



**Proceedings of the
DOE/ORNL Heat Pump Conference:
RESEARCH AND DEVELOPMENT ON HEAT PUMPS
FOR SPACE CONDITIONING APPLICATIONS**

**WASHINGTON MARRIOTT HOTEL
WASHINGTON, D.C.
DECEMBER 11-13, 1984**

**Sponsored by
U.S. DEPARTMENT OF ENERGY
OFFICE OF BUILDING ENERGY RESEARCH AND DEVELOPMENT**

A COMPARATIVE ANALYSIS OF SINGLE- AND CONTINUOUSLY VARIABLE-CAPACITY HEAT PUMP CONCEPTS*

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Objective

This work is an initial assessment of the potential benefits of continuous-capacity-modulation in electric-driven, air-to-air heat pumps for residential application. The purpose of the project was to provide a quantitative estimate of the possible annual performance gains of advanced continuously modulating heat pumps relative to single-speed designs at comparable levels of development.

Background

Previous analytical design work in this area at ORNL dealt with single-design-point, heating-mode optimization of single-speed heat pumps.¹ For that work the ORNL Steady State Heat Pump Design Model² was connected to a constrained numerical optimization code.³

The present work represents an extension of the earlier work in two directions. First, seasonal (heating and cooling) and annual performance factor (APF) analysis capability was added to allow direct evaluation of annual energy use from heat pump performance data generated by the ORNL heat pump model. Secondly, a modulating version of the heat pump model was developed to provide a means for simulating the steady state performance of continuously variable-speed (CVS) systems. With these tools, the APFs of both single- and continuously variable-capacity (CVC) concepts could be studied as basic heat pump design variables were varied.

Approach

These ORNL steady state heat pump models and the APF model were used with numerical optimization routines to find design configurations of currently marketed heat pump com-

ponents that give best annual performance for some typical house loads. In this way, the annual performance of single-capacity and CVC heat pump concepts were each optimized (maximized) to evaluate the energy-saving potential of likely competing candidates of the two types.

The use of an optimization program ensured that the maximum potential of each operating concept (given certain assumptions)[†] would be found that was consistent with the applied constraints. For this analysis, engineering constraints reflecting current practice were applied to:

- total heat exchanger size,
- compressor and fan performance, and
- design cooling capacity.

The first two types of constraints were imposed directly within the program, while the cooling capacity constraints were externally maintained by the optimizer. The design variables that were optimized for the best performance subject to the constraints included heat exchanger frontal area, tube rows, circuits, and air flow rates; compressor displacement; and condenser subcooling.

Assumptions

Overview Of Cases

Four primary cases were considered to provide values for comparison of the efficiency and energy-savings potential of optimized CVS vs single-speed heat pumps. These cases were:

1. single-speed systems with regular‡ cycling and frosting/defrosting losses,⁴

[†]E.g., it was assumed that speed modulation had the highest efficiency potential for the continuous-capacity-modulation concept.

[‡]The cycling loss factors assumed⁴ were larger than the alternative DOE default C_D factors⁵ of 0.25 (ranging in equivalent values from 0.25 to 0.35).

*Research sponsored by the Building Equipment Research Division, U.S. Department of Energy, under contract DE-AC05-84OR21400 with Martin Marietta Energy Systems, Inc.

2. single-speed systems with reduced** cycling losses (through reduced loss factors),
3. CVS systems with advanced inverter-motor combinations, regular‡ cycling losses, and estimated frosting/defrosting losses, and
4. CVS systems with advanced inverter-motor combinations and reduced** cycling losses (through reduced loss factors).

Case 4 was not reoptimized, but was obtained from case 3 by simply applying the different loss assumptions to that optimized configuration. The effect of this shortcut was estimated to be minimal, based on the results of case 1 and 2 optimizations.

Care was taken to make consistent assumptions regarding component performance between the two operating concepts. The only component efficiency advantages allowed the CVS systems were (1) those due to inherent flexibility and (2) those resulting in significant performance gains unique to that concept. These allowed advantages were:

1. use of three-phase compressor and indoor fan motors for the CVS systems because the inverters convert residential service to three-phase power as an added benefit, and
2. use of advanced synchronous, permanent-magnet, electronically commutated, motor-inverter combinations (PM-ECMs) for the CVS systems because they have unique application advantages over conventional induction motor, inverter-driven systems—by eliminating compressor motor slip losses (which increase significantly at lower speeds), by reducing motor harmonic losses, and by better maintaining fan efficiency at reduced fan speeds.

Speed-Change Ranges

For the CVS systems, we assumed the speed ranges used in the York ENMOD CVS heat pump,⁶ which give a continuous modulation of:

- the compressor over the speed ranges of
 - 6 to 1 in heating mode (15 to 90 Hz) and
 - 4 to 1 in cooling mode (15 to 60 Hz), and
- the indoor fan over a speed range of
 - 2.5 to 1 in either mode (24 to 60 Hz).

The outdoor fan speed was held fixed based on earlier unpublished work which indicated that any gains resulting from modulation of this fan were minimal.

Inverter-Driven Induction Motors Vs PM-ECMs

During the course of the study, we concluded that the present commercially available inverter-motor combinations do not provide the technology to closely approach the potential of continuous-

**C_D factors of 0.05 were assumed in both heating and cooling modes.

capacity modulation. This is because conventional three-phase induction motors have:

- motor slip losses that increase substantially (reducing the motor efficiency from about 4 to 16 percentage points) as the operating frequency is reduced (from 60 to 15 Hz) under constant-torque conditions typical of compressor applications, and
- motor losses due to choppy inverter waveforms that have been measured at 2 to 5 percentage points (depending on frequency and inverter type).⁷

These losses can be greatly reduced through the use of a PM-ECM because:

- all motor slip losses are eliminated because slip is not used to generate torque—elimination of the rotor and other losses associated with slip results in a 4 to 5 percentage point motor efficiency boost at rated torque at all speeds, and
- losses due to the choppy (nonsinusoidal) inverter waveforms are significantly reduced (estimated at 1.5 percentage points) with the use of PM motors.

Because we were interested primarily in the potential of the CVC concept and it appears highly probable that the cumulative efficiency gain of the PM motors would more than offset any reasonable cost premium, we assumed PM-ECMs in the CVS systems.

The inverter performance was modeled as that of an efficient, present-day inverter⁸ as shown in Fig. 1 with a peak efficiency of 95% at rated power and with a reduction of performance at reduced output power.

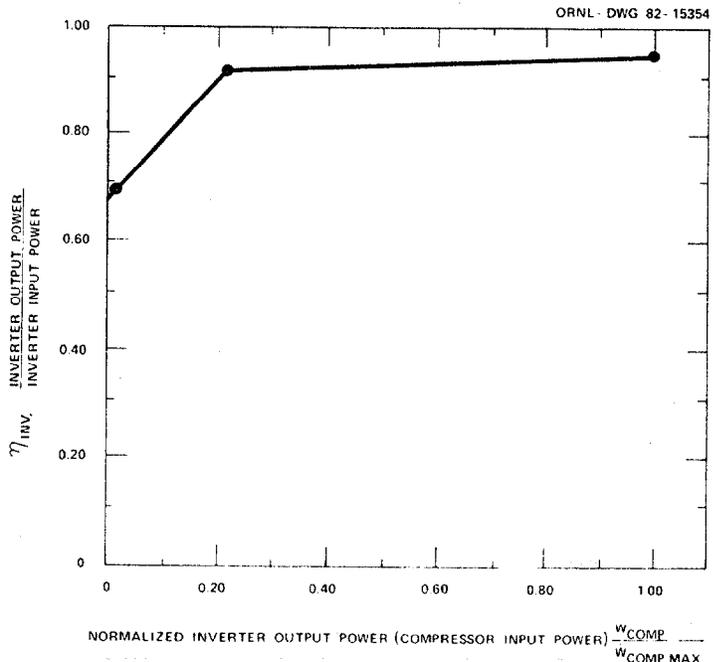


Fig. 1. Assumed frequency-inverter efficiency.

Compression Efficiency Change With Speed

A residential-size, high-efficiency reciprocating compressor was assumed for the analysis. We also included in the CVS compressor

model limited data⁸ which indicated that, at the lower pressure ratios which would exist across a compressor at lower speed operation, the basic compressor efficiency (exclusive of the motor) could increase by as much as 15%. This is presumably due to a reduction in flow losses. Note that other types of modulating compressors may not provide this same net benefit at lower capacity operation (e.g., stroke-change schemes or rotary compressors). No data were available for the overspeed operation range (60 to 90 Hz); in this region 60 Hz compressor efficiencies were assumed.

Results

The design configurations for the four systems were computer optimized, using these assumptions, in order to maximize the APF of each system. Comparative results for the optimum configurations and system performance are summarized as follows:

Optimum Configurations

The optimized single-speed configurations with high and low cycling loss factors were nearly identical, suggesting that cycling-loss reduction techniques could be retrofitted to existing designs with little loss in performance potential.

Optimum heat pump configurations for the single- vs variable-speed concepts showed the following notable configuration changes for the variable-speed case:

- a 20% larger compressor displacement that, with overspeed operation, reduced the heating balance point from 25°F to 11°F with the unit still capable of precisely tracking the cooling load down to 82°F and the heating load up to 42°F;
- a 25% smaller outdoor air flow rate;
- lower indoor air flow rates in the cooling mode; and
- half as many refrigerant circuits in the indoor coil.

The first result indicates that, even with overspeed capability, the variable-speed systems can use a larger compressor to good advantage in a moderate climate (60% heating / 40% cooling loads) to minimize backup heat use. Apparently, the unit can slow down sufficiently to meet the lighter loads at milder ambients without significant cycling loss penalties.

In Fig. 2, the load matching characteristics and balance points for optimized cases of the two operating concepts (case 1 and case 3) are shown with frosting/defrosting effects included. This figure shows how the CVS system dramatically improves the matching between heat pump capacity and house load requirements over the range of heating and cooling mode ambients.

Figure 3 shows the optimum speed variations vs ambient for the indoor fan and compressor over the range of heating and cooling loads shown in Fig. 2. Superimposed in Fig. 3 are the normalized annual heating and cooling energy outputs required of the heat pumps in each ambient temperature bin—outputs which are the product of the load line values in Fig. 2 multiplied by the hours of temperature occurrence in each bin. Comparison of the required output distribution against the modulation range shows that the unit modulation range is shifted off center of the output

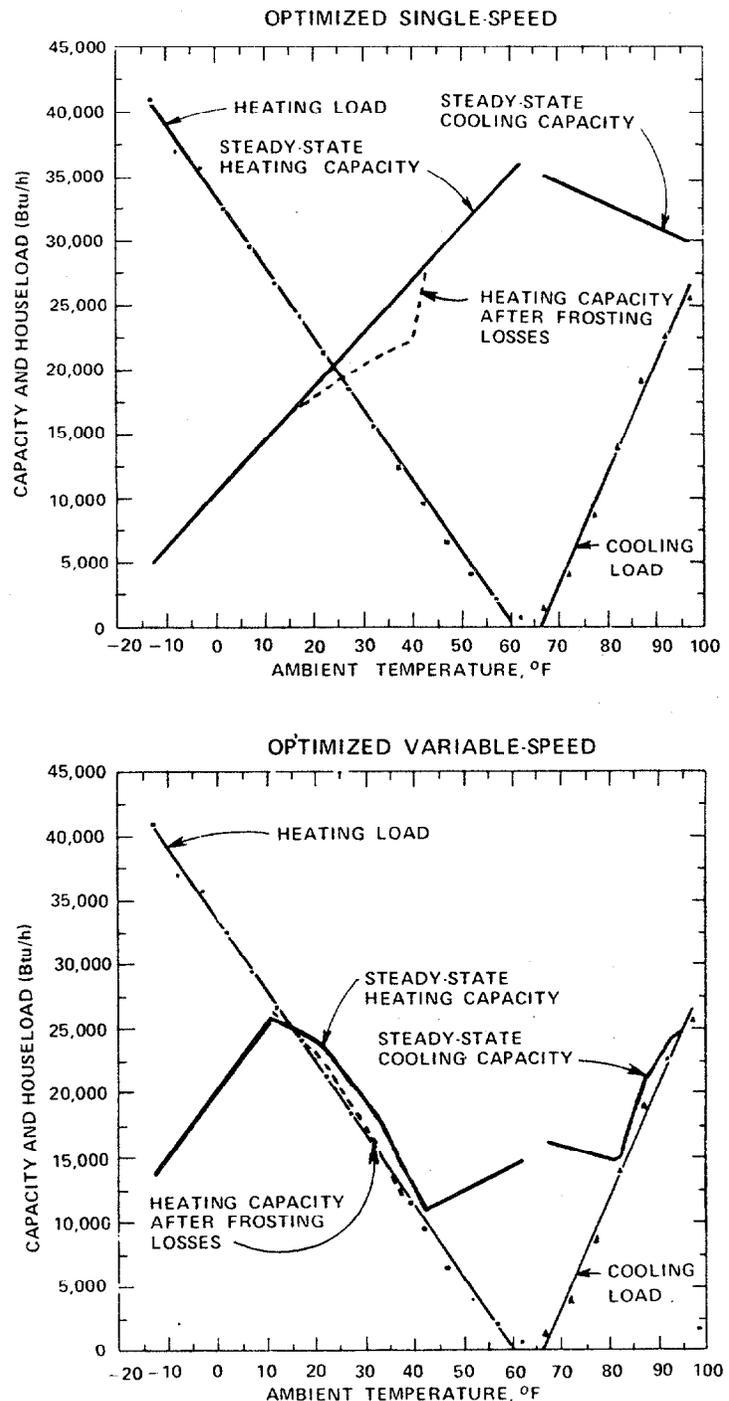


Fig. 2. Load matching characteristics of optimized single- vs variable-speed heat pumps.

distribution to obtain the benefits of reduced backup resistance heat use at the colder ambients. This was found to maximize the annual performance even though it results in more cyclic operation at the milder ambients in both heating and cooling modes.

Maintenance of Comfort Conditions

With regard to comfort criteria, the optimized single-speed cases had better than acceptable indoor-air delivery temperatures (a minimum of 89°F at the single-speed balance point) and indoor air dehumidification rates (sensible-to-total cooling capacity ratios

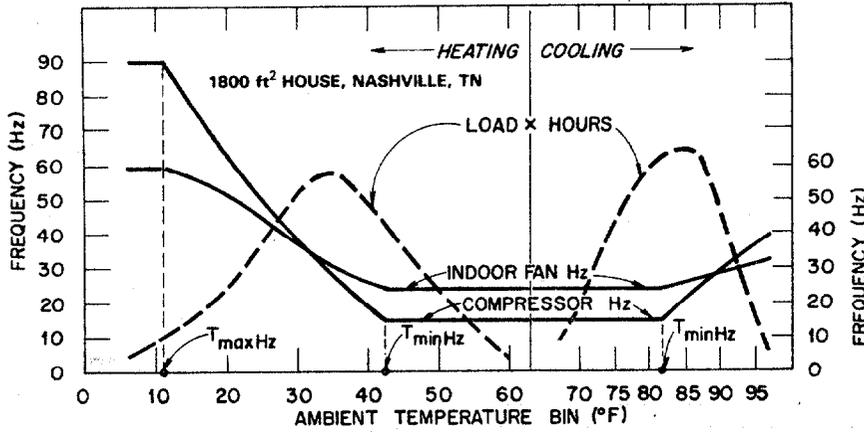


Fig. 3. Optimal inverter frequency vs load-hour distribution and ambient temperature.

of 0.69 to 0.71). These results call into question the notion that high-efficiency, single-speed heat pumps must sacrifice comfort.

The variable-speed cases had dehumidification rates that were low by conventional measures (sensible-to-total ratios of 0.72 to 0.82) but that we judged acceptable for the chosen house and location because of the nearly continuous operation of the unit. Continuous, or nearly continuous operation, should:

- achieve the actual steady-state, sensible-to-total cooling ratio instead of a higher net value that would be expected with a cycling single-speed unit;
- allow a higher tolerable humidity level because air is in continual circulation and temperature and humidity swings are minimized; and
- minimize re-evaporation of water from the coil because of cyclic operations—especially with continuous fan operation.

Delivery air temperatures for the optimized CVS configuration were, however, too low in the heating mode (remaining between 85 to 86°F for all ambients). This indicates the need for an additional comfort constraint in future analyses. Preliminary analysis indicates that such a constraint will significantly reduce the maximum indoor flow rate in heating mode but will not have that large an effect on the APF values.

Comparison of Maximum Performance Predictions

All the performance and energy savings values which follow were predicted for an 1800-ft² house with HUD minimum levels of insulation in Nashville, Tennessee. The first sets of comparisons will be made between cases 1 and 3—single- vs variable-speed systems with regular dynamic losses—to present the general performance differences between the two basic concepts. Following this, cross-comparisons between all four cases will be made.

The net coefficients of performance (COPs) vs ambient for cases 1 and 3 are illustrated in Fig. 4. The net COPs include the effects of backup resistance heat, frosting/defrosting, and cycling. Significant performance differences are evident throughout the heating and cooling ambients, with the largest percentage differences occurring at the lower ambients in the cooling mode. In

1800 ft² HOUSE, NASHVILLE, TENNESSEE
CVS—CONTINUOUSLY VARIABLE-SPEED

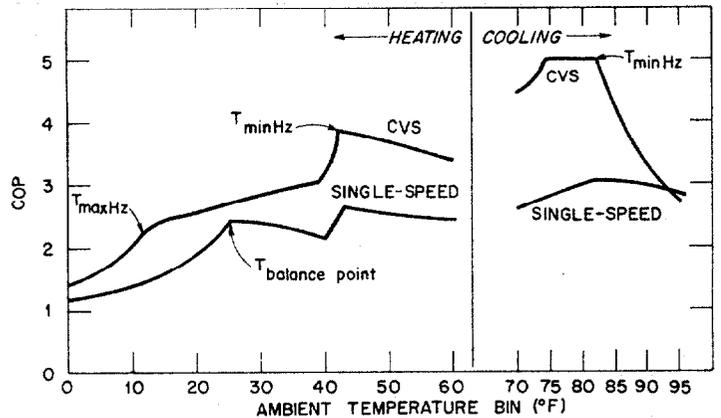


Fig. 4. Net COP comparison between single-speed and CVS systems—regular dynamic losses.

heating mode, only near the single-speed balance point do the two approach each other. Here both units are supplying the same instantaneous output and there are no heat exchanger unloading nor resistance heat displacement advantages for the CVS case; the slight COP advantage remaining in the absence of these benefits is because of the higher efficiency three-phase PM motors which could be used (and possibly also because of the slightly lower compressor speed with the larger displacement). In cooling, the CVS case has much better performance except near the cooling design point, where apparently a poorer performance configuration was acceptable to gain the higher performance at the milder ambients.

The resultant effect of these COP differences on the energy use between the two systems is given in Fig. 5. Here the distance between the two curves at each ambient gives the energy savings per 1F° temperature bin from the CVS system performance advantage. In the heating mode, the absolute savings are greatest around the peaks of the required system output curves of Fig. 3 and next largest at the milder ambients in both heating and cooling. Only at the higher ambients in cooling are there no savings with the modulating systems.

1800 ft² HOUSE, NASHVILLE, TENNESSEE

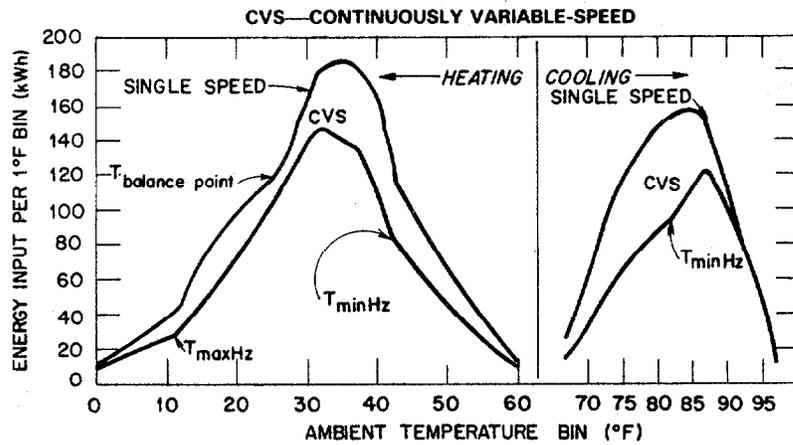


Fig. 5. Net COP comparison between single-speed and CVS systems—regular dynamic losses.

The summation of the differences between the curves in Fig. 5 gives the net annual energy savings of the modulating system. This difference is shown in Fig. 6 as a portion of the total energy use for the nonmodulating case, which has been subdivided into various steady state and dynamic loss components. The energy use (26.7% less) and breakdown for the CVS case is given in the remaining portion of the pie. The savings are shown as subdivided into four categories. These are as follows:

- reduced cycling losses because of better load following (reduced 68% from the nonmodulating case),
- heat exchanger unloading because of lower speed operation (reduced compressor and fan consumption by 11%),
- reduced backup heat use because of compressor oversizing and overspeed operation (reduced by 82%), and
- reduced frosting/defrosting losses (estimated) resulting from heat exchanger unloading (reduced by 51%).

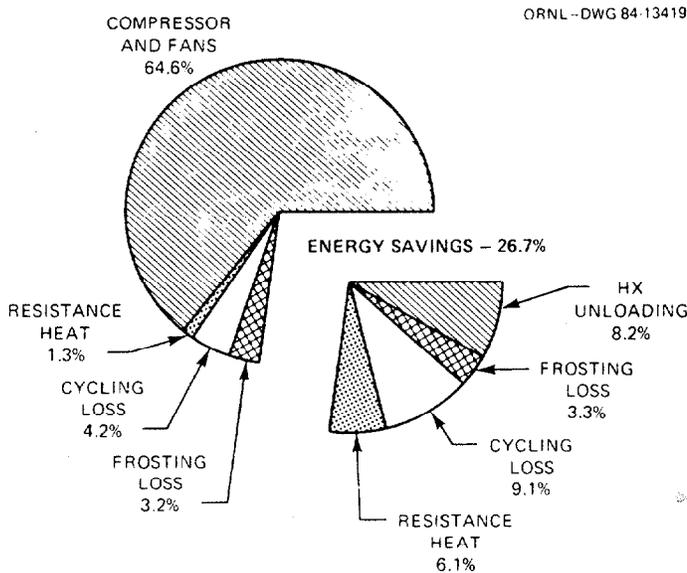


Fig. 6. Energy use and savings for a continuously modulating heat pump.

The relative contributions of the four factors to the 26.7% total energy savings are indicated in Fig. 7. While reduced cycling losses are the largest single percentage of savings, they represent only one-third of the total savings, with primary and secondary benefits of heat exchanger unloading (amounting to 43.3%) and reduced backup heat use (saving 22.8%) providing the remaining two-thirds.

This completes the more detailed comparative analysis between the nonmodulating and modulating systems with regular dynamic losses. The comparative results between all four cases are summarized in Figs. 8 and 9 where the potential for energy savings and efficiency improvement by each path is given. These results are again for the house and location with a 60/40 split between heating and cooling loads.

In Fig. 8, the 27% (26.7%) energy savings of the CVS system with relatively large cyclic loss factors (case 3) are compared to an 11% savings for the single-speed system with cycling losses decreased by assuming greatly reduced cyclic loss factors (case 2). When continuous modulation is combined with reduced cyclic loss coefficients (case 4), the net effect is a 30% savings over the optimized base case (case 1). The individual savings (from cases

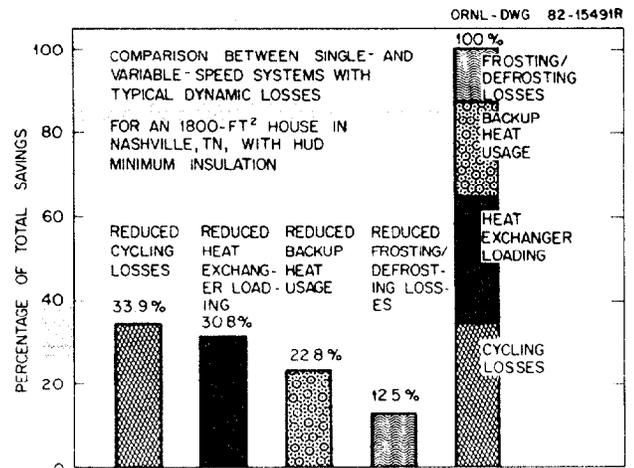


Fig. 7. Comparison of relative contributions to the total energy savings with a CVS system.

ENERGY SAVINGS POTENTIAL

ONE HOUSE, ONE LOCATION

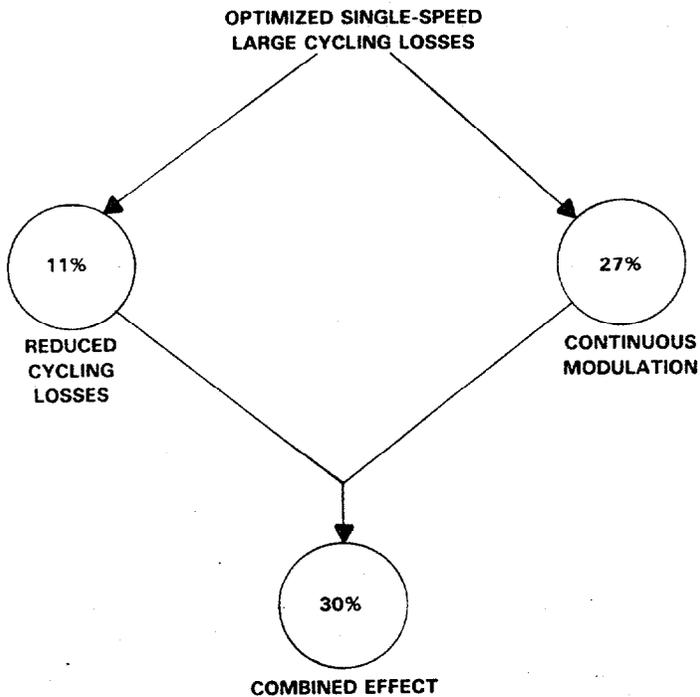


Fig. 8. Energy savings potential relative to optimized reference system.

2 and 3) are not additive because the cycling losses are reduced significantly by either path; thus these reductions cannot be counted twice.

The efficiency improvement potential for the various cases is similarly shown in Fig. 9. By definition, the efficiency improvement numbers are always higher than the energy savings percentages. Performance improvements of 36 to 43% are thus predicted for the continuous modulation cases when compared to the optimized "regular" dynamic loss, single-speed system.

Allowable Consumer Cost Premiums

Based on the energy savings computed for the various cases for the average-insulated 1800-ft² house in Nashville, maximum-allowable, consumer first-cost premiums to obtain 3- and 5-year simple paybacks were computed. These are shown in Fig. 10, where the values within the circles are again relative to the optimized base case. An electricity cost of 5 cents/kWh was assumed. For a 3-year simple payback, a consumer could pay \$145 more for a nonmodulating heat pump with reduced cycling losses or \$386 more for a similar modulating system. Allowable premiums for 5-year paybacks are proportionally increased (by 5/3 to 67% larger than the 3-year values).

The values in Fig. 10 are absolute numbers as opposed to the percentages given in Figs. 8 and 9. As such, they will have a much wider variation for different houses, climates, and electricity costs than the percent energy savings and performance improvements. Nevertheless, the numbers in Fig. 10 should provide ballpark

EFFICIENCY IMPROVEMENT POTENTIAL

ONE HOUSE, ONE LOCATION

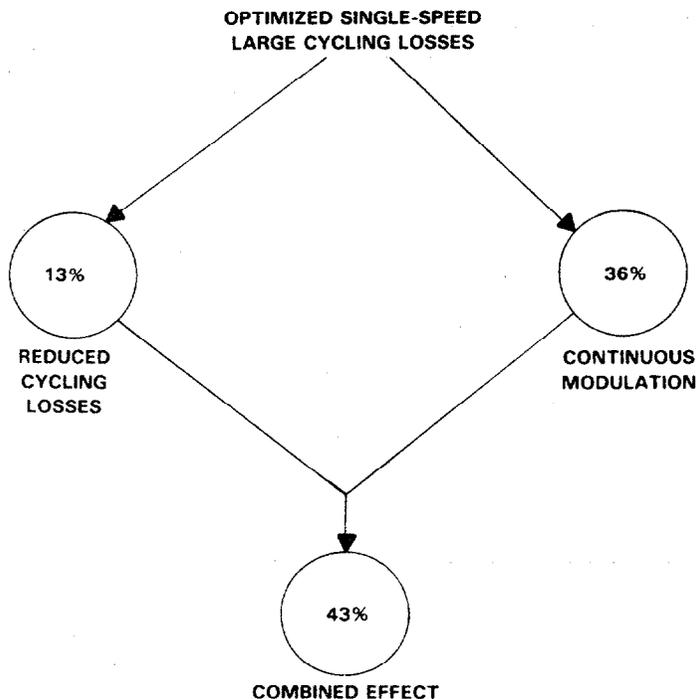


Fig. 9. Efficiency improvement potential relative to optimized reference system.

ALLOWABLE CONSUMER COST PREMIUMS FOR 3- AND 5-YEAR PAYBACKS

ONE HOUSE, ONE LOCATION

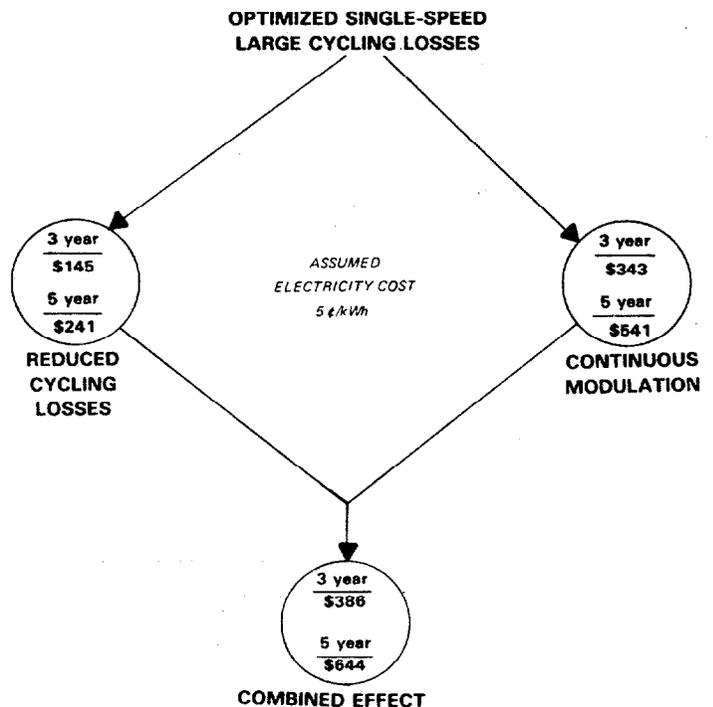


Fig. 10. Allowable consumer cost premiums for 3- and 5-year paybacks relative to optimized reference system.

values for the prices that the speed control and/or the cycling loss control devices must approach for short paybacks unless other concomitant benefits are considered.

Additional Advantages Of Continuous-Modulation-Systems

Continuous-capacity-modulation does offer other benefits that either are difficult to quantify with specific dollar figures, or relate to comfort and noise (which are somewhat subjective), or both. These include the potential for:

- improved compressor reliability since there would be far fewer on/off cycles and a much smaller inrush of motor current during startup;
- improved indoor temperature and humidity control because of the almost continuous operation of the unit when heating or cooling is required;
- multipurpose use of required microprocessor-based thermostats for setback, smart setup, and humidity control;
- reduced outdoor noise because of smaller required air flow rates;
- oversizing on cooling in northern climates to further reduce the backup heat needs while still providing adequate dehumidification in cooling, and
- significant additional operating cost savings through zone control (possibly by using a microprocessor thermostat to automatically adjust the capacity of the central unit by monitoring the average rate of temperature rise in the conditioned rooms).

Continuously modulating heat pumps would be uniquely well suited to zoning applications. For those whose lifestyles would allow them to reduce their annual house loads by zoning by 25%, the allowable first cost for case 3 would rise 69% to \$580 for a 3-year payback and 73% to \$936 for a 5-year payback (assuming the same annual performance factors apply for a zoned vs a non-zoned application⁹). This zoning application has been studied specifically for various possible lifestyles and is reported in the following paper.¹⁰

Benefits of System Optimization

In Fig. 11, the performance of available nonmodulating and modulating equipment is shown alongside comparable optimized results. It is estimated that all the cases have roughly equivalent levels of component efficiency. The solid dots in Fig. 11 show performance results for specific manufacturers' models of the various modulation types.

For all cases in Fig. 11, an ORNL binned APF model¹¹ was applied to either the manufacturers' or the heat pump model-generated steady state data. Dynamic loss factors (cycling and frosting/defrosting factors) for the commercial models were obtained directly from the manufacturers whenever available. For cases where data were unavailable or incomplete, consistent assumptions were made between the commercial and optimized

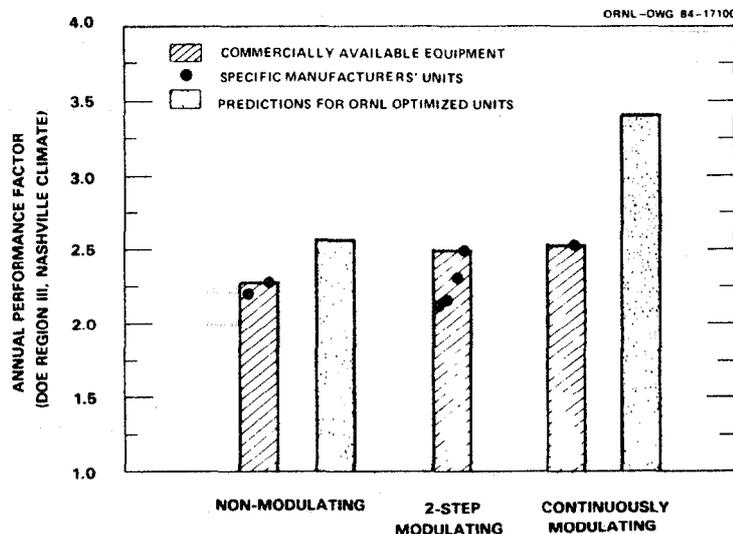


Fig. 11. Relative performance of current product lines vs optimized configurations—typical high-efficiency components.

cases. Default cyclic degradation coefficients (C_{Ds}) of 0.25⁵ were used for the optimized cases (for Fig. 11 only) and where manufacturers' values were not available.

From these comparisons, it is seen that there is a relatively small performance difference between the "optimized" and available nonmodulating equipment (13%) as compared to the much larger (35%) difference between the optimized and currently available CVS systems. Clearly, heat pump manufacturers are much closer to optimum designs (as would be expected) for the existing single-speed technology—where the manufacturers have had many years of design experience and product evolution. But for the newer modulating technology, our results using computer-aided design optimization indicate that substantial further benefits are obtainable. Such design tools become more important in the modulating heat pumps because these systems are:

- more complex—more design variables to integrate, more design flexibility to capitalize on;
- in higher performance areas—where design integration is more important in obtaining maximum performance; and
- under a more restrictive cost-vs-benefit situation—where designers must get the most efficiency out of the system to justify the cost of the improved components.

This seems to hold true even when higher-efficiency components are used and dynamic losses are reduced. In Fig. 12, the relative performances of the most advanced single-speed and step-modulating equipment are shown†† relative to comparable optimized CVS systems. In both CVS systems, compressor and indoor fan efficiency levels comparable to the Westinghouse/DOE heat pump¹² were assumed. In one case, dynamic loss factors equivalent to those measured for the Westinghouse/DOE unit were used. For the other CVS case, the lowest-loss combination of cycling and frosting/defrosting factors from the two advanced products was

††The performance of the step-modulating unit relative to the nonmodulating unit would be better in a more northern climate for which the unit was primarily intended.¹²

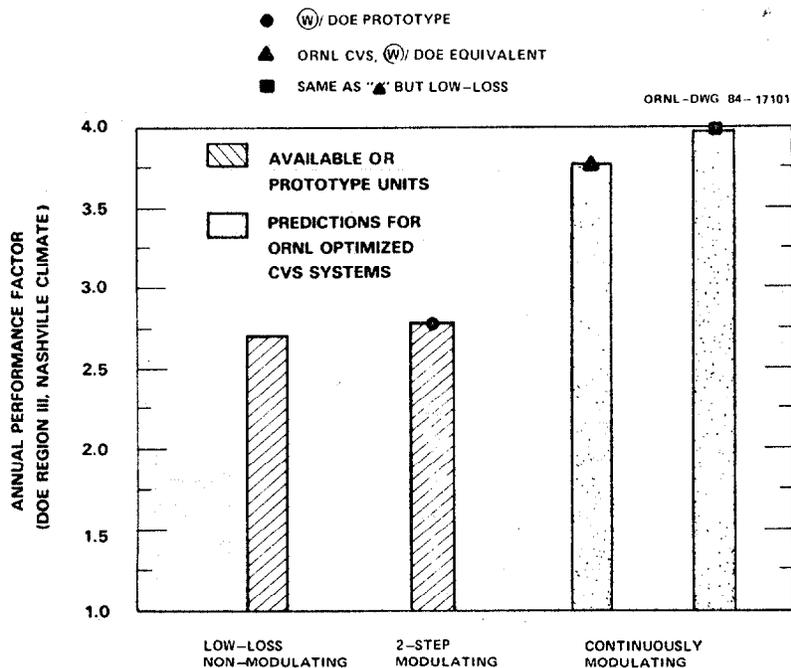


Fig. 12. Relative performance of advanced products vs optimized CVS configurations—best high-efficiency components available.

used. The Westinghouse/DOE-comparable CVS case shows a 35% performance advantage and the best CVS system has a 47% higher performance than the best nonmodulating system.†† These results suggest that system optimization benefits are equally available at both moderate and high component efficiency levels.

Recommendations

Based on this initial evaluation of CVS systems and considering the potential for electronics costs to further decrease as electricity prices rise, we see such advanced CVS systems as a strong future competitor to single-speed systems and as decidedly superior in energy conservation potential.

To achieve this position, it seems especially important that such systems be computer optimized to take full advantage of the increased design flexibility available. Further, development of PM-ECMs or equivalent technology must continue to the point of providing speed controllers of

- moderate to low cost,
- moderate to high performance, and
- high reliability

for successful penetration of this technology into the marketplace.

Status of Related Work and Future Activities

Since this initial assessment work was completed, efforts have been largely devoted to APF modeling, system design validation work, and collection of information to complete a more refined evaluation of CVS system benefits.

††Note that no judgements are being made here regarding the relative cost-effectiveness of the various advanced designs.

The APF model used in the initial capacity-modulation study has been greatly broadened to include step-modulation cases (as those shown in Figs. 11 and 12); a more general treatment of continuous-capacity modulation; a variety of frosting, defrosting, and cycling loss schemes; and thermally activated heat pumps. This program has been combined with a binned loads program¹³ and a 117-city weather data base¹⁴ to provide a convenient tool for scoping and optimization studies and a common comparison basis, including loss breakdowns, for a variety of heat pump types. Validation and documentation of the APF/Loads program has just recently been completed.¹¹ Documentation of a set of computer design tools for use with these models is well underway.¹⁵

The system design validation work includes further heat pump model validation against laboratory data on high-efficiency single-speed heat pumps and assessment of the effect of charge inventory assumptions and of optimized capillary tube sizing relative to more idealized refrigerant flow control schemes.¹⁶ Some further laboratory validation tests of the modulating heat pump model are planned as well.

Efforts have begun to more thoroughly evaluate the potential of continuous-capacity-modulation. More current and complete data are being obtained for the performance of PM-ECMs and reciprocating compressors as a function of operating speed and torque. Current plans are to optimize a PM-ECM-driven, reciprocating compressor system with this new data for a broader range of climates—with sufficient constraints added to ensure acceptable comfort levels. This analysis should (1) provide a good estimate of the national energy savings and application potential of one type of advanced modulating system, and (2) provide a credible demonstration of the use of a set of computer-aided design tools on a problem of interest to the heat pump industry as a whole.

Concurrent with these activities are efforts to transfer the programs and methodology to potential users in industry. Cooperative

formal and informal working arrangements are being actively pursued with individual manufacturers of heat pumps and compressors.

References

1. C. K. Rice, et al., *Design Optimization and the Limits of Steady-State Heating Efficiency for Conventional Single-Speed Air-Source Heat Pumps*, ORNL/CON-63, Oak Ridge National Laboratory, October 1981.
2. S. K. Fischer and C. K. Rice, *The Oak Ridge Heat Pump Models: I. A Steady-State Computer Design Model for Air-to-Air Heat Pumps*, ORNL/CON-80/R1, Oak Ridge National Laboratory, August 1983.
3. Numerical Analysis Group, *NAG Fortran Library, Mark 7*, Atomic Energy Establishment, Harwell, England (1978).
4. W. H. Parken, et al., "Factors Affecting the Performance of a Residential Air-to-Air Heat Pump," *ASHRAE Trans.* **83**(1), 839-850 (1977).
5. Department of Energy, "Test Procedures for Central Air Conditioners, Including Heat Pumps," 10 CFR Part 430, *Federal Register*, Vol. 44, No. 249, pp. 76700-76723, December 1979.
6. "ENMOD Split System Heat Pump," *York Application Data*, Form 515.50-AD1 (380).
7. J. D. Lloyd, "Variable-Speed Compressor Motors Operated on Inverters," *ASHRAE Trans.* **88**(1) (1982).
8. Ken Cooper, York Division, Borg-Warner, York, PA., personal communication with C. K. Rice, Oak Ridge National Laboratory, 1982.
9. J. C. Moyers and E. A. Nephew, *Scoping Evaluation of Potential Benefits of Zoning with Residential Space-Conditioning Applications*, ORNL/CON-167, Oak Ridge National Laboratory (in press).
10. J. C. Moyers and E. A. Nephew, "Analytical Study of the Benefits of Zone-Control Heat Pumps," presented at the DOE/ORNL Heat Pump Conference in Washington, D.C., Dec. 11-12, 1984.
11. C. K. Rice, et al., *The Oak Ridge Heat Pump Models: II. An Annual Performance Factor/Loads Model for Residential Air-Source Heat Pumps*, ORNL/CON-160, Oak Ridge National Laboratory (to be published).
12. S. E. Veyo and T. J. Fagan, *Advanced Electric Heat Pump Dual-Stroke Compressor and System Development, Final Report*, ORNL/Sub/79-24712/3, December 1983.
13. M. L. Ballou, et al., *MAD: A Computer Program for ACES Design Using Monthly Thermal Loads*, ORNL/CON-51, Oak Ridge National Laboratory, March 1981.
14. U. S. Air Force, *Engineering Weather Data*, AFM-88-29, July 1978.
15. S. K. Fischer, et al., *The Oak Ridge Heat Pump Models: III. Design Tools for Heat Pump Development*, ORNL/CON-175, Oak Ridge National Laboratory (to be published).
16. S. K. Fischer and C. K. Rice, "System Design Optimization and Validation for Single-Speed Heat Pumps," submitted for presentation at the ASHRAE 1985 Summer Meeting.