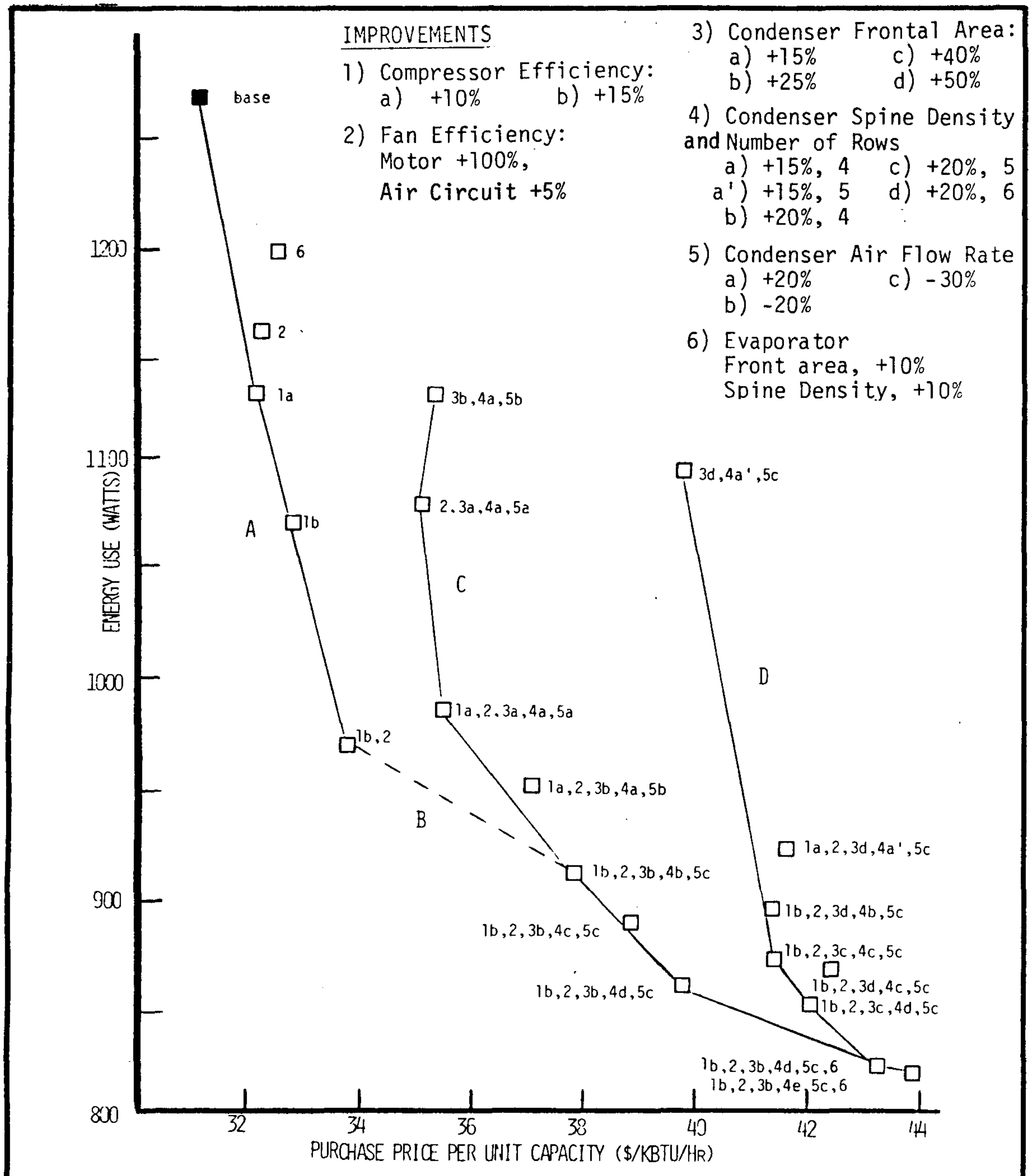
2. Lower cost energy savings were realized by improving the compressor and fan motors. Replacing a SP fan motor with a PSC motor and a two-pole compressor motor with a four-pole one increased RAC efficiency by 30% for a purchase price increase of 8%.

3. Compressor fan motor improvements combined with increased condenser and evaporator dimensions are more expensive methods of conserving energy. Such an improvement increased EER by 53% at a 40% increase in purchase price.

4. Payback period for the more expensive energy saving design improvements was 3.6 yr in New York, N.Y., and 2.6 yr in Miami, Fla. (see Appendix 7.3).

Table 3. Summary of Energy Savings and Additional Cost for Alternative Designs

Single Improvements	Energy Savings % (ΔW/W) @ 95°F	Additional Cost % (ΔP/P)
1. Compressor motor efficiency		
a) improved 2-pole motor	-10	3
b) improved 4-pole motor	-19	5
2. Permanent split capacitor fan motor and improved air circuit	-13	3
3. Heat exchanger variations:		
a) evaporator (A S V N)		
(+10%, +10%, -20%, 4)	-5	4
b) condenser (+25%, +15%, -20%, 4)	-10	13
(+50%, +15%, -30%, 5)	-13	27
Combined Improvements		
4. Compressor (4-pole) and fan motors	-23	8
5. #4 plus heat exchanger improvements		
condenser (A S V N), evaporator (A,S)		
a) (25%, 20%, -30%, 4)	-28	21
b) (25%, 20%, -30%, 5)	-31	24
c) (40%, 20%, -30%, 6)	-32	35
d) (40%, 20%, -30%, 6), (10%, 10%)	-35	39
e) (40%, 30%, -30%, 6), (10%, 10%)	-35	40
6. Evaporator (A S)		
(10%, 10%)	-5	4



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PRODUCTION POSSIBILITIES CURVES FOR 8000 BTU/HR ROOM AIR CONDITIONERS

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6. RECOMMENDATIONS

1. Energy-efficiency ratio vs temperature results should be used to estimate annual energy use for a given geographic location. Temperature-time profiles from weather records could be used to obtain EER vs time and power vs time profiles.

2. Application of this model over a wide range of capacities should be tested before incorporating it into the ORNL residential energy model.

3. More detailed component cost information should be obtained from manufacturers.

7. APPENDIX

7.1 Development of Energy Model

7.1.1 Heat Transfer Across Evaporator Coils

The rate of heat absorption by the evaporator coils was assumed to be given by

$$Q_E = (UA)_E \, \Delta T_E \tag{1}$$

where $(UA)_E$ is the heat transfer coefficient of the evaporator times its area, and ΔT_E is the arithmetic average temperature driving force which is within 4% of the logarithmic average and is given by

$$\Delta T_E = \frac{(T_{in} - T_E) + (T_{in,d} - T_E)}{2} = T_{in} - T_E - \frac{1}{2}\Delta T_{in} \tag{2}$$

where:

$$\Delta T_{in} = T_{in,d} - T_{in} \, .$$

Also,

$$\Delta T_{in} = \frac{Q_E \, f}{(\rho C_p)_{in} \, V_E}\left(\frac{1}{60}\right) \tag{3}$$

where f is the fraction of heat absorbed by the condenser that goes towards changing the temperature of the air. Combining Eqs. (1), (2), and (3) gives

$$Q_E = (UA)_E\left[T_{in} - T_E - \frac{1}{2}f \, \frac{Q_E}{(\rho C_p)_{in} V_E (60)}\right]$$

or

$$Q_E = \frac{(UA)_E (T_{in} - T_E)}{1 + (UA)_E \, R} \, , \quad R = \frac{f}{(120)(\rho C_p)_{in} V_E} \tag{4}$$

7.1.2 Heat Transfer Across Condenser Coils

The average temperature driving force across the condenser coil, similar to Eq. (2) is

$$\Delta T_c = \frac{(T_c - T_{out}) + (T_c - T_{out,d})}{2} = T_c - T_{out} - \frac{1}{2}\Delta T_{out} \quad (5)$$

At the same time the condenser load is related to ΔT_{out} by

$$\Delta T_{out} = \frac{Q_c}{(\rho C_p)_{out} V_c (60)} \quad (6)$$

Rearranging Eqs. (5) and (6) and using $Q_c = (UA)_{out} \Delta T_c$ yields

$$Q_c = \frac{(UA)_c (T_c - T_{out})}{1 + UA_c G} , \qquad G = \frac{1}{(\rho C_p)_{out} V_c (120)} \quad (7)$$

7.1.3 Heat Transfer Coefficients

These are adapted from SAI reports (14) and for the condenser are given by the following empirical correlation:

$$(UA)_c = \alpha (V_c)^{0.37} \left(\frac{F_c}{8}\right)^{0.59} (A_c)^{0.63} \quad (8)$$

where α = 20.15, 29.29, 36.95, and 56, if N_C = 2, 3, 4, and 5, respectively. Similarly for the evaporator:

$$(UA)_E = \beta (V_E)^{0.37} \left(\frac{F_E}{8}\right)^{0.59} (A_E)^{0.63} \quad (9)$$

where β = 20.44, 31.0, and 38.24, if N_E = 2, 3, and 4, respectively.

Since spines are more efficient heat transfer surfaces than fins, it is desirable to find the equivalence so that improvements due to spine use can be calculated. A fin density of 13/in. in the base model leads to a spine density of 433/in. and the following relationship was calculated (10):

$$(\frac{F}{8})^{0.59} = 1.9(\frac{S}{266})^{0.59} \quad (10)$$

Substituting Eq. (10) into Eqs. (8) and (9) and rearranging gives:

$$(UA)_C = 0.623\ e^{0.34N_C}\ (V_C)^{0.37}\ (A_C)^{0.63}\ (S_C)^{0.59} \quad (11)$$

$$(UA)_E = 0.710\ e^{0.31N_E}\ (V_E)^{0.37}\ (A_E)^{0.63}\ (S_E)^{0.59} \quad (12)$$

If spines are made of copper instead of aluminum, (UA) is increased by a factor of the ratio of the square of the thermal conductivities (9):

$$(UA)_2 = \sqrt{\frac{k_2}{k_1}}\,(UA)_1 = 1.35(UA)_1 \quad \text{(for Cu-Al)} \quad (13)$$

This relationship is accurate for spine efficiencies less than 0.75.

7.1.4 Compressor Work

According to Moyers (7),

$$\frac{W_{comp}}{Q_E} = K(\frac{T_C - T_E}{T_E + 460}) \quad (14)$$

Here W_{comp} is the compressor work and K is the ratio between the Carnot efficiency and the actual efficiency of the compressor.

7.1.5 Solving for Condenser Temperature

Now that the fundamental equations have been presented, it is possible to rearrange them into a workable form. An energy balance yields,

$$W_{comp} = Q_C - Q_E \quad (15)$$

So, using Eq. (14), Eq. (15) can be put into the form:

$$\frac{Q_c}{Q_E} = K\left(\frac{T_c - T_E}{T_E + 460}\right) + 1 \tag{16}$$

From Eqs. (4) and (7),

$$\frac{Q_c}{Q_E} = \frac{(UA)_c\ (T_c - T_{out})}{(UA)_E\ (T_{in} - T_E)} \cdot \frac{[1 + (UA)_E \cdot R]}{[1 + (UA)_c \cdot G]} = M\frac{(T_c - T_{out})}{(T_{in} - T_E)} \tag{17}$$

where:

$$M = \frac{(UA)_c}{(UA)_E} \cdot \frac{[1 + (UA)_E R]}{[1 + (UA)_c G]} \tag{18}$$

Combining Eqs. (16) with (17) yields:

$$M\,\frac{(T_c - T_{out})}{(T_{in} - T_E)} = K\,\frac{(T_c - T_E)}{(T_E + 460)} + 1 \tag{19}$$

Solving for T_c results in the following:

$$T_c = \frac{\dfrac{M\ T_{out}}{(T_{in} - T_E)} - \dfrac{K\ T_E}{(T_E + 460)} + 1}{\dfrac{M}{(T_{in} - T_E)} - \dfrac{K}{(T_E + 460)}} \tag{20}$$

7.1.6 Estimating Evaporator Temperature

The outdoor temperature is set at 95°F for the evaporator at 45°F. The evaporator temperature varies approximately linearly in a 1:10 ratio with the outdoor temperature. Therefore a T_{out} of 85° corresponds to T_E = 44, and T_{out} of 105° corresponds to T_E = 46. When improvements are made in the evaporator coils, T_E is solved for using Eq. (4) and set within the limits 40-50°F. This limits the improvements in $(UA)_E$.

7.1.7 Fan Work

Fan work is given by (10),

$$W_{fan} = (V_C \Delta P_C + V_E \Delta P_E) \frac{\eta_{base}}{\eta} \quad (21)$$

where ΔP_C and ΔP_E are the pressure drops across the condenser and evaporator, and both are given by

$$\Delta P + \frac{bNV^2}{Re^{0.32}} \quad (22)$$

Substituting Eq. (22) into Eq. (21) and rearranging yields:

$$W_{fan} = b_C N_C V_C^{2.68} + b_E N_E V_E^{2.68} \quad (23)$$

The constants b_C and b_E are 8.95 x 10^{-6} and 1.93 x 10^{-6}, respectively, based on the base model.

7.1.8 Total Work and EER Calculation

Total work is obtained by summing fan work and compressor work:

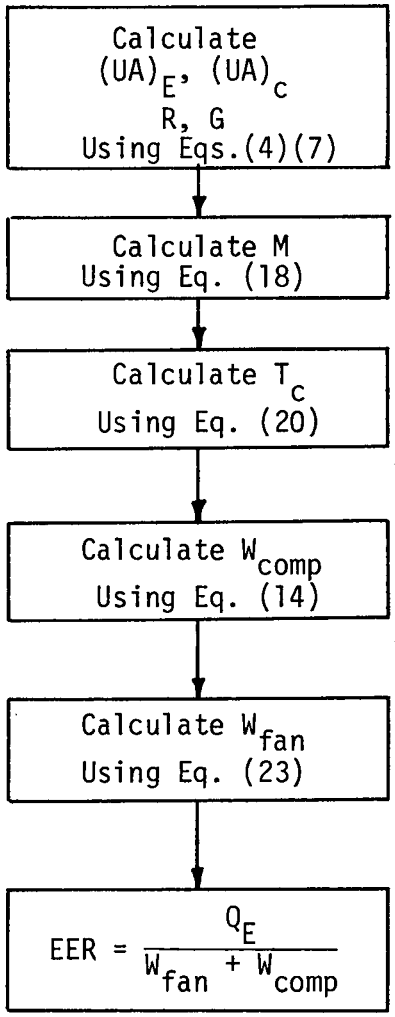
$$W_{total} = W_{fan} + W_{comp} \quad (24)$$

EER is defined by

$$EER = \frac{Q_E}{W_{total}} \text{ Btu/Whr} \quad (25)$$

7.1.9 Using the Model

The following algorithm was developed.



7.2 Price Estimation Model

7.2.1 Purchase Price (1975 dollars)

The cost of an improved air conditioner is given by:

$$P = P_o + \Delta P_{cont} + \Delta P_{he} + \Delta P_{comp} + \Delta P_{fan} \qquad (26)$$

where:

P_o = \$250

ΔP_{cont} = cost due to increase in container size

ΔP_{he} = cost to improve heat exchanger

ΔP_{comp} = cost to improve compressor

ΔP_{fan} = cost to improve fan

7.2.2 Price Increase Due to Container Size Increase

Carrier Corp. (16) suggests that 29% of the cost of a low EER air conditioner is attributed to the cost of the container. Thus, the initial cost of the container, $P_{o,cont} = 0.29(250) = \$72.5$. Half this cost is markup due to labor.

Scaling-up the base unit to an EER of 9.4 increases the container cost from \$72.5 to \$128. This is caused by increasing the outside area of the machine from 1778 to 2475 in.^{2} (16). Thus,

$$\frac{P_{cont}}{P_{o,cont}} = \left(\frac{2475}{1778}\right)^n \qquad (27)$$

or

$$\frac{128}{72.5} = (1.4)^n \qquad (28)$$

so

$$n = 1.7 \qquad (29)$$

For the purpose of this model, it is assumed that the container area is proportional to the condenser frontal area. A 40% increase in frontal area is required to bring the EER to 9.4, so this approximation should be reasonable. Thus,

$$\frac{P_{cont}}{P_{o,cont}} = \left(\frac{A_c}{A_{c,o}}\right)^{1.7} \qquad (30)$$

$$\frac{P_{cont} - P_{o,cont}}{P_{o,cont}} = \left(\frac{A_c}{P_{c,o}}\right)^{1.7} - 1 \qquad (31)$$

or

$$\Delta P_{cont} = P_{o,cont} [(\frac{A_c}{A_{c,o}})^{1.7} - 1] \quad (32)$$

$P_{o,cont}$ is taken to be the material cost of the container, i.e., $\frac{\$72.5}{2}$ = \$36.25. Furthermore, it is assumed that labor cost does not increase as container size increases. Thus,

$$\Delta P_{cont} = 36.25[(\frac{A_c}{A_{c,o}})^{1.7} - 1] \quad (33)$$

7.2.3 Price Increase Due to Heat Exchanger Improvements

The price increase of the heat exchanger surfaces is assumed to be

$$\frac{P_{he}}{P_{o,he}} = (\frac{A_{he}}{A_{o,he}})^{m} \quad (34)$$

Based on information from Moyers (7), m = 0.88. The Carrier estimate (16) places $P_{o,cond}$ at 17% of the total cost and $P_{o,evap}$ at 7% of total cost. Thus,

$$\Delta P_{he} = 25[(\frac{A_c}{A_{c,o}})^{0.88} - 1] + 17.5(\frac{A_E}{A_{E,o}})^{0.88} - 1] \quad (35)$$

If condenser fins are made of copper, Eq. (35) should be modified to

$$\Delta P_{he} = [25 \frac{P_{cu}}{P_{al}}][(\frac{A_c}{A_{c,o}})^{0.88} - 1] + 17.5[(\frac{A_E}{A_{E,o}})^{0.88} - 1] \quad (36)$$

7.2.4 Price Increase Due to Compressor Improvement

The cost of improving the compressor is taken from NBS (11).

$$\Delta P_{comp} = \$7 \text{ for a 10\% efficiency improvement} \quad (37)$$

$$\Delta P_{comp} = \$12 \text{ for a 15\% efficiency improvement} \quad (38)$$

7.2.5 Price Increase Due to Fan Improvement

The cost of improving the fan motor is also taken from NBS (11).

$$\Delta P_{fan} = \$8 \text{ for an efficiency increase from 35 to 70\%.} \qquad (39)$$

7.3 Payback Period for Room Air Conditioner Improvements

Power savings for the most efficient design alternative presented are $\Delta W = 0.5$ kW. The total cost increase (ΔP_{tot}) due to these improvements is \$100. If p = electricity cost (\$/kWh) and t = annual compressor operating hours, then payback period (PB) is given by

$$PB = \frac{\Delta P_{tot}}{\Delta Wpt} \qquad (40)$$

The following is the payback period for three cities:

City	p (\$/kWh) (17)	t (hr) (8)	PB (yr)
New York	0.073	755	3.6
Miami	0.035	2169	2.6
Atlanta	0.033	983	6.2

7.4 Nomenclature

A	frontal area of heat exchanger, ft^2
C_p	air heat capacity, Btu/lb-°F
EER	energy-efficiency ratio, Btu/hr
f	fraction of heat absorbed by the evaporator that is used to change the air temperature
F	fin density, $in.^{-1}$
K	compressor efficiency constant
k_i	thermal conductivity of substance i

N	number of rows of cooling tubes
p	price of electricity
P	air conditioner purchase price (1975 dollars)
PB	payback period
ΔP	pressure drop (Sect. 7.1 only)
ΔP_i	change in price due to improvement in component i
PPC	production possibilities curve
PSC	permanent split capacitor
Q	rate of heat transfer across heat exchanger surfaces, Btu/hr
RAC	room air conditioner
S	spine density, in.$^{-1}$
SER	seasonal efficiency ratio
SP	shaded pole
t	annual compressor operating hours
T	temperature, °F
ΔT	temperature drop, °F
UA	product of overall heat transfer coefficient and total heat exchanger area, Btu/hr-°F
V	volumetric flow rate of air across heat exchanger, ft^3/min
W	power consumption, W
ΔW	power reduction
η	fan efficiency
ρ	air density, lb/ft^3

<u>Subscripts</u>

al	aluminum
c	condenser
comp	compressor

cont	container
cu	copper
d	pertaining to air leaving the heat exchangers
E	evaporator
fan	fan
he	heat exchanger
in	indoor
o	initial
out	outdoor
tot	total

7.5 Literature References

1. Association of Home Appliance Manufacturers, "American National Standard," Room Air Conditioner Volume RAC-1, p. 7, Chicago, Ill.

2. Association of Home Appliance Manufacturers, "1977 Directory of Certified Room Air Conditioners, Edition No. 3 - June 1977, Chicago, Ill.

3. Bird, R.B., W.E. Stewart, and E.N. Lightfoot, "Transport Phenomena," p. 391, Wiley, New York, 1960.

4. Hamilton, J.F., and A.B. Newton, "Energy Conservation Thru Use of Variable Capacity Compressors," Proc. Technical Opportunities for Energy Conservation in Appliances, p. 227, Arthur D. Little, Inc., Cambridge, Mass. (1976).

5. *Ibid.*, p. 232.

6. Hirst, E., J. Cope, S. Cohn, W. Lin, and R. Hoskins, "An Improved Engineerng Economic Model of Residential Energy Use," ORNL/CON-8 (1977).

7. Moyers, J.C., "The Room Air Conditioner as an Energy Consumer," ORNL-NSF-EP-59 (1973).

8. Pilati, D.A., "Room Air-Conditioner Lifetime Cost Considerations: Annual Operating Hours and Efficiencies," ORNL-NSF-EP-85 (1975).

9. Rohsenow, W.M., and J.P. Hartnet, "Handbook of Heat Transfer," p. 3-116, McGraw-Hill, New York, 1973.

10. *Ibid.*, pp. 18-81 through 18-97.

11. "Appliance Energy Efficiency Improvement Target for Room Air Conditioners," National Bureau of Standards Center for Consumer Product Technology, April 15, 1977.

12. "Energy Efficiency Program for Room Air Conditioners, Central Air Conditioners, Dehumidifiers, and Heat Pumps," p. A8, Science Applications, Inc., La Jolla, CA (1975).

13. *Ibid.*, pp. B6-B25.

14. *Ibid.*, pp. B145-B149.

15. *Ibid.*, pp. B176-B205.

16. Carrier Corp., personal communication, Syracuse, NY, Feb. 1978.

17. Oak Ridge City Hall, personal communication, Feb. 1978.

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