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R&D Opportunities for Commercial HVAC Equipment

S. A. Chiu
F. R. Zaloudek

March 1987

Prepared for the U.S. Department of Energy
under Contract DE-AC06-76RLO 1830

Pacific Northwest Laboratory
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R&D OPPORTUNITIES FOR COMMERCIAL HVAC EQUIPMENT

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Richland, Washington 99352

SUMMARY

A substantial amount of energy is consumed annually to satisfy space heating, air conditioning and ventilation (HVAC) requirements in commercial buildings. According to the U.S. Department of Energy (DOE), the total primary energy consumption for commercial buildings was 10.72 quadrillion Btu (quad) in 1983, compared with 14.74 quad for the residential sector. About 60% of the commercial sector energy consumption was used for HVAC applications (DOE 1985). The DOE has funded considerable research on residential HVAC equipment and is now examining similar types of research for the commercial sector. As part of this investigation, the DOE has contracted with the Pacific Northwest Laboratory (PNL) to conduct a study to identify research and development (R&D) opportunities for large commercial HVAC equipment.

Commercial buildings as a class are diverse in size and business functions. They have complex HVAC systems which are made up of the HVAC equipment, the distribution system, and the control system. As an initial task, the focus of the current effort is on the HVAC equipment. HVAC systems as a whole will be addressed in a later project.

Based on 1984 HVAC equipment shipment statistics (U.S. Department of Commerce 1984, ACHRN 1985), the equipment types identified to have the most impact on national energy use are the electric centrifugal chiller, electric packaged and split air conditioners, oil boiler, gas boiler, and gas furnace. Since oil boiler research will be covered under the DOE Oil-Fueled Equipment Program and substantial gas-fired equipment research is being sponsored by the Gas Research Institute (GRI) (GRI 1985), the current study addresses the electric air conditioners and centrifugal chillers.

The air conditioner market is dominated by small-sized units. The average size of air conditioner shipped, excluding packaged terminal units, is 10 to 15 ton, and the majority of the units are 5 to 10 ton. On the other hand, centrifugal chillers are commonly used in large buildings. The average size of chiller shipped is 400 to 500 horse power (hp), and the most popular size is 300 to 400 hp.

In this study, the performance and improvement potentials of the current-market centrifugal chillers and air conditioners have been assessed based on analysis and discussions with manufacturers' field representatives and sales officers and also on information published by the Air-Conditioning and Refrigeration Institute (ARI) (ARI 1986). The conclusion is that established methods are presently available to produce high performance water chillers. For example, for medium-sized centrifugal chillers, compressor energy consumption rates of as low as 0.5 kW/ton, or an equivalent energy efficiency ratio (EER) of 24 Btuh/W, may be attained using established technologies. However, because of the corresponding increases in capital equipment cost and other institutional impediments, energy-efficient features are often not incorporated. For the highly competitive air conditioner market, an even more drastic economic barrier is in place because this market is strongly attuned to equipment first-cost. The average EERs for 10-ton air conditioners listed in the 1986 ARI Directory (ARI 1986) cover a range from 7.1 to 10.6 Btuh/W (including auxiliary power). Nonetheless, established methods analogous to those for chillers may be used to produce higher performance air conditioners.

A literature search was also conducted to review past and ongoing research that may be applied to improve the energy efficiency of the air conditioners and chillers. A computer search of the DOE Energy Database was performed to aid the effort, and a manual search was conducted to focus on publications from the American Society of Heating, Refrigeration and Air Conditioning Engineers (ASHRAE), DOE reports, Electric Power Research Institute (EPRI) reports, GRI reports, conference and workshop proceedings, abstracts from the American Society of Mechanical Engineers (ASME) Applied Mechanics Review, and some relevant U.S. patents. In addition, HVAC researchers and practitioners from various organizations and communities, including academia, research laboratories, manufacturers, trade organizations, and consulting firms, have been contacted to solicit their expert opinions on the subject.

The combined results from this study of current equipment and from this review of past and current research indicate that because air conditioner and chiller technologies are mature technologies, the potential for their performance improvement is not stymied by technical limitations. The factors

constraining the energy efficiency of the current equipment are basically non-technical and primarily economic. If cost were not a factor and if the design features used to improve operating efficiency were uniformly applied, substantial reduction in energy consumption would result. Obviously, the most advantageous technology advancements to improve energy efficiency are those that would reduce the incremental cost of the established energy-efficient features. In addition, the promotion of life-cycle costing in the context of a DOE equipment research program will also encourage the use of energy-efficient equipment. Based on the current assessment, PNL recommends that DOE address the following specific equipment-level R&D on electric air conditioners and centrifugal chillers:

- promote better understanding and engineering of heat exchanger coils
- support R&D for variable capacity drives, including AC inverters for chillers and fans, and brushless DC units for air conditioners
- support R&D on nonazeotropic refrigeration cycle research for chillers for low-temperature application and improved part-load performance
- assess environmentally benign refrigerants and their respective refrigeration cycles.

Developing more effective or lower cost heat exchangers will have immediate impact on the performance of air conditioning equipment. In addition, since heat exchanger surface fouling is the most common cause of long-term equipment performance degradation, studies on innovative ways to retard its buildup and to prevent the potential subsequent equipment failure will prove useful to equipment design and operation.

Variable-capacity inverter drives are becoming very popular in Japan. Among the U.S. manufacturers, interest in inverter drive applications, particularly to large chillers, is increasing because of the decline in the cost of inverters. Competition will likely increase as the Japanese gain more shares of the market. Currently, strong private-sector development is taking place. However, supporting R&D from DOE is needed. Moreover, because of the high energy requirements of the fans used in commercial buildings, inverter-driven

variable-speed fans should also be studied. Finally, the brushless DC units used in smaller sized air conditioners will also benefit from further research.

Applying the nonazeotropic refrigeration cycle will potentially improve the performance of chillers. The nonazeotropic system is complex but may yield important performance improvements in producing low-temperature chilled water. Subsequently, the distribution system size and energy requirements may be reduced. In addition, better load-matching and part-load efficiency will result.

The two most often used refrigerants in commercial air conditioning equipment are R-11 and R-12. However, they are considered the most environmentally detrimental among chlorofluorocarbon refrigerants. Investigations must be conducted to assess alternative, more environmentally benign refrigerants and their respective refrigeration cycles.

Although the primary objective of this study was to identify the R&D opportunities of air conditioners and chillers at the equipment level, it was also the first step in an effort to determine the overall R&D needs in commercial HVAC systems. Although the comments from HVAC researchers and practitioners identified a number of different R&D needs, the consensus emphasized the importance of system-level problems, part-load issues, and control issues. It has been repeatedly suggested by the experts that system-level HVAC problems play an important role in energy conservation in commercial buildings. These system problems include both the problems of incorporating individual equipment components to produce an operating unit as well as problems with operating the equipment, together with the distribution and control systems as part of the whole building system. Better equipment part-load performance will substantially impact seasonal performance. Because of the decline in cost of electronics, U.S. manufacturers are increasingly interested in systems control. Both better systems integration and part-load performance will benefit by improved control strategies and hardware.

As a result of the current assessment and of input from HVAC researchers throughout this research project, recommendations are also made here to address commercial HVAC systems. System issues in commercial buildings are more important to energy consumption and thermal comfort than is the efficiency of the

air conditioner or chiller. The keys to energy conservation in commercial buildings are the integrated design of the HVAC equipment, distribution system, control system, and building envelope, and the optimization of this integrated system. The system-level research required has two elements. First, detailed knowledge and basic information on many of the system-related issues, especially the fundamental engineering issues, are lacking. They should be added to the research agenda of the DOE Building Equipment Programs. Secondly, detailed trade-offs of various energy conservation designs have not been well studied. The DOE Building System Programs are beginning to address this area. Computer and measurement tools now exist for developing detailed analyses in the system area. The improvements on commercial HVAC system performance offer tremendous opportunity for energy conservation. Therefore, research should be started in order to structure and specify research problems in commercial HVAC systems.

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1.0 INTRODUCTION

A substantial amount of energy is consumed annually to satisfy space heating, air conditioning and ventilation (HVAC) requirements in commercial buildings. According to the U.S. Department of Energy (DOE), the total primary energy consumption for commercial buildings was 10.72 quadrillion Btu (quad) in 1983, compared to 14.74 quad for the residential sector. About 60% of the commercial sector energy consumption is used for HVAC applications (DOE 1985).

The DOE has funded considerable research on residential HVAC equipment and is now examining similar types of research for the commercial sector. As part of the investigation, the DOE has contracted with the Pacific Northwest Laboratory (PNL)[a] to conduct a study to identify the research and development (R&D) opportunities for large commercial HVAC equipment.

Commercial buildings as a class are diverse in size and business functions, and have complex HVAC systems because of the dynamic interactions among the building structure, HVAC equipment, distribution system, and control system. To gain insight into HVAC systems, the system as a whole must be addressed. As an initial task, the focus of the current effort is solely on the HVAC equipment. HVAC systems in commercial buildings will be analyzed in a later project.

1.1 OBJECTIVE AND APPROACH

The overall objective of this project is to identify and characterize generic HVAC equipment research that will provide the best investment opportunities for DOE R&D funds. The prerequisites of a DOE research program include research efforts that are potentially significant in energy conservation impact and that are cost-effective, long-term, and high risk. These prerequisites form the basic guidelines for the R&D opportunities assessed. The assessment excludes the R&D areas that have potential or current private sector sponsors. Finally, R&D areas which are included in DOE programs generally are not addressed.

(a) PNL is operated for the U.S. Department of Energy by Battelle Memorial Institute under Contract DE-AC06-76RLO 1830.

To target the candidates for equipment research, the initial task of this project was to identify the equipment types whose performance improvements indicated the greatest potential impact on national energy. For the equipment types identified, the performance of the units available in the marketplace was then assessed to determine improvement potentials of each. Finally, the advanced research areas applicable to the targeted equipment candidates were identified. The scope of this effort required that evaluations of the advanced technologies be preliminary rather than detailed. The detailed analyses will be performed as part of the future research based on the R&D recommendations generated from this project.

This project was composed of the following three tasks:

Task 1: Selection of Equipment Research Candidates--The objective of this task was to target three equipment types for research. These equipment types had to have a large potential energy impact as a result of improvements in performance. Their potential energy impacts were examined based on available equipment shipment statistics and energy ratings.

Task 2: Performance Assessment of Current-Market Equipment--The objective of this task was to evaluate the improvement potential of the equipment currently available on the market. Performance was assessed, and opportunities for improvement were identified based on information gathered from manufacturers.

Task 3: Assessment of Equipment Research--The objective of this task was to assess advanced research which might be applicable to improving the performance of the targeted equipment. The potential research areas were identified based on a literature search and discussions with HVAC equipment researchers.

1.2 ORGANIZATION OF REPORT

This report summarizes the results of the three tasks performed in the assessment of R&D opportunities for commercial HVAC equipment. The report is organized into six chapters. Chapter 2.0 documents the results of the equipment shipment inventory and presents the three types of equipment selected. Observations on the performance and improvement potentials of units that are

available in the marketplace are discussed in Chapter 3.0. Chapter 4.0 summarizes the findings from a literature search and from discussions with HVAC researchers and practitioners on potential R&D opportunities. Among the opportunities identified, PNL's recommendations on DOE-sponsored commercial HVAC research are presented in Chapter 5.0. The references are listed in Chapter 6.0, followed by Appendix A, which includes a discussion on the generic energy inefficiencies of vapor compression equipment, and Appendix B, which speculates on the national energy savings resulting from the recommended research.

2.0 SELECTION OF COMMERCIAL HVAC EQUIPMENT

The first step in identifying the R&D opportunities for large HVAC equipment is to determine the equipment types whose performance improvements will most significantly impact future energy use. Assuming no major changes in the trend of equipment sales, these are the same equipment types recently sold and installed which have the highest collective energy consumptions.

The HVAC equipment shipment statistics for 1984 are obtained from data published by the U.S. Bureau of the Census (U.S. Department of Commerce 1984) and by Air Conditioning, Heating & Refrigeration News (ACHRN 1985). These data reflect statistics of equipment shipped and presumably sold and installed in 1984. They include a detailed breakdown of the number of units shipped for various equipment types and sizes.

In addition to performance characteristics, the energy requirements of the HVAC equipment depend on the characteristics of the distribution systems used and the buildings served. Commercial buildings are diverse in size as well as in business functions. In order to make use of the shipment data to properly characterize the energy impact of the various types of HVAC equipment, an extensive effort is required. For each equipment type, a detailed inventory is required regarding the number of units sold, their capacity and efficiency characteristics, the type of distribution system used, and the function, size, operation, and location of the buildings in which they are installed. Moreover, computer simulations are needed to estimate the seasonal energy consumption of the equipment for each particular application.

Time and cost constraints precluded such a detailed analysis to calculate the energy impacts of the equipment. Therefore, a simplified alternative approach has been used. For the simplified approach, the relative energy impacts of the various equipment types are compared based on their collective rated power requirements. The equipment types thus selected are the electric centrifugal chillers and the electric packaged and split air conditioners. The approach used and the results are described in the following sections.

2.1 APPROACH

To be consistent with the published shipment data, all available equipment types are classified into the following categories (U.S. Department of Commerce 1984):

- CHILLERS
  - Centrifugal Units
    - Hermetic Units
    - Open Units
  - Absorption Units
  - Reciprocating Units
- AIR CONDITIONERS
  - Packaged Units
    - Cooling Only Units
    - Year-Round Units
  - Split Units
  - Packaged Terminal Units
- FURNACES
  - Gas-Fired Units
  - Oil-Fired Units
- BOILERS
  - Gas-Fired Units
  - Oil-Fired Units
- HEAT PUMPS
  - Air-Source Units
    - Packaged Units
    - Split Units
  - Water-Source Units
  - Packaged Terminal Units

Table 2.1 summarizes the equipment shipment data used in the analysis (U.S. Department of Commerce 1984, ACHRN 1985). The available inventory is further divided into equipment sizes. The size data are expressed in terms of the rated horse power (hp) inputs for the chillers and rated capacities for the other equipment types. For this study, air conditioners and heat pumps with capacities below 5 tons, and furnaces and boilers with capacities below 300 kBtuh were considered residential-sized equipment and were not included in the following analysis.

TABLE 2.1. HVAC Equipment Shipment Data

HVAC Equipment	Units Shipped/Year
CHILLER	17,005
Centrifugal	3,089
Hermetic	3,089
Below 200 hp	489
201 to 300	638
301 to 400	754
401 to 500 (EST)[a]	600
501 to 1000	408
Over 1000 (EST)	200
Open Type	?
Absorption	153
Reciprocating	13,763
Below 20 hp	6,729
21 to 49	2,236
50 to 75	2,049
Over 75	2,749
AIR CONDITIONER	314,246
Packaged--Cooling Only	34,331
Horizontal	24,634
65 to 96.9 kBtu[b]/hr	8,207
97 to 134.9	6,745
135 to 184.9	3,815
185 to 249.9	1,636
250 to 319.9	922
320 to 379.9	897
380 and Over	2,412
Other than Horizontal	9,697
65 to 96.9 kBtu/hr	2,244
97 to 134.9	2,370
135 to 184.9	1,665
185 to 249.9	1,020
250 and Over	2,398
Packaged--Year-Round	54,648
65 to 134.9 kBtu/hr	35,424
135 to 184.9	9,867
185 to 229.9	4,276
250 to 319.9	1,930
320 to 379.9	804
380 to 539.9	656
540 to 639.9	738
640 and Over	953

TABLE 2.1. (contd)

HVAC Equipment	Units Shipped/Year
Split	40,487
65 to 96.9 kBtu/hr	15,649
97 to 134.9	10,676
135 to 184.9	7,141
185 to 249.9	2,702
250 to 319.9	1,270
320 to 379.9	971
380 to 539.9	858
540 to 639.9	539
640 and Over	681
Packaged Terminal	184,780
FURNACE	11,178
Gas	7,430
300 to 400 kBtu/hr (EST)	5,928
Over 400	1,502
Oil	3,748
300 to 400 kBtu/hr (EST)	3,072
Over 400	676
BOILER	16,221
Gas	7,660
300 to 450 kBtu/hr (EST)	4,447
450 to 950	1,909
950 to 1,550	873
1,550 and Over	431
Oil	8,561
300 to 450 kBtu/hr (EST)	3,425
450 to 950	2,066
950 to 1,550	1,155
1,550 and Over	1,915
HEAT PUMP	112,271
Air-Source	16,908
Packaged	
65 kBtu/hr and Over	9,715
Split	
65 kBtu/hr and Over	7,193
Water-Source (EST)	1,814
Packaged Terminal	93,549

(a) Estimated values are identified with (EST).
(b) kBtu = 10^3 Btu.

The inventory data for some items were unavailable, and either estimations were made or they were excluded from further analysis. In particular, the data for open-type centrifugal chillers have not been included in industry figures since 1982 in order to protect individual manufacturers' statistics (ACHRN 1985). In addition, the size breakdown for the small number of absorption chillers sold is not available. These two types of equipment have been excluded in this study.

The basic assumptions underlying the simplified analysis approach are that the average seasonal operation hours of the various equipment types and their various sizes and applications are approximately the same. The relative annual energy consumptions of the various equipment types were compared based on their power input requirements, which are calculated using Equations (2.1) and (2.2) for chillers and all other equipment, respectively:

$$[\text{INPUT}] = [\text{UNIT SHIPPED}] * [\text{RATED INPUT}] \tag{2.1}$$

$$[\text{INPUT}] = [\text{UNIT SHIPPED}] * [\text{RATED CAPACITY}] / [\text{EFFICIENCY}] \tag{2.2}$$

The averaged seasonal efficiencies used for the various equipment were estimated based on information reported in the Air Conditioning, Heating and Refrigeration News (ACHRN 1985). In summary, the following efficiencies were assumed:

Equipment Type	Efficiency
Air Conditioners or Heat Pumps	2.50
Furnaces	0.75
Boilers	0.70

Since the heat pumps operate in both space heating and cooling modes, their annual run times are higher than those for other equipment types. This has not been considered in the analysis. However, because of the substantially smaller inventory of heat pumps, the energy impacts are relatively insignificant regardless of run time (see Section 2.2).

Admittedly, the approach used is crude but offers an indication of the relative energy impacts of different HVAC equipment without involving building stock and equipment inventory characterizations or building equipment performance simulations.

2.2 COMMERCIAL HVAC EQUIPMENT SELECTED

Table 2.2 gives the tabulated results showing the input power requirements calculated using Equations (2.1) and (2.2). These are power consumption rates on site. The following six equipment types have been identified as most energy intensive in terms of their annual power inputs:

- oil boiler
- gas boiler
- electric packaged air conditioner
- gas furnace
- electric centrifugal chiller
- electric split air conditioner.

This observation is consistent with the statistics reported in the Gas Research Institute (GRI) 1986 to 1990 R&D Plan (GRI 1985). This GRI report indicates that 75% of the commercial buildings have a floor area under 10,000 sq. ft., and most average from 1,000 to 5,000 sq. ft. The space conditioning of these buildings is usually provided by rooftop gas furnaces and electric air conditioners. On the other hand, high-rise buildings of over 50,000 sq. ft. account for over 40% of the total floor area in the commercial sector; and they are generally conditioned by large central boiler and chilled water systems (GRI 1985). Electric heat pumps make up only 5% and 6% of the commercial heating and cooling systems, respectively (Brodrick 1986).

Among the six types of equipment identified above, the current assessment has been conducted for the three electric equipment types only, namely centrifugal chillers and packaged and split air conditioners. Oil boiler research will be covered under the DOE Oil-Fueled Equipment Program. Substantial gas-fired equipment research is presently being sponsored by GRI. According to GRI's 1986 to 1990 R&D plan (GRI 1985), their strategy focuses on

TABLE 2.2. HVAC Equipment Energy Requirement Summary

	Units Shipped/yr	Capacity kBtu[a]/ hr/Unit	Power kBtu/ hr/Unit	Efficiency	Energy Input MMBtu[b]/ hr/yr
CHILLER					
Centrifugal					
Hermetic	2,864				3,217
Below 200 hp	489		341.0		167
201 to 300	638		636.4		406
301 to 400	754		890.9		672
401 to 500 (EST)[c]	375		1,145.4		430
501 to 1,000	408		1,909.1		779
Over 1,000 (EST)	200		3,818.1		764
Reciprocating	13,763				1,542
Below 20 hp	6,729		34.1		229
21 to 49	2,236		89.1		199
50 to 75	2,049		159.1		326
Over 75	2,749		286.4		787
AIR CONDITIONER					
Packaged--Cooling Only	34,331				2,484
Horizontal	24,634				1,746
65 to 96.9 kBtu/hr	8,207	81.0		2.50	266
97 to 134.9	6,745	116.0		2.50	313
135 to 184.9	3,815	160.0		2.50	244
185 to 249.9	1,636	217.5		2.50	142
250 to 319.9	922	285.0		2.50	105
320 to 379.9	897	350.0		2.50	126
380 and Over	2,412	570.0		2.50	550
Other than Horizontal	9,697				738
65 to 96.9 kBtu/hr	2,244	81.0		2.50	73
97 to 134.9	2,370	116.0		2.50	110
135 to 184.9	1,665	160.0		2.50	107
185 to 249.9	1,020	217.5		2.50	89
250 and Over	2,398	375.0		2.50	360
Packaged--Year-Round	54,648				3,414
65 to 134.9 kBtu/hr	35,424	100.0		2.50	1,417
135 to 184.9	9,867	160.0		2.50	631
185 to 249.9	4,276	217.5		2.50	372
250 to 319.9	1,930	285.0		2.50	220
320 to 379.9	804	350.0		2.50	113
380 to 539.9	656	460.0		2.50	121
540 to 639.9	738	590.0		2.50	174
640 and Over	953	960.0		2.50	366
Split	40,487				2,522
65 to 96.9 kBtu/hr	15,649	81.0		2.50	507
97 to 134.9	10,676	116.0		2.50	495
135 to 184.9	7,141	160.0		2.50	457
185 to 249.9	2,702	217.5		2.50	235
250 to 319.9	1,270	285.0		2.50	145
320 to 379.9	971	350.0		2.50	136
380 to 539.9	858	460.0		2.50	158
540 to 639.9	539	590.0		2.50	127
640 and Over	680	960.0		2.50	262
Packaged Terminal (C EST)[d]	184,780	12.0		2.50	887
FURNACE					
Gas	7,430				3,968
300 to 400 kBtu/hr (EST)	5,928	350.0		0.75	2,766
Over 400	1,502	600.0		0.75	1,202
Oil	3,748				1,974
300 to 400 kBtu/hr (EST)	3,072	350.0		0.75	1,434
Over 400	676	600.0		0.75	541

TABLE 2.2. (contd)

	Units Shipped/yr	Capacity kBtu/ hr/Unit	Power kBtu/ hr/Unit	Efficiency	Energy Input MMBtu/ hr/yr
BOILER					
Gas	7,660				7,282
300 to 450 kBtu/hr (EST)	4,447	375.0		0.70	2,382
450 to 950	1,909	700.0		0.70	1,909
950 to 1,550	873	1,250.0		0.70	1,559
1,550 and Over	431	2,325.0		0.70	1,432
Oil	8,561				12,324
300 to 450 kBtu/hr (EST)	3,425	375.0		0.70	1,835
450 to 950	2,066	700.0		0.70	2,066
950 to 1,550	1,155	1,250.0		0.70	2,063
1,550 and Over	1,915	2,325.0		0.70	6,361
HEAT PUMP					
Air-Source	16,908				659
Packaged					
65 kBtu/hr and Over	9,715	97.5		2.50	379
Split					
65 kBtu/hr and Over	7,193	97.5		2.50	281
Water-Source (EST, C EST)	1814	97.5		2.50	71
Packaged Terminal (C EST)	93,549	12.0		2.50	449

(a) kBtu = 10^3 Btu.
(b) MMBtu = 10^6 Btu.
(c) Estimated shipment data are identified with (EST).
(d) Estimated capacity data are identified with (C EST).

the adaptation of the state-of-the-art residential-sized pulse and condensing technologies, which promise high-performance gas furnaces and boilers with efficiencies of above 90%. On the other hand, research on the three commercial-sized electric equipment types is not being specifically addressed in the Electric Power Research Institute (EPRI) 1985-1989 R&D Program Plan (EPRI 1984). Although some of the EPRI research conducted for heat pumps will be applicable to these cooling-only equipment types, no specific research work on the chillers and air conditioners has been planned.

The R&D opportunities to be assessed in this study are generic in terms of equipment capacities. However, representative sizes were necessary for the detailed survey of the performance of equipment currently on the market. The representative sizes chosen were based on their relatively large energy impacts as well as on shipment inventory. The air conditioner market is dominated by small-sized units. The average size of an air conditioner shipment, excluding the packaged terminal units, is 10 to 15 ton; the majority of the units are

from 5 to 10 tons. Conversely, the centrifugal chillers have large-tonnage applications. The average size of chiller shipment is 400 hp to 500 hp, and the most popular size is 300 hp to 400 hp. As a result, relatively small sized packaged and split air conditioners of 10 tons were selected. For the chillers, medium sized units of 350 hp (or an equivalent of 375 tons for an estimated efficiency of 0.7 kW/ton) were used.

3.0 ASSESSMENT OF CURRENT-MARKET AIR CONDITIONERS AND CENTRIFUGAL CHILLERS

This section describes assessments of the commercial air conditioning equipment selected in Chapter 2.0. The types of equipment assessed were:

- packaged and split air conditioners of approximately 10-ton refrigeration capacity
- centrifugal water chillers of 350- to 375-ton capacity.

This assessment was performed on the basis of information published by the Air-Conditioning and Refrigeration Institute (ARI 1986), manufacturers' catalogs, and personal communication with manufacturers' field representatives and sales officers.

The ARI rating conditions and the performance of vapor compression equipment are described in Appendix A, and the specifics will not be repeated in this chapter.

3.1 PACKAGED AND SPLIT AIR CONDITIONERS

This section describes the results of an assessment of commercial 10-ton capacity, water- and air-cooled, packaged and split air conditioners. These results are described from the viewpoint of energy efficiency for the purpose of evaluating the improvement potential.

3.1.1 Performance Data and Methods Used to Obtain Energy Efficiency

The ARI Directory of Certified Unitary Air-Conditioners, Unitary Air-Source Heat Pumps and Sound-Rated Outdoor Unitary Equipment (January 1 to June 30, 1986) (ARI 1986) lists over 135 air conditioners with approximately 10-ton capacity. The EERs of listed units range from a low of 7.1 Btuh/W to a high of 10.6 Btuh/W, with an average of 8.3 Btuh/W (including auxiliary power).[a] The ideal Carnot EERs for air- and water-cooled units operating

(a) For 10 ton air conditioners, evaporator fans are typically 1.5 to 2 hp and condenser fans are 1 to 1.5 hp. For a unit with EER of 8.5, for example, auxiliary power would range from 13 to 18% of the total requirement.

with perfect heat transfer at the evaporator and condenser at ARI inlet conditions are 123 and 369 Btuh/W, respectively; the corresponding basic ideal vapor compression cycle EERs using refrigerant R-22 are approximately 100 and 270 Btuh/W, respectively. However, because of the need to provide for the dehumidification of the cooled air, the evaporator must be operated at considerably lower temperatures than the dew point temperature of the inlet air. The EERs of air- and water-cooled units operating at ARI standard conditions while maintaining a 40°F evaporator temperature for dehumidification purposes are 27.6 and 34.7 Btuh/W, respectively.

The EER specifications of the top 10 units taken from the manufacturers' catalogs are summarized in Table 3.1. Also listed for comparison purposes are the specifications of units with average and below average EERs. However, the condenser and evaporation pressures during operation at ARI conditions are not available. Table 3.1 shows that the highest EERs are obtained by water-cooled units. In fact, in the entire ARI listing of 135 units, all 7 water-cooled units listed fell in the top 16 units of all types listed, which is partly because of the difference in the standard rating inlet conditions of 95°F air and 85°F water for the air- and water-cooled equipment, respectively. For air-cooled air conditioners, the features that seem to characterize the units with highest EERs include large evaporator heat transfer surface, large condenser heat transfer surface, high condenser air flowrates, or all of these. These general characteristics are consistent with the features required for high energy efficiency discussed in Appendix A, which, however, typically result in increased equipment costs.

Sales and engineering representatives of the manufacturers of units with top EERs were contacted and catalog data were surveyed to determine the reasons for the better performance of these units. Reasons obtained include:

- efficient heat transfer surfaces, e.g., large frontal area coils with low pressure drop, deep coils, small fin pitch, shaped pins, large air flow, and external/internal fins on water-cooled condensers
- special liquid subcooling heat exchangers which increase system capacity without additional compressor horsepower

TABLE 3.1. Summary of Selected 10-Ton Air Conditioner Design Features

					Evaporator						Concenser						Compressor		
Manufacturer	Model	Type	EER (Btuh/W)	Refrigerant	Fin Type	Face Area ft^2	Materials (Fins/Tubes)	Fins/in.	Tube Rows	CFM/HP Nominal	Fin Type	Face Area ft^2	Materials (Fins/Tubes)	Fins/in.	Tube Rows	CFM/HP (GPM)	Type	Special Part-Load Regulatio	Comments
York	C2ED120A25	Water-Cooled Unitary	10.6	22	plain	10.2	Al[a]/Cu[b]	13	3	4000/2	----------Water Cooled----------					(8-40)	Hermetic(2)	NA	
Carrier	50BT012	Water-Cooled Unitary	10.45	22	plain	8.5	Al/Cu	12.1	3	4000/2	----------Water Cooled----------					(11-18)	Semi-Hermetic	Unload cylinders (2 of 6)	
Trane	SWUB-C10	Water-Cooled Unitary	9.8	22	wavy fin	9.8	Al/Cu	12	3	4000/2	----------Water Cooled----------					NA	Hermetic(2)	NA	
Carrier	50BU12 90E016 (coil)	Air-Cooled Separate Cond. Coil	9.3	22	plain	8.5	Al/Cu	12.1	3	4000/2	plain	29.2	Al/Cu	15	3	9600/1.5	Semi-Hermetic	Unload cylinders (2 of 6)	Special Circuits for 15°F Subcooling
Carrier	38AE(Cond.) 40RR16 (evap.)	Air-Cooled Separate Units	9.3	22	plain	16.8	Al/Cu	15	3	4000/2	plain	29.2	Al/Cu	15	2	4000/1	Semi-Hermetic	Bypass	Large port area for higher compressor efficiency
Carrier	50BU12 9AB12 (coil)	Air-Cooled Separate Cond. Coil	9.2	22	plain	8.5	Al/Cu	12.1	3	4000/2	plain	18.0	Al/Cu	13.5	3	8800/1	Semi-Hermetic	Unload cylinders (2 of 6)	Special circuits for subcooling
Sunbeam	1264-120-1-3 1591 (evap).	Air-Cooled Separate Coil	9.2	22	plain	9.6	Al/Cu	NA	4	NA/2	NA	23.94	Al/Cu	NA	NA	3000-5000/1	Hermetic	NA	Large condenser coil area
Carrier	38AE (cond.) 40RR12 (evap.)	Air-Cooled Separate Coil	9.0	22	plain	10.7	Al/Cu	15	2	4000/2	plain	29.2	Al/Cu	15	2	4000/1	Semi-Hermetic	Bypass	Large port area for higher compressor efficiency
Carrier	50BU012 9AB8(coil)	Air-Cooled Separate Coil	8.9	22	plain	8.5	Al/Cu	12.1	3	4000/2	plain	18.0	Al/Cu	13.5	3	6600/1	Semi-Hermetic	Unload cylinders	Special circuits for subcooling
Sunbeam	RC1202	Air-Cooled Unitary	8.6	22	plain	9.5	Al/Cu	15	4	4000/NA	plain	17.4	Al/Cu	13	4	8900/NA	Hermetic(2)	Bypass	
York	D1EA120A25	Air-Cooled Unitary	7.1	NA	plain	7.8	Al/Cu	12	5	4000/1.5	plain	11.62	Al/Cu	10	5	4000/3	Hermetic(2)	NA	
Trane	SAUA-C10	Air-Cooled Unitary	6.85	22	wavy fin	8.0	Al/Cu	12	3	NA/2	wavy fin	10.38	Al/Cu	12	6	6580/3	Hermetic(2)	NA	

(a) Aluminum
(b) Coppper

- efficient compressor performance because of large gas passageways and valves that reduce turbulence losses
- the effective matching of large condenser and evaporator coils with an efficient compressor.

All of the air conditioners examined used refrigerant R-22 and only the simple basic refrigeration cycle.

Compressor types used include both hermetic and semi-hermetic (i.e., capable of being disassembled for repair). The most frequently used method of capacity regulation was intermittent operation of the compressor. Some units used two hermetic compressors, with only one compressor used at part load to reduce the frequency of intermittent operation. One manufacturer using semi-hermetic compressors employed cylinder unloading (two to six cylinders) for regulation. Another manufacturer offered the hot gas bypass method as an option. This method diverts a part of the compressed gas from the high- to the low-pressure side of the compressor. Although the hot gas method may effectively limit the capacity of the machine, it does not limit power consumption correspondingly. With the bypass to a cylinder open, the power consumption of that cylinder may still be 65% to 90% of the consumption when the bypass is closed.

From ASHRAE tables of refrigerant performance (ASHRAE 1985), it is apparent that another way to influence the energy efficiency of a unit is by the selection of the refrigerant, and variations in EER of up to 20% can be obtained. For example, if R-11 is used instead of R-22, ideal EER may be improved by approximately 8%. However, the choice of refrigerant for a particular application is not dictated by energy efficiency alone, but by its heat transfer properties, evaporator pressure, density, availability, toxicity, and more recently, environmental impacts.

Manufacturers' representatives indicate that cost and reliability strongly influence whether these energy-efficiency improvements are used in air conditioners. Available methods to improve energy efficiency are well established. All components of the basic refrigeration cycle can be improved somewhat by reducing thermodynamic inefficiencies. The basic cycle itself may also be

improved by introducing complex cycle arrangements. Part-load operation efficiency may be improved by using variable-speed or displacement compressors. However, such improvements can be obtained only by additional development costs and/or by using more costly components. These costs, of course, must be amortized over the market lifetime of the product and in turn, increase the price of the product. In the marketplace, customers are strongly influenced by initial low cost of equipment. Therefore, the manufacturer may or may not pursue costly improvements depending on the analysis of the competition in the marketplace.

3.1.2 Conclusions

The EERs of commercially available air conditioners with 10-ton capacity vary from 7.1 to 10.6 Btuh/W with an average EER of 8.3 Btuh/W. These EERs are significantly lower than those theoretically attainable in ideal cycles. If the design features which improve operating efficiency were uniformly employed, important improvements in these air conditioners as a class can be attained. The features include, among others, the use of large heat exchangers and efficient compressors. However, the marketplace for these units is very competitive and attuned to first-cost rather than lifetime operating costs. This first-cost selection criteria is reflected in the designs of units using only simple cycles, minimum capacity control features, and minimum heat exchangers. The most effective research to immediately impact energy efficiency would be to develop low-cost and effective condenser coils, evaporator coils, and control systems.

3.2 CENTRIFUGAL CHILLERS

In this section, the results are described from an assessment of the energy efficiency of centrifugal water chillers having a 350- to 375-ton capacity. These water chillers are manufactured with capacities in excess of 1000 tons; a 375 ton unit is classified as "medium-sized".

3.2.1 Description of Centrifugal Chillers

Packaged centrifugal water chillers generally use the basic vapor compression cycle and consist of the following equipment mounted on common skid or

framework: 1) one or more parallel compressors, 2) a water-cooled (chilled water) evaporator, 3) a water- or air-cooled condenser, 4) an expansion valve, and 5) controls. Ancillary equipment required for a typical air conditioning application might include a water pump or system of pumps to transport the chilled water to the conditioned space and to return it to the chiller, and depending on the type of condensing system, the cooling tower, condensing cooling tower, or air cooling coils. Because clean water-side heat transfer surfaces are important to the performance of a unit, water treatment equipment may also be required. Refrigerants used by the manufacturers surveyed include R-11, R-22, and R-500.

A centrifugal compressor may be either directly driven or driven through a gear drive by an electric motor. The electric motor can be either externally mounted air-cooled or hermetically enclosed with the compressor and cooled by the flowing refrigerant. The compressor must operate within a large range of flow conditions to meet the variety of air conditioning loads throughout a typical year. Therefore, some are equipped with automatically controlled inlet vanes, variable diffusers, variable rpm, or hot-gas bypass systems to prevent compressor surging and inefficient operation which may otherwise occur at low flow conditions. Also, multiple compressors (usually two) are used to accomplish the same purpose.

One manufacturer uses a positive displacement rotary compressor of helical-axial design with high-pressure refrigerants. The compressor consists of two intermeshing helically grooved rotors in a stationary housing with suction and discharge ports. Capacity modulation for part-load operation is made possible by a slide valve mechanism operated by an electronic/hydraulic control system in response to the system load. The motor drives the compressor directly and is enclosed in a hermetic housing. The principal advantages of this type of compressor are efficiency and quiet operation.

Several types of condensers may be used: 1) a water-cooled heat exchanger, 2) an air-cooled heat exchanger, or 3) a condensing cooling tower. However, in this study, attention was focused on water-cooled condensers only. These condensers are generally shell-and-tube units with water flowing on the inside of the tubes. Heat transfer surface enhancement, i.e., fins, may be

used on the outside, inside, or both. Because cleanliness of the heat transfer surface is important for efficient chiller performance, provisions are generally included to facilitate manual water-side cleaning, e.g., removable heads. One manufacturer uses an automatically activated brush cleaning system for cleaning the water-side surfaces during operation, thereby eliminating the inconvenience and expense of periodic shutdowns for manual cleaning.

Evaporators are generally flooded shell-and-tube units with the chilled water on the tube side. Surface enhancements are used on the water side, refrigerant side, or both. Like the condensers, evaporators are arranged to facilitate cleaning of the water-side surfaces.

Thermostatically controlled, pilot operated valves or float-controlled valves expand the refrigerant from the condenser to the evaporator. Sometimes flash tanks are used to remove part of the vapor from the flashed refrigerant before it enters the evaporator. This vapor is bypassed directly to the compressor. Since the remaining liquid is at lower enthalpy than the mixture, it provides an improved capacity for heat absorption in the evaporator, thereby improving efficiency.

Controls and motor starters are mounted on either the skid or nearby. Most manufacturers will supply highly sophisticated controls that provide for the detailed electronic monitoring of the operation of the equipment with annunciation of malfunctions as well as control of the refrigeration system's process variables.

3.2.2 Performance Data

Compilations of performance characteristics of commercial units were not available. Therefore, data on the performance of these water chillers were obtained by direct contact with the field representatives of the following five major manufacturers of these units:

- Carrier Corporation
  Syracuse, New York
- Dunham-Bush, Inc.
  Harrisonburg, Virginia
- McQuay, Inc.
  Minneapolis, Minnesota

- The Trane Company
  LaCross, Wisconsin
- York International Corporation
  York, Pennsylvania

Manufacturers of this size water chiller usually do not produce standard units. Instead, they make standard components which are then assembled in combination per special order to provide units designed to meet the customers' requirements. For example, one manufacturer offers 100,000 combinations of components allowing almost any customer requirement to be met, whether it be specific operational parameters, low initial cost, energy efficiency or lifetime economy. Because of the large number of components available for selection, these chillers are normally designed by in-house computer programs in response to customers' specifications. Their catalogs describe performance characteristics of the most representative or most frequently ordered units. These catalogs provide guidelines only to enable a prospective customer to estimate performance of units under a narrow range of operational parameters. Table 3.2 was obtained from several catalogs and gives some typical performance data of 375-ton units operating under similar conditions.

The energy economy of the units given in Table 3.2 (expressed in kW/ton, excluding auxiliary equipment) cannot be directly compared nor conclusions made regarding the manufacturer's success in achieving equipment energy efficiency. Energy economy is achieved by a number of methods and equipment choices; and these choices can significantly affect the acquisition cost of the chiller unit. For example, the most effective method of obtaining energy economy from a chiller is to provide a large or very effective condenser and evaporator. Using such components will significantly increase the cost of a chiller. An example of the relationship between energy economy and first-cost is illustrated in Figure 3.1. The designer's criteria for unit selection is usually economic, e.g., low construction cost. Therefore, the most popular designs reflected in the catalogs are not necessarily the most energy efficient. Rather, they may be the least efficient but lowest initial cost designs that compromise on energy efficiency.

TABLE 3.2. Summary of 375-Ton Centrifugal Chiller Energy Performance[a]

Manufacturer	Refrigerant	Chilled Water Temperature In/Out (°F)	Cond. Water Temperature In/Out (°F)	Nominal Size (kW)	Efficiency (kW/ton)
1	R-11	54/44	85/95	400	0.68
2	R-22	54/44	85/95	398	0.67
				422	0.66
				470	0.66
3	R-11	54/44	85/95	NA	NA
4	R-11	54/44	85/95	400	0.64 to 0.65[b]
				450	0.58 to 0.62[b]
5	R-11	54/44	85/95	400	0.57 to 0.68[b]

(a) All performance is based on two-pass evaporators and condensers with 0.0005 fouling factor; chiller produces 375 tons of refrigeration.
(b) Range of performance of typical units referenced in catalogs.

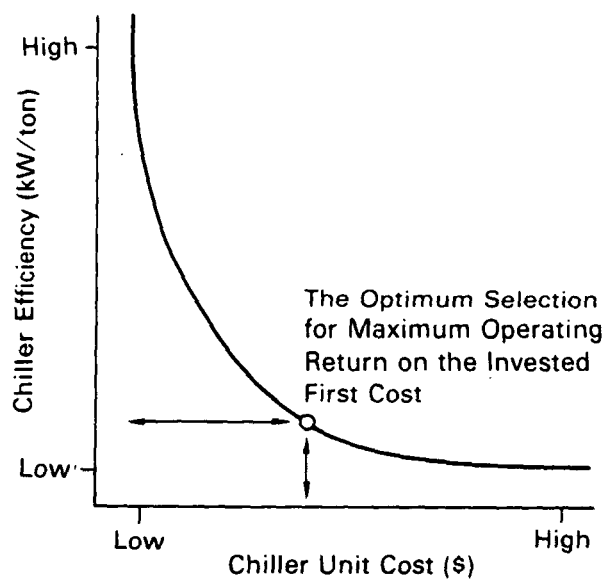


FIGURE 3.1. Relation Between the First-Cost and Operating Cost of a Centrifugal Chiller

In discussions with manufacturers' representatives, estimates were obtained of the minimum kW/ton that could be obtained using available components if costs were unrestrained. These estimates varied from 0.50 kW/ton to 0.55 kW/ton for standard ARI conditions. In comparison, the Carnot refrigerator has a kW/ton of 0.29 for uniform evaporator and condenser temperatures of 44°F and 85°F, respectively. The ideal vapor compression KW/ton is 0.31 and 0.33 for R-11 and R-12, respectively, under the same conditions. Therefore, commercial water chiller technology and practice allow a closer approach to the ideal performance than do air conditions. Further R&D on the basic refrigeration cycle may not greatly improve what is currently available.

3.2.3 Methods Used to Obtain Energy Efficiency

All of the water chiller manufacturers' literature examined provides considerable detail on methods used to increase energy efficiency. The primary methods include 1) modifying chilled water and condenser water temperatures, 2) enhancing heat transfer performance of the evaporator and condenser, and 3) improving compressor efficiency. Additional methods include 4) using multi-compression and flashtanks and 5) using alternative refrigerants.

Modifying Chilled Water and Condenser Water Temperatures

The design condenser and chilled water temperature are usually fixed by the application. However, in many instances, it may be possible to use a different source of condenser water to obtain lower condenser inlet temperatures. For example, one manufacturer's data indicate that substituting city or well water at 65°F for cooling tower water at 85°F could reduce the kW/ton to 70% of the original values. Further lowering the condenser water temperature to 55°F would decrease the kW/ton to 62% of the value at 85°F. Some decrease in kW/ton could also be obtained by increasing the chilled water temperature. However, any energy economy obtained in this manner may be offset by increased acquisition and operating costs of the peripheral chilled water coils and the required larger chilled water circulating pumps. Several manufacturers' representatives indicated a trend to lower chilled water temperatures to conserve on recirculating pump operating costs. Although the power to operate these pumps is not

generally charged against the energy efficiency of the chiller, it could nevertheless be an important part of the overall energy costs of operating the chiller system.

Enhancing Heat Transfer Performance of the Evaporator and Condenser

One of the reasons for energy loss in a refrigeration cycle is the transfer of heat across a finite temperature difference in the condenser and the evaporator. This lost energy must be supplied by additional compressor work, and this significantly affects the kW/ton of the cycle. To reduce the amount of energy lost, the temperature difference between the evaporating refrigerant and the chilled water and between the condensing refrigerant and the condenser water must be minimized. Newton's Law of Cooling implies that for a given quantity of heat transferred, reducing the temperature difference between a surface and a fluid requires an increase in the surface area available for heat transfer or in the heat transfer coefficient or in both. The former is accomplished by using a large number of tubes in the condenser and evaporator, or by using an extended heat exchange surface such as externally or internally finned heat exchanger tubing. The latter is accomplished by using high fluid velocities or special surface enhancements. However, the use of measures to improve the heat transfer coefficients may be unattractive because such measures unavoidably cause corresponding increased pumping losses which can partially or totally negate any gain in energy efficiency. Also, to maintain low temperature differences across the surfaces of condensers, it is important to minimize fouling of the water side from mineral deposition and algae growth. This requires that the condensers be arranged for easy manual or automatic cleaning and that the water quality be monitored and treated as necessary.

From discussions with the various manufacturers' representatives, it was concluded that the improvement of heat transfer in the evaporator and in the condenser were the favored methods available to improve chiller performance. By overdesigning these components, the energy efficiency of chillers at standard ARI conditions could be reduced to an estimated 0.50 kW/ton. However, this improvement may be costly because large, expensive heat exchangers are

required which can significantly increase the overall initial cost. One manufacturer supplied the computer design information shown in Table 3.3, which gives the performance improvements and the cost penalties associated with these improvements.

Methods and means used by the various manufacturers to improve the heat transfer characteristics of their condensers and evaporators include the following:

- integral externally finned heat exchanger tubing
- internal tube heat transfer enhancement (details not specified)
- multiple water passes to produce uniform condensing/evaporation conditions
- internal refrigerant (gas and liquid) distribution lines to produce even condensation/evaporation and thereby more effectively use available area
- removable water heads for manual cleaning of water-side surfaces
- automatic brush cleaning systems

TABLE 3.3. Performance Improvements and Associated Costs for 375-Ton Centrifugal Chillers

Refrigerant	Evaporator In/Out (°F)	ΔP (psi)	Condenser In/Out (°F)	ΔP (psi)	No. of Passes	Efficiency (kW/ton)	Relative Cost (N.D.)
R-11	55/45	19.1	85/95	15.1	2	0.796	1.000
R-11	55/45	14.6	85/94.8	15.4	2	0.737	1.005
R-11	55/45	29.7	85/94.7	15.4	2	0.677	1.016
R-11	55/45	19.7	85/94.6	15.4	2	0.637	1.030
R-11	55/45	28.7	85/94.6	27.2	4	0.622	1.250
R-500	55/45	19.7	85/94.6	24.6	2	0.633	1.076
R-500	55/45	13.2	85/94.6	12.6	2	0.592	1.165

- automatic feed-controlled chemical treatment systems for cooling water
- larger allowances for fouling (i.e., more available surface) and special nonfouling materials of construction.

Improving Compressor Performance

The manufacturers surveyed indicated that they use centrifugal compressors with both single and multiple impellers. Single-impeller compressors are generally driven through a gear box, while multiple-impeller units are directly driven by an electric motor. The compressor capacity is varied in response to the refrigeration load by automatically controlled inlet guide vanes, variable diffuser vanes, or motor speed. These compressors have very high efficiencies, ranging up to approximately 95% at rated maximum design conditions. However, at less than maximum refrigeration load, the efficiency can drop off rapidly, e.g., to around 70% for a single-impeller unit at 55% load as illustrated in Figure 3.2. To partially compensate for this drop in efficiency, one manufacturer supplies models using two compressors in parallel that are operated in stages in response to the refrigeration load. The efficiency map for these

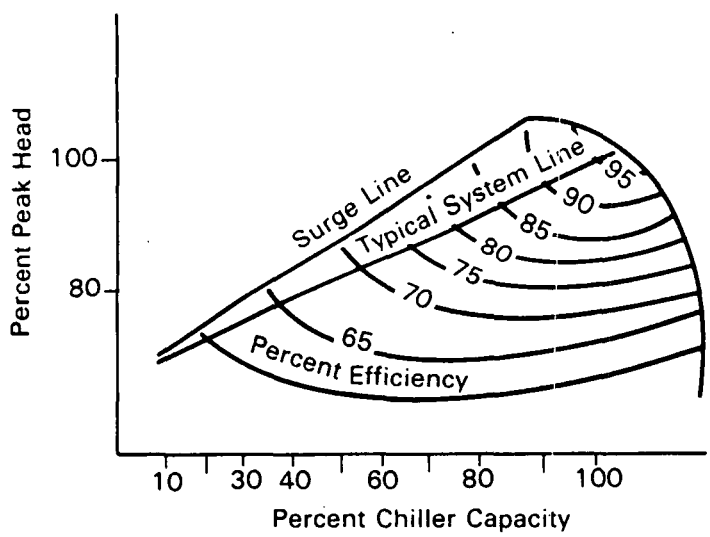


FIGURE 3.2. Compressor Efficiency Map for Single-Compressor Operation

two-compressor arrangements is shown in Figure 3.3. Note that at the same 55% load, the lead compressor operates at about 90% efficiency and the lag compressor operates at about 70%.

The multiple-stage compressor has two additional advantages from the standpoint of energy economy. First, it does not require a gear transmission to provide the necessary head. This eliminates the gear losses. Secondly, it facilitates the application of the so-called economizer cycle to enhance energy economy. The economizer cycle will be described in the next section.

(For comparison, the screw compressors are classified as positive displacement devices. However, the use of an automatic inlet slide valve mechanism can vary the axial length of the pump rotors engaged in the pumping action thus enabling response at reduced refrigerant flow rates. Furthermore, at part load, this compressor type is inherently efficient, enabling the manufacturer of these units to achieve kW/ton performance as low as 0.49 with a 0.59 average.)

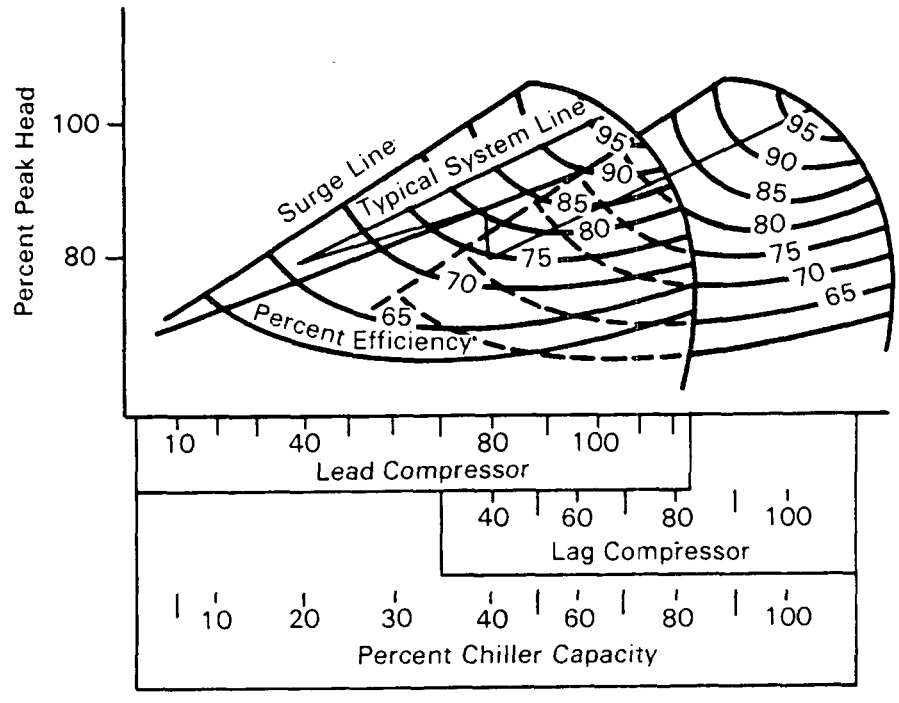


FIGURE 3.3. Compressor Efficiency Map for Two-Compressor Operation

Other Improvement Methods

The expansion of the refrigerant from the condenser to the evaporator is accomplished by float valves or thermostatically controlled throttling valves. In expanding the refrigerant across such valves, energy is degraded because the throttling process is irreversible. However, such energy degradation is unavoidable unless an expansion turbine is used, and this is economically impractical in vapor compression units because of the relatively small amounts of energy that could be recovered.

Two of the manufacturers surveyed use the economizer cycle with multicompression systems and flashtanks in the expansion process to enhance the refrigeration effect and thereby improve efficiency. Figure 3.4 illustrates a modified vapor compression cycle using one flashtank. In a conventional basic refrigeration system, as the liquid refrigerant is throttled through an expansion valve, a portion of the liquid is immediately vaporized. This vapor passes through the evaporator without further change in state and, therefore, without providing any refrigeration effect. The compressor must then compress

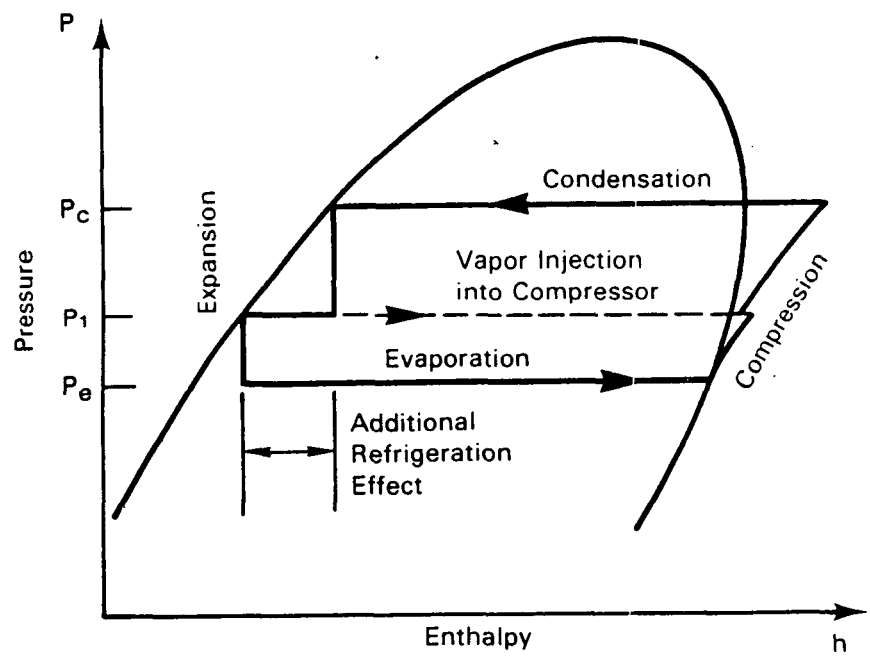


FIGURE 3.4. Pressure-Enthalpy Diagram for a Single-Flashtank System

this flash vapor along with the working vapor that is formed by the absorption of heat in the evaporator. To relieve the compressor from compressing this noneffective flash vapor, a two-stage compressor and flashtank are used. The liquid is flashed to an intermediate pressure, and the flash vapor, which is useless for refrigeration purposes, is passed to the second stage of the compressor where it is compressed back to the condenser pressure. Therefore, less work is required because the flash vapor removed at the intermediate pressure needs to be compressed over only a part of the pressure ratio. A two-stage flashtank system is illustrated in Figure 3.5. The manufacturer claims that this two-stage system can produce a 7% energy saving compared with conventional systems.

Small benefits can also be obtained through the choice of refrigerant. Approximately 5% improvement can be obtained by using R-12 instead of R-11 in a basic ideal refrigeration system operating between 44°F and 85°F. However, the pressures involved with R-12 are greater. As a result of the additional structural capability required, higher costs are associated with R-12 systems.

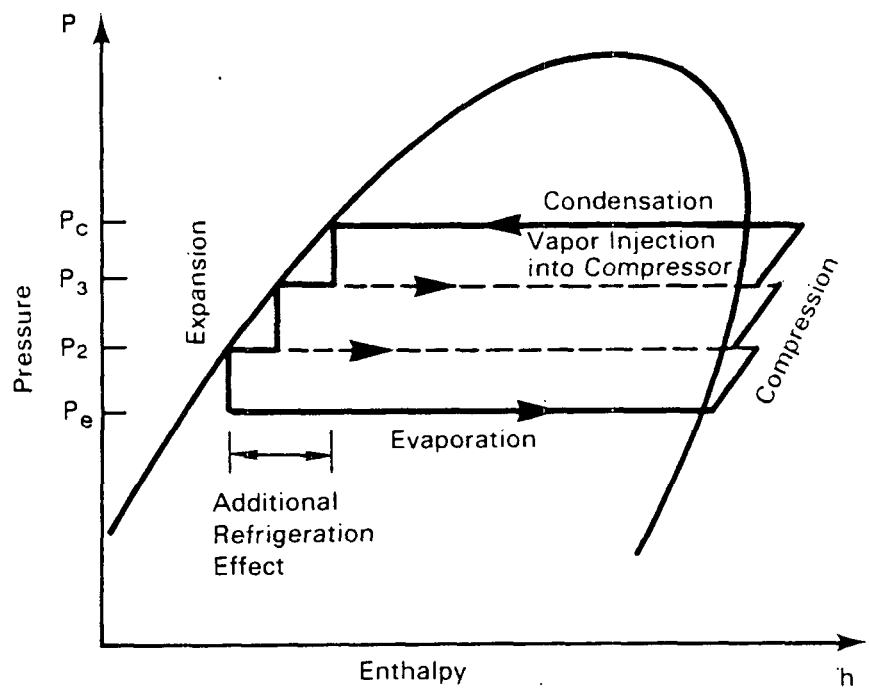


FIGURE 3.5. Pressure-Enthalpy Diagram for a Two-Flashtank System

3.2.4 Conclusions

Methods are already available to manufacture energy-efficient water chillers in the 375-ton range. The kW/ton performance attainable by most manufacturers approached 0.50 for ARI standard conditions. However, the features necessary to obtain this performance are not always used because they are expensive, and the system designers are usually more concerned with initial cost of equipment. To encourage the use of these features, it would be necessary to reduce the incremental cost of these energy-saving features. Inexpensive heat transfer surfaces that would encourage the use of efficient condensers and evaporators would be especially advantageous. Reducing the capital cost of energy-efficient designs both improves equipment lifetime economy and encourages energy efficiency.

4.0 ASSESSMENT OF R&D OPPORTUNITIES FOR AIR CONDITIONERS AND CENTRIFUGAL CHILLERS

Although the primary objective of this project is to identify R&D opportunities for chiller and air conditioner equipment, it is also the first step in an effort to determine the overall R&D needs in the commercial HVAC area. This chapter summarizes the R&D needs and opportunities assessed with specific emphasis on air conditioners and centrifugal chillers, but noting other related topics as well. PNL's recommendations and interpretation of this information are given in Chapter 5.0.

The overall R&D needs were identified based on discussions with HVAC equipment researchers from various organizations and communities, including academia, research laboratories, manufacturers, trade organizations, and consulting firms. In addition, a literature review was conducted to focus on specific concepts which may be applied to improve the energy performance of the air conditioner and chiller equipment. A computer search of the DOE Energy Database was performed to aid the effort, and a manual search was conducted to focus on ASHRAE publications, DOE reports, EPRI reports, GRI reports, conference and workshop proceedings, abstracts from ASME Applied Mechanics Reviews, and some relevant U.S. patents. The research concepts reviewed address the performance improvements at the equipment level only and do not treat the integration with the distribution and control systems. Such integration will be addressed in another project. This review excludes R&D areas that have potential or current private sector sponsors. Finally, R&D areas already included in DOE programs are not discussed. However, the nonazeotropic refrigeration cycle is discussed because its specific application to large chillers is not currently being researched.

The following sections summarize the results of the current assessment. First, the general R&D needs in the commercial HVAC area are described. From discussions with the researchers, important topics include the following:

- reducing building energy consumption by better systems integration
- improving equipment part-load performance

- improving equipment and system performance by better control
- applying established performance improvement features
- fostering high-efficiency equipment via performance standards
- extending HVAC research focus to indoor air quality and humidity control
- assessing alternatives to conventional chlorofluorocarbon (CFC) refrigerants.

The final section describes the research concepts in the literature for improving energy performance of air conditioners and chillers. These concepts include the following:

- reduce evaporation and condensation heat transfer inefficiencies via heat transfer augmentation or nonazeotropic refrigerants
- improve part-load performance using variable capacity drives or nonazeotropic refrigerants
- reduce compression energy requirement using isothermal compression
- reduce throttling process irreversibility using a regenerative cycle.

4.1 OVERVIEW OF R&D NEEDS FOR COMMERCIAL HVAC EQUIPMENT AND SYSTEMS

In discussions with HVAC equipment researchers, comments on the R&D needs in chillers and air conditioners identified a number of different subject areas. In particular, it was emphasized that vapor compression technology is a mature technology. As such, the potential applications of various improvement features have already been analyzed and traded off on a cost basis. Although the knowledge base for various performance improvements technologies is established, it is rarely used because of nontechnical, often economic, reasons. HVAC researchers and practitioners recommended that systems-level research should be addressed rather than component-level research. The following sections provide an overview of the suggested R&D needs for commercial HVAC systems identified by this project.

4.1.1 Systems Integration

Vapor compression equipment represents a well developed technology. As such, a considerable base of information and knowledge for improving performance is already established. Many experts have suggested that future HVAC research should emphasize system-level design issues, rather than the equipment components. A report prepared by TRW in 1982 which identified R&D opportunities for large air conditioners and heat pumps concluded that the first part of an energy-conserving system, i.e., efficient equipment, was available and was not a major limiting factor. Instead, the area of system concept development presented extensive R&D opportunities (MacDonald, Goldenberg and Hudgins 1982).

Systems integration problems cover both the problem of putting together the chillers and air conditioners from heat exchangers, expansion devices, compressors etc. and the problem of installing the chillers and air conditioners into a building. At the equipment level, systems integration focuses on the incorporation of individual components to produce an optimal operating unit. An example of the equipment-level problem is the poor air distribution over heat exchanger coils in packaged air conditioners because of the configuration of the components. This configuration is often governed by the limited space within the equipment package. It has been reported from field operation that as much as half of the heat exchanger coil was not used because of uneven air distribution. Another example is that of the difficulty in designing variable capacity units for properly controlling the amount of refrigerant charge in the evaporator. The amount of charge has to maximize the evaporating surface while maintaining a desired level of superheat entering the compressor. This type of equipment system-level problem tends to worsen in the field under off-design conditions. The performance of field-installed systems is seldom known. However, it is recognized that significant performance degradations occur; this must be quantified and improved.

Systems integration at the building level focuses on issues that arise as a result of the interactions among the HVAC equipment, distribution equipment, and building envelope. System problems arise in commercial buildings, particularly in large commercial buildings, because of the size and complexity of the

dynamic thermal systems involved. Compared with residential dwellings, commercial buildings have more complicated heating and cooling requirements, including zoning, coincident heating and cooling, and ventilation. This complex and dynamic HVAC system on one hand may result in considerable wasted energy if not properly designed and controlled. On the other hand, it may offer abundant opportunity for energy conservation.

An example of the building-level system issue is the fan energy required to deliver and circulate air around the conditioned space. The electricity used by the air-handling unit is generally very high. For some large office buildings, fans are reported to be the largest single consumer of energy. The reduction of fan energy may be addressed in the context of whole building system design. Improved design of ductwork to reduce air turbulence and to improve system aerodynamics can help lower the fan energy requirement by as much as half. Different flow control techniques and control algorithms can also result in significant differences in the fan energy required. Reducing the amount of air circulated obviously lowers the fan energy as well, which leads to another system-level issue on the selection of the optimal evaporator temperature.

The optimal evaporator temperature must minimize the total energy consumed by the compressors, fans, and pumps. Recent developments in HVAC equipment and system design illustrate the complexity of this issue. On one hand, the system using low evaporator temperature, and hence low-temperature air, is gaining popularity among HVAC designers because it reduces distribution energy and system size requirements and because it may be used with cool storage systems. On the other hand, new high-efficiency equipment often employs high evaporator temperature, which results in increased total cooling capacity, reduced latent cooling capacity, and reduced compressor energy requirements. The third issue is not derived from efficiency considerations, but instead from the ventilation requirements which dictate the minimal amount of outdoor air brought in regardless of the evaporator temperature. The optimization of the evaporator setpoint needs to trade off the various system parameters including compressor energy, fan and pump energy, distribution equipment first-cost, and ventilation requirements. This represents an important systems integration issue.

In addition to the above examples, several system-level energy conservation opportunities in commercial buildings include:

- optimal use of outdoor air for ventilation and economizer cooling
- reduced energy use for reheating cold air supply
- proper sizing of energy conversion equipment to reduce part-load losses
- proper balance or adjustment of equipment
- proper use of system control
- reclaiming heat from cooling zones and transferring to heating zones.

This system-related energy waste cannot be reduced via improved energy conversion equipment but must be addressed as part of the integrated system. The integrated design of the HVAC equipment, distribution system, control system, building envelope, and optimization of the integrated system hold the key to energy conservation of commercial buildings. Not only are these issues often not addressed in the system design by the architects and engineers, but detailed knowledge and information on many of the system-related issues are often lacking. Improved understanding of the comprehensive building equipment system is essential. These system issues and the role of basic and applied research results will be addressed in another PNL project.

4.1.2 Part-Load Operation

Equipment designs are normally optimized at the rating conditions. However, equipment operations are often at off-design conditions, which degrades performance. The problem is compounded because most equipment installations are sized for the design loads, and some are even further oversized. Consequently, for most of the operating season, equipment part-load operations prevail. Part-load performance plays an important role in determining the seasonal energy efficiency of the equipment. Improving equipment part-load performance is the major concern of the HVAC manufacturers.

The simplest and most commonly used method for capacity control is to cycle the compressor on and off. Cycling is required as long as the equipment capacity is greater than the load. Cycling reduces the average efficiency of

the equipment because of transient losses. For improved equipment efficiency and reliability, the amount of compressor cycling must be reduced. If necessary, rated performance may be compromised to incorporate the performance upgrade at part-loads. An example is that of the inverter-driven units, whose performance at full-load is degraded because of the inverter's inefficiency. However, reduction in equipment cyclic losses results in overall seasonal energy conservation. There is general agreement that for better equipment designs, it is important to improve the varying-load performance. With the decline in cost of the inverters, the potential for its widespread application should be explored. Brushless DC motors also provide an important option for small units. In addition, optimal staging of parallel compressors or multiple unit operations should also be examined. (Section 4.2.5 discusses the variable capacity equipment in further detail.)

4.1.3 Control

Research in equipment control offers potential energy conservation opportunities. In the past, HVAC manufacturers have not extensively used electronic controls in the equipment primarily because they were not confident about the reliability of the controls and they were concerned about the added cost. However, with the microprocessor revolution, HVAC control has become one of the most rapidly changing areas in the HVAC field.

For energy-efficient operation of the equipment, the key area of focus should be research for proper control to optimize part-load performance of the equipment, including compressors, fans, and pumps. Other control issues which need to be addressed include optimization of equipment staging, multiple unit operations, and interaction with the HVAC distribution system and building envelope. In addition, equipment diagnostics, predictive, and adaptive control are areas that have great potential for energy conservation.

Besides energy efficiency, the research in HVAC control should also address reliability, better performance, ease of maintenance, and thermal comfort. Three specific research areas in equipment control suitable for government involvement have been identified in a workshop on research needs in HVAC systems. The general control issues include cost and benefit analysis of control schemes, subsystem dynamics, control of interacting subsystems, adaptive

control, predictive control, and control strategies. Specific control issues include control of indoor air quality, control of thermal energy storage, control of humidity, and field versus rated performance of equipment. Finally, knowledge-based system issues include diagnostics and expert systems [ASHRAE/ National Society of Professional Engineers (NSPE) 1986].

4.1.4 Application of Established Improvement Features

As evident from the performance assessment of the current-market equipment, state-of-the-art performance improvement technologies are already established. However, they are often not incorporated into the equipment designs because of nontechnical, mostly economic, reasons. Since the established equipment improvement concepts are not currently being applied to most of the air conditioning equipment, extensive research on next-generation technologies seems premature. Instead, development work is recommended to facilitate effective application of established energy-efficient technologies. Specifically, the emphases should be to apply performance improvement features currently used in chillers to the air conditioners and to reduce costs of the advanced technologies.

As discussed in Chapter 3.0, because of the highly competitive market, air conditioner designs are more attuned to the equipment first-cost compared with designs for the large chillers. Not only are improvement features not used, but energy efficiencies are sometimes compromised in favor of lower capital investment by using, for example, undersized heat exchangers and undercharged refrigerant. On the other hand, complex cycle arrangements and improved equipment designs are more often incorporated in large chillers. R&D work is important in order to apply some of the proven energy-efficient features to the air conditioners. Of particular interest is the multi-stage throttling and compression feature, i.e., the economizer cycle, used in centrifugal chillers (see Section 3.2.3). The supercharged reciprocating compressor being researched uses the same basic concept. For this design, a portion of condensed refrigerant is flashed and introduced directly into the reciprocating compressor. Figure 4.1 is a schematic of the design arrangement (Katsenelenbogen 1986). Other basic design modifications such as parallel compressor staging to improved part-load performance should also be investigated.

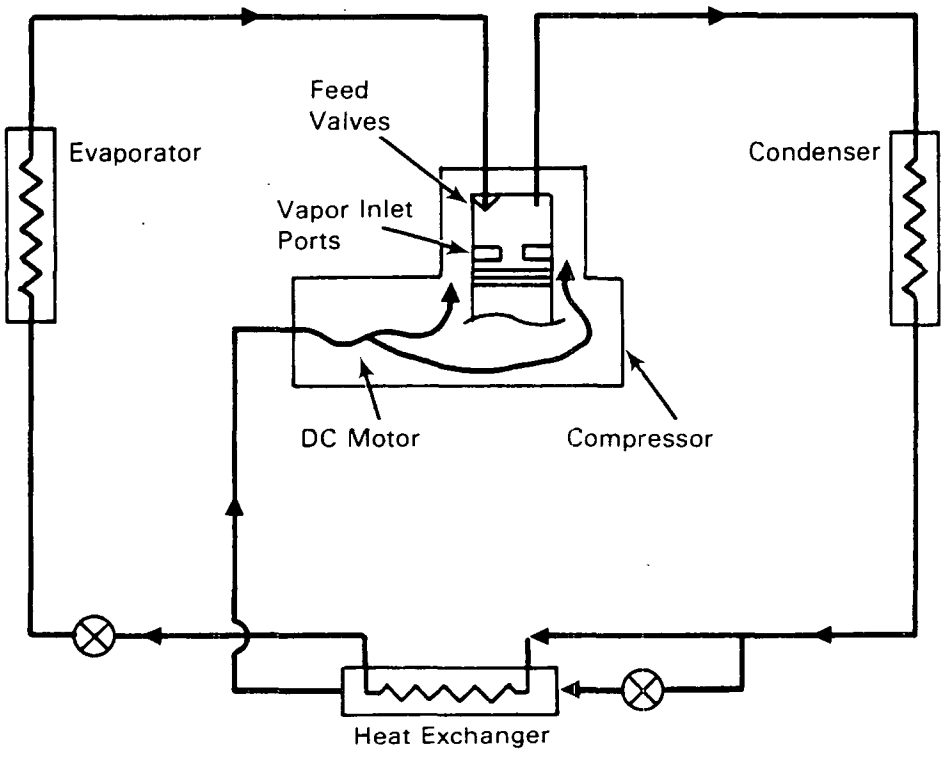


FIGURE 4.1. Supercharged Reciprocating Compressor

From the manufacturer's stand-point, cost-effectiveness of the equipment is the main concern. To promote high-efficiency equipment in the marketplace, the added product costs which are the consequence of advanced technologies must be reduced. Among the various equipment components, inexpensive heat exchangers used as evaporators and condensers will immediately impact the equipment performance. In addition, electronics offer substantial potential to improve equipment energy efficiency through better controls. The development of inexpensive electronic controls will foster their large-scale application.

4.1.5 Equipment Performance Standards

The factors impeding the development of energy-efficient equipment designs are nontechnical, as discussed above. It was pointed out in Chapter 3.0 that the capital cost of the equipment plays a very important role in the purchase decision-making, and manufacturers must respond to this. In addition, other nontechnical impediments, including institutional, educational, informational,

and psychological constraints, also discourage the use of energy-efficient designs. For detailed discussion on these nontechnical constraints, refer to MacDonald, Goldenberg, and Hudgins (1982).

Since the factors constraining current equipment designs are nontechnical, it has been suggested that equipment performance standards be established in order to promote and foster energy-efficient commercial-sized air conditioning equipment. It is argued that minimal performance standards will facilitate the use of energy-efficient equipment. For the residential sector, the National Appliance Energy Conservation Act, which sets minimum energy-efficiency standards for a wide variety of appliances, has been proposed. For the HVAC equipment, it has been proposed that for the central air conditioners and heat pumps below 65 kBtuh, a minimal seasonal energy efficiency ratio (SEER)[a] of 10 Btuh/W is required for split systems effective in 1992; and a minimal SEER of 9.7 Btuh/W for packaged systems is required effective in 1993 (Mahoney 1986; ACHRN 1986). Although the proposed act has recently been vetoed, it is likely to be reintroduced next year. It has been suggested that similar rating standards for commercial-sized equipment should also be implemented.

However, the practical difficulties in establishing standards for testing and rating large HVAC equipment cannot be overlooked. First, many large commercial-sized pieces of equipment, particularly the chillers, are designed according to the customers' specifications by mixing and matching equipment components; some are even site-built. Because of the lack of standard equipment, it is not possible to test the numerous units that are configured from the various combinations of components. Moreover, because of the physical size of commercial equipment, transportation of these units to and from the factories and test facilities will probably be a major effort. Finally, development of large test facilities to accommodate these units will involve substantial investment.

(a) The definition of SEER is similar to EER, except that the ratio of the equipment capacity and energy requirement is evaluated on the seasonal basis, instead of at the standard rating condition.

4.1.6 Indoor Humidity and Air Quality

The mission of the HVAC industry is to provide indoor comfort. In the past, the issue of comfort with HVAC equipment focused exclusively on space temperature. Although cooling equipment is used for dehumidification, the humidity level is not specifically controlled. The control of indoor humidity is quickly becoming an area of focus in comfort conditioning among HVAC researchers.

The emphasis on high-efficiency cooling equipment often results in higher evaporator temperatures with decreased latent cooling capacity. For geographical areas with high humidity or for buildings with high latent loads, indoor conditions and thermal comfort may suffer unless dehumidification is accomplished using alternative methods. GRI is currently developing an early entry integrated desiccant dehumidifier and an air conditioner or chiller of 20 ton to 100 ton for commercial buildings (GRI 1985). Figure 4.2 shows a schematic of the hybrid system in which the sensible load is met by the evaporator and the latent load by the desiccant wheel (Anderson 1984). On the other hand, using low-temperature air to reduce distribution requirements or using a

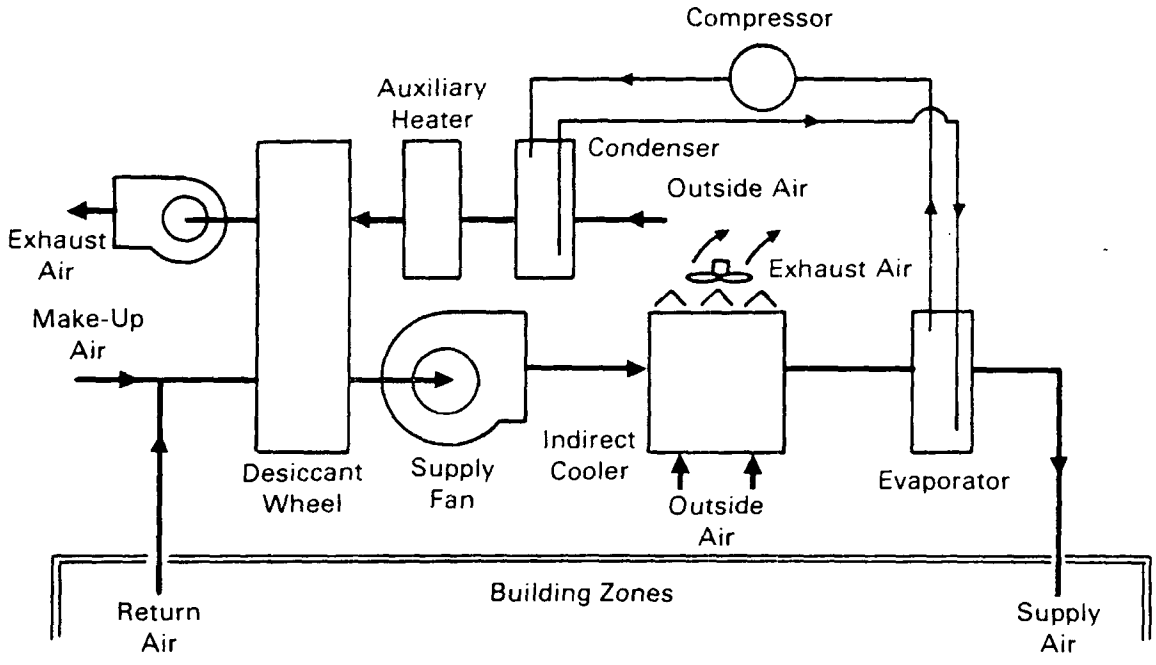


FIGURE 4.2. Desiccant-Vapor Compression Hybrid System

cool-storage system results in more latent cooling but lower equipment efficiency. The low-temperature air approach is gaining in popularity among designers. With the opposing trend in evaporation setpoints, the problem of indoor humidity control must be properly addressed.

The scope of needs in humidity control research includes identifying comfort issues and control options that may be viable to provide improved performance. For example, an available option would be changing blower speed during high humidity periods to provide greater humidity removal (ASHRAE/NSPE 1986). Specific equipment research needs are humidity sensors, dehumidification devices, and automatic control.

Except for the treatment of odors, indoor air quality has not been a major concern of the HVAC industry until the past decade. With increased evidence of the presence of detrimental indoor air pollutants and the general awareness of the topic, indoor air quality control is now an important issue. Commercial buildings with poor air quality have prompted concern for the health and productivity of the building occupants.

The scope of air quality control research consists of evaluating the use of ventilation energy recovery equipment such as air-to-air heat exchangers and air cleaning devices such as electronic air cleaners. Their application as stand-alone units versus integrated central units should be evaluated. The development of air pollutant measurement sensors is also important and needs to be addressed. In addition, ventilation effectiveness should be analyzed for various supply and return air configurations in air distribution systems (ASHRAE/NSPE 1986).

4.1.7 Alternative Environmentally Benign Refrigerants

In recent environmental studies, CFCs have been linked to the depletion of the protective stratospheric ozone. Studies have shown that the contribution of CFCs to ozone depletion ranges up to an estimated 5%. However, it is important to note that the chemical effects of the CFCs are strongly coupled with other trace substances in the atmosphere, including carbon monoxide, carbon dioxide, methane, nitrous oxide, and nitrogen oxides. These chemical effects should not be considered by themselves. In addition, the CFCs are also linked

to the "greenhouse effect" of global warming. For the greenhouse effect, CFCs are predicted to be contributing an amount comparable to the effect of carbon dioxide (Duffy 1986a, Duffy 1986b).

Because of the reported environmental impacts of the release of CFCs from recent assessments, there is increased attention in the HVAC industry focusing on the potential shifting to more environmentally benign refrigerants. R-11 and R-12, often used in commercial air conditioning equipment, are considered to be the most environmentally detrimental among other refrigerants. The 1980 release rates of R-11 and R-12 were reported to be 680 million to 953 million lb/year, respectively (Duffy 1986b). The unitary air conditioning industry, for example, has already made some switch from R-11 and R-12 to R-22, which is considered more environmentally benign.

Researchers expressed the need for specific evaluations of the level of environmental impacts by the various refrigerant types and assessment of alternative refrigerants which may pose less environmental threat. The effect of the alternative refrigerants on the performance of the equipment, particularly on the compressor, and the subsequent hardware modifications must also be addressed.

4.2 R&D CONCEPTS FOR AIR CONDITIONERS AND CENTRIFUGAL CHILLERS

This section summarizes the research areas reported in the literature for improving the performance of chillers and air conditioners. To be consistent with DOE's charter, this review addresses only advanced research topics.

The literature review revealed that there has been substantially more research on heat pumps than on cooling-only equipment such as chillers and air conditioners. However, research on the improvement of the basic vapor compression heat pump cycle is also applicable to chillers and air conditioners. Market-related issues determine the direction of HVAC research and the market favors the improvement of heat pumps. From the manufacturer's stand-point, heat pumps can more quickly amortize the cost increase resulting from added energy-efficient features because of their capabilities for both space heating and cooling. The new features will shorten payback and are consequently more

attractive to a potential customer. From the electric utilities' point-of-view, electricity already dominates 95% of the commercial building air-conditioning market. After years of steady increase, electricity has recently attained almost half of the commercial heating market (Brodrick 1986). Heat pumps offer excellent opportunity for the electric utilities to continue to improve their share of the heating market. In addition to these market-related issues, the relatively poor performance, and therefore greater potential for improvement, of the electric heat pumps in the heating mode at low ambient temperatures has also prompted significant research in areas such as ground-source units, dual-source evaporators, etc.

The operational inefficiencies of the vapor compression refrigeration cycle are discussed in detail in Appendix A. All components of the basic vapor compression refrigeration cycle can be improved by further reducing thermodynamic irreversibilities, and the basic cycle itself can be improved by introducing complex cycle arrangements. The following section focuses on longer term research concepts that may be applied to improve the energy performance of the air conditioners and chillers. These research concepts address performance improvements at the equipment level only, not treating integration with the distribution and control systems.

4.2.1 Nonazeotropic Refrigerants

Nonazeotropic refrigerants refer to fluids consisting of multiple components of different volatilities that, when used in refrigeration cycles, change composition as they evaporate or condense. As a result, they offer two generic characteristics not available from single-component fluids, namely gliding temperature phase-change and variable composition.

Figure 4.3 shows an operating nonazeotropic cycle on a temperature-composition plot (Atwood 1985). Except for the two heat transfer processes, the operation theory is similar to the single component cycle. As condensation proceeds, the total vapor fraction is reduced; and, as shown in Figure 4.3, the vapor phase composition shifts to the right along the dew point line, reflecting the gradual depletion of the component with the higher boiling point (component A). At the same time, the liquid phase, which is initially rich in component A, shifts to the right along the bubble point line until all vapor is

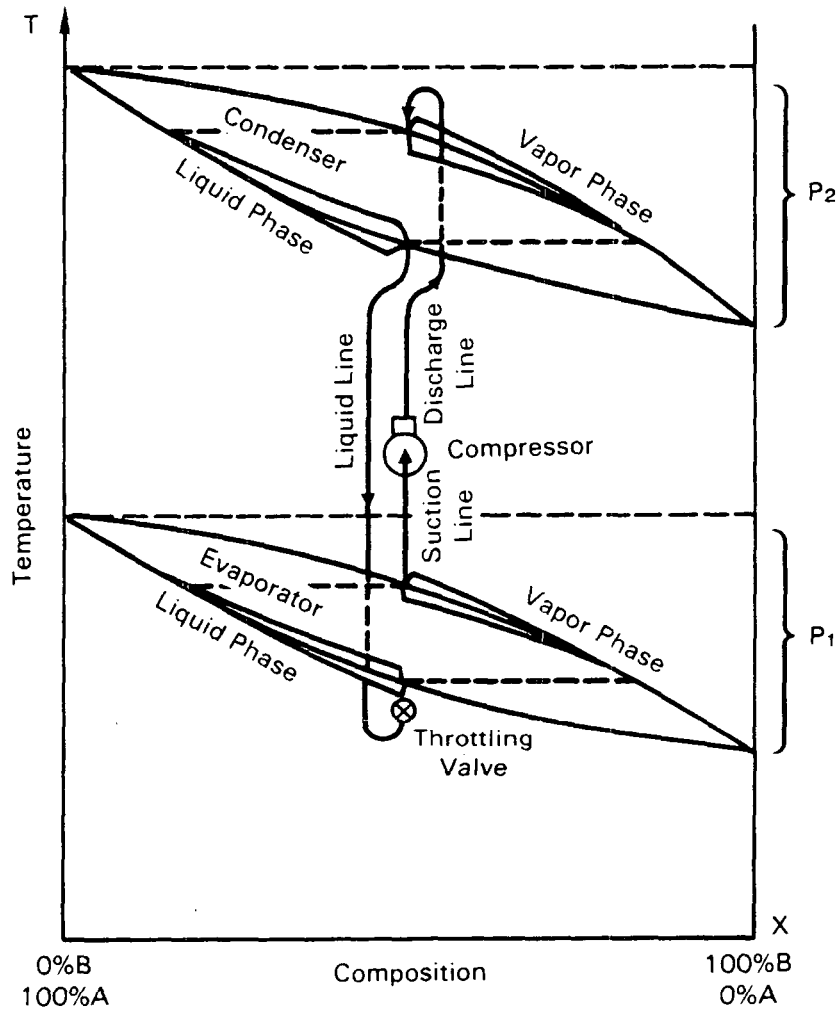


FIGURE 4.3. Temperature-Composition Diagram for a Nonazeotropic Refrigeration Cycle

condensed. At this point, the liquid composition is the same as the original entering vapor. The evaporator process is the reverse with the more volatile component (component B) evaporating initially at a higher rate. Note that the temperature changes continuously throughout the phase-change processes, and the removal of a portion of either phase during phase change results in a composition shift (Atwood 1985).

One of the unique attributes of a nonazeotropic refrigerant system is its "gliding temperature", which refers to its nonisothermal phase change. With proper matching of the temperature gradients of the heat source/sink and the refrigerant, this attribute demonstrates significant improvement in steady-

state coefficient of performance (COP). This is done by narrowing the log mean temperature difference between the refrigerant and the fluid being sensibly heated or cooled, which therefore reduces the cycle irreversibility. On a temperature-entropy diagram, the ideal single-component cycle (Carnot cycle) and the ideal nonazeotropic mixture cycle (Lorentz cycle) are compared in Figure 4.4, showing the additional work required by the Carnot cycle (Didion 1986). Figure 4.5 shows the break-down of the COP degradations of sample single-component and nonazeotropic mixture cycles. For the conditions specified, a theoretical COP improvement of 36% was calculated (Atwood 1985). The actual expected savings are lowered, however, because of the reduction in

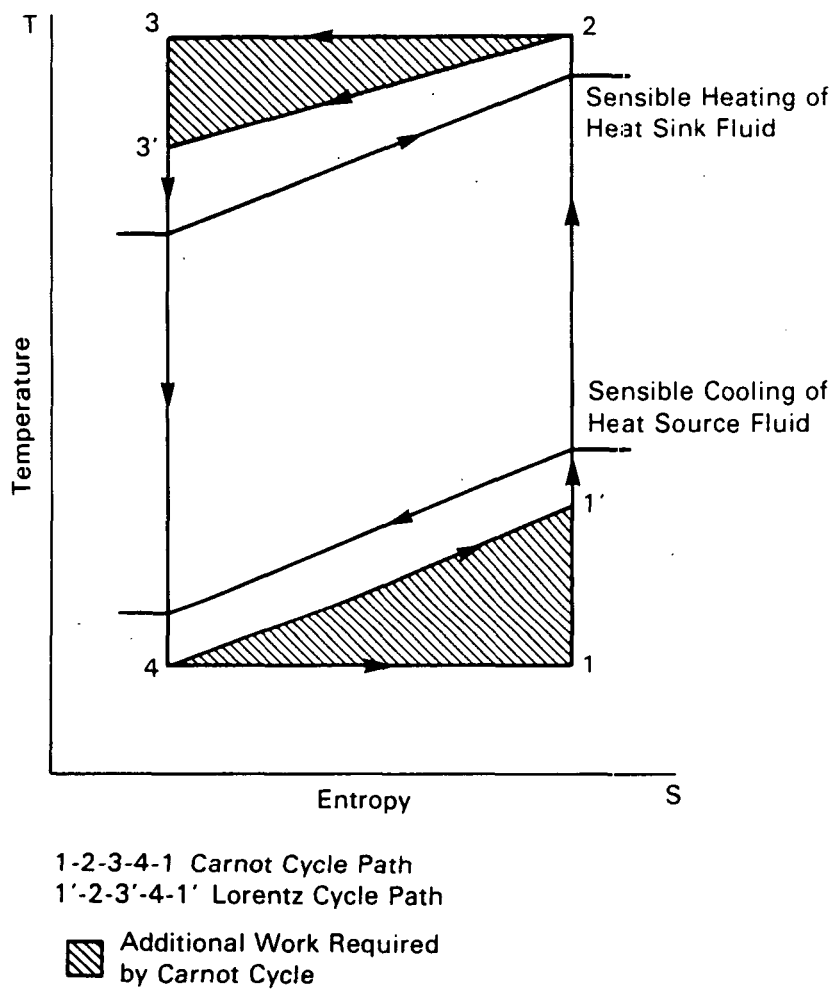


FIGURE 4.4. Temperature-Entropy Diagram for Carnot Cycle Versus Lorentz Cycle

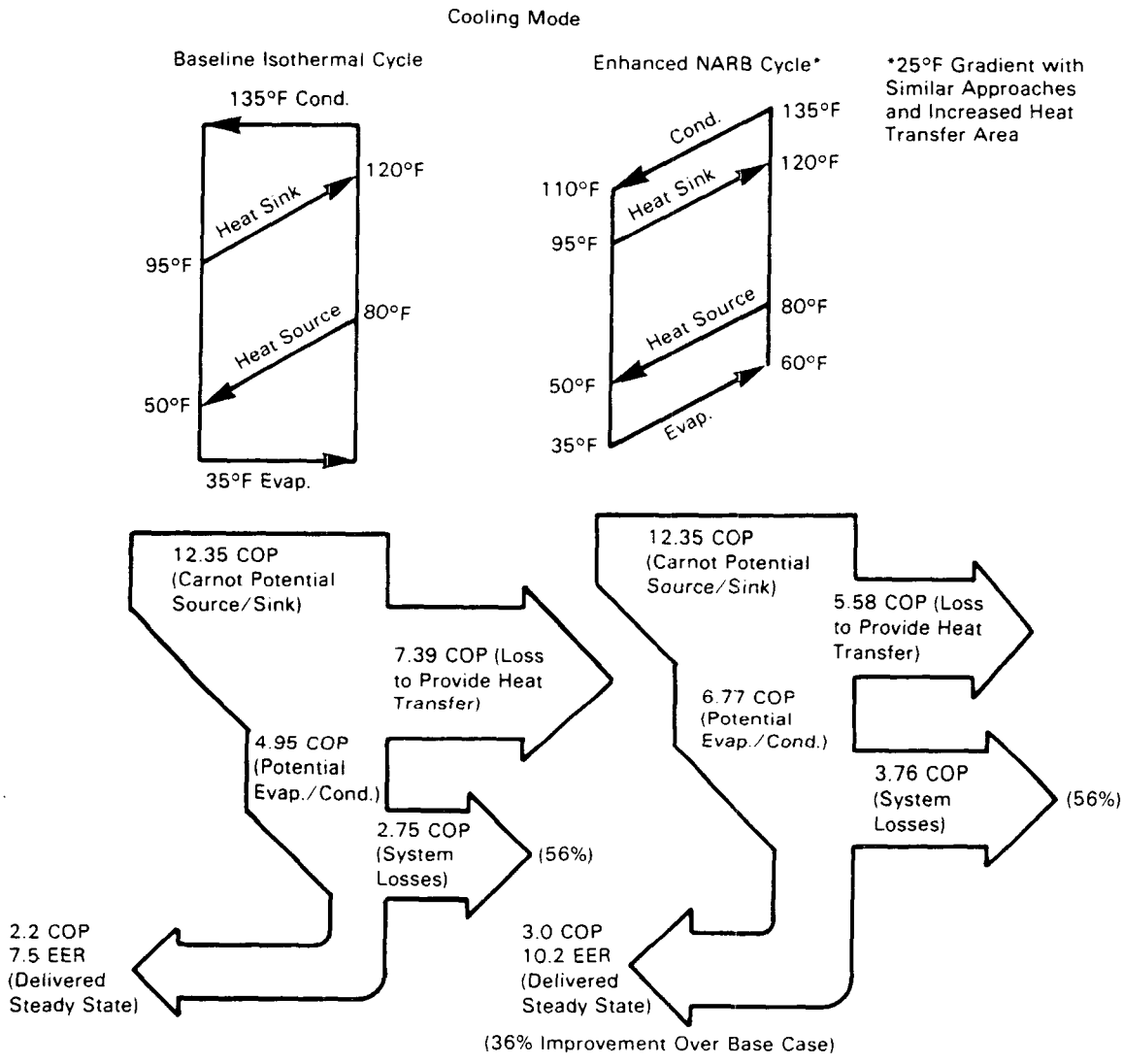


FIGURE 4.5. Performance of Sample Single-Component Cycle Versus Nonazeotropic Refrigeration Cycle

heat transfer coefficients as the two refrigerants are mixed (Stoecker and McCarthy 1984). In addition, to realize the maximum COP improvement, the heat exchange processes must be counterflow with increased surface area (Atwood 1985).

"Variable composition" is the second unique attribute of the nonazeotropic refrigerant that promises improved equipment performance. Variable composition, and thus variable density, provides an opportunity for independent control of the equipment capacity. A number of schemes for composition shifting

have been proposed, ranging from selective absorption or immiscible solution to distillation. The simplest method relies on the distillation effect of the cycle itself during phase change. The increase in energy efficiency, because of composition shifting, stems from reduced equipment cycling and for the heating mode, from increased capacity and thus a decreased auxiliary heat requirement at low evaporator temperatures (Atwood 1985).

To effectively capitalize on the attributes of a nonazeotropic mixture, hardware modifications to current equipment are required instead of simple refrigerant substitutions. For example, additional hardware is required to remove and contain the inactive portion of the refrigerant charge. The degree of improvement for a given refrigerant mixture system is application-specific. In particular, for the opportunities to exploit the advantages of gliding-temperature evaporation and condensation, nonazeotropic refrigerants appear to be more attractive in systems that require high-temperature glides, such as heat pump water heaters, community heat pump systems, industrial heat pumps, and multi-stage refrigeration systems (Calm and Didion 1985). The potential COP improvement depends on the temperature gradient of the heat source and sink along the heat exchanger. The larger the gradient, the greater the potential improvement. For example, for a pure latent heat source or sink, the temperature gradient is zero and the nonazeotropic refrigerant will offer no COP upgrade. It has been suggested that gradients of at least 10°F are required to justify consideration of a nonazeotropic mixture cycle (Atwood 1985).

For low-temperature chiller application, the potential improvement can benefit from both higher COP as well as from reduced distribution energy. Energy conservation through capacity control in chillers and air conditioners is also a prospect, although difficulties in achieving improved efficiency become apparent from the on-going research. The practical use of a nonazeotropic mixture will not be demonstrated for at least several more years. Both basic and applied research are needed to identify and establish the viability of specific mixtures and associated cycle modifications. This research must develop the necessary data, design tools, and thermodynamic impact of selected cycles for specific applications (Calm and Didion 1985).

4.2.2 Heat Exchangers

In the discussion on the inefficiencies of vapor compression refrigeration equipment in Appendix A, it is pointed out that substantial performance degradation results from the loss of available energy associated with the transfer of heat across a finite temperature, which subsequently increases the compressor energy required. Newton's law of cooling governs the heat transfer across the evaporator/condenser between the refrigerant and the heat source/sink. This may be expressed as follows:

$$Q = U * A * LMTD \tag{4.1}$$

Through the reduction of the log mean temperature difference (LMTD) between the two heat transfer media, the evaporator and condenser temperatures are brought closer to the heat source and sink temperatures, respectively, and the cycle COP can be improved. However, to maintain the same heat transfer capacity (Q), reduced temperature difference subsequently requires the increase of the heat transfer surface area (A) and/or the overall heat transfer coefficient of the evaporator/condenser (U).

Improving the energy performance of current HVAC equipment is most commonly accomplished by "overdesigning" the evaporators and condensers. Within economic constraints, this is the method most favored by manufacturers. In Chapter 3.0, it has been shown that the most advantageous technology advancement is that of inexpensive heat transfer surfaces which encourage the use of efficient evaporators and condensers. However, the upper limit of evaporator temperature must be consistent with the dehumidification function of the coil. Unless alternate dehumidification methods, such as desiccation, are used, the potential improvement in energy efficiency through evaporator coil design is limited when compared with that of the condensers. Nonetheless, the improvement in evaporator effectiveness will result in capital cost reduction.

In addition to heat exchange between working fluid and heat source/sink, many types of vapor compression equipment use a special refrigerant-to-refrigerant heat exchanger to transfer heat from the subcooled liquid to the suction gas. In systems with capillary tubes for thermal expansion, for

example, the heat is often transferred through a solder connection, which is called a capillary tube-suction line heat exchanger. Since the liquid refrigerant is warmer than the suction gas, heat is transferred from the liquid resulting in a decreased irreversibility of the expansion process, and a corresponding increase in COP.

Using more effective air-to-refrigerant, water-to-refrigerant, or refrigerant-to-refrigerant heat exchangers all increases the potential for improved equipment performance. Numerous examples of heat transfer research are reported in the literature, some of which specifically address air conditioning applications. The collective objectives of the research focus on reducing heat exchanger size, reducing pumping power by lowering the pressure drop, increasing heat transfer capacity, or reducing approach temperature difference. The study of heat transfer performance is referred to as heat transfer augmentation, enhancement, or intensification. Depending on whether external power is required, augmentation techniques may be classified as passive or active. Passive enhancement techniques include the use of treated surfaces, rough surfaces, extended surfaces, displaced enhancement devices, swirl flow devices, coiled tubes, surface tension devices, additives for liquids, and additives for gases. Active enhancement techniques include use of mechanical aids, surface vibration, fluid vibration, electrostatic fields, injection, suction, and jet impingement. In addition, compound enhancement techniques use two or more enhancement techniques simultaneously to promote heat transfer augmentation (Bergles et al. 1983).

Standard types of heat exchangers currently used in chillers and air conditioners are shell-and-tube and tube-and-fin. For basic economic reasons (including material cost, production cost, fluid pumping cost), only the simpler among the various passive surface enhancement techniques are used. Standard methods used were discussed in Chapter 3.0. Manufacturers have invested substantially in research on standard heat transfer surfaces. Making the heat exchanger coils more compact, thus lowering first-cost, and reducing pressure drops through the coils, thus lowering operating cost, are their primary concerns.

4.2.3 Isothermal Compression

For a given compressor pressure ratio, the lowest power requirement is the result of an isothermal compression process. Improved performance of the isentropic compression process is achieved by discrete interstage cooling, which is a step-wise approximation to an isothermal process. Theoretically, the more cooling stages, the closer the approximation, however this is often not cost-effective. Alternatively, isothermal compression may be approached by injection of a second component into the compressor to quasi-continuously cool the working fluid.

In one design, a multicomponent flow in the compressor is tailored by controlled seeding of micro-sized particles into the compressor medium. Theoretically, the particles act as heat sinks, affecting a near isothermal compression process. After the compression process is completed, the particles are separated from the gas, cooled and then recycled. The greatest difficulties for successfully using this design include the following:

- handling and injection of micro-sized particles
- separation of particles from the working fluid
- slip velocities in strongly accelerated rotational flows
- erosion of compressor surfaces
- cooling of separated particles.

This particle-seeding compressor concept is an on-going effort of the DOE's Energy Conversion and Utilization Technologies Program. Current development is geared for radial-flow compressors (Minardi 1984).

In another design, the cooling effect on the centrifugal compressor is achieved by the injection of a coolant, usually water. The liquid is admitted at atmospheric conditions to the rotating shaft along the path of compression. The injected liquid cools the gas, or superheated vapor, by full or partial vaporization and keeps it below adiabatic compression temperature, again theoretically effecting a near-isothermal compression process (Pinkus 1983). The key problems with liquid-injection are the possible slugging of the compressor and erosion of the compressor's surface.

Current studies on isothermal compression focus on the radial-flow compressor for use in gas turbines. Although isothermal compression is more energy efficient than isentropic compression, a basic incompatibility exists when using an ideal isothermal compression process in a vapor compression cycle because of the required temperature lift between the condenser and evaporator. One potential way to incorporate the isothermal compression concept into the vapor compression cycle is to use a two-stage compression process, with both isentropic and isothermal compression in order to eliminate the superheat at the compressor discharge. Ideally, the gas is first compressed isentropically to the condenser temperature and then compressed isothermally in the second compressor to the condenser pressure. Alternatively, this two-stage compression may potentially be completed in a single step by precise control of the amount and distribution of the secondary component injected.

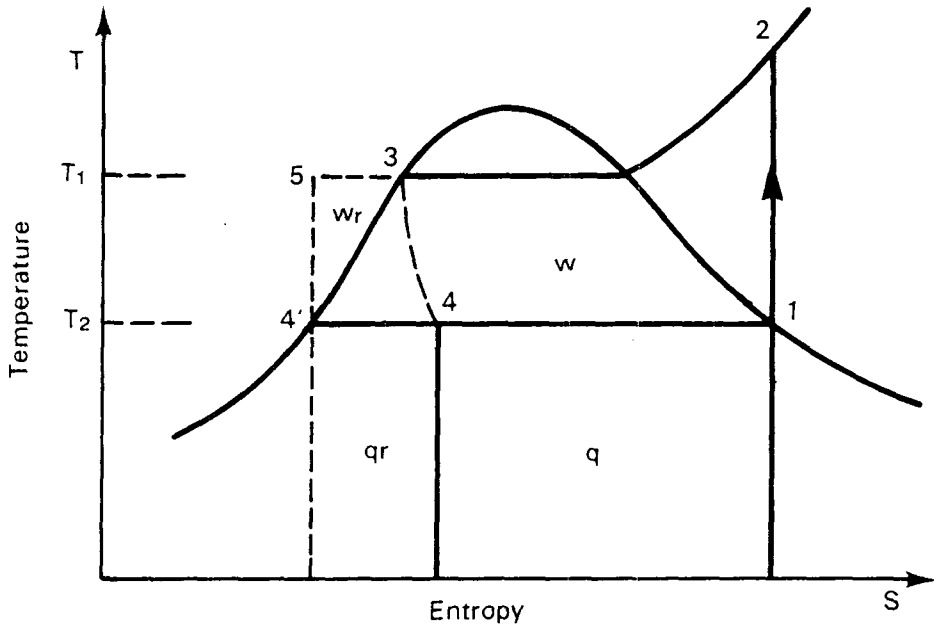
4.2.4 Regenerative Refrigeration Cycle

The objective of this modified vapor compression cycle is to reduce throttling losses by subcooling the refrigerant liquid before it enters the evaporator. The subcooling process reduces the irreversibility of the throttling valve and increases the cooling capacity.

In the temperature-entropy diagram in Figure 4.6, the theoretical COP of the regenerative cycle and its improvement over the basic cycle are demonstrated. The conventional cycle follows the path 1-2-3-4-1, with the areas q and w representing the cooling capacity and isentropic compression energy requirement, respectively. Thus, the COP of the conventional cycle is:

$$COP = q / w \tag{4.2}$$

The ideal regenerative cycle follows the path 1-2-3-4'-1, and the refrigerant liquid is subcooled from temperature T_1 at the inlet of the heat exchanger to the temperature T_2 at the exit. The areas q_r and w_r represent, respectively,



1-2-3-4-1 Reversed Rankine Cycle Path

1-2-3-4'-1 Regenerative Refrigeration Cycle Path

FIGURE 4.6. Temperature-Entropy Diagram for Reversed Rankine Cycle Versus Regenerative Refrigeration Cycle

the additional cooling capacity and the additional energy requirement of the modified cycle. Thus, the COP of the regenerative cycle is increased to:

$$COP = (q + q_r) / (w + w_r) \tag{4.3}$$

The special feature of the regenerative cycle design is a regenerative heat exchanger, or regenerator, in which the liquid refrigerant is subcooled. The heat exchanger acts as a "regenerative economizer." The principle of the cycle is to break the subcooling process into time intervals for two different modes of operation. Their circuit diagrams are shown in Figure 4.7. During normal operation, the liquid refrigerant is subcooled by passing through the regenerative heat exchanger, which has been precooled in the regenerative cycle. After a period of normal operation, the regenerator material is heated. When the temperature before the throttling valve rises above its setpoint, the regenerative cycle is activated. During the regenerative cycle,

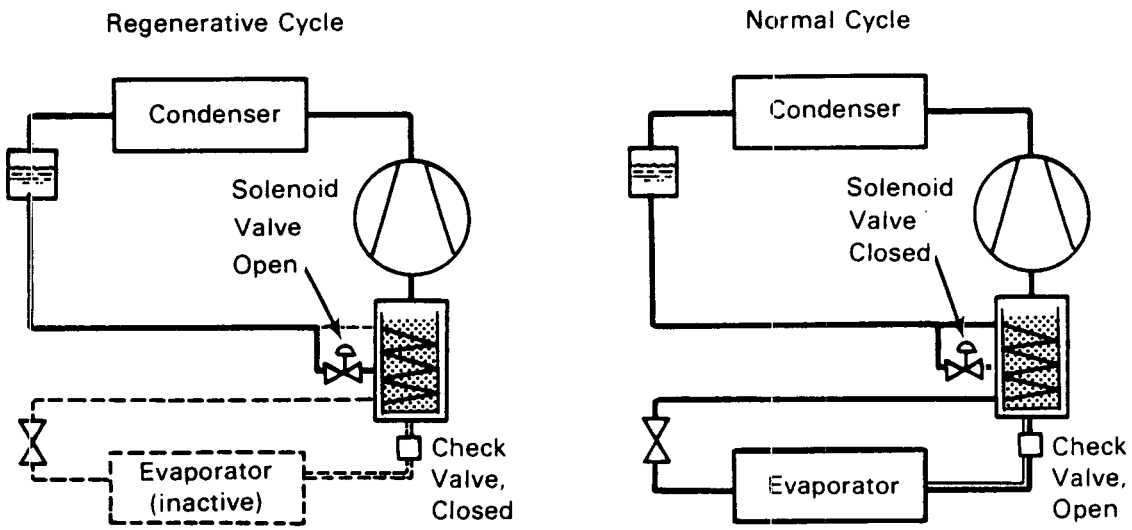


FIGURE 4.7. Circuit Diagram for the Regenerative Refrigeration Cycle

the liquid refrigerant is channeled into the warm heat exchanger, is rapidly boiled off, and cools the regenerative heat exchanger. During this period, the check valve to the evaporator is closed, and the evaporator is basically inactive. When the regenerator is cooled to a preset temperature level, the normal operation is resumed.

A computer program was developed to predict the performance of this cycle. As an example, for an R-502 system with a temperature lift of 108°F, the cooling capacity was theoretically calculated to increase by 50% while the power increased by 5%, resulting in a 43% increase in COP. However, the performance improvement declined significantly at lower temperature lifts (e.g., for 54°F temperature lift, the capacity was increased by about only 12%). Moreover, the actual performance improvement reported from laboratory tests was only about 30% to 50% of the theoretical values. These tests were conducted on small hermetic compressors of 3 kW to 5 kW (Granryd 1975, 1977 and 1985).

One of the two crucial concerns about the regenerative cycle concept is that of the impact on the compressor due to sudden inlet pressure fluctuations at the periodic switch-over of normal and regenerative cycles. Although no signs of malfunction have been detected during laboratory tests if the compressor is sufficiently sized, actual operation over the lifetime of the equipment still causes concern. In addition, because of the periodic switch-over of the

two modes, the operation never attains a steady-state condition. Of particular concern is that during the regenerative cycle, the space cooling effect occurs only because of the thermal inertia of the evaporator material, which has to be recovered at the beginning of each normal cycle.

4.2.5 Variable Capacity Equipment

Over the lifetime of any HVAC equipment, the majority of its operation is at part-load condition. Part-load performance of single-speed units results in reduced energy efficiency because of transient losses. To reduce cyclic performance degradations, variable capacity designs have been implemented, including multiple capacity, AC inverters, and brushless DC.

Multiple speed compressors and equipment with multiple compressor or cylinder unloading provide two or more discrete levels of capacity. Part-load operation is accomplished by load-sharing between the different capacity levels. Although the amount of equipment on-off cycling is reduced, the equipment still cycles between speeds, number of active cylinders, or number of compressors operating. In addition, problems exist because of the inflexibility of capacity modulation ratio.

Another way to achieve multiple capacity is to use the dual-stroke concept, which has been incorporated into a design for a residential heat pump with a positive displacement compressor. Compressor capacity modulation is achieved by stroke change using a circular, eccentrically-bored cam between crankpin and connecting rod. The cam drives against one of the two rotational stops attached to the crankpin, depending on the direction of the crankshaft. Field test results indicated that the annual average performance was improved by about 20% compared with the performance of two-speed heat pumps (Veyo 1982, Veyo 1984).

The inverter-driven variable capacity units are gaining in popularity primarily because of the declining cost of inverters. Figure 4.8 shows the decline in cost for large inverters compared with costs for equipment using other variable speed modulation methods (Stoecker 1986). Within the context of air conditioning technology, an inverter is a device that generates AC power of variable frequencies in order to change the rotational speed of the electric

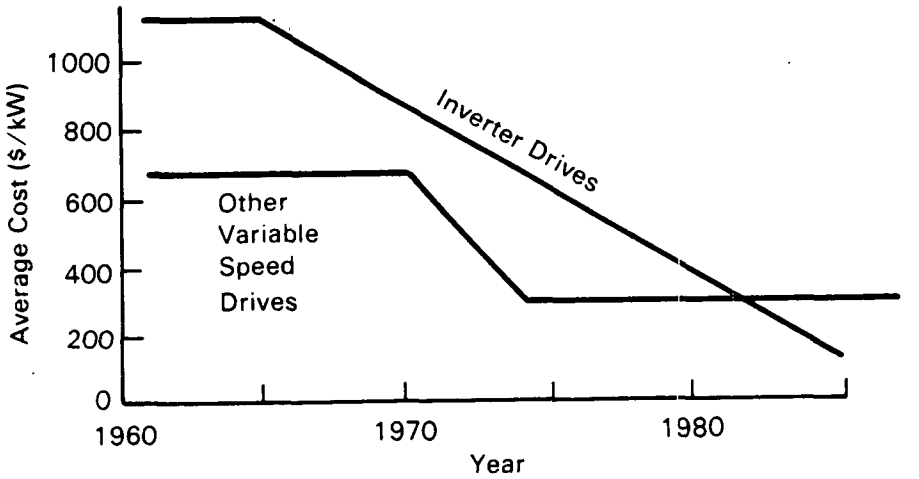


FIGURE 4.8. Average Cost of Inverter Drives Compared with Costs of Other Variable Speed Drives for Large Equipment

motor for compressor capacity modulation. The feasibility of AC inverter drives for heat pump capacity modulation has been investigated in an EPRI-sponsored project. Six advanced inverter concepts have been studied:

- square wave, voltage source inverter (VSI)
- square wave, current source inverter (CSI)
- pulse-width-modulated, voltage source inverter (PWM)
- electronically commutated synchronous motors with a permanent magnetic field (ECM)
- pole-amplitude-modulation in induction motors (PAM)
- high-frequency, high-speed motors using low-loss magnetic materials (HFM).

Preliminary results indicate that the VSI and PWM inverter-driven induction motors appear to be the most versatile for continuous speed adjustments of 6 to 1 (Katsenelenbogen 1986; DOE 1982). For the variable capacity equipment, the compressor power input is low at low-speed and fan power becomes a significant fraction of the total power if fan speed is held constant. To reduce fan energy requirements, the inverters can also be applied.

The market share of inverter-driven residential heat pumps in Japan was estimated to be 40% for 1985. In particular, Toshiba uses inverters for motors

ranging from 3/4 to 300 hp for a variety of applications. The effective speed modulation range is 3 to 1 (Katsenelenbogen 1986). Because the cost of electricity in the U.S. is lower than in Japan, U.S. manufacturers feel that inverter prices must be further reduced before they can be implemented on a large scale. However, interest has increased and the pace is likely to pick up as the Japanese units begin to gain increasing shares in the U.S. market. York has already marketed inverter-driven large tonnage chillers as well as residential heat pumps.

The variable-speed brushless DC motor design has been incorporated into residential heat pumps in another EPRI-sponsored project. The brushless DC motor with electronic commutation claims high efficiency and long service life. Speed modulation through electronic control is achieved over a wide power range not easily attained with AC induction motors (Katsenelenbogen 1986; DOE 1982).

To improve energy efficiency of equipment and perceived thermal comfort, variable-capacity designs seek to improve the capacity modulation ratio. However, the transient control of the refrigerant circuit over the full capacity range often presents a problem. Specifically, it is very important to match the throttling mechanism with the operational range of capacities. The task of the throttling device is to provide optimal equipment operation. To improve the equipment's seasonal performance, a high level of performance must be maintained over the entire operational range. For high reliability, compressor overheat and an abrupt liquid back flow must be prevented (Nakashima et al. 1985). Optimal throttling device operation is defined in terms of the following three elements:

- maximum use of evaporator surface
- proper superheat at evaporator exit to prevent compressor slugging
- stable operation with no hunting (Thuesen and Winter 1985).

To improve the response of the throttling device and to secure a high operation reliability over the capacity range, microprocessor electronic control systems are required. Studies on the refrigerant circulation gain, time constants, control algorithms, and related factors for dynamic control are needed (Nakashima et al. 1985; Gruhle 1985; Gruhle and Isermann 1985).

5.0 R&D RECOMMENDATIONS

This study assessed the performance and improvement potentials of commercial-sized centrifugal chillers and packaged and split air conditioners. The conclusion is that the factors constraining the energy efficiency of the current equipment are basically nontechnical, mostly economic, in nature. The chillers and air conditioners both use the well-established vapor compression refrigeration technology. As such, they represent a mature technology where higher efficiency is achieved through increased complexity and thus increased first-cost. Further technical advancements should focus on reducing the incremental cost of the established energy-efficient features and on improving the application of the technology for better part-load performance and load-matching.

If the design features used in high-efficiency units were uniformly implemented by all air conditioners and chillers, substantial reduction in energy consumption would result. For instance, established technologies are available to manufacture high performance water chillers, with energy consumption rates as low as 0.5 kW/ton (excluding auxiliary energy requirements), or an equivalent EER of 24 Btuh/W for medium-sized units of 375 tons. These technologies include extended heat exchanger surfaces, flash tanks, etc. However, because of the corresponding increase in equipment first-cost, these features are seldom incorporated. Even more drastic economic barriers are in place for the air conditioners. The average EERs for 10 ton air conditioners are from 7.1 to 10.6 Btuh/W (including auxiliary power), with the vast majority of units sold at the lower efficiencies.

Among the various R&D needs and opportunities described in Chapter 4.0, PNL's recommendations for DOE-sponsored chiller and air conditioner research are discussed in the following sections. In addition to the information already presented, more specific research topics will be identified and further elaborated. The recommendations include the following:

- to better understand and engineer the heat exchanger coils
- to support R&D for variable capacity drives, including AC inverters for chillers and fans, and brushless DC for air conditioners

- to support R&D for nonazeotropic refrigeration cycle research for chillers with low-temperature application and improved part-load performance
- to assess environmentally benign refrigerants and refrigeration cycles.

In addition, because of the first-cost constraints for the equipment, it is recommended that life-cycle costing be promoted within the context of an equipment research program. Finally, based on input throughout this research project from HVAC researchers, recommendations are also made for work addressing commercial HVAC systems. Systems issues in commercial buildings are more important to energy consumption and thermal comfort than is the efficiency of the air conditioner or chiller. Moreover, systems issues strongly affect the undertaking of the best types of central plant equipment research.

5.1 HEAT EXCHANGERS

The development of inexpensive and effective heat exchangers for use as evaporators and condensers is a straightforward if uninspiring research need. The discussion here will bring out a few long-term and more fundamental topics best suited to DOE research.

Commonly used heat exchangers in chillers and air conditioners include shell-and-tube types for water-to-refrigerant heat transfer, and tube-and-fin types for air-to-refrigerant heat transfer, and tube-to-tube types for refrigerant-to-refrigerant heat transfer. Performance improvements of these heat exchangers will result in improved energy efficiency of the vapor compression equipment. The standard method currently used to promote the effectiveness of these heat exchangers is passive surface enhancement, including treated surfaces, rough surfaces, and extended surfaces.

For the tube-and-fin evaporators and condensers used in air conditioners and the shell-and-tube evaporators used in chillers, performance is satisfactorily understood; the primary R&D need is that of production methods to reduce the cost of heat exchangers. If the heat exchanger can be made with less cost, a larger heat exchanger can be used, leading to overall equipment performance

improvement. Similarly, if a more effective heat exchanger can be made at no added cost, its implementation will also result in improved equipment performance.

However, for the shell-and-tube chiller condensers, some basic heat transfer research is needed. First, most data available on shell-side refrigerant condensation are for single-tubes. Since additional factors such as condensate dripping from tube to tube within the bundle (called inundation), vapor shear, number of tube passes, tube layouts, etc., also affect the performance of tube-bundles, the condensing coefficients are different from those of single tubes. It is normally expected that tube bundles have lower condensation heat transfer performance compared to the single tubes, but only very limited information has been gathered.

In addition, doubly-enhanced tubes for both shell- and tube-side enhancements and innovative enhanced fin geometries have recently been promoted. However, because of the strong effect of capillary retention of condensate between fins, it appears that more knowledge is needed to determine the optimal fin spacing as a function of the fin shape, height, thickness, etc.

Contaminants may further complicate the condensing heat transfer. Noncondensable gases may exist in the condenser, especially if the evaporator operates below atmospheric pressure. These gases produce added thermal resistance, increasing condenser pressure and, consequently, compressor power. Moreover, compressors often transport oil to the condenser, affecting its heat transfer performance. For improved integrated design of the condenser tube-bundle and shell, basic data on the above subjects are essential (Webb 1984).

Finally, another very important area of research with reference to all types of heat exchangers is surface fouling. The performance of field-operated evaporators and condensers often deteriorates with time because of surface fouling, and this is the most common cause of long-term performance degradation of air conditioning equipment. In particular, severe fouling problems may be present in the condenser tubes of water chillers from the recirculation of cooling tower water. For the air conditioners, dust and lint deposits also reduce the coil performance. Studies on the heat exchanger surface fouling,

particularly on innovative ways to retard its buildup, and controls to detect early warnings of potential equipment failure will prove useful for the design and operation of these units.

5.2 VARIABLE CAPACITY DRIVES

The consensus of HVAC researchers and practitioners is that the part-load performance of air conditioning equipment must be improved. Different variable capacity designs are used to reduce equipment part-load losses. Among the various methods used, the variable speed drives have the greatest potential. The two types of variable speed drives that are currently drawing special attention are the inverters and the brushless DC motors. They could both benefit from further research. The AC inverter drive is of particular interest for chillers, while brushless DC drives are suitable for relatively small air conditioners.

The application of small inverter drives in residential heat pumps began in Japan, and their market share in 1985 was estimated to be 40%. Even though the cost of inverters is declining, energy costs in the U.S. are low enough that U.S. manufacturers are in general still very concerned about the added cost to the equipment. In addition, the reliability of the electronics also prompts concern about their large-scale implementation. Nonetheless, it is generally agreed that inverter drives should be applied to large tonnage chillers. In 1984, inverters were capable of reaching from 35-75 Hz and currently the range is 30-90 Hz (Miller 1986). To effectively take advantage of the inverters for commercial building applications, the capacity modulation range required is expected to be higher than for residential applications. The development of inverters with wide modulation ranges applicable to large chillers is an important research topic. In addition, the application of inverter drives to fans also offers considerable energy conservation potential.

Initial assessment of brushless DC motors with electronic commutation shows high efficiency and long service life. In addition, speed modulation can be achieved over a wide power range which is not easily achieved with AC

motors. Currently, the brushless DC application is cost-effective only for small-sized units. However, its use in light commercial air conditioners is potentially feasible. Further development to reduce costs is necessary.

Overall, there is substantial activity in the private-sector to develop the variable speed drive. Nonetheless, some supporting R&D by DOE is essential.

Finally, for all variable capacity vapor compression equipment, the transient control of the refrigerant circuit over the full capacity range often presents a problem. To effectively improve the performance of the variable capacity designs, basic refrigerant thermal and flow characteristics at transient conditions should be studied to facilitate the proper design and control of the throttling device. Considerable work is being done in Japan on this topic with application to small machines. Study of the refrigerant circulation gains, time constants, control algorithms, and related factors for dynamic controls is essential.

5.3 NONAZEOTROPIC REFRIGERATION CYCLE

The application of the nonazeotropic refrigeration cycle will potentially enhance the performance of water chillers. The nonazeotropic system is rather complex but may yield important improvements in both steady-state EER and part-load performance. Although the nonazeotropic cycle research is currently included in the DOE program, its application to centrifugal chillers is not specifically addressed.

For standard chiller application, the cycle performance improvement potential using nonazeotropic refrigerants may be limited because of the low temperature gradient of the chilled water. However, the temperature glide may be intentionally increased, for example, by reducing the chilled water flowrate. It can then be used to produce low-temperature chilled water, perhaps in connection with cool storage systems. In addition to improved cycle efficiency from the gliding temperature heat transfer, the low-temperature nonazeotropic chiller can also enjoy the advantage of low-temperature air distribution. As

noted earlier, advantages of using low-temperature space conditioning include reducing the chilled water pump energy, reducing fan energy, and reducing distribution equipment first-cost.

The nonazeotropic refrigeration system also offers potential for capacity modulation by composition shifting. Large-sized chillers normally require a wider range of capacity modulation than residential units. The nonazeotropic system may be used to complement a variable speed unit in order to extend its modulation range, once the basic utility of nonazeotropic systems has been demonstrated. However, the additional difficulties involved in nonazeotropic refrigerant operation with a variable speed compressor should be anticipated.

An added potential benefit of nonazeotropic refrigerants is derived from their possible substitutions for the environmentally detrimental R-11 and R-12.

Before design and development of the hardware, research in this area should begin with the screening of compatible refrigerant components, thorough examination of potential system configurations, and the resultant performance improvements (Calm & Didion 1985).

5.4 ENVIRONMENTALLY BENIGN REFRIGERANTS AND REFRIGERATION CYCLES

Because of the reported environmental impacts from recent assessments regarding the release of CFCs, increased attention in the HVAC industry is focusing on the potential shifting to more environmentally benign refrigerants. It is important for the industry to prepare for any necessary changes and to assess alternative refrigerants and refrigeration cycles which may pose less environmental threat. In addition to the nonazeotropic refrigeration cycle discussed above, two kinds of cycles, the multi-component/mixed phase cycle and the reverse Stirling cycle, are noted here.

The multi-component/mixed-phase refrigeration cycles may be a potential candidate. The advantage over standard single-component vapor compression cycles is that by injecting the phase-change component into the gaseous carrier, the work of compression required to reach a given pressure ratio is decreased and the recoverable expansion energy is increased. The most commonly examined system of this type is an air cycle with the injection of water.

Although its performance evaluation based on thermodynamic analysis shows a significant decline in cycle efficiency with decreasing compressor and expander efficiencies, application of the concept to alternative refrigerants may prove beneficial. An in-depth theoretical thermodynamic analysis with specific attention directed to the array of compatible components should be conducted first. In addition to basic thermodynamic analysis for ideal conditions, cycle actualization sensitivity related to pressure drops and component inefficiencies should be included, since this is where the current concept appears to have shortcomings. If the concept proves to be thermodynamically feasible, further issues to be addressed include the design and control of the injection of the phase change component and its effect on the reliability and service life of the compressor (Creswick 1981; Edwards 1978; Ecker, Edwards, and Wark 1978).

The reverse Stirling cycle has potential for providing cooling, and its use is well established in specialty applications in the aerospace and defense industry for refrigerating infrared sensors (Walter 1986). The cycle uses a working fluid of hydrogen, helium, or air. Its performance efficiency is not considered capable of competing with the vapor compression cycle at typical space cooling conditions, but it has advantages for delivering lower temperatures. Though Stirling cycle machinery is difficult to develop and still needs substantial research, its examination from the point of view of a non-CFC situation should be attempted. After all, one of the basics of thermodynamics is that the Stirling is the only cycle ideally capable of reaching Carnot. In Stirling engine work, efficiencies have been improved significantly, even doubled in the twenty years since serious research began, and there may be related potential in refrigeration.

5.5 HVAC SYSTEMS

Although the HVAC system area was not formally addressed by this study, throughout the development of the current study there was considerable evidence that systems problems are viewed by HVAC researchers and practitioners as the key to the energy conservation in commercial buildings. System-level HVAC

issues do not simply mean equipment selection, such as the selection of combinations of compressor sizes for better load-matching. Instead, it refers to thermal problems affecting the performance and interactions of different equipment subsystems. This type of problem is typically not addressed by current DOE programs.

Systems integration at the building level focuses on issues that result from the interactions among the HVAC equipment, distribution equipment, and building envelope. Systems problems arise in commercial buildings, particularly large ones, because of the size and complexity of the dynamic thermal systems involved. Compared with residential dwellings, commercial buildings have more complicated heating and cooling requirements, including zoning, coincident heating and cooling, and ventilation requirements. A considerable amount of wasted energy may result if this complex and dynamic HVAC system is not properly designed and controlled. On the other hand, it may offer substantial opportunity to conserve energy.

Examples of system level energy conservation opportunities in commercial buildings include:

- optimal evaporator temperature setpoint to minimize energy conversion equipment and distribution system energy requirements
- reduction of fan and pump energy requirements by optimal design layout of distribution system
- proper sizing of energy conversion equipment to reduce part-load losses
- optimal use of outdoor air for ventilation and economizer cooling
- reduction of energy used for reheating cold air supply
- heat reclamation from cooling zones and transfer to heating zones
- proper balance or adjustment of equipment
- proper use of system control.

These opportunities appear simple, and are indeed well-known, but often either the knowledge and/or incentive to implement them does not exist. The system-related energy waste cannot effectively be reduced with improved energy conversion equipment, but must be addressed as part of the integrated equipment and building system.

The integrated design of the HVAC equipment, distribution system, control system, and building envelope, and the optimization of the integrated system hold the key to energy conservation in commercial buildings. The system-level research required is two-fold. First, detailed knowledge and information on many of the system-related issues are basically lacking. For example, the data on the convection heat transfer between zones, loss coefficients in standard duct configurations, applied data on the characteristics of control systems, field data on the load-imposed operation requirements of equipment, etc. are unavailable. Second, detailed trade-off of various energy conservation designs has not been well-studied. An example of a system-level trade-off is the selection of the evaporator setpoint to optimize efficiency of the energy conversion equipment and of distribution energy requirements of fans and pumps while providing a proper amount of outdoor air for ventilation and economizer cooling. These system-level issues are generally not thoroughly addressed in the system design process by the architects and engineers. They are often only treated from the point of view of capital cost reduction rather than of energy efficiency.

Computer and measurement tools now exist for developing detailed analyses in the system area. The improvements on commercial HVAC system performance offer tremendous opportunities for energy conservation. It is therefore recommended that research be started in order to structure and specify research problems in commercial HVAC systems.

6.0 REFERENCES

Air-Conditioning and Refrigeration Institute (ARI). 1981. Standard for Unitary Air-Conditioning Equipment. ARI Standard 210-81, Arlington, Virginia.

Air-Conditioning and Refrigeration Institute (ARI). 1983. Standard for Centrifugal or Rotary Water-Chilling Packages. ARI Standard 550-83, Arlington, Virginia.

Air-Conditioning and Refrigeration Institute (ARI). 1986. Directory of Certified Unitary Air-Conditioners, Unitary Air-Source Heat Pumps and Sound-Rated Outdoor Unitary Equipment (Effective January 1 to June 30, 1986). ARI, Arlington, Virginia.

Air-Conditioning, Heating and Refrigeration News (ACHRN). 1985. "Statistical Panorama." ACHRN. April 8, 1985.

Air-Conditioning, Heating and Refrigeration News (ACHRN). 1986. "Manufacturers, Environmentalists Praise Minimum Standards." Air-Conditioning, Heating and Refrigeration News, August 25, 1986.

American Society of Heating, Refrigeration and Air Conditioning Engineers (ASHRAE), Inc. 1984. ASHRAE Handbook of Systems. ASHRAE, New York.

American Society of Heating, Refrigeration and Air Conditioning Engineers (ASHRAE), Inc. 1985. ASHRAE Handbook of Fundamentals. ASHRAE, New York.

American Society of Heating, Refrigeration and Air Conditioning Engineers (ASHRAE)/National Society of Professional Engineers (NSPE). 1986. Proceedings of a Workshop on Research Needs in HVAC Systems. Co-Sponsored by University of Colorado and ASHRAE/NSPE.

Anderson, W. M. 1984. "Desiccant-Based Air Conditioning for Commercial Buildings." Presented at the 11th Energy Technology Conference, 1984, Washington, D.C.

Atwood, T. 1985. "The ABCs of NARBs (Nonazeotropic Refrigerant Blends)." ASHRAE Transactions. 91(2).

Bergles, A. E., V. Nirmalan, G. H. Junkhan, and R. L. Webb. 1983. Bibliography on Augmentation of Convective Heat and Mass Transfer-II. ISU-ERI-AMES-84221, National Technical Information Service, Springfield, Virginia.

Brodrick, J. R. 1986. Commercial Buildings, Energy Consumption, and Natural Gas Markets. Gas Research Institute (GRI), Chicago, Illinois.

Calm, J. M., and D. A. Didion. 1985. "Research and Development of Heat Pumps Using Nonazeotropic Mixture Refrigerants." ASHRAE Transactions. 91(2).

Chemistry and Engineering News (CEN). 1986. "Trace Gases Altering Atmosphere's Composition. Chemistry and Engineering News, January 20, 1986, pp. 5-6.

Creswick, F. A. 1981. "The Theoretical Thermal Performance of Air-Cycle Heat Pumps with Water Injection." ASHRAE Transactions. 87(1):1025-1035.

Didion, D. A. 1986. "Nonazeotropic Mixture as Working Media for Refrigeration Cycles." Presented at a Short Course on Recent Advances in Refrigeration Machinery: Design, Manufacturing and Application, Sponsored by the U.S. National Committee of the International Institute of Refrigeration, August, 1986, Purdue University, West Lafayette, Indiana.

Duffy, G. 1986a. "CFC Net Ozone Depletion Rate Still Not Established but Greenhouse Effect Added." Air-Conditioning, Heating and Refrigeration News (ACHRN), February 17, 1986, p. 29.

Duffy, G. 1986b. "EPA to Decide on Regulating CFCs by May 1987." Air-Conditioning, Heating and Refrigeration News (ACHRN), February 17, 1986, p. 3.

Ecker, A. L., T. C. Edwards, and K. Wark. 1978. "A Tabular Data Boundary-State Analysis of the Edwards Cycle." ASHRAE Transactions. 84(2):173-184.

Edwards, T. C. 1978. "A New Air Conditioning, Refrigeration and Heat Pump Cycle." ASHRAE Transactions. 84(2):150-172.

Electric Power Research Institute (EPRI). 1984. 1985-1989 Research and Development Program Plan. EPRI, Palo Alto, California.

Gas Research Institute (GRI). 1985. 1986-1990 Research and Development Plan and 1986 Research and Development Program. GRI, Chicago, Illinois.

Granryd, E. 1975. "A Regenerative Refrigeration Cycle." Presented at the 15th International Congress of Refrigeration, 1975, Moscow, USSR.

Granryd, E. 1977. "Method of Improving Refrigerating Capacity and Coefficient of Performance in a Refrigerating System, and a Refrigerating System for Carrying Out Said Method," U.S. Patent 4,014,182, March, 1977.

Granryd, E. 1985. "A Regenerative Refrigeration Cycle." STU Energy Technology. No. 3, pp. 8-12.

Gruhle, W. D. 1985. "Digital Control of a Refrigerant Evaporator." Paper presented at the 7th IFAC Conference on Digital Computer Applications to Process Control, September, 1985, Vienna, Austria.

Gruhle, W. D., and R. Isermann. 1985. "Modeling and Control of a Refrigerant Evaporator." Journal of Dynamic Systems, Measurement, and Control. 107:235-240.

Harris, L., S. Katsenelenbogen, and J. Bluestein. 1986. Status of the Space Cooling Equipment Market in the Commercial Sector. GRI 86-0085, Gas Research Institute, Chicago, Illinois.

Hurley, J. R., K. C. Shukla, and M. P. Grimanis. 1986. Commercial Appliance Energy Use Update. GRI-86/0184, Gas Research Institute, Chicago, Illinois.

Jennings, B. H., and S. R. Lewis. 1959. Air Conditioning and Refrigeration. International Textbook Co., Scranton, Pennsylvania.

Katsenelenbogen, S. 1986. Status of Compressors and Electric Heat Pumps: Review of Literature. GRI 86-0014, Gas Research Institute, Chicago, Illinois.

MacDonald, M., D. Goldenberg, and E. Hudgins. 1982. RD&D Opportunities for Large Air Conditioning and Heat Pump Systems. ORNL/Sub/80-13817/1&20, prepared by TRW for Oak Ridge National Laboratory, Oak Ridge, Tennessee.

Mahoney, T. 1986. "Manufacturers, Environmentalists Agree on National Appliance Standards." Air-Conditioning, Heating and Refrigeration News (ACHRN), August 18, 1986, pp. 1-2.

Miller, M. 1986. "Product Refinement is Evident at Japan HVACR Exposition." Air-Conditioning, Heating and Refrigeration News (ACHRN), April 21, 1986, pp. 4-6.

Minardi, J. E. 1984. Multicomponent Flow Compressor: A New Concept for Efficient Energy Conversion and Utilization. DOE Unsolicited Proposal No. R-4989, University of Dayton, Dayton, Ohio.

Nakashima, Y., H. Iijima, M. Umehara, and F. Matsuoka. 1985. "Reversible-Flow-Type Linear Expansion Valves for Heat Pumps." ASHRAE Transactions. 91(2).

Pinkus, O. 1983. "Liquid Particles Dynamics and Rate of Evaporation in the Rotating Field of Centrifugal Compressors." Journal of Engineering for Power. 105:80-87.

Senshu, T., A. Arai, K. Oguni, and F. Harada. 1985. "Annual Energy-Saving Effect of Capacity-Modulated Air-Conditioner Equipped with Inverter-Driver Scroll Compressor." ASHRAE Transactions. 91(2).

Shimma, Y., T. Tateuchi, and H. Sugiura. 1985. "Inverter Control Systems in the Residential Heat Pump Air Conditioner." ASHRAE Transactions. 91(2).

Stoecker, W. F. 1986. "Current Trends in Refrigeration." Presented at a Short Course on Recent Advances in Refrigeration Machinery: Design, Manufacturing and Application, Sponsored by the U.S. National Committee of the International Institute of Refrigeration, August 1986, Purdue University, West Lafayette, Indiana.

Stoecker, W. F., and C. I. McCarthy. 1984. The Simulation and Performance of a System Using an R-12/R-114 Refrigerant Mixture. ORNL/Sub/81-7762/3&01, Oak Ridge National Laboratory, Oak Ridge, Tennessee.

Thuesen, S. E., and J. Winter. 1985. "Improved Control of Liquid Injection into Display Case Evaporators." ASHRAE Transactions. 91(1A):80-91.

U.S. Department of Commerce, Bureau of the Census. 1984. Air-Conditioning and Refrigeration Equipment, Current Industrial Reports. MA35M(84)-1, U.S. Department of Commerce, Washington, D.C.

U.S. Department of Energy (DOE). 1982. U.S. Heat Pump Research and Development Projects. DOE/CE-0035, U.S. Department of Energy, Washington, D.C.

U.S. Department of Energy (DOE), Office of Buildings and Community Systems. 1985. Overview of Building Energy Use and Report of Analyses--1985. DOE/CE-0140, U.S. Department of Energy, Washington, D.C.

Veyo, S. E. 1982. "An Optimized Two-Capacity Advanced Electric Heat Pump." ASHRAE Transactions. 88(2):988.

Veyo, S. E. 1984. "Dual-Stroke Heat Pump Field Performance." In Proceedings of the DOE/ORNL Heat Pump Conference: Research and Development on Heat Pumps for Space Conditioning Applications, ORNL-CONF-841231, pp. 27-36. Oak Ridge National Laboratory, Oak Ridge, Tennessee.

Walter, G. 1986. "Stirling Cryocoolers: Trends in Development". In Proceedings, 21st IECEE, paper 860192, p. 446. American Chemical Society, Washington, D.C.

Webb, R. L. 1984. "Shell-Side Condensation in Refrigerant Condensers." ASHRAE Transactions. 90(1B).

APPENDIX A

DESCRIPTION OF VAPOR COMPRESSION EQUIPMENT PERFORMANCE

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DESCRIPTION OF VAPOR COMPRESSION EQUIPMENT PERFORMANCE

Both the air conditioner and chiller are vapor compression equipment. Their performance is rated at standard conditions and uses methods established by the Air Conditioning and Refrigeration Institute (ARI). For unitary air conditioners, the rating conditions are given in ARI Standard 210-81 (ARI 1981) which specifies the following evaporator and condenser temperatures:

Air entering evaporator:	80°F dry bulb
	67°F wet bulb
Air-cooled condenser inlet air:	95°F
Water-cooled condenser inlet water:	85°F
outlet water:	95°F

For centrifugal water chillers, the standard rating conditions are given in ARI Standard 550-83 (ARI 1983). The standard rating conditions are specified as the following:

Leaving chilled water temperature:	44°F
Chilled water flowrate:	2.4 gpm/ton
Water-cooled condenser inlet water:	85°F
Condenser water flowrate:	3.0 gpm/ton
Air-cooled condenser inlet air:	95°F
Evaporative-cooled condenser inlet air:	75°F wet bulb

The energy efficiency of air conditioners is commonly expressed in terms of the energy efficiency ratio (EER), which is defined as the ratio of the cooling capacity of a unit in Btuh divided by the compressor power consumed in Watts. With water chillers, the energy efficiency is expressed as kilowatts per ton (kW/ton), i.e., kilowatts of compressor power consumed to produce 1 ton of refrigeration. The EER or kW/ton for a given air conditioning refrigeration

cycle depends on 1) the temperature at which the cycle receives heat from the conditioned space (i.e., the evaporator temperature), 2) the temperature at which the cycle rejects heat (i.e., the condenser temperature), and 3) the thermodynamic irreversibilities (i.e., inefficiencies) that are present in the design.

A.1 THEORETICAL PERFORMANCE CAPABILITIES

To evaluate the performance of commercial air conditioning units, it is useful to note the EERs attainable by the referenced refrigeration cycles without thermodynamic inefficiencies, such as the reversed Carnot cycle and the ideal basic vapor compression cycle (often called reversed Rankine cycle). For given heat source and sink temperatures, a thermodynamically reversible cycle, such as the reversed Carnot cycle, will have the highest possible EER. The ideal EER of the basic vapor compression cycle accounts for the irreversibilities resulting from the expansion process at the throttling valve and the heat transfer processes across finite temperature differences.

The ideal Carnot EERs for air-cooled and water-cooled air conditioners operating at ARI inlet conditions are 123 Btuh/W and 369 Btuh/W, respectively; the corresponding basic reversed Rankine cycle EERs using refrigerant R-22 are approximately 100 Btuh/W and 270 Btuh/W, respectively. However, because of the need to dehumidify the conditioned air, the evaporator must operate at lower temperatures than the dew point temperature of the inlet air. Table A.1 gives the EERs of air conditioners with evaporator temperatures ranging from -20°F to 40°F. As the table shows, the EERs of air-cooled and water-cooled units operating at ARI standard conditions are 27.6 Btuh/W and 34.7 Btuh/W, respectively, in order to maintain a 40°F evaporator temperature for dehumidification.

The ARI Directory of Certified Unitary Air-Conditioners, Unitary Air-Source Heat Pumps and Sound-Rated Outdoor Unitary Equipment (January 1 to June 30, 1986) (ARI 1986) lists over 135 air conditioners with approximately 10-ton capacity. The EERs of the listed units range from a low of 7.1 Btuh/W to a high of 10.6 Btuh/W, with an average of 8.26 Btuh/W (including auxiliary

TABLE A.1. EERs for the Ideal Basic Vapor Compression Cycle with Low Temperature Evaporation for Dehumidification (Refrigerant R-22)

Evaporator Temperature (°F)	Condenser Temperature (°F)				
	80	85	95	100	120
-20	11.7	11.1	9.8	9.1	7.2
0	15.9	14.9	13.0	12.0	9.3
20	23.3	21.6	18.1	16.4	12.2
40	38.3	34.7	27.6	24.0	16.6

power).[a] These EERs are considerably lower than the theoretical maximums represented by the reversed Carnot and the ideal basic vapor compression cycle values. The lower EERs are a result of the inefficiencies or irreversibilities present in the design of these commercial units.

For centrifugal water chillers, the Carnot cycle efficiency is 0.29 kW/ton for uniform evaporator and condenser temperatures of 44°F and 85°F, which correspond to ARI standard outlet chilled water and inlet cooling water conditions. The ideal vapor compression cycle efficiency is 0.31 kW/ton and 0.33 kW/ton for R-11 and R-12, respectively, under the same conditions. Although the performance summary for commercial water chillers is not currently available, the current assessment determined that the best obtainable performance is 0.50 kW/ton to 0.55 kW/ton at similar ARI conditions using available technologies (see Chapter 3.0). Therefore, commercial water chiller technology and practice allow a closer approach to the ideal performance, indicating that the inefficiencies present in commercial chiller designs are fewer than those in the air conditioner designs.

A.2 VAPOR COMPRESSION CYCLE LOSSES

According to the Second Law of Thermodynamics, for given condenser and evaporator temperatures, a thermodynamically reversible cycle such as the reversed Carnot cycle will have the highest energy efficiency. The reversed

(a) For 10 ton air conditioners, evaporator fans are typically 1.5 to 2 hp and condenser fans are 1 to 1.5 hp. For a unit with EER of 8.5, for example, auxiliary power would range from 13 to 18% of the total requirement.

Carnot cycle is assumed to operate under ideal conditions such as negligible temperature differences in heat transfer processes and no fluid or mechanical friction. However, in an actual refrigeration cycle, departures from ideal conditions can cause the loss of available energy and require that more energy be consumed to produce the same refrigeration effect.

How this energy is lost is illustrated qualitatively in the temperature-entropy diagram given in Figure A.1. In this diagram, 1'-2-3-4- 4"-1' represent a basic vapor compression refrigeration cycle with an average condensation temperature T_c, an average evaporation temperature T_e, an isenthalpic expansion

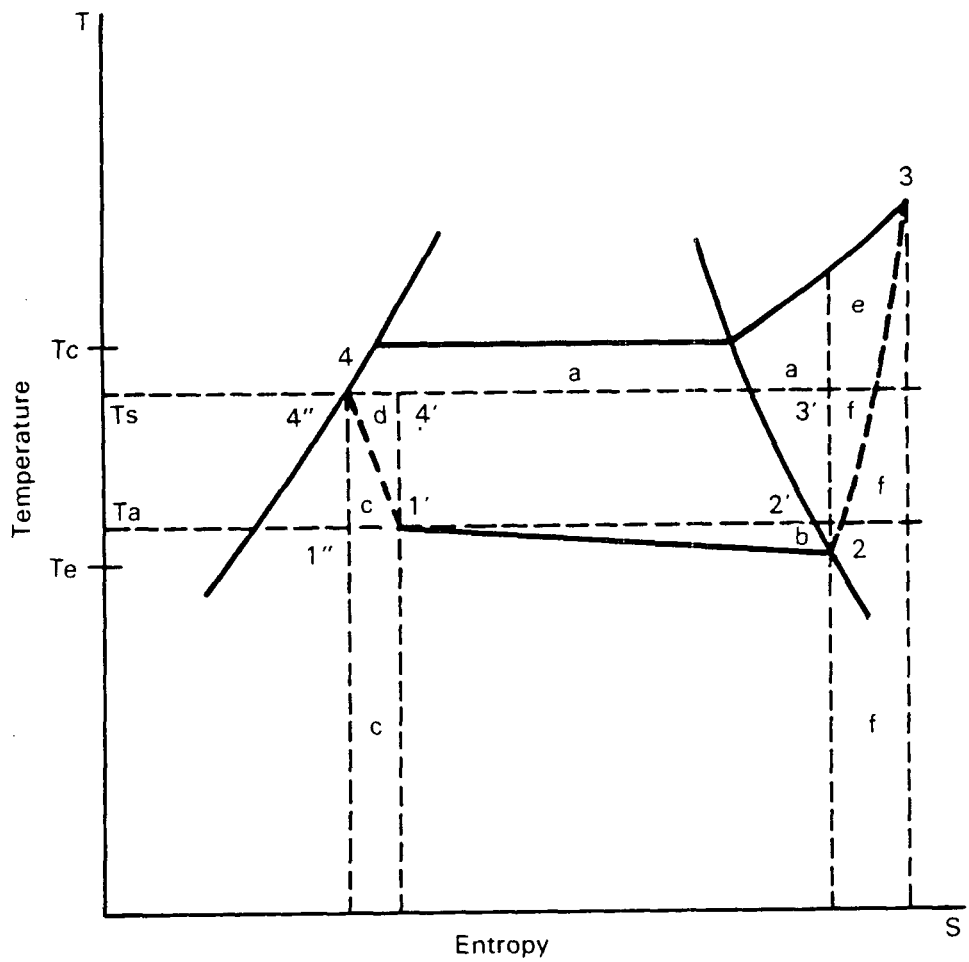

FIGURE A.1. Temperature-Entropy Diagram for the Basic Refrigeration Cycle

4"-1', and a nonisentropic compression 2-3. The temperature of the refrigerated space is T_a and the temperature of the heat sink is T_s. The differences between T_c and T_s and between T_e and T_a occur because of the temperature differential required for heat exchange at the condenser and evaporator, respectively.

The associated ideal Carnot cycle operating with no irreversibilities between the same refrigerated space temperature and sink temperature is shown as 1'-2'-3'-4'. The areas "b" and "a" represent the energy lost or degraded in the evaporator and condenser, respectively, because of heat transfer processes involving finite temperature differences. The areas "c" and "d" represent the energy degraded in the expansion valve which is largely unavoidable in this type of cycle. The areas "e" and "f" represent the energy degraded due to compressor inefficiency and transfer of heat across a finite temperature difference. To improve the energy efficiency of a cycle operating between given temperatures, the degraded energy must be minimized where possible in a viable way.

A.3 COMPRESSOR LOSSES

Reciprocating compressors are used in the packaged and split air conditioners, and centrifugal compressors are used in the centrifugal chillers. Losses in the compression process, represented by the areas "e" and "f" in Figure A.1, can be a major contributor to the overall energy loss in the cycle. The average isentropic efficiency of a reciprocating compressor ranges from 60% to 75%. Of the excess work required over the isentropic work, 0.2 times isentropic work originates from the irreversibilities in the compression process, e.g., turbulence in the inlet and outlet streams, the nonreversible compression and expansion of the gas in the clearance volume and heat transfer. The remainder is due to mechanical inefficiencies (0.8 to 0.85 is a typical range of mechanical efficiency). In addition, the inefficiency of the electrical motor must be superimposed (Jennings and Lewis 1959). On the other hand, the centrifugal compressors have very high efficiencies, up to 95% at rated maximum design conditions. Several common designs which enhance compressor efficiency include:

- large, wide opening valves
- large inlet and outlet gas passages to reduce gas velocities
- close tolerances to reduce the effective clearance volume and improve volumetric efficiency.

Methods which improve the effectiveness of the compression process include:

- intercooling between compression stages for multi-stage units
- external cooling to the compressor
- more elaborate cycles with compound compression.

Cooling and intercooling reduce the specific volume of the gas under compression thereby reducing turbulence losses, increasing the capacity of the compressor, and reducing losses in the condenser due to higher gas superheat. These benefits are partially counteracted by the irreversibilities associated with the heat transfer process.

The compressors used in air conditioners and chillers are in most cases driven by constant-speed electric motors. Constant-speed operation makes regulation of the output difficult under variable load conditions, and some of the methods used to regulate compressors can cause substantial energy losses. Methods commonly used to regulate compressors include:

- intermittent operation
- hot gas bypass
- multiple compressors
- cylinder unloading
- variable displacement or speed
- movable inlet and diffuser guide vanes (for centrifugal compressors).

Intermittent operation of the compressor(s) is the most commonly used method of regulation with small machines. However, frequent starting and stopping of compressors waste energy and are undesirable because of mechanical and electrical reasons. Hot gas bypass, i.e., the bypass of part of the discharged gas directly to the evaporator or to the compressor suction, can be very inefficient because reducing the refrigeration load is not accompanied by a corresponding reduction of power consumed. The remaining methods, i.e., multiple compressors, cylinder unloading, and variable displacement or speed, are more

desirable because they allow the compressor capacity to match or approximate the refrigeration load. With multiple compressors (usually two to four), only the number of compressors needed for part-load requirements are operated. Cylinder unloading is affected by holding a valve open, thereby incapacitating one or more cylinders in a multiple cylinder compressor. Variable displacement compressors, such as screw compressors or constant displacement compressors driven by variable speed motors, can continuously vary the capacity to closely match part-load requirements, thereby providing the maximum economy of operation.

A.4 HEAT TRANSFER LOSSES

According to the Second Law of Thermodynamics, the energy efficiency of a refrigeration system can be improved by reducing the condensation temperature of the refrigeration to the lowest possible value consistent with the application. One way this can be done is to employ an available condenser cooling medium (air or water) with low temperature. However, for fixed condenser cooling air/water conditions, the only way to accomplish this is to design a more effective heat transfer process, thereby minimizing the temperature difference between the condensing refrigerant and the cooling medium.

The energy loss caused by the temperature difference between the condensing refrigerant and the cooling medium is represented by the area marked "a" in Figure A.1. By minimizing this area, the amount of compressor work will also be minimized resulting in a more energy effective process. For a fixed quantity of heat exchanged, the Newton's law of cooling states that to reduce the associated mean temperature difference requires an increase in 1) the surface area across which the transfer occurs, 2) the overall coefficient of heat transfer, or 3) both. A common method to increase the available surface area in condensers is to employ "extended surface" coils which provide large surface heat transfer areas. Air-cooled condensers normally use fins on the air side only, because of the relatively poor surface-to-air heat transfer coefficient compared with the phase-change process occurring on the refrigerant side. Water-cooled condensers may or may not have extended surfaces on the water

side. The coefficient of heat transfer can be improved by introducing turbulence into the air flow through the finned coil. This can be done by using wavy fins, interrupted fins, or fins containing dimples.

Another method to improve energy efficiency of a refrigeration unit is to arrange the condenser so that the condensate is heavily subcooled before it reaches the expansion coil. All condensers provide some subcooling, but this can be increased by cooling further in special coils. With cooler condensate, a corresponding increase occurs in the irreversibility in this process. However, this is offset by the decrease in the irreversibility of the expansion valve; and the net effect is a decrease in the total irreversibility of the system.

Obviously, with given cooling medium temperature, the way to achieve maximum EERs is to maximize condenser surface. If infinite heat transfer surface were possible and coolant flow rates were without limit, significant improvements would be made in energy performance. However, such condensers would be neither economically possible nor practical from space considerations. In actual design practice, the size of the condenser coil is selected on an economic basis. The manufacturer usually uses the largest or most effective condenser coil possible that is consistent with marketing a competitive product.

Many of the comments made regarding condensers are applicable to the evaporator. However, it is desirable to maintain as high an evaporator temperature as possible consistent with the cooling and dehumidification function of the coil. Theoretically, if infinite heat transfer surfaces were available, the evaporator temperature could be kept nearly equal to the cooled space. However, as with the condenser, this is neither economically feasible nor practical. Furthermore, the need to provide a dehumidification capacity requires that surface temperatures be maintained below the airstream dew point temperature. Therefore, it appears that energy efficiency improvement through evaporator coil design is more limited.

A.5 OTHER CYCLE LOSSES

The losses during the irreversible isenthalpic expansion of the refrigerant through the expansion valve are unavoidable in the basic refrigeration cycle. However, through cycle design features, these losses can be minimized. Some of these design features include:

- replacement of the expansion valve with a power recovery turbine
- use of compound compression, with flashed vapor returned to intermediate compressor stages
- compound compression with intercooling
- system compounding.

These design features can fractionally improve the system energy efficiency. However, they also significantly complicate an air conditioner or water chiller and increase its cost. Although some cycle design features are used on larger chillers, they normally cannot be justified for units of the most popular size identified in Chapter 2.0 unless they are means to achieving performance targets for special order applications.

Finally, from ASHRAE tables of refrigerant performance (ASHRAE 1985), another way to influence the energy efficiency of a unit is by the selection of the refrigerant. However, the choice of refrigerant for a particular application is not dictated by energy efficiency alone, but by heat transfer properties, evaporation pressure, density, availability, and toxicity, etc.

APPENDIX B

ESTIMATED ENERGY SAVINGS OF EQUIPMENT R&D

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ESTIMATED ENERGY SAVINGS OF EQUIPMENT R&D

This appendix provides estimates of the potential energy savings which may result from the equipment R&D recommendations made in Chapter 5.0. The five recommended R&D items are:

- heat exchangers
- variable capacity drives
- nonazeotropic refrigeration cycle
- environmentally benign refrigerants and refrigeration cycles
- HVAC systems.

Among them, only the energy saving potentials for the first three equipment-level R&D items are addressed in this section. The objectives in the assessment of environmentally benign refrigerants is the potential substitution of R-11 and R-12 for environmental protection rather than for energy conservation reasons. The HVAC system research covers a wide spectrum of system issues. Not until a specific system research agenda is specified can its energy savings be estimated.

The estimated energy savings are summarized in Table B.1 and are based on estimated EER improvements and market penetration of the new improved technologies. Successful results of the particular research and their implementation in future equipment designs are assumed. Since the performance improvements and market shares are fundamentally determined by many factors, including added equipment first-cost, equipment sizes in relation to loads, climates, building types, construction trends, energy prices, etc., these projections should be considered only as a hypothesis.

Referring to Table B.1, the 28% EER improvements resulting from heat exchanger research assume that the efficiencies of the chillers and the air conditioners will improve from the present average to the best ones currently available. This is possible because of the heat exchanger work which will lead to less expensive heat transfer surfaces. Seasonal cycling degradations of

TABLE B.1. Potential Annual Energy Savings of Equipment R&D

R&D Items	Estimated EER Improvement (%)	Estimated Market Penetration (%)	Annual[a] Energy Savings (quad/year)
heat exchangers	28	30	0.06
variable capacity drives	20	15	0.02
nonazeotropic refrigeration cycles	25	15	0.03

(a) The electricity consumed by the commercial sector for space cooling was 1.1 quad (primary energy) in 1983 (DOE 1985). Approximately 85% of this was consumed by air conditioners and chillers (Hurley et al. 1986) for an equivalent of 0.94 quad.

equipment vary typically from 10% to 30%. Including the efficiency penalty when incorporating the variable speed drives (for example even high performance inverters result in 5% to 10% inefficiency), an average estimated EER improvement of 20% is used. Finally, the nonazeotropic refrigeration cycles offer two performance improvements to the chiller itself, namely steady-state EER improvement and cycling losses reduction. It has been shown that the efficiency improvements of nonazeotropic refrigerants in laboratory tests are considerably lower than those from theoretical calculations because of the lowered heat transfer coefficients compared with the pure fluids (Stoecker and McCarthy 1984). A steady-state EER improvement of 10% and cycling losses reduction of 15% are estimated.

The service lives of the centrifugal chillers and air conditioners average 23 years and 15 years, respectively (ASHRAE 1984). Some time is necessary before the new technologies achieve sizable market penetration. The heat exchanger improvement has been assigned a high market penetration factor of 30%. It is assumed that the new heat exchanger technology will result in minimal equipment first-cost increases. For both the variable-speed drives and nonazeotropic refrigerants, a lower market penetration factor of 15% is assumed considering the expense and complexity added to the system.

Based on the assumed efficiency improvements and on the assumed market penetration factors, it is estimated that heat exchanger research will result in the greatest energy savings or an annual amount of 0.06 quad/year. For the nonazeotropic cycle and variable capacity drives, estimated savings are 0.03 quad/year and 0.02 quad/year, respectively.

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