APPENDIX N: SMALL DUCT, HIGH VELOCITY SYSTEMS

N.1 INTRODUCTION

The U.S. Department of Energy (DOE) issued a final rule on May 23, 2002 (67 FR 36368), increasing the minimum efficiency standards for most residential central air conditioners and heat pumps to 12 SEER/7.4 HSPF. DOE also determined that small duct, high velocity (SDHV) air conditioner and heat pump systems were unlikely to meet the revised efficiency standards adopted for conventional products and created a separate product class for them. DOE concluded that additional analysis on the cost and technical issues related to SDHV air conditioner and heat pump products was needed to determine appropriate minimum efficiency standards for this class of product.

Small-duct, high-velocity (SDHV) systems are a type of air-cooled, split-system central air conditioning and air conditioning/heat pump systems considered by DOE to be "space-constrained," thus requiring special consideration in determining the technical feasibility and economic justification criteria required for a new efficiency standard. However, because of their small share of the unitary air conditioner and heat pump market, opportunity for significant energy savings to the nation resulting from increased minimum efficiency standards for the SDHV product class are limited. Consequently, DOE's approach to conducting the engineering analysis and consumer life-cycle-cost (LCC) analysis was guided by the principle to maximize the use of publically available information and to conduct new analysis only when warranted because of fundamental differences between SDHV and conventional systems.

The engineering and life-cycle-cost analyses presented in this appendix supplement the summary information published in Federal Register announcements of proposed and final rules for SDHV systems. The Engineering Analysis is presented in Section N.2 and the consumer Life-Cycle-Cost analysis is presented in Section N.3 of this appendix. This appendix supersedes two draft reports; "Engineering Analysis", and "Life-Cycle-Cost Analysis", which were prepared for DOE's December 13, 2002, Public Workshop on SDHV standards.

N.2 ENGINEERING ANALYSIS

The engineering analysis develops the relationship between the efficiency and cost of a central air conditioner or heat pump. This relationship serves as the basis for the subsequent life-cycle cost analysis. Determining the cost-efficiency relationship for an SDHV system involves analysis of the options available to manufacturers for increasing the efficiency of a baseline product (i.e., one that just meets the current 10 SEER minimum efficiency standard). The cost-efficiency relationship has to consider a range of system efficiencies, varying from a baseline system to a system that DOE considers to be the maximum technologically feasible, or "Max

Tech" level.

Based on DOE's guidelines to maximize use of existing data and perform analyses only when warranted by differences between SDHV and conventional systems, the steps in the engineering analysis consisted of a) obtaining technical and cost information for current SDHV systems in the marketplace; b) identifying a range of possible system and component changes that increase efficiency; c) selecting a methodology to estimate the effect of component and system changes on SDHV performance; d) developing a spreadsheet model to analyze cost and performance; and e) providing an estimate of a nearly optimal cost versus performance relationship for input to the LCC analysis. The remainder of the engineering analysis follows the outline of the above steps.

N.2.1 Data Sources

Publically-available information useful for SDHV rulemaking includes DOE's analyses from the recent Central Air Conditioner (CAC) rulemaking, procedures used in testing and rating central air conditioning systems, technical information from manufacturers' catalogs, and certified performance data from industry data bases.

Considerable amounts of technical and cost data relevant to the SDHV engineering analysis are available in other chapters and appendices of this Technical Support Document (TSD) for the CAC rulemaking. Of particular importance are Section 4.6.2, of Chapter 4, which provides a general overview of all space constrained residential central air conditioning products, and Appendix L, which provides a detailed quantitative discussion of the analyses and results for the through-the-wall air conditioner and heat pump product class. All of the analyses conducted for the CAC rulemaking are available at DOE's web site: http://www.eere.energy.gov/buildings/appliance_standards/residential/ac_central.html.

Two companies currently produce SDHV systems and certify performance: Unico, Inc., and the SpacePak Division of Mestek Inc. These companies typically out-source production of blower coil units' components, limiting manufacturing operations to assembly, functional testing, packaging and shipping. These manufacturers also out-source production of many components used in the duct distribution system, limiting production to those components that are unique to their designs. More information is available on each manufacturer's web site. Unico's web site is at http://www.UnicoSystem.com Spacepak's web site is at http://www.SpacePak.com.

DOE requires that manufacturers rate the performance of the air conditioning systems they produce. The Air Conditioning and Refrigeration Institute (ARI) publishes certified performance data in an on-line product directory that is useful for establishing the performance levels of currently available SDHV systems. The ARI directory is available at: http://www.ariprimenet.org/ari-prog/direct.nsf?Open.

Because of the proprietary nature of manufactured products, the underlying production cost data used in the current Engineering Analysis were obtained under non-disclosure agreements between each SDHV manufacturer and DOE's contractor, Navigant Consulting Inc. Proprietary technical and cost information were also obtained during previous CAC rulemaking activities by DOE's contractor, Arthur D. Little, Inc., under non-disclosure agreements with several large unitary air conditioner manufacturers. Proprietary information obtained under these non-disclosure agreements are not a part of the public record.

N.2.2 SDHV Systems Description

SDHV systems are used primarily to retrofit older residential type buildings that were constructed without conventional air duct systems that might be used for central air conditioning. In such applications, installation of a SDHV system is often preferable to operating multiple window-air conditioners or installing bulky conventional ductwork and obtrusive air supply registers. Manufacturers claim the special features of SDHV systems reduce disruption during installation and preserve the interior and exterior appearance of traditional buildings.

An SDHV system consists of a conventional outdoor (condensing) unit, produced by other (unitary systems) manufacturers, and a special indoor (blower-coil) unit and air distribution system produced by one of two manufacturers currently specializing in SDHV applications. The blower coil unit and the return air duct and supply plenum are usually installed in an attic, closet or crawl space, and the branch ducts that distribute conditioned air to the various building spaces are concealed within wall or ceiling cavities. All SDHV blower coil units are compactly designed to pass through a 16 inch space between floor joists or wall studs, thereby facilitating retrofit installation. SDHV blower coil units, which are either of modular or single-package configuration, include a direct-expansion refrigerant coil and a blower wheel and motor capable of developing the static pressure requirements of the air distribution system.

The air distribution system consists of a ceiling-mounted return grill, air filter and flexible return duct connected to the blower coil unit's air inlet, and a supply duct system connected to the blower coil unit's air outlet. The supply duct system consists of a rigid supply plenum and a series of small branch ducts. Each branch duct terminates in a special high-velocity air outlet designed to entrain room air and deliver a mixture of supply and room air to the space.

N.2.2.1 Blower Coil Units

Blower coils units used in SDHV cooling-only applications are given the ARI designation "RCU-A-CB". Many indoor units in conventional air conditioning applications are used in conjunction with forced-air heating furnaces. Such indoor units are referred to as cased-coils and have the ARI designation "RCU-A-C". These indoor coils are not equipped with their own blowers, but use the furnace's blower. The furnace blower is often operated at different motor speeds, depending on whether the system is in the cooling or heating mode. Other indoor units in conventional systems are equipped with their own blowers. These units are also referred to as blower coils and given the same ARI designation as SDHV blower coils. Some of these conventional blower coil units have variable speed blower motors. Because variable speed blower motors operate at high efficiency over a range of speeds, power consumption is often less than that for conventional, single-speed systems. Variable speed blower motors are usually found in conventional systems rated at the higher end of the efficiency spectrum.

The primary differences between conventional cased-coil units and SDHV blower coil units are certain design features of the coils, the air flow rates and the static pressures produced by the blowers. SDHV blower coil units include a direct-expansion refrigerant coil sized to meet the unit's design capacity, a thermostatic expansion valve, and a blower wheel and blower motor that are capable of delivering the required volumetric flow rate at the static pressure requirements of the air distribution system. Other component parts of a blower coil unit include an insulated cabinet, a blower control module and a condensate drainage system. SDHV blower coil units are either built-up from separate blower and coil modules as with the Unico design or have the blower and coil in the same module as with the Spacepak design.

Conventional indoor coils are usually available in fixed capacity sizes of 1.5, 2, 2.5, 3, 3.5, 4 and 5 tons, while SDHV blower coil units are available only in three sizes with capacities that vary over the nominal 1.5 to 5 ton range. Similar to conventional indoor units, SDHV blower coils have air flow directions that are either horizontal or vertical.

N.2.2.1.1 SDHV Coil Geometry and Air Flow Rates

SDHV refrigerant coils are constructed with several features that are similar to those in conventional refrigerant coils, including use of copper for the tubes and aluminum for the fins. SDHV tubing includes flow enhancement features and the fins have heat transfer enhancement features. Fin spacing and tube geometry are similar to those of conventional coils. Distinguishing features of SDHV coils are that they are either 4 or 6 rows deep and have flat coil configurations compared with conventional indoor coils, which are either 2 or 3 rows deep and have an "A" or "V" configuration. Conventional indoor coils often have a greater "face area" than that of SDHV coils of the same nominal capacity.

SDHV coils operate at volumetric air flow rates that range from 200 to 280 cubic feet per minute (CFM) per <u>nominal ton</u> (12,000 Btu/h), which is 50 to 70 percent of the nominal 400 CFM per rated ton air flow rate of conventional systems. Because the pairing of an SDHV blower coil with a conventional outdoor unit reduces its rated capacity by 10 to 20 percent, the CFM per <u>actual ton</u> of a SDHV system is in the range of 220 to 310 CFM/ton.

A key performance attribute for an SDHV system is the product of the number of coil rows and the coil face area. This attribute is termed "coil size" or alternately, "core volume", because the number of coil rows, which is a surrogate for coil depth, multiplied by the coil's face area is related to volume (and weight). As will be discussed in Sections N.2.4.1 and N.2.5.3.1, core volume has a major impact both on performance and cost. Indeed, the ratio of core volume for an SDHV coil to the core volume for the matched indoor coil, i.e., the indoor coil with which a specific outdoor unit is paired, has a major impact on SDHV system performance.

N.2.2.2 SDHV Air Distribution System and Blowers

An SDHV air distribution system is similar to that of conventional systems in that the system consists of a return grill and a return duct which delivers room air to the blower coil unit and a system of supply air ducts and registers which delivers conditioned air to various spaces within the building. The differences between the two system types are in the duct sizes and air velocities.

An SDHV supply system includes an insulated, rigid plenum, 7 to 9 inches in diameter, and a series of flexible two-inch diameter branch ducts, which are small enough to be concealed within wall spaces. In contrast, conventional supply air ducts and plenums are considerably larger and are sized to minimize static pressure loss and noise. Consequently, concealment of conventional ducts within finished interior spaces is difficult. Because of their smaller sizes, SDHV supply ducts operate with air velocities in the 2000 to 3000 feet per minute (FPM) range compared with conventional duct systems that operate with air velocities in the 500 to 700 FPM range. Each SDHV branch duct terminates with a special supply outlet that delivers air at a velocity of 1000 FPM or greater. These supply outlets are designed to entrain and mix the high velocity supply air with room air. The supply velocity is much greater than the normal 200 to 300 FPM for conventional supply registers.

Because of the higher velocities in the air distribution systems, SDHV systems experience higher frictional losses and require blowers capable of delivering air at higher static pressure than that for conventional systems. SDHV blowers are rated at 1.2 inches of water static pressure and must be capable of developing even higher static pressures in the field. In comparison, conventional indoor blowers are usually rated at 0.2 inches of water static pressure although they must be capable of developing higher static pressures to ensure proper field operation.

N.2.2.3 Outdoor Units Used With SDHV Systems

This section presents a summary of certified performance ratings that quantify the performance reductions of SDHV systems relative to nominal performance ratings of conventional unitary system air conditioners. SDHV manufacturers pair each model of blower coil with a number of different outdoor (condensing) units. The condensing units are selected based on compatibility of the SDHV blower coil's refrigerant flow requirements to provide a desired cooling capacity and efficiency. The particular condensing unit with which an SDHV blower coil unit is paired can have a large impact on the cooling capacity and efficiency of the system. This is the case even when the selection of condensing unit is made from a group of unitary system condensing units with the same nominal cooling capacity and efficiency.

Table N.2.1 summarizes the cooling capacities and efficiencies for each of the currently available SDHV models as obtained from the 2003 ARI Directory of Certified Equipment.

Table N.2.1 SDHV Systems in 2003 ARI Directory

SDHV Manufacturer	Model No.	No. of ARI Listings	Capacity (Btu/h)		Efficiency (SEER)	
			min	max	min	max
Spacepak	ESP2430+	156	18.4	27.2	10.00	12.45
	ESP3642+	182	28.4	40.0	10.00	12.50
	ESP4860+	123	34.8	57.5	10.00	12.30
Unico	M1218	39	12.8	17.6	10.00	12.00
	M2436+	136	17.6	29.4	10.00	12.00
	M4260+	177	27.0	47.5	10.00	11.00

The range of capacities and efficiencies shown in Table N.2.1 is of interest for this rulemaking because the data illustrate the problems faced by SDHV manufacturers with new efficiency standards. For example, the table shows that Spacepak's Model ESP3642 has certified performance ratings for 182 different condensing units. For this model, SDHV system performance ratings vary significantly; with cooling capacities ranging from 28,400 to 40,000 Btu/h and efficiencies varying from 10.0 to 12.5 SEER. The ARI data base shows that to achieve this range of capacity and efficiency, ESP3642 models are paired with outdoor units that have nominal capacities of 36,000 and 42,000 Btu/h, corresponding to 3 and 3.5 tons, and have nominal efficiencies[a] varying from 12 to 14 SEER. The ARI data base also shows that the particular condensing unit paired with the highest efficiency SDHV system, i.e., the 12.5 SEER, Model 3642, has a range of performance values depending on the particular indoor coil, varying from 33,000 to 36,000 Btu/h capacity and from 13.0 to 15.5 SEER. Thus, to just meet today's minimum efficiency standard of 10 SEER, an ESP3642 SDHV blower coil unit rated at 28,400 Btu/h must often be paired with a nominal 12 SEER, 36,000 Btu/h condensing unit. Table N.2.1 shows a similar reduction in cooling capacity and efficiency for all the other SDHV blower coil models. Depending on the minimum efficiency standard selected in the SDHV rulemaking, a number of the current condensing units will no longer be viable.

N.2.2.4 SDHV System Capacity and Efficiency

The section discusses the technical reasons for the reduction in SDHV capacity and efficiency relative to the nominal performance ratings of conventional unitary systems. As

[a]Nominal efficiency for the condensing unit based on the highest sales volume combination of condensing unit and indoor coil.

previously discussed, SDHV blower coil units operate with reduced air flow rates and higher static pressures compared to conventional air conditioning systems. Because of the reduced air flow rate, the supply air temperature delivered to conditioned spaces is lower (and its humidity ratio is lower) than that of conventional systems. The reduced air flow through the evaporator coil also decreases the temperature and pressure of the refrigerant leaving the evaporator coil and returning to the compressor in the outdoor unit. This in turn decreases the compressor's capacity and efficiency. Also, the higher static pressure loss in the SDHV ducts increases the blower motor's power consumption. The combination of lower compressor efficiency and greater blower power consumption results in reduced SDHV system efficiency and capacity compared to conventional air conditioning systems described in Section N.2.2.3.

Having provided general background information on SDHV systems and mentioned the major differences between SDHV and conventional systems, the following section describes the methodology used to estimate the effects of SDHV component changes (or design options) on SDHV system performance.

N.2.3 System Performance Methodology

Section 2.4, Chapter 2 of the TSD discusses a range of possible methodologies for developing the cost efficiency relationship. For the SDHV rulemaking, DOE selected the design option approach in which individual or combinations of design options are identified that result in increased energy efficiency.

Within the design option approach, there are alternative methods to determine the effects of design options on system performance. DOE determined that physical testing of SDHV systems or performing computer simulations would require more resources than could be justified on the basis of potential energy savings resulting from higher efficiency standards. Consequently, DOE selected a calculation procedure that is currently available for industry's use in rating system performance, and modified the procedure to meet the special characteristics of SDHV systems and the constraints of the rulemaking process.

N.2.3.1 Modeling SDHV System Performance

Estimating system performance is a key factor in the Engineering Analysis. To estimate SDHV system performance, DOE took advantage of an existing rating method that is currently available for use by industry to rate split-system air conditioners.

In rating split-system air conditioners and heat pump systems for efficiency, DOE requires a unitary system manufacturer to test the highest likely sales volume combination of indoor and outdoor units they produce[b]. For combinations of indoor and outdoor units that are not the highest likely sales volume, DOE allows the efficiency rating to be determined by an

[b] 10 CFR part 430, section 430.24(m)(2)

alternative rating method. The alternative rating method, which must be requested by a manufacturer and approved by DOE, must be supported by test data, as specified in the Code of Federal Regulations. One such rating method, developed by the National Institute of Standards and Technology (NIST), is accepted as a standard rating method by DOE, available to any manufacturer to use for rating untested combinations. Because the NIST alternative rating method is in the public domain, is well documented, has been in use for many years and is sufficiently general to cover the desired range of SDHV parameters, DOE selected this method as the basis for determining SDHV system performance.

N. 2.3.2 NIST Rating Method for Split Air Conditioning Systems

The NIST rating method for air conditioning systems is summarized in the U.S. Department of Commerce Report, NISTIR 89-4071[1]. The NIST rating method uses measured performance data for a "matched system", which is a tested combination of specific outdoor and indoor units, and calculates the performance of a "mixed system", which is a non-tested combination of the same outdoor unit and a different indoor unit. The stated purpose of the NIST rating method is to account for the interaction of three components deemed most likely to be changed in a matched system: the indoor evaporator coil, the indoor fan, and the expansion device. The technical basis and methodology used to develop the NIST rating method is described elsewhere[2]. For reference purposes, details of the NIST rating method, including specific equations and constraints, are excerpted from the 1989 NIST report and provided in the Annex to this Appendix.

In the following, the nomenclature and terminology used in the NIST method are retained, and the equations are renumbered. As with the NIST method, the subscript notation 'm' indicates the matched system and the subscript 'x' indicates the mixed or SDHV system. Similar to performance equations 4.1 to 4.4 in NISTIR 89-4071 (Annex), the following equations are used in the SDHV analysis:

$$Q_x(95) = [Q_m + (3.413 \times P_{F,m})] \times F_c^{0.37} - (3.413 \times P_{F,x}) \quad \textbf{(N.2.1)}$$

$$SEER_x = SEER_m \times (Q_x/Q_m)_{82} / (P_x/P_m)_{82} \quad \textbf{(N.2.2)}$$

$$(Q_x/Q_m)_{82} = [(1 + 3.25 \times P_{F,m}/Q_m) \times F_c^{0.35}] - [3.25 \times P_{F,x}/Q_m] \quad \textbf{(N.2.3)}$$

$$(P_x/P_m)_{82} = (0.8 \times F_c^{0.14}) + (0.1 \times P_{F,x}/P_{F,m}) + 0.1 \quad \textbf{(N.2.4)}$$

where:

$Q_x(95)$ = cooling capacity of a mixed or SDHV system (at 95 °F) [Btu/h],
Q_m = cooling capacity of a matched system (at 95 °F) [Btu/h],
$SEER_x$ = efficiency of a mixed or SDHV system [Btu/W-h],
$SEER_m$ = efficiency of a matched system [Btu/W-h],
$(Q_x/Q_m)_{82}$ = ratio of the mixed and matched system cooling capacities at 82 °F ambient

temperature,

$(P_x/P_m)_{82}$ = ratio of total power input to the mixed and the matched system at 82°F ambient temperature,

F_c = indoor coil scaling factor described in the NIST report,

$P_{F,x}$ = indoor unit fan power for the mixed system [W], and

$P_{F,m}$ = indoor unit fan power for the matched system [W].

Note that for the SDHV analysis, Equations N.2.1 and N.2.3 assume that F_e, the expansion device scaling factor, is unity. Equation N.2.2 assumes that F_{TXV}, the thermostatic expansion valve factor, is also unity, so these terms are omitted from the equations for clarity.

Because of their importance in the SDHV Engineering Analysis, it is useful to depict the relationship between the important system performance variables, Q_x and $SEER_x$ in the above equations, as a function of F_c, for various values of mixed and matched blower powers, $P_{F,x}$ and $P_{F,m}$. This is accomplished by casting Equations N.2.1 and N.2.2 into non-dimensional form and by plotting the dependent variables as a function of the indoor coil scaling factor, F_c, for various ratios of SDHV blower power to matched system blower power, $P_{F,x}/P_{F,m}$. To reduce the number of parameters, for the remainder of this report DOE assumed that conventional (matched) system performance is based on an indoor coil air flow rate of 400 CFM per ton of cooling and on indoor blower power of 0.365 W/CFM, the default blower power prescribed by the DOE test procedure.

Figures N.2.1 and N.2.2, respectively, plot the ratio of SDHV capacity to matched system capacity (at 95 F), and SDHV system efficiency to matched system efficiency, as a function of F_c,

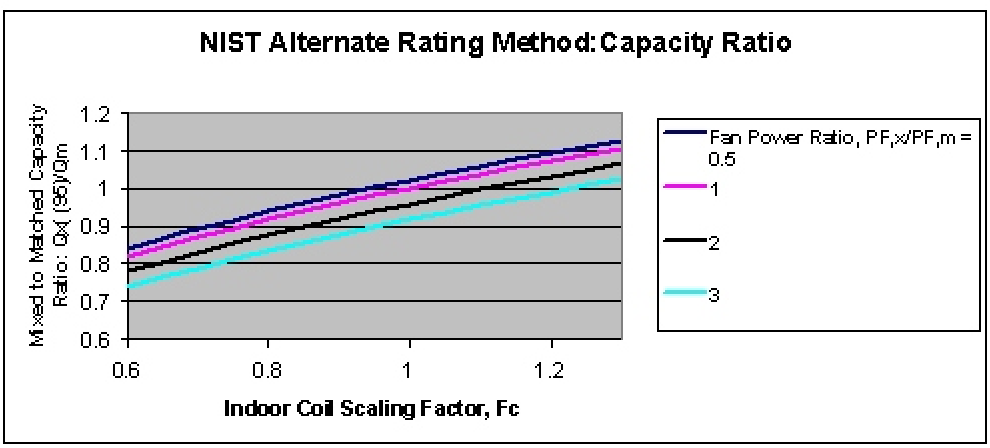

Figure N.2.1 Capacity Ratio vs F_c for $P_{F,M}/CFM_m$ = 0.365 W/CFM, CFM_m/Q_m = 400 CFM/ton

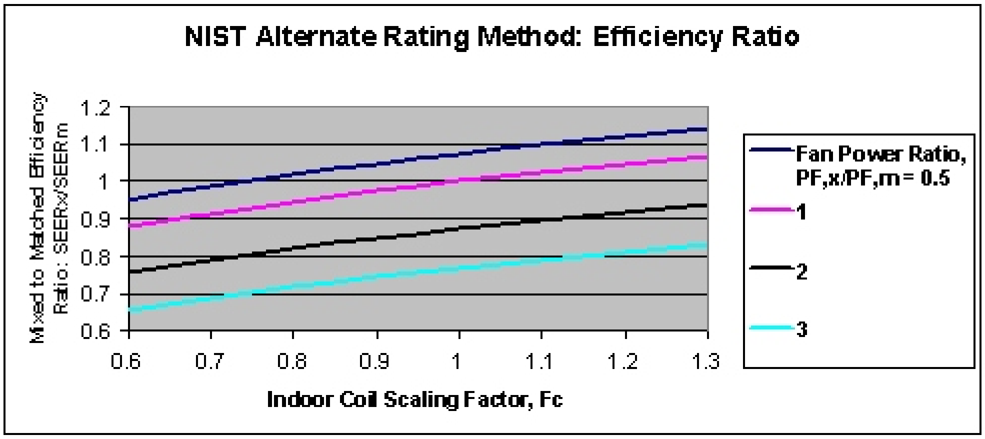

Figure N.2.2 Efficiency Ratio vs Fc for $P_{F,M}/CFM_m$ = 0.365 W/CFM, CFM_m/Q_m = 400 CFM/ton

for different blower power ratios between 0.5 and 3.0. As expected from Equations N.2.1 to N.2.4, Figures N.2.1 and N.2.2 show that the capacity ratio $Q_x(95)/Q_m$ and the efficiency ratio $SEER_x/SEER_m$ are both unity when both the coil factor, F_c, and blower power ratio $P_{F,X}/P_{F,M}$ are unity. Since SDHV systems are likely to have F_c values that are less than 1 and blower power ratios of the order 2 to 3, the two plots show that both capacity and efficiency of an SDHV system is likely to be less than that of conventional matched systems, and that higher SDHV fan power has a somewhat greater impact on reducing system efficiency than on reducing system capacity.

There are two small and probably insignificant departures from the NIST assumptions in the data plotted in Figs. N.2.1 and N.2.2. First, NIST recommended that where air flow rate information for the matched system is not known, a volumetric flow rate of 425 CFM/ton be assumed. However, DOE assumed a slightly lower value of 400 CFM/ton, which is believed to be more representative of current design practice. Second, although NIST recommended limiting the rating procedure to values of F_c between 0.8 and 1.3 because the test data did not include values less than 0.8, experimental data[3] subsequent to NIST's 1989 publication of the rating method suggested the F_c range could be lowered to 0.6 or possibly 0.5 without significantly altering the accuracy of the original NIST correlation. Thus, the plots show a values of F_c that vary over a range between 0.6 and 1.3.

N.2.3.3 NIST Rating Method Specialized for SDHV Rulemaking

This section describes the specific assumptions used in the Engineering Analysis for estimating SDHV system capacity and efficiency. These assumptions specialize the general equations of the NIST method and are coded as equations and data in the EXCEL spreadsheet that has been posted to the DOE web site at: http://www.eere.energy.gov/buildings/appliance_standards/residential/docs/sdhv_calculator_112502.xls.

Additional information on the spreadsheet is provided in Section 2.6 of this appendix. Because of the restrictive nature of these assumptions and the specificity of the data, DOE does not recommend use of the spreadsheet for purposes other than that of SDHV rulemaking.

Distinguishing features for the SDHV methodology include the method for estimating the indoor coil scaling factor, F_c, and the assumed values for indoor blower power, $P_{F,x}$. In the next sections, the assumptions are stated and differences with the NIST method are noted. In section 2.2.2.3, the rationale behind these assumptions is discussed.

N.2.3.3.1 Approximation For Indoor Coil Scaling Factor, F_c

The NIST method states that the coil capacities for the matched and mixed coils must be obtained using a "verified method" at the same set of thermodynamic conditions for the air and refrigerant. Also, air side flow rates must be the same as those specified for the mixed and the matched systems. Further, non-tested coil capacities must be determined using the same methodology, i.e., both coils determined from the same computer simulation program which must be based on fundamental heat transfer phenomenon; or both coils determined from similar coil catalog data, and the methodology must explicitly consider both the coil material properties as well as the geometrical features of the fins and tubing.

In determining values of F_c for the SDHV rulemaking analyses, DOE assumed that the above NIST requirements could be relaxed, thereby avoiding a need to perform detailed computer analyses of conventional coils and SDHV coils to obtain more accurate values for the F_c variable in the NIST equations. DOE assumed that the indoor coil scaling factor can be based on a "rule of thumb", which is frequently used by design engineers and given by:

$$F_c = [(CFM_x/CFM_m) \times (FA \times Rows)_x/(FA \times Rows)_m]^{0.5} \qquad \textbf{(N.2.5)}$$

where:

$(FA \times Rows)_x$ =	SDHV core volume (face area times number of rows),
$(FA \times Rows)_m$ =	matched system core volume,
CFM_x =	SDHV coil volumetric air flow rate (CFM), and
CFM_m =	matched system volumetric air flow rate (CFM).

Figure N.2.3 is a plot of Equation N.2.5 showing the relationship between indoor coil scaling factor, F_c, and core volume ratio, $(FA \times Rows)_x/(FA \times Rows)_m$, for values of air flow ratio, CFM_x/CFM_m between 0.5 and 0.9. The plot shows that F_c increases as the ratio of SDHV core volume to matched core volume increases and as the ratio of SDHV CFM to matched system CFM increases.

The intention of the previous discussion was to show the relationship between the design parameters and variables affecting SDHV system performance in as general a manner as possible and to depict the importance of the indoor coil scaling factor, F_c. This variable could have been eliminated between Figures N.2.1 and N.2.3, and between Figures N.2.2 and N.2.3, which provide plots of capacity ratio and efficiency ratio, respectively, as a function of core volume ratio for various fan power ratios. However, because Equation N.2.5 is an approximation that is believed to be accurate for SDHV systems analysis, a different relationship for F_c could be substituted and new results displayed.

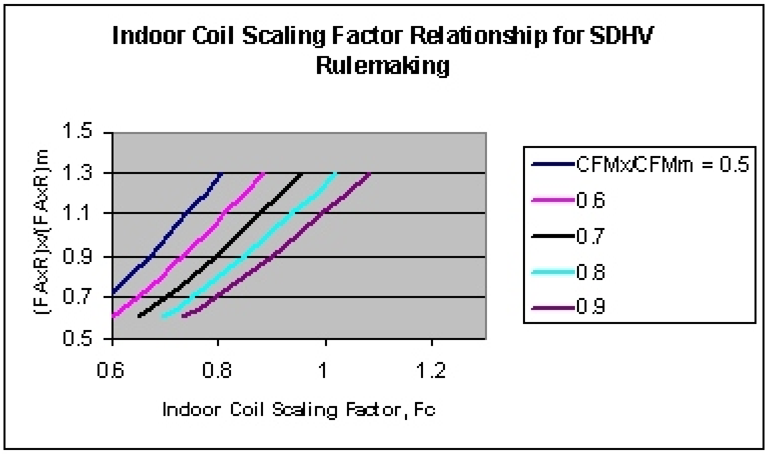

Figure N.2.3 Assumed Relationship for F_C vs Core Volume Ratio and CFM Ratios of SDHV Systems

N.2.3.1.2 Assumptions for Indoor Blower Power, $P_{F,x}$ and $P_{F,M}$

Equations N.2.1 to N.2.4 and Figures N.2.1 and N.2.2 explicitly depict the importance of blower power in the general NIST rating method. Thus, estimating SDHV blower power for this rulemaking is considered important. One approach to estimating blower power uses fundamental thermodynamic principles relating air flow rate and fan total pressure rise and makes assumptions for blower-motor efficiency. Another approach uses measured performance data. DOE's original approach was to use the first method since measured performance data were not available; however, stakeholders raised concerns at the December 13, 2002, workshop regarding DOE's assumed values for blower motor efficiencies. At the same time, an SDHV manufacturer recommended specific values based on ARI certification tests. The two approaches are further discussed below and a comparison made between the two approaches.

N.2.3.1.2.1 Fundamental Principles and Assumed Efficiency Approach

Blowers and their motors convert electrical energy into mechanical work done on an air stream in increasing total pressure and imparting velocity. For blower motors used in heating, ventilation and air conditioning (HVAC) applications, work is proportional to the product of the volumetric air flow rate and the fan total pressure rise. The ratio of the work done on the air stream to the thermal equivalent of the electrical energy input to the motor is the blower-motor's "wire-to-air" efficiency, η. For both the SDHV system and the matched system, the total pressure of the fan is assumed to be the sum of the static pressure losses in the coil and in the supply ducts. Thus, the SDHV system fan power is determined from the matched system fan power by the following equation:

$$P_{F,x} = P_{F,m} [CFM_x \times (ESP_x + CPD_x)]/[CFM_m \times (ESP_m + CPD_m)]\ \eta_m/\eta_x \qquad \textbf{(N.2.6)}$$

where,

ESP_m = the external static pressure imposed on the indoor unit of the matched system prescribed by rating standards,
ESP_x = external static pressure imposed on the indoor unit of the mixed (SDHV) systems,
CPD_m = air side pressure drop of the wet indoor coil for the matched system,
CPD_x = air side pressure drop of the wet indoor coil of the mixed system,
η_x = combined blower and motor efficiency of the SDHV system, and
η_m = combined blower and motor efficiency of the matched system.

Table N.2.2 lists the assumed values and resulting calculations from Equation N.2.6.

Table N.2.2 System Assumptions Specific for SDHV Rulemaking Engineering Analysis

	System	
	matched	**mixed (SDHV)**
Air flow rate (CFM/rated ton)	400	220-350
External static pressure (in. H_2O)	0.15	1.2
Coil air side pressure drop (in. H_2O)	0.3	0.3
Blower/motor efficiency (percent)	45	65
Indoor fan power (W/CFM)	0.365	0.842

As previously discussed, most matched systems are of the "coil-only" type, so performance ratings are based on the test procedure's "default" value (0.365 W/CFM) for blower power. Matched systems are also typically rated with a nominal air flow of 400 CFM/ton. SDHV systems differ because the rating method requires use of the actual SDHV fan power and an air flow rate that can range between 220 to 350 CFM/ton. Therefore, to estimate the SDHV fan power from the known information, an assumption regarding blower-motor efficiencies is needed.

The source of the default 0.365 W/CFM is unknown, but DOE believes it based on data obtained many years ago from residential forced air heating systems. At that time, furnace blowers used either shaded pole motors or split capacitor motors which were less efficient than today's motors. The actual value of default fan power is a matter of some controversy, with some arguing that the value is too high, and others taking the opposite position. SDHV blower motors are based on permanent split capacitor (PSC) motors, which generally have higher efficiency. Absent specific data, DOE assumed efficiency values of 45 percent and 65 percent for matched systems and SDHV systems, respectively.

As shown in Table N.2.2, the resulting indoor blower power per unit volumetric air flow rate calculated using Equation. N.2.6 are:

matched system blower power: 0.365 W/CFM
mixed system blower power: 0.84 W/CFM

N. 2.3.2.1.2 SDHV Manufacturers' Test Data

______One SDHV manufacturer suggested basing the SDHV blower power on the following default values:

matched system blower power: 0.365 W/CFM
mixed system blower power: 0.80 W/CFM

Figure N.2.4 shows a plot of SDHV blower power (W) vs. volumetric air flow rate (CFM) in which the calculated value of 0.84 W/CFM is used. Figure N.2.4 also shows discrete blower power data obtained from ARI certification testing and used with permission of the SDHV manufacturers. Comparing DOE's assumed value and the test data suggests that there is only a small difference between the DOE's original assumption and the test results, consequently, DOE did not believe it necessary to revise the results of the Engineering Analysis and the LCC Analysis for this rulemaking.

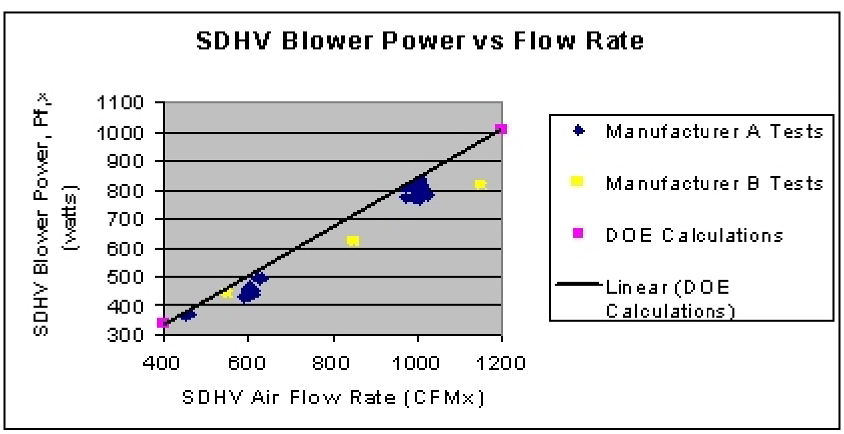

Figure N.2.4 SDHV Blower Power vs Air Flow Rate

N.2.4 SDHV System and Component Changes to Increase Efficiency

An SDHV manufacturer has several options to improve system efficiency. These are categorized as either changes to the design and/or operation of current SDHV blower-coil units, or changes to the set of condensing units currently paired with SDHV blower coil units. In considering changes to the SDHV blower coil unit, DOE includes design changes which a) increase blower wheel or blower motor efficiency; or b) increase coil heat transfer capacity. In the category of operation changes, DOE includes adjusting SDHV air flow. In considering changes to currently available condensing units, DOE includes: a) selecting higher efficiency condensing units; and b) restricting the type of condensing units to those whose matched systems performance ratings are not obtained through use of excessively large indoor coils.

While it is theoretically possible for unitary systems manufacturers to redesign their outdoor units to optimize performance for the reduced indoor air flow rates associated with SDHV systems, DOE believes that outdoor unit redesign is unlikely because of the very small market for SDHV systems.

In the following section DOE discusses each of these options in a qualitative sense and in subsequent sections estimates the costs associated with those options that make sense from a basis of efficiency improvement.

N.2.4.1 Increased Efficiency of Blower Coil Components

Changes to certain SDHV components can increase the efficiency of an SDHV system. These include improved blowers and improved coils. The efficiency of the blower can be increased in several ways, typically by:

1) substituting a higher efficiency, electronically commutated motor (ECM) for the current permanent split capacitor (PSC) motor;

2) substituting a blower wheel with air foil design or with backward-inclined blades for the current blower wheel;

3) increasing the effective heat transfer area of the refrigerant coil by increasing coil face area, by adding tubing rows to the coil, or by improving the heat transfer performance of the refrigerant tubing and/or fin design.

Motor type change could potentially be cost effective in modulating systems, because ECMs are substantially more efficient than baseline permanent split capacitor (PSC) motors at partial load. However, because SDHV systems must deliver air to the terminal outlets at a prescribed velocity (to ensure proper mixing with room air) modulation of the blower is not considered to be a reasonable approach. Further, at full load conditions, the potential efficiency improvement for ECM motors is not much different from that of PSC motors; consequently, the much greater cost of the ECM motors is not justified. Thus, replacing PSC motors with ECM motors is not considered to be an economically viable option and is not discussed further in this analysis.

Blower wheel air foil changes could provide an increase in blower efficiency where space is not an issue, but backward-inclined blowers are considerably larger than conventional forward-inclined blowers. Substitution of larger blower wheels would likely require enlargement of the blower coil housings, which are already space-constrained. Therefore, DOE did not consider blower-wheel design change as a viable option for the purpose of this analysis.

Increasing coil surface area is one of the simplest design options to increase SDHV system efficiency. Since current SDHV designs have either four or six rows of coil, and the tube and fin designs have near optimal features and configurations, DOE believes that adding coil face area is the only viable design option. A larger coil has a beneficial effect on both efficiency and capacity. Since current SDHV models have coils which fill the available cabinet or module space, enlarging the coil face area requires the coil housings to be enlarged as well as adding cabinet insulation materials and larger external condensate drain pans. However, field installation constraints provide a practical limit to indoor cabinet size increases.

The effects of increased coil size on performance can be illustrated by further specializing the NIST equations for SDHV applications and plotting the results. Besides the previous assumptions of matched system air flow rate of 400 CFM/ton and blower power of 0.365 W/CFM, DOE assumed a matched system performance rating of 12 SEER efficiency and 36,000 Btu/h (3 ton) capacity, with corresponding air flow rate of 1200 CFM. Also, the indoor coil size (core volume) of 12 is assumed, which corresponds to a 3 row coil with 4.0 sq. ft of face area. The matched system indoor coil size of 12 is believed to be typical of single-speed, 12 SEER, 3-ton split systems based on a sampling of unitary air conditioning equipment for the CAC rulemaking.

Figures N.2.5 and N.2.6, respectively, show the capacity and efficiency for an SDHV system as a function of SDHV coil size for an air flow rate of 720 CFM and for blower powers of 0.73, 0.80 and 0.84 W/CFM. The plots show a range of SDHV coil sizes from 8 to 20 which span the range of currently available blower coil models. The 720 CFM flow rate assumed in the plots is midway between the nominal 600 to 840 CFM range of current SDHV designs. The values of blower power indicate the range from the average of the manufacturer's test data to the value used in the Engineering Analysis.

Fig. N.2.5 shows the significant affect that coil size (core volume) has on SDHV system capacity, with blower power having a smaller impact. Fig. N.2.6 shows a similar impact of coil size on SDHV efficiency with blower power having a somewhat greater impact on efficiency than on capacity. Of significance is that the plots show that both SDHV capacity and efficiency range from about 5/6ths that of the nominal matched system performance rating at a coil size of 8 to almost the nominal rating at a coil size of 20. These coil sizes represent the range of current

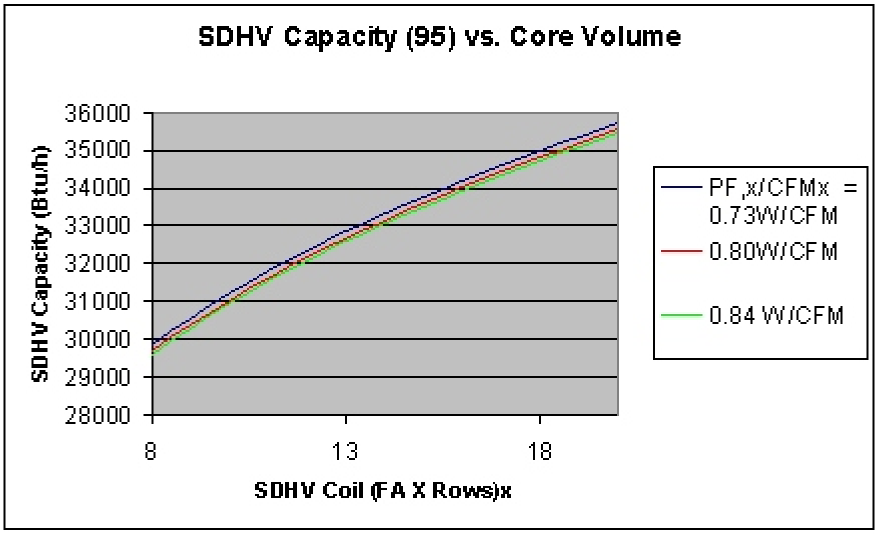

Figure N.2.5 SDHV Capacity vs Core Volume: CFMx = 720, Matched Core Volume = 12

SDHV systems, albeit the coils are used in systems having nominal capacity ratings varying from 1.5 to 5 ton instead of the nominal 2.5 to 3 ton system depicted.

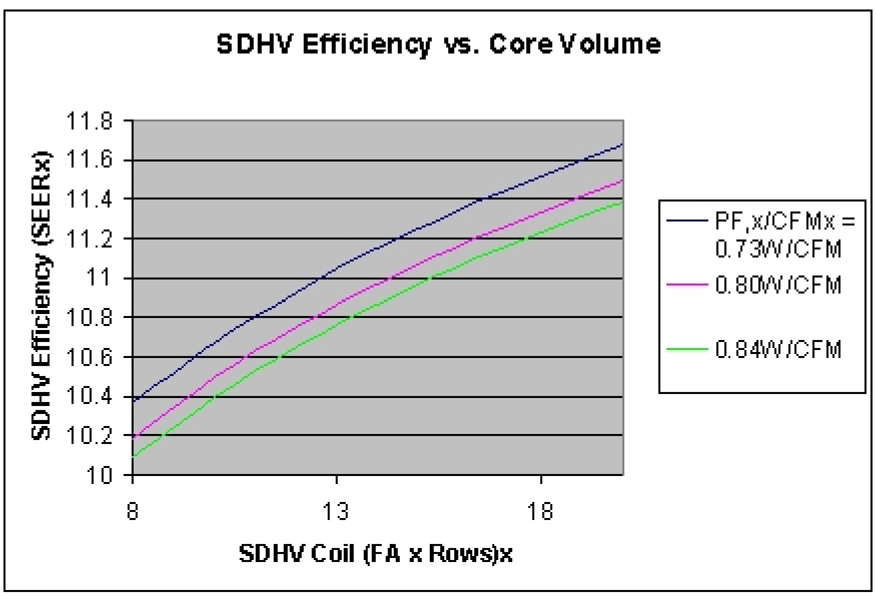

Figure N.2.6 SDHV Efficiency vs Core Volume: CFMx= 720 CFM, Matched Core Volume = 12

N.2.4.2 Reduced Air Flow

As previously mentioned, a nominal 3-ton SDHV system has an indoor airflow rate varying from 600 to 840 CFM. In principle, when the airflow of an SDHV system is decreased, the F_c ratio also decreases. This results in capacity decrease and both compressor power and blower power decrease. Thus, if the airflow rate for a particular system is decreased from 840 CFM to 600 CFM, because the total power decreases at a faster rate than the capacity decreases, the efficiency of the SDHV system <u>increases</u>.

Figures N.2.7 and N.2.8, respectively, show the effects of air flow rate on SDHV system capacity and efficiency for an SDHV coil size of 12.0 (the same size as the matched system coil)

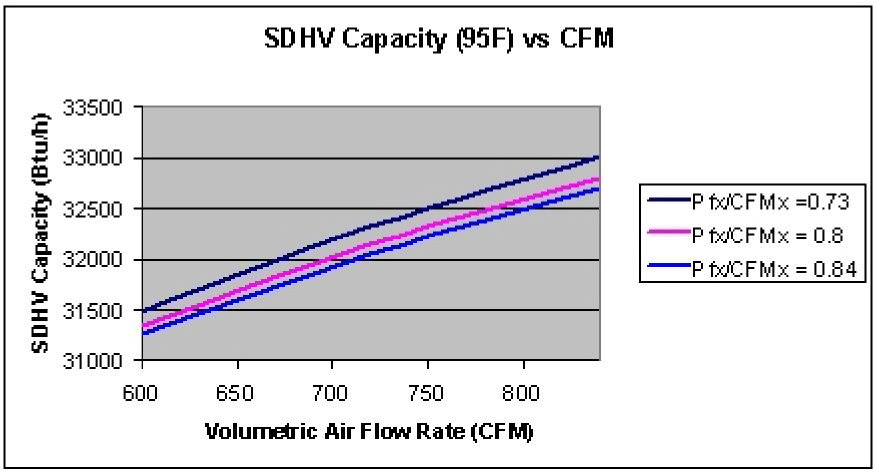

Figure N.2.7 SDHV Capacity vs CFM: SDHV Core Volume = 12 , Matched Core Volume = 12

for values of blower power raging from 0.73 to 0.84 W/CFM. For that system, as the air flow decreases from 840 to 600 CFM, capacity decreases by about 1500 Btu/h or 5 percent and efficiency increases by about 0.2 SEER or 2 percent. DOE does not believe that major changes to indoor air flow rate such as using the range of 600 to 840 CFM is a viable option for SDHV systems. Increasing airflow rate by any substantial amount could result in excessive pressure drop in the small duct distribution system. Decreasing air flow rate could result in poor air distribution at the terminal outlets because the design of these outlets requires the air velocity to be sufficiently high to entrain and mix with room air. Also extending the range in either direction could cause the CFM per rated-ton of the SDHV system to drift outside the mandated limits of 220 to 350 CFM per rated ton.

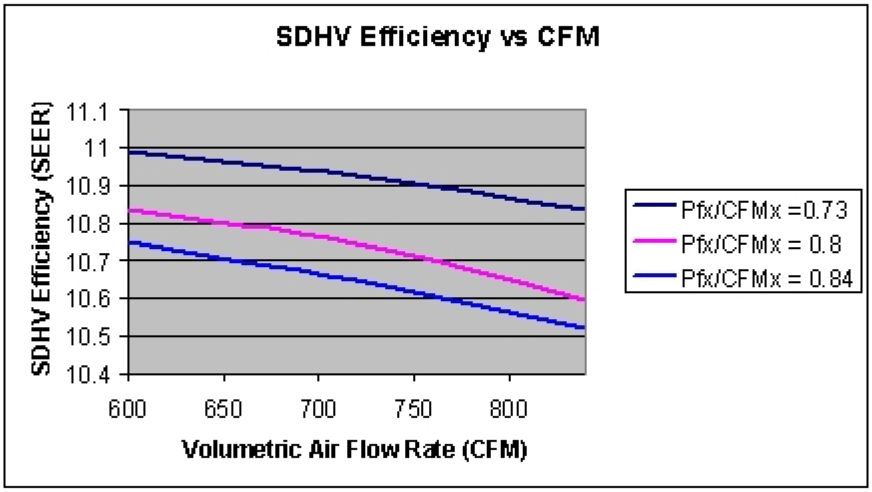

Figure N.2.8 SDHV Efficiency vs CFM: SDHV Core Volume = 12 , Matched Core Volume = 12

N.2.4.3 Selection of Condensing Units

Changes to currently available condensing units can also increase SDHV system efficiency. Two changes are considered possible:

1) Select outdoor units rated at higher efficiency.

2) Select outdoor units at lower rated efficiency, but limit the selection to those units with design features that will not significantly degrade SDHV system efficiency.

In the system performance methodology discussed previously, Equation N.2.2 shows that SDHV efficiency is directly proportional to efficiency of the matched system:

$$SEER_x = SEER_m \times (Q_x/Q_m)_{82} / (P_x/P_m)_{82}$$

Therefore, increasing the efficiency of a matched system by increasing the efficiency of the outdoor unit from, say a 12 SEER to a 13 SEER nominal rating, would increase the SDHV efficiency by 8.33 percent, provided all other terms in Equation N.2.4 remain the same. Assuming that a baseline 10 SEER SDHV system is paired with a condensing unit with a nominal efficiency of 12 SEER, a new SDHV system with a 13-SEER condensing unit will have an efficiency of 10.83. However, the coil capacity ratio and the blower power ratio are not necessarily constant with increased outdoor unit efficiency, so the coil scaling factor, F_c, must also be considered. For example if the greater efficiency of a matched system is achieved simply by use of a larger indoor coil without any changes to the outdoor unit, the improvement in SDHV performance will be less than that suggested by Equation N.2.2.

Another option is to be more selective about the matched system indoor coil size. It is possible to limit the selection of condensing units to those that achieve efficiency ratings with relatively small indoor coils. For a given SDHV coil size, a smaller matched system indoor coil size increases F_c. Avoiding selection of condensing units that achieve efficiency ratings with relatively large indoor coils is helpful in providing SDHV system performance ratings that are closer to the nominal rating of the paired outdoor unit.

One way of examining the effects of matched system coil size on reduction in SDHV performance is to plot the efficiency vs capacity curve for an SDHV system with different matched system coil sizes. Fig. N.2.9 show the effects of matched system coil sizes of 8, 12 and 16 on the efficiency vs. capacity curves of a system with a fixed SDHV coil size of 12, fixed blower power of 0.84 W/CFM, and with CFM varying from 600 to 840 CFM. A measure of SDHV performance degradation is the reduction in capacity from the nominal 36,000 Btu/h and in efficiency from the nominal 12 SEER assumed for the matched system. As shown in Fig. N.2.9, the largest reduction in SDHV performance occurs for a matched system with the largest indoor coil size. These curves illustrate DOE's concern regarding the pairing of certain outdoor units with SDHV blower coil units, in that if the matched unitary system achieved its performance rating by using a relatively large indoor coil, the performance of the SDHV system will experience a greater reduction in capacity and efficiency.

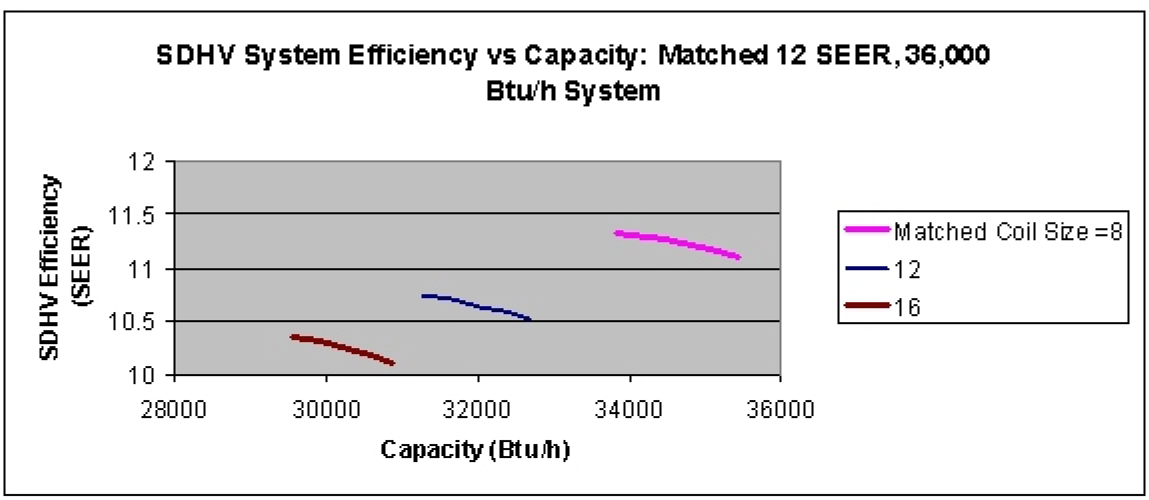

Figure N.2.9 SDHV System Efficiency vs Capacity for SDHV Coil Size = 12 and $P_{F,x}/CFM_x$ = 0.84

However, DOE believes that limiting the selection of condensing units might cause certain distributors to move away from SDHV products. Therefore, this option negatively impacts SDHV manufacturers in terms of marketing and sales opportunities and is not considered further in the analysis.

N.2.4.4 Use of Emerging Technologies

Section 4.5 of Chapter 4 discusses emerging technologies, including variable-speed motor control, advanced compressors, and microchannel heat exchangers considered for conventional air conditioning systems. Of the three emerging technologies, only variable-speed motor controls and microchannel heat exchangers might be appropriate for SDHV systems. However, as discussed in Section N.2.4.1, variable speed motor controls are deemed unsuitable. With regard to microchannel heat exchangers, while the compact nature of this emerging technology could assist SDHV systems in overcoming size constraints for the evaporator coil, DOE does not consider them a viable option for SDHV systems because of condensate removal concerns.

N.2.4.5 Conclusions on Design Options

In consideration of the above, DOE concludes that SDHV manufacturers have only two choices for meeting higher efficiency standards:

1) Increased indoor coil size and
2) Higher efficiency outdoor unit.

N.2.5 Cost Efficiency Relationship

As previously mentioned, proprietary technical and cost information obtained under non-disclosure agreements between the two SDHV manufacturers, and DOE's contractor, Navigant Consulting Inc., provided the information needed for the engineering analysis. Both SDHV manufacturers participated effectively in this process and DOE acknowledges the importance of the manufacturers' efforts.

Each manufacturer provided cost breakdowns for labor and materials for each of the blower coil units in production. In addition, markups at the manufacturer level, the distributor level and the contractor level enabled DOE to aggregate the cost data to the average price paid by a consumer. In this way, DOE has protected the confidentiality of the manufacturers data because neither manufacturers production costs or markups are identified.

The cost efficiency relationship has to be established over a range of system efficiencies, varying from a baseline system to a system identified as Max Tech.

N.2.5.1 Description of Baseline SDHV System

The starting point for the analysis of efficiency standards is the definition of a baseline system, which just meets the current minimum efficiency standard of 10 SEER. An examination of certified performance ratings in the ARI directory indicates that a significant number of SDHV systems just meet the current 10.0 SEER minimum efficiency standard. Rather than arbitrarily selecting any particular SDHV system as a baseline, DOE defined a baseline system assuming the attributes described in Table N.2.3.

Table N.2.3 Physical Characteristics of Baseline SDHV System

SDHV System Features				Matched System Features			
Efficiency (SEER)	Capacity (Btu/h)	Air Flow Rate (CFM)	Core Volume (Face Area (sqft × Rows)	Efficiency (SEER)	Capacity (Btu/h)	Air Flow Rate (CFM)	Core Volume (Face Area (sqft × Rows)
10.0	30,000	800	2.0 x 4	12.0	36,000	1200	4.0 x 3

Indoor blower powers were assumed to be 672 W for the SDHV system and 438 W for the matched system.

N.2.5.2 Max Tech

For rulemaking purposes, DOE must identify a system with the highest efficiency level that is technologically feasible, or "Max Tech". Technical feasibility implies that a system is not only theoretically possible, but is capable of being designed, constructed and operated. A technically feasible system may be quite costly or may exceed constraints such as size or comfort and may, therefore, not be commercially practical.

Section 4.4 of Chapter 4 of the TSD notes that "....we are considering 18 SEER to be the Max Tech level. Some niche products and all coil-only products face a Max Tech that is considerably lower." The section goes on to note that "....only about half of all air conditioner condensers sold are paired with fancoils, with the other half ... paired with cased coils and therefore are not capable of using evaporator blower energy savings or fan modulation to raise their SEER ratings. For these cased coil systems, the Max Tech selection would be substantially less than the 18 SEER identified for fancoil systems. A closer estimate would be approximately 14.5 SEER." Section 4.5.1 discusses variable speed motor controls noting that "Because of practical limitations on equipment efficiency as measured by EER, variable speed (VS) systems dominate the market over 14 SEER, and it is rare to find a system over 13 SEER that does not incorporate a variable speed indoor blower."

For SDHV rulemaking purposes, determination of Max Tech has to consider options available to SDHV manufacturers as well as constraints in conducting the Engineering and LCC analyses; consequently, DOE relied heavily on conclusions in TSD Chapter 4 in its determination of Max Tech. Based on these considerations, DOE concluded that 14 SEER is the nominal efficiency of the most efficient matched system for SDHV selection, although there are single-speed, fixed capacity conventional systems available at 15 SEER.

The major constraint facing SDHV manufacturers is believed to be the 16 inch spacing of attic joists and closet studs which limits the size of SDHV blower coils. Indeed, the largest current coil size, i.e., the Spacepak Model 4860, a six-row coil of 3.25 ft^2 face area is housed in a cabinet that is 14 ½ inches high by 43 inches wide by 30 inches deep. Different Spacepak models maintain the cabinet's 14 ½ inch height and 30 inch depth as cabinet width is varied to accommodate the different coil lengths in each model. Discussions with both SDHV manufacturers lead DOE to conclude that coil air flow distribution uniformity constrains the SDHV manufacturers from simply making the coil longer. Adding length would increase non-uniformity of air flow across the coil, which could negate the potential performance benefits of a larger coil.

Based on the above considerations, DOE determined the Max Tech to be based on the following considerations:

1) Outdoor units with relatively high efficiency ratings, i.e., 15 SEER and greater, often have performance features that are deemed unsuitable to SDHV systems. The most efficient, generally available outdoor units that can be successfully paired with SDHV blower-coils have a 14-SEER rating.

2) Blower coil size is limited by physical constraints. Therefore, SDHV blower coil units cannot have coils larger than the largest coil currently on the market.

DOE defined the Max Tech SDHV system assuming the attributes described in Table N.2.4. This system, which incorporated the previously mentioned design option features, results in an efficiency of 13.4 SEER and a capacity of 34,800 Btu/h. Based on the noted air flow rate, this system just meets the minimum air flow rate of 220 CFM/rated ton of capacity assumed in the SDHV system definition.

Table N.2.4 Physical Characteristics of Max Tech SDHV System

SDHV System Features				Matched System Features			
Efficiency (SEER)	Capacity (Btu/h)	Air Flow Rate (CFM)	Core Volume (Face Area (sqft × Rows)	Efficiency (SEER)	Capacity (Btu/h)	Air Flow Rate (CFM)	Core Volume (Face Area (sqft × Rows)
13.4	34,800	638	3.25 x 6	14.0	36,000	1200	4.0 x 3

Indoor plower powers were assumed to be 538 W for the SDHV system and 438 W for the matched system.

In the public workshop conducted on December 13, 2002, DOE identified the above criteria for Max Tech and requested stakeholder commentary on its rationale. Because no comments were tendered, DOE concluded that the criteria for selecting Max Tech were reasonable.

N.2.5.3 Estimated Cost Impacts

DOE relied on several sources to estimate manufacturing costs for each of the two selected design options: the Residential Central Air-Conditioning System Engineering Analysis, previously mentioned; confidential information provided to DOE's contractor, Navigant Consulting, Inc., by the SDHV manufacturers; and estimates based on industry expertise and "rules of thumb."

N.2.5.3.1 Increased Indoor Coil Size

As coil size increases, more materials and labor are needed and the cost increases. While coil material cost is essentially proportional to the product of coil face area and number of rows (core volume), other blower coil unit components, including cabinet panels, cabinet insulation and the external drain pan provided in all attic installations, also experience cost increases in a similar fashion. The confidential technical and cost data provided by the SDHV manufacturers to DOE's contractor allowed the relationship between manufacturer cost and core volume, i.e., coil's face area × rows for the entire blower coil unit to be estimated. The confidential markups data allowed the estimate of consumer price information to be determined from manufacturer costs. Because of the confidential nature of this data, DOE is not disclosing the manufacturer cost increases for larger coil sizes. In the sensitivity analysis, discussed in section N.2.8, the impact of incremental coil cost on the cost-efficiency curves is analyzed.

N.2.5.3.2 Higher Efficiency Outdoor Unit

To estimate the cost of more efficient outdoor units, DOE relied largely on the analysis performed for the residential air conditioner rulemaking.

First, DOE evaluated the high volume production costs of 12, 13, and 14 SEER outdoor units. As described in Appendix B of the TSD, the manufacturing costs of split-system air conditioners are divided into direct costs (labor and materials) allocated to the outdoor unit and to the indoor unit and costs allocated to overhead. To determine the total cost of the outdoor unit only, DOE added a fraction of the overhead cost to the direct cost of the outdoor unit. The fraction of overhead cost added to the outdoor unit cost was based on the ratio of direct cost attributed to the outdoor unit. For example, if the direct cost of the outdoor unit is 78 percent of the total direct cost (outdoor unit plus indoor unit), the overhead cost for the outdoor unit is also 78 percent of the total overhead cost. The same procedure was followed for 13 and 14 SEER split systems and the results are presented in the Table N. 2.5.

Because these costs were obtained assuming that each standard level evaluated was the minimum allowable efficiency level, the costs do not account for lower production volumes and higher profit margins assumed by DOE when systems exceed minimum efficiency levels. Since the new minimum efficiency level adopted by DOE is 12 SEER, cost data for 13 and 14 SEER systems in Table N.2.5 are not correct and DOE used correction factors to estimate the cost of 13 and 14 SEER outdoor units under a 12 SEER standard from the data in Table N.2.5. DOE's correction factors account for the lower volume and higher markups applied to non-baseline systems and are based on knowledge of the unitary air conditioning industry.

In particular, correction factors of 1.05 and 1.10 were applied to the 13 and 14 SEER condensing units respectively to account for lower production volumes. Also, different gross margins were applied across the efficiency levels, since higher-efficiency units are typically sold at a premium. Gross margins were estimated to be 15, 25 and 35 percent respectively for 12, 13 and 14 SEER condensing units. Average markups were then applied to estimate the consumer costs. The results of this analysis are summarized in Table N.2.6.

The sensitivity analysis discussed in section N.2.8 estimates the impact of different assumptions on the cost-efficiency curves. The cost estimates in Table N..2.6 are judged to be reasonable based on comparison with confidential data provided by the SDHV manufacturers.

Table N.2.5 High Production Volume Cost Allocation for Outdoor Units

		12	13	14
Outdoor Unit - Direct Costs	Coil Materials	$59.31	$69.63	$94.85
	Coil Labor	$4.76	$4.92	$5.40
	Electrical Materials	$31.99	$35.22	$35.17
	Electrical Labor	$1.07	$1.07	$1.07
	Miscellaneous Materials	$5.66	$5.98	$6.30
	Miscellaneous Labor	$1.73	$1.76	$1.78
	Fan Materials	$4.47	$5.11	$4.86
	Fan Labor	$0.25	$0.25	$0.25
	Cabinet Materials	$18.83	$20.55	$21.88
	Cabinet Labor	$2.45	$2.51	$2.55
	Plumbing Materials	$11.32	$13.55	$14.46
	Plumbing Labor	$3.72	$3.99	$4.21
	Compressor Materials	$152.87	$167.25	$167.78
	Compressor Labor	$0.65	$0.65	$0.65
Overhead Costs	Refrigerant Matl	$5.60	$6.11	$8.08
	Refrigerant Labor	$0.36	$0.36	$0.36
	Indirect Labor	$12.74	$12.85	$15.63
	Indirect Material	$11.28	$10.52	$11.60
	Equipment Depreciation	$5.99	$6.59	$7.47
	Building Depreciation	$11.70	$13.59	$15.47
	Maintenance	$3.07	$3.20	$3.41
	Utilities	$3.82	$4.36	$4.95
	Taxes	$4.21	$4.76	$5.29
	Insurance	$3.74	$4.23	$4.70
	Freight-Out	$45.21	$63.91	$62.31
		$406.80	**$462.91**	**$500.49**

Table N.2.6 Estimated Consumer Prices for Outdoor Units Used in SDHV Systems

CONDENSING UNIT EFFICIENCY ----------------->	12	13	14
Production cost for high volume production rates	$406.80	$462.91	$500.49
Adjustment for actual production rates	1.00	1.05	1.10
Adjusted production cost	$406.80	$486.06	$550.54
Gross Margin, Percent	15.00	25.00	35.00
Manufacturer's Markup	1.18	1.33	1.54
Selling Price to Distributor	$478.59	$648.07	$846.98
Distributor Markup	1.27	1.27	1.27
Selling Price to Dealer	$607.81	$823.05	$1,075.67
Dealer Markup	1.30	1.30	1.30
Dealer Selling Price	$790.15	$1,069.97	$1,398.37
Sales Tax	1.07	1.07	1.07
Consumer Cost	**$845.5**	**$1,144.9**	**$1,496.3**

N.2.6 Spreadsheet Model

As previously mentioned, DOE prepared an EXCEL spreadsheet to demonstrate the development of the cost vs. efficiency relationship for the purpose of the SDHV system rulemaking analysis. The assumptions, results and formulas used in the Engineering Analysis, with the exception of data covered by non-disclosure agreement, are available for stakeholders' review in the form of worksheets posted on the DOE web site.[c] The worksheet is designed to permit users to verify assumptions, analyze sensitivities, review methodologies and submit any comments to DOE.

The formulas are based on the NIST rating method described in Section N.2.3. Because of the mathematical form of the NIST equations, which provide explicit relationships for capacity and efficiency as functions of indoor coil scaling factor, F_c, raised to fractional powers that differ in each equation, operations involving unknown values for F_c require an iterative solution. For example, one possible need is to determine the coil size required to provide a system having a fixed value of capacity. The spreadsheet has been programmed to consider such operations and, thus, much useful information can be obtained.

As further described in Section N.2.11, three user-selected computational scenarios are programmed into the spreadsheet. The particular scenario selected depends on the information that is known and what information is desired. For example, the effect on SDHV efficiency of substituting a larger coil for an existing coil, while all other variables remain constant, is that the system capacity also increases. However, the extra capacity may not be needed. In that case, an alternate approach would fix the SDHV capacity (say, at the 30,000 Btu/hr baseline) and let the SDHV airflow rate vary to meet this requirement.

The introductory worksheet of the spreadsheet provides the user with specific instructions for each scenario. DOE used a combination of three different scenarios and a combination of the two design options to develop the cost-efficiency curve that minimized cost. Additional information is contained in the worksheet itself.

N.2.7 Distribution Chain Markups

SDHV systems are typically sold in the replacement market. DOE assumes that the distribution chain for SDHV systems is identical to the distribution chain for conventional products in the replacement market. SDHV manufacturers sell blower coil units and air circulation systems to a distributor (wholesaler). The distributor sells the blower coil units and the air circulation systems to a contractor (dealer) along with an outdoor unit purchased from a unitary systems manufacturer. The contractor sells all of the components to a consumer

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[c] http://www.eere.energy.gov/buildings/appliance_standards/residential/docs/sdhv_calculator_112502.xls

(homeowner), applies the sales tax and installs the system in the residence. Therefore, the total distribution chain markup is the product of four factors: manufacturer markup, distributor markup, contractor markup and sales tax.

Because of the limited market and specialty nature of the product, SDHV manufacturer markups tend to be somewhat greater than markups for conventional systems. Through its non-disclosure agreements, the DOE's contractor obtained confidential information from which to estimate an average SDHV blower coil manufacturer markup. DOE estimated other distribution markups, which appear to be similar to that for conventional product markups. Because of the confidential nature of the data, the manufacturer's markups for SDHV blower coils cannot be discussed.

N.2.8 Sensitivity Analysis

The Process Rule requires the Department to consider uncertainty. The initial cost-efficiency relationship does not portray the uncertainty and variability in the assumptions. Uncertainty arises when the precise model parameters cannot be determined. Variability arises when the precise values can be determined, but when they vary among manufacturers, suppliers, or processes. To quantify the uncertainty and variability in the consumer price estimates, DOE used Crystal Ball Pro™ to run Monte Carlo simulation analyses.

Distributions were defined for nine SDHV system characteristics. All other characteristics were considered to be single-point values, either because of low uncertainty, or because any variation would not have significant impact on the results. Uniform distributions represent the assumption that no single value predominates across the range of possible values. Triangular distributions represent that assumption that a particular value does predominate, and that other values are increasingly less likely the farther they are from that value. Table N.2.7 summarizes the ranges considered in the sensitivity analysis.

Table N.2.7 Variability and/or Uncertainty Ranges for the Sensitivity Analysis Assumptions

Variable	*Min*	*Mean*	*Max*	*Type of Distribution*
Matched Coil Size $(FA \times Rows)_m$	10	12	14	Normal
Blower Efficiency (η_{blower})	50%	65%	80%	Triangular
Indoor Airflow Rate of Matched System $(CFM/Ton)_m$	400	412.5	425	Uniform
Incremental Coil Cost ($)	**CONFIDENTIAL**			
Actual Condensing Unit Production Rate Factor at 13 SEER	1.0	1.05	1.1	Triangular
Actual Condensing Unit Production Rate Factor at 14 SEER	1.05	1.1	1.15	Triangular
Condensing Unit Manufacturer Gross Margin at 13 SEER	20%	25%	30%	Triangular

Condensing Unit Manufacturer Gross Margin at 14 SEER	30%	35%	40%	Triangular
SDHV Distribution Markups	**±20% - CONFIDENTIAL**			

To run a Monte Carlo simulation analysis, Crystal Ball™ selects inputs randomly according to the distributions and tracks the effects on consumer price and system efficiency. Since two final results were tracked, a new Price/SEER parameter that combines the two values was defined. The result is a bell-shaped probability distribution for the Price/SEER value of each sample. Rather than predicting a single consumer price, the distribution describes the likelihood that the actual consumer price is equal to a predicted value. Thus, the uncertainty in the consumer price estimates can be quantified. Figure N.2.10 illustrates a typical Crystal Ball™ output. On average, Price/SEER values were well within 20 percent of the predicted values with a 95 percent confidence. Therefore, the Department applied a 20 percent uncertainty to the cost-efficiency curve. This uncertainty is then considered in the Life-Cycle-Cost analysis described in Section N.3.

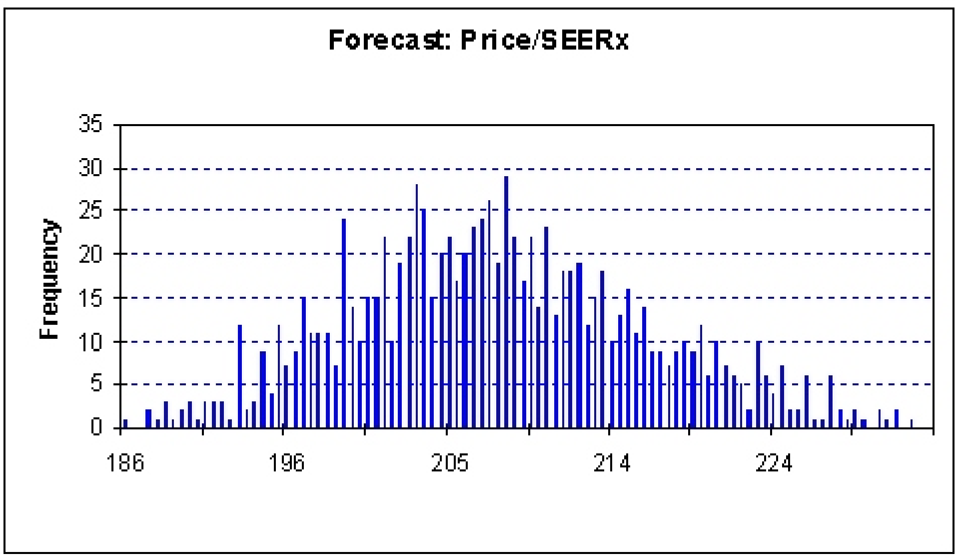

Figure N.2.10 Typical Crystal Ball Output for a SDHV Sample

N.2.9 Heat Pumps

Chapter 4, Section 4.8.1 of the TSD discusses the relationship between cooling efficiency (SEER) and heating efficiency (HSPF) in air conditioning heat pump systems. Based on heat pump performance data from the Spring 1998 ARI Unitary Directory, DOE proposed the following relationship for heat pumps with cooling efficiencies between 10 SEER and 14 SEER:

$$HSPF = 6.8 + 0.3 \times (SEER - 10)$$

In previous proposed and final rules, DOE explained the rationale for accepting this relationship. In the May 23, 2002, final rule, DOE adopted this relationship for split system and single package heat pumps and for split system and single package through-the-wall air conditioners and heat pumps.

As previously discussed, a SDHV blower coil unit can be operated as a heat pump system by substituting a reverse-cycle heat pump outdoor unit for an air conditioning outdoor unit and by providing the indoor unit with a backup heat source. Currently, one SDHV manufacturer offers a heat pump option. In this case, the coil unit has the same face area but a six-row coil is substituted for the four-row coil used in air conditioning applications to provide the refrigerant management function needed for a heat pump.

While DOE has no data on the market size of SDHV heat pumps, it is believed to be small. This is because the common application for SDHV air conditioning systems is in retrofits to existing homes, which are already equipped with a steam or hydronic heating system. The added cost of a heat pump system is usually not justified.

Consequently, DOE assumes that the above SEER vs HSPF relationship adopted for conventional heat pumps and for through-the-wall air conditioners and heat pumps is valid for the purpose of SDHV heat pump standards.

N.2.10 Air-Distribution Systems and Installation Costs

The Engineering Analysis assumes that the cost of the air-distribution system and the installation cost for all the SDHV equipment do not vary with efficiency level. DOE estimated the average cost paid by a consumer for the SDHV air-distribution system materials, exclusive of installation, is $2720. This cost includes distribution markups and sales tax. In addition, the Department is using the same installation cost that was used for the analysis of conventional systems ($1279).

N.2.11 Additional Discussion on SDHV Cooling Capacity

Previous sections discussed how SDHV manufacturers can improve SDHV system efficiency by increasing the blower-coil size and by pairing with more efficient condensing units.

However, the discussions did not address changes to the cooling capacity of the SDHV system. The purpose of this section is to describe how to address variability in SDHV cooling capacity to develop an optimal cost-efficiency relationship. Section N.2.6 describing the EXCEL spreadsheet mentioned the spreadsheet's capability to analyze variable capacity.

As previously mentioned, if an SDHV manufacturer were to simply substitute a larger size coil for an existing smaller coil and keep all other variables constant, both the SDHV system's capacity and efficiency would increase. This is a very basic scenario and is illustrated in the 'Fixed CFM Ratio' (scenario 1) spreadsheet (Section N.2.6). In this situation, a manufacturer would set the airflow rate at a certain value and let the capacity increase as the coil size increases. DOE used this method to estimate performance values in the baseline case by fixing the SDHV airflow rate to 800 CFM (or, alternatively, fixed the CFM ratio at 0.67 which is the ratio of 800 CFM and 1200 CFM).

However, as the size of the coil increases, there is now extra capacity which might not be needed. In this situation, the least-cost approach is to fix the SDHV capacity (say at the 30,000 Btu/hr baseline) and let the SDHV airflow rate vary to meet this requirement. SDHV SEER values are typically higher than the 'Fixed CFM Ratio', because the extra capacity is re-used to increase efficiency. Conversely, costs are lower at a given SEER. This scenario is illustrated in the 'Fixed SDHV Capacity' spreadsheet (scenario 2).

However, by increasing the coil size above a certain value (the threshold varies depending on other assumptions), using the 'Fixed SDHV Capacity' approach, CFM/Ton values fall outside of the definition of SDHV systems (SDHV systems have an airflow rate between 220 and 350 CFM/Ton). Therefore, when reaching the 220 CFM/Ton limit, the CFM/Ton value remains constant and all other variables are free of constraints. This scenario is illustrated in the 'Fixed SDHV CFM/Ton' spreadsheet (scenario 3). The capacity will increase as well, but is accounted for in the fact that the manufacturer has no way to take advantage of this capacity surplus. Using a combination of three different scenarios and a combination of the two design options, an optimal cost-efficiency relationship was developed which is discussed in the next section.

N.2.12 Summary of Results

The cost-efficiency relationships are summarized in Table N.2.8 and are plotted in Figure N.2.11. In Table N.2.8, the results are presented in terms of consumer price increase relative to baseline price at different SDHV efficiency levels. A given SDHV system efficiency can be obtained using a combination of condensing unit efficiency, which varies from 12 to 14 SEER, and SDHV coil size, which varies from 8 to 20. For each SDHV efficiency level, the least expensive combination was selected. Grey areas in Table N.2.8 represent combinations that are not cost effective. For each combination, the minimum, average and maximum consumer prices are reported.

Table N.2.8 Consumer Costs as a Function of SDHV Efficiency and Coil Size

Condenser Unit SEER ----------->	12				13				14			
		Consumer Price ($)				Consumer Price ($)				Consumer Price ($)		
Coil Size (Face Area (ft^2) x Rows)	SEER	Min	Ave	Max	SEER	Min	Ave	Max	SEER	Min	Ave	Max
8	10.00	0	0	0								
8.5	10.15	21.1	26.3	31.6	11.00	260.6	325.8	390.9				
9	10.29	42.2	52.7	63.2	11.15	281.7	352.1	422.5				
9.5	10.41	63.2	79.1	94.9	11.27	302.8	378.5	454.2	12.14	583.9	729.8	875.8
10	10.51	84.3	105.4	126.5	11.39	323.8	404.8	485.8	12.26	605.0	756.2	907.4
10.5	10.59	105.4	131.8	158.1	11.48	344.9	431.2	517.4	12.36	626.0	782.5	939.1
11	10.65	126.5	158.1	189.7	11.54	366.0	457.5	549.0	12.43	647.1	808.9	970.7
11.5	10.71	147.6	184.5	221.3	11.60	387.1	483.9	580.6	12.50	668.2	835.2	1002.3
12	10.77	168.6	210.8	253.0	11.66	408.2	510.2	612.3	12.56	689.3	861.6	1033.9
12.5	10.82	189.7	237.2	284.6	11.72	429.2	536.6	643.9	12.62	710.4	887.9	1065.5
13	10.87	210.8	263.5	316.2	11.78	450.3	562.9	675.5	12.68	731.4	914.3	1097.2
13.5	10.92	231.9	289.9	347.8	11.83	471.4	589.3	707.1	12.74	752.5	940.6	1128.8
14	10.97	253.0	316.2	379.4	11.88	492.5	615.6	738.7	12.80	773.6	967.0	1160.4
14.5					11.93	513.6	642.0	770.4	12.85	794.7	993.3	1192.0
15					11.98	534.6	668.3	802.0	12.90	815.8	1019.7	1223.6
15.5					12.03	555.7	694.7	833.6	12.95	836.8	1046.0	1255.3
16					12.08	576.8	721.0	865.2	13.00	857.9	1072.4	1286.9
16.5									13.05	879.0	1098.7	1318.5
17									13.10	900.1	1125.1	1350.1
17.5									13.14	921.2	1151.4	1381.7
18									13.19	942.2	1177.8	1413.4
18.5									13.23	963.3	1204.1	1445.0
19									13.27	984.4	1230.5	1476.6
19.5									13.32	1005.5	1256.8	1508.2
20									13.36	1026.6	1283.2	1539.8

The data of Table N.2.8 are used to develop the incremental consumer cost-efficiency curve, as shown in Figure N.2.11. This relationship is the input to the consumer LCC analysis in the following sections.

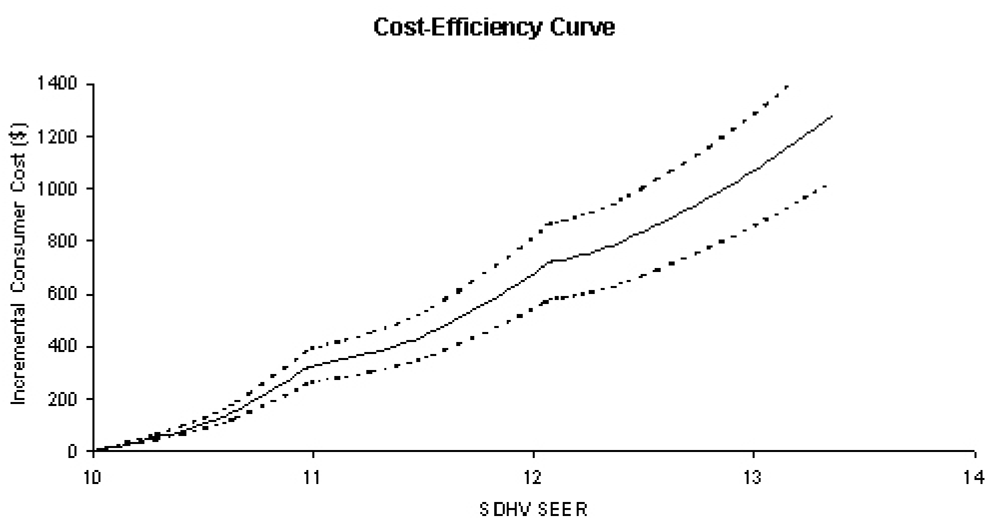

Figure N.2.11 Incremental Cost-Efficiency Relationship for SDHV Systems

N.3 LIFE-CYCLE COST ANALYSIS

The life-cycle cost (LCC) analysis for small duct, high velocity (SDHV) products was conducted with the same spreadsheet model that was used for the LCC analysis of conventional central air conditioner (CAC) product classes. Minor modifications were made to the spreadsheet model to incorporate inputs specific to SDHV products. The LCC model was developed using Microsoft Excel spreadsheets combined with Crystal Ball (a commercially available add-in). The LCC analysis models both the uncertainty and the variability in the model's inputs using Monte Carlo simulation and probability distributions. The LCC results are displayed as distributions of impacts compared to specified baseline conditions. Results are presented later and are based on 10,000 samples per Monte Carlo simulation run. The conventional CAC product LCC analysis and spreadsheet model are detailed in Chapter 5, *Life-Cycle Cost and Payback Period Analysis*, of the TSD.

N.3.1 Inputs to LCC Analysis

The LCC analysis for SDHV products utilizes the same inputs as those used in the LCC analysis conducted for conventional CAC products but with the following exceptions: 1) consumer equipment prices for baseline and standard-level SDHV products are based on a detailed engineering analysis conducted specifically for these products; 2) inclusion of an additional consumer price for the installation of the duct work; 3) electricity price trends are based on projections from the Annual Energy Outlook 2002 (*AEO2002*); and 4) repair costs are based on SDHV consumer equipment prices. All prices used in the SDHV LCC analysis are in 2001 dollars. Prices carried over from the conventional CAC product analysis to the SDHV analysis were converted from 1998 dollars to 2001 dollars using the annual consumer price indices (CPI) for all urban consumers[d] from the Bureau of Labor Statistics[4].

N.3.1.1 Baseline and Standard-Level Consumer Equipment Prices

Table N.3.1 presents the *weighted-average* baseline and standard-level consumer equipment prices for SDHV products based on the optimal cost-efficiency relationship of Section N.2.12. Incremental equipment price increases due to an efficiency increase are actually characterized with normal probability distributions. The consumer equipment prices were determined with markups and sales taxes different from those used for the conventional CAC product analysis. The prices in Table N.3.1 are for the equipment only. The installation prices of the equipment and the duct work are not included.

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[d]The 1998 and 2001 annual CPIs equal 163.0 and 177.1, respectively.

Table N.3.1 Weighted-Average Baseline and Standard-Level SDHV Consumer Equipment Prices (2001$)

SEER	Incremental Price Increase	Total Consumer Equipment Price
10.0	-	$2,067
10.15	$26.35	$2,093
10.29	$52.70	$2,120
10.41	$79.05	$2,146
10.51	$105.40	$2,172
10.59	$131.75	$2,199
10.65	$158.10	$2,225
10.71	$184.45	$2,251
10.77	$210.80	$2,278
10.82	$237.15	$2,304
10.87	$263.50	$2,330
10.92	$289.85	$2,357
10.97	$316.20	$2,383
11.0	$325.76	$2,393
11.5	$431.16	$2,498
12.0	$668.31	$2,735
12.5	$835.25	$2,902
13.0	$1,072.40	$3,139
13.4	$1,283.20	$3,350

N.3.1.2 Duct Total Installed Cost

The total installed cost of the duct work was included in the SDHV LCC analysis. As discussed in Section N.2.10, the total installed cost is the cost to the consumer of the labor and materials needed to install the duct work and was determined to be $2720.

N.3.1.3 Electricity Price Trend

In order to forecast electricity prices, a projected price trend in national average electricity prices is applied to each household's and commercial building's energy prices. The SDHV LCC analysis allows for the following scenarios to be analyzed:

- Constant energy prices
- Energy Information Administration *AEO 2002*[5]: Reference Case; High Economic Growth Case; and Low Economic Growth Case
- Gas Research Institute (GRI) 1998 Baseline Projection[6]

Figure N.3.1 shows the trends for the last four of the above projections. The values in later years (i.e. after 2015 for GRI and after 2020 for *AEO 2002*) are extrapolated from their relative sources. Extrapolation is needed because the sources used do not forecast beyond 2015 for GRI and 2020 for *AEO 2002*. For the *AEO 2002* trends, extrapolations are based on the average growth rate from 2010 to 2020. For the GRI trend, the extrapolation is based on the average growth rate from 2005 to 2015. LCC results presented later are based on the *AEO 2002* Reference Case price trend.

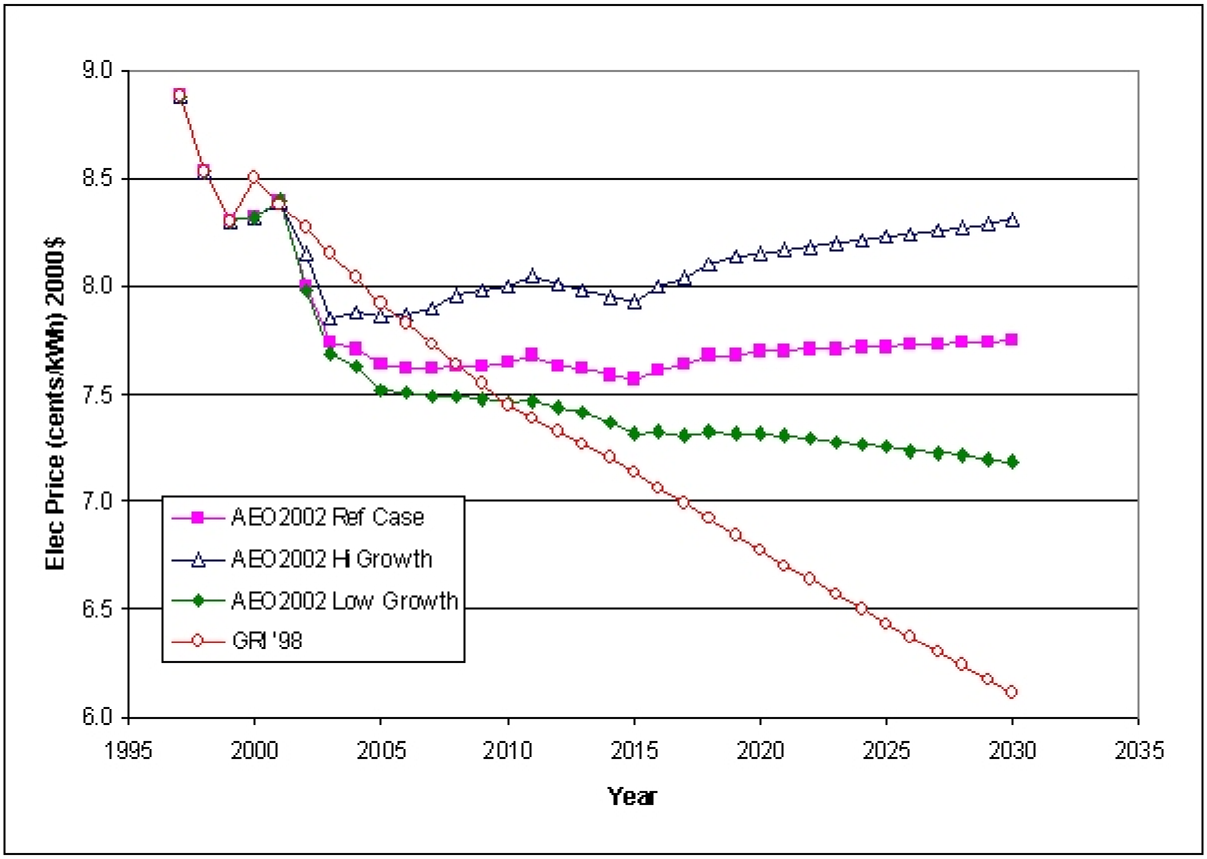

Figure N.3.1 Electricity Price Trends

N.3.1.4 Repair Costs

The repair cost is the cost to the consumer for replacing or repairing components which have failed in the SDHV product. The assumed annualized repair cost for baseline (10 SEER) efficient SDHV products (i.e., the cost the consumer pays annually for repairing equipment) is based on the following expression:

$$RC = \frac{0.5 \cdot EQP}{LIFE}$$

Where,

RC = repair cost,
EQP = equipment price (consumer price for only the equipment), and
LIFE = the average lifetime of the equipment (18.4 years).

If the efficiency of the condensing unit packaged with the SDHV unit has a seasonal energy efficiency ratio (SEER) rating of 13 or less, then the SDHV product is assumed to incur a one percent increase in repair cost over the baseline efficient equipment. If the condensing unit has an SEER rating greater than 13, the assumed annualized repair cost of the SDHV product is based on the equation presented above.

The rationale for assuming essentially flat repair costs for SDHV units coupled with 13 SEER or less condensing units pertains to the level of technology being used in the condensing units. For condensing units with ratings up through 13 SEER, system technology generally does not incorporate sophisticated electronic components which are believed to incur higher repair costs. Increases in condensing unit SEER are generally achieved through more efficient single-speed compressors or more efficient and/or larger heat exchanger coils. Condensing units with efficiencies beyond 13 SEER start to incorporate compressors which are generally believed to be more susceptible to failure.

Table N.3.2 shows the *weighted-average* repair costs by standard level. Since equipment prices are a function of variables which are represented by distributions rather than single point-values, repair costs are actually represented by a distribution of values rather than just the *weighted-average* values shown in Table N.3.2.

Table N.3.2 SDHV Weighted-Average Annualized Repair Costs (2001$)

SDHV SEER	Annualized Repair Cost [a]
10.0 through 10.99	$61
11.0 through 12.2	$62
greater than 13.4	$86

[a] SDHV SEERs of 10 to 10.99 based on 12 SEER condensing units; SDHV SEERs of 11 to12.13 based on 13 SEER condensing units; SDHV SEERs greater than 12.13 SEER based on 14 SEER condensing units.

N.3.1.5 Summary of Conventional CAC Product Weighted-Average Inputs

Table N.3.3 lists the *weighted-average* values for the various LCC inputs (other than consumer equipment prices and repair costs) taken from the conventional CAC product analysis that are being used in the SDHV LCC analysis. Note that although the *weighted-average* values are being provided, for most inputs, probability distributions are used to characterize the input. The sections of the conventional CAC product TSD from which the data are being drawn are also presented in Table N.3.3.

Table N.3.3 Weighted-Average LCC Input Values taken from the Conventional CAC Product Analysis

LCC Input	Weighted-Average Value	Source
Percent of units used in commercial applications [a]	10%	CAC TSD, Section 5.1.4
Equipment Installation Cost (2001$)	$1390	CAC TSD, Section 5.2.2.8
Baseline (10 SEER) Energy Use (kWh/yr)[b]	Residential = 1947 Commercial = 5824 Combined = 2305	CAC TSD, Section 5.2.3.1
Average Electricity Price (cents/kWh)	Residential = 9.46 Commercial = 8.49 Combined = 9.36	CAC TSD, Section 5.2.3.5
Marginal Electricity Price (cents/kWh)	Residential = 9.16 Commercial = 8.64 Combined = 9.11	CAC TSD, Section 5.2.3.6
Maintenance Cost (2001$)	$40	CAC TSD, Section 5.2.3.9
Lifetime (years)	18.4	CAC TSD, Section 5.2.3.10
Compressor Replacement Cost (2001$)	10 - 10.99 SEER = $363[c] 11- 12.13 SEER = $391[d] >12.13 SEER = $498[e]	CAC TSD, Section 5.2.3.10
Discount Rate	5.6%	CAC TSD, Section 5.2.3.11

[a] Value is a single-point value.

[b] Standard-level energy use values are based on the ratio of the baseline efficiency (10 SEER) to the standard-level efficiency. Refer to TSD, Section 5.2.3.2 for more details.

[c] Range of SEERs based on 12 SEER condensing unit. As a result, 12 SEER conventional unit replacement cost is used.

[d] Range of SEERs based on 13 SEER condensing unit. As a result, 13 SEER conventional unit replacement cost is used.

[e] Range of SEERs based on 14 SEER condensing unit. As a result, 18 SEER conventional unit replacement cost is used.

N.3.2 LCC Results

LCC results are presented here for the standard levels specified in Table N.3.1. Results presented here are based on the inputs described above. As has been discussed earlier, the value of most LCC inputs are uncertain and are represented by a distribution of values rather than a single point-value. Thus, the LCC results will also be a distribution of values.

N.3.2.1 Baseline LCC

As stated earlier, the Monte Carlo method of analysis relying on Crystal Ball (i.e., random sampling from distributions) was used to conduct the LCC analysis. The following results presented here are based on 10,000 samples per Monte Carlo run.

The first step in developing LCC results is to develop the baseline LCC. Figure N.3.2 shows the frequency chart for the baseline LCC for SDHV products. A frequency chart shows the distribution of LCCs with its corresponding probability of occurrence. Table N.3.4 summarizes the baseline distributions depicted in Figures N.3.2 by showing the mean, median, minimum, and maximum LCCs.

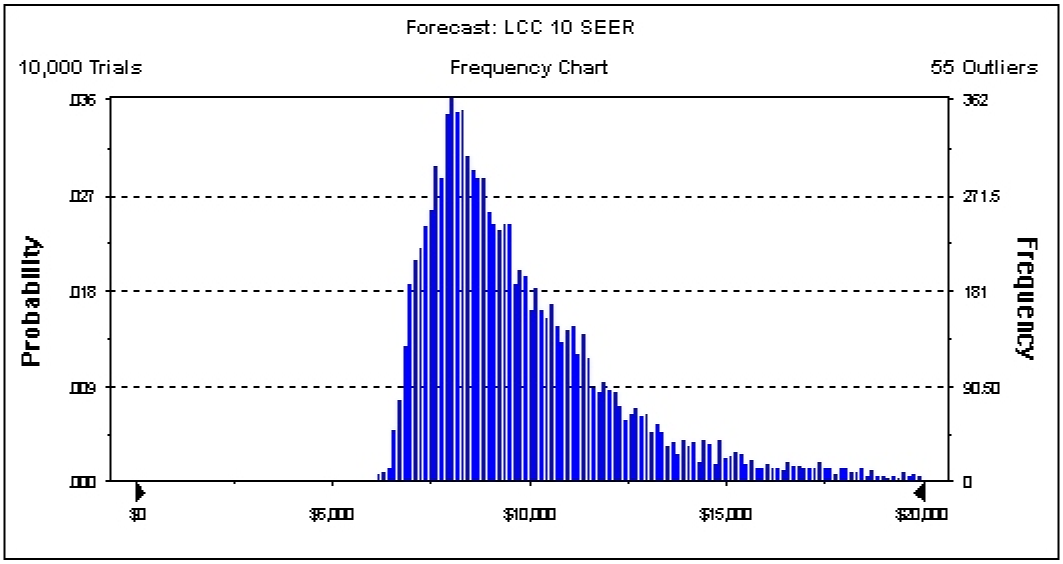

Figure N.3.2 Baseline LCC Distribution

Table N.3.4 Baseline LCC: Mean, Median, Minimum, and Maximum Values (2001$)

	Minimum	Median	Mean	Maximum
Baseline LCC	$6,160	$9,159	$9,843	$26,432

N.3.2.2 Change in LCC

The changes in LCC results are presented as differences in the LCC relative to the baseline central air conditioner or heat pump design. The LCC differences are depicted as a distribution of values. The primary results are presented in two types of charts within Crystal Ball: 1) a *frequency chart* showing the distribution of LCC differences with its corresponding probability of occurrence and 2) a *cumulative chart* showing the cumulative distribution of LCC differences along with the corresponding probability of occurrence. In each chart, the mean LCC difference is provided along with the percent of the population for which the LCC will decrease.

In the explanation below, the two charts depicting the case for an 10.51 SEER efficiency level are used (Figures N.3.3 and N.3.4). In either chart (frequency or cumulative), the mean change (reduction of $4 in the examples here) is shown in a text box next to a vertical line at that value on the x-axis. The phrase “Certainty is 35.41% from -Infinity to $0” means that 35.41

percent of households will have reduced LCC with the increased efficiency level compared to the baseline efficiency level (i.e., 10 SEER).

Figure N.3.3 is an example of a *frequency chart.* The y-axes show the number of households ("Frequency" at right y-axis) and percent of all households ("Probability" at left y-axis). In this example, 10,000 households were examined ("10,000 trials") and almost all the results are displayed (there are 312 "outliers"). The x-axis is the difference in LCC between a baseline efficiency level and a higher efficiency level (in this example, 10.51 SEER). The x-axis begins with negative values on the left, which indicates that standards for those households provide savings (reduced LCC). Reduced LCC occurs when reduced operating expenses more than compensate for increased purchase expense. In Figure N.3.3, going from the baseline efficiency level (10 SEER) to the 10.51 SEER efficiency level provides buildings with an average LCC reduction of $6, and range from reductions of $300 (at the left) to increases of $126 (at the right) depending upon the building. (The minimum and maximum values cannot be read with precision from the graph, but rather, the program provides them in a statistical summary. It should be noted that in this example, reductions in LCC extend to $1285 but, because they are considered outliers, are not shown.)

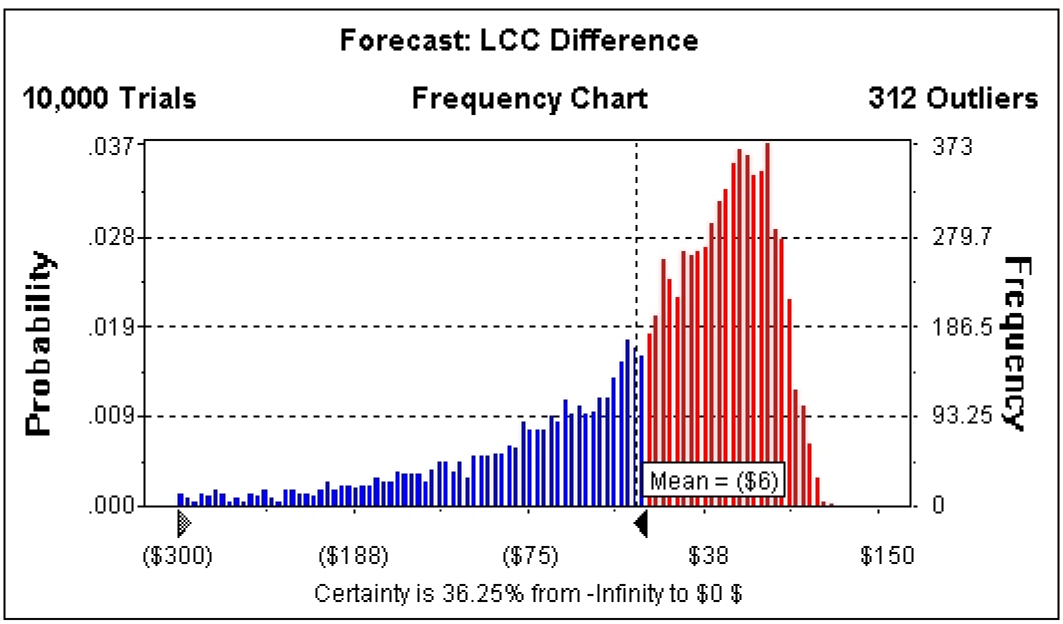

Figure N.3.3 Frequency Chart of Differences for 10.51 SEER

The vertical axis in Figure N.3.4 is the cumulative probability (left axis) or frequency (right axis) that the LCC difference will be less than the value on the horizontal axis. Starting at the left, there is a 0 percent probability that a household will have a reduction in LCC larger than $300 in absolute value (excluding outliers). At the right, there is a 100 percent probability that a

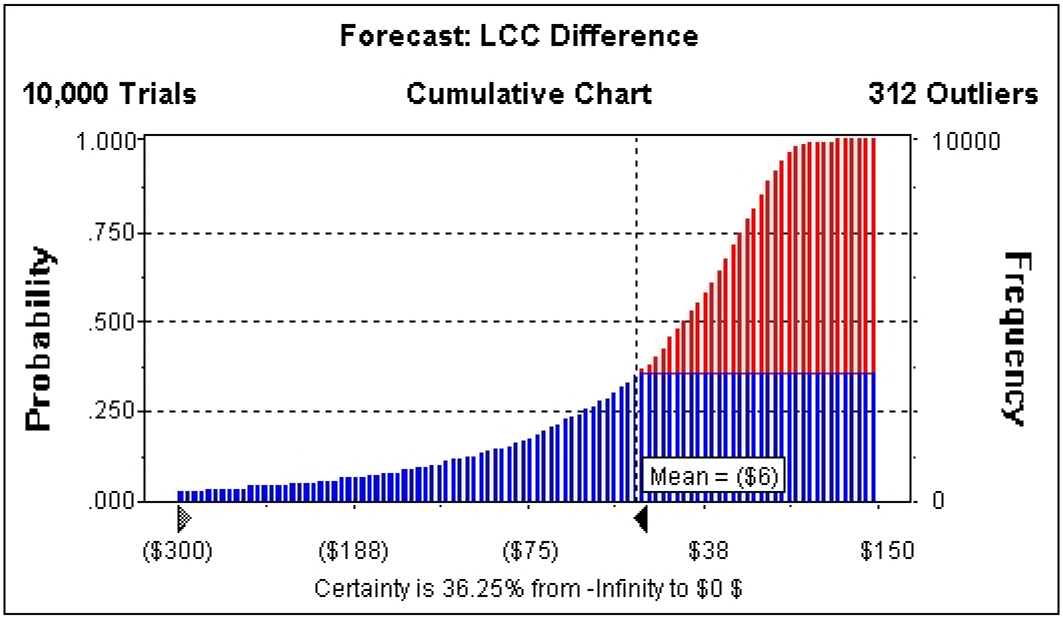

Figure N.3.4 Cumulative Chart of LCC Differences for 10.51 SEER

household will have either a decrease in LCC or an increase in LCC of less than $126.

A summary of the change in LCC from the baseline by percentile groupings (i.e., of the distribution of results) is provided below in Table N.3.5. The mean and the percent of LCCs that are reduced for each standard-level are also shown. As an example of how to interpret the information in Table N.3.5, the 10.51 SEER efficiency level is reviewed. The 10.51 SEER efficiency level in Table 1.3.2 (row 4) shows that the maximum (zero percentile column) change in LCC is a savings of $1285. (Negative values are net savings.) For 90 percent of the cases studied (90th percentile), the change in LCC is a cost of $84 or less. The largest increase in LCC is $126 (100th percentile). The mean change in LCC is a net savings of $6. The last column shows that 36 percent of the sample have reduced LCC (i.e., change in LCC less than or equal to zero).

The results in Table N.3.5 indicate that efficiency levels up through 10.51 SEER achieve mean LCC savings.

Table N.3.5 Summary of LCC Results

SEER	Change in LCC from Baseline Shown by Percentiles of the Distribution of Results (values in 2001$)												Percent of Buildings with reduced LCC
	0%	10%	20%	30%	40%	50%	60%	70%	80%	90%	100%	Mean	
10.15	($331)	($49)	($25)	($12)	($3)	$3	$8	$12	$16	$20	$30	($8)	45%
10.29	($716)	($87)	($43)	($17)	($2)	$9	$18	$26	$33	$40	$61	($11)	42%
10.41	($1,206)	($115)	($53)	($19)	$2	$18	$32	$42	$52	$62	$91	($10)	38%
10.51	($1,285)	($139)	($60)	($17)	$10	$28	$45	$59	$71	$84	$126	($6)	36%
10.59	($1,242)	($149)	($59)	($11)	$21	$43	$61	$78	$93	$108	$158	$4	33%
10.65	($1,152)	($147)	($50)	$5	$37	$62	$82	$99	$115	$132	$191	$20	29%
10.71	($1,827)	($148)	($37)	$21	$56	$81	$103	$121	$139	$157	$219	$35	26%
10.77	($1,494)	($142)	($29)	$34	$71	$100	$121	$141	$161	$182	$256	$51	24%
10.82	($1,622)	($141)	($22)	$42	$87	$119	$143	$164	$184	$206	$282	$64	23%
10.87	($1,712	($133)	($4)	$62	$107	$137	$163	$186	$209	$233	$319	$83	21%
10.92	($1,536	($130)	$5	$75	$122	$157	$185	$209	$232	$256	$348	$98	19%
10.97	($1,928	($127)	$12	$86	$137	$176	$205	$231	$255	$282	$384	$115	19%
11.0	($2,426)	($102)	$43	$123	$170	$204	$233	$257	$280	$307	$406	$141	16%
11.5	($2,407)	($182)	$27	$131	$199	$248	$290	$326	$358	$396	$530	$161	18%
12.0	($3,440)	($150)	$130	$275	$367	$431	$487	$532	$578	$628	$837	$318	14%
12.5	($2,860)	$238	$533	$678	$773	$843	$908	$970	$1,048	$1,149	$1,648	$749	6%
13.0	($4,239)	$330	$681	$854	$968	$1,050	$1,116	$1,183	$1,261	$1,366	$1,843	$921	6%
13.4	($2,598)	$438	$824	$1,023	$1,140	$1,230	$1,309	$1,383	$1,466	$1,572	$2,175	$1,093	5%

N.3.2.3 LCC Results Based on ±2 Percent Threshold

The results in Table N.3.5 show the percent of households with reduced LCC. But considering that the baseline LCC is significantly greater than the LCC differences shown in Table N.3.4, it is more useful to demonstrate which consumers experience significant net LCC savings or costs due to a higher standard-level. We define significant as those consumers experiencing net LCC savings or costs which are greater than 2 percent of the baseline LCC.[e] Since for SDHV products the *weighted-average* baseline LCC is $9843, this translates to an LCC increase or decrease of approximately $197 or an annual expense of approximately $10 over the lifetime of the system.

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[e]The use of the ±2 percent threshold to express LCC results was used in the conventional CAC product analysis. Refer to section 5.2.4.4 of the conventional product CAC TSD.

Table N.3.6 depicts the LCC results based on the above defined 2 percent threshold. The table shows the average LCC values for the baseline level (10 SEER) and the various standard levels analyzed. As presented earlier in Table N.3.5, Table N.3.6 also provides the difference in LCC at each efficiency level relative to the baseline. The differences represent either an LCC savings or an LCC cost increase. In addition, each table shows the subset of consumers (both residential and commercial) at each efficiency level who are impacted in one of three ways: consumers who achieve *significant* net LCC savings (i.e., LCC savings greater than 2 percent of the baseline LCC), consumers who are impacted in an insignificant manner by having either a small reduction or small increase in LCC (i.e., within ±2 percent of the baseline LCC), or consumers who achieve a *significant* net LCC increase (i.e., an LCC increase exceeding 2 percent of the baseline LCC). Accompanying each percentage value is the average LCC savings or increase that corresponds to each subset of consumers. For example, in the case of the 10.51 SEER efficiency level, the percentage of consumers with significant net savings is 6 percent and the corresponding average LCC savings for those consumers is $326. At 10.51 SEER, an overwhelming majority of the consumers (94 percent) are not impacted significantly by the efficiency increase. The efficiency level where a majority of consumers (52 percent) begin to incur significant LCC increases is 11 SEER.

Table N.3.6 Summary of LCC Results based upon ±2 Percent Threshold criterion

SEER	Average LCC	Average LCC (Savings) or Costs (2001$)	Percent of Consumers with Net Savings (>2%)	Average LCC (Savings) or Costs (2001$)	No significant impact (<2%)	Average LCC (Savings) or Costs (2001$)	Net Costs (>2%)	Average LCC (Savings) or Costs (2001$)
10.0	$9,843	-	-	-	-	-	-	-
10.15	$9,835	($8)	0%	($236)	100%	($8)	0%	-
10.29	$9,832	($11)	2%	($272)	98%	($5)	0%	-
10.41	$9,833	($10)	5%	($301)	95%	$5	0%	-
10.51	$9,837	($6)	6%	($326)	94%	$16	0%	-
10.59	$9,847	$4	7%	($342)	93%	$31	0%	-
10.65	$9,863	$20	7%	($352)	93%	$49	0%	-
10.71	$9,878	$35	8%	($370)	92%	$67	0%	$204
10.77	$9,894	$51	7%	($383)	88%	$77	5%	$211
10.82	$9,907	$64	8%	($395)	79%	$82	14%	$219
10.87	$9,926	$83	7%	($408)	67%	$81	25%	$229
10.92	$9,941	$98	8%	($424)	57%	$79	35%	$241
10.97	$9,958	$115	8%	($419)	50%	$74	43%	$255
11.0	$9,984	$141	7%	($444)	41%	$75	52%	$268
11.5	$10,004	$161	10%	($543)	30%	$58	60%	$325
12.0	$10,161	$318	9%	($624)	15%	$35	76%	$483
12.5	$10,592	$749	4%	($632)	5%	$30	91%	$852
13.0	$10,764	$921	4%	($718)	3%	$24	92%	$1,032
13.4	$10,936	$1,093	4%	($708)	3%	$19	93%	$1,202

REFERENCES

1. Domanski, Piotr A., *Rating Procedure for Mixed Air-Source Unitary Air Conditioners and Heat Pumps in the Cooling Mode - Revision 1*, U.S. Dept of Commerce Report, NISTIR 89=4071,May 1989.

2. Domanski, Piotr A., "Rating of Mixed Split Air Conditioners", *Proceedings of 5th Annual Symposium on Improving Energy Efficiency in Hot and Humid Climates*, Houston, TX; September, 1988.

3. Palani, M., O'Neal, D., and Haberi, J., "The Effect of Reduced Evaporator Air Flow on the Performance of a Residential Central Air Conditioner", *Proceedings of the Eighth Symposium on Improving Building Systems in Hot and Humid Climates*, Dallas TX, May, 1992.

4. U.S. Department of Labor, Bureau of Labor Statistics, *Consumer Price Index - All Urban Consumers (Current Series)*. Washington, DC. <www.bls.gov>

5. U.S. Department of Energy-Energy Information Administration, *Annual Energy Outlook 2002*, December, 2001. Washington, DC. Report No. DOE/EIA-0383(2002).

6. Gas Research Institute, *1998 Edition of the GRI Baseline Projection*, 1998. Arlington, VA.

ANNEX : NIST METHOD FOR RATING MIXED AIR CONDITIONING SYSTEMS

To explain the NIST method for rating mixed air conditioning systems, relevant sections have been excerpted from Chapter 4 of the U.S. Department of Commerce Report, "Rating Procedure for Mixed Air-Source Unitary Air Conditioners and Heat Pumps in the Cooling Mode-Revision 1," Domanski, Piotr A., May 1989. (NISTIR 89-4071) retaining all the notation and equation numbering:

4. PROCEDURE FOR RATING MIXED SYSTEM

4.1 Rating Correlations

Mixed system capacity at DOE Test A condition, Q_x shall be calculated using equation 4.1.

$$Q_x = [Q_m + 3.413 \cdot P_{F,m}] \bullet F_c^{0.37} \bullet F_e^{\alpha} - 3.413 \; P_{F,x} \qquad (4.1)$$

Mixed system Seasonal Energy Efficiency Ratio, $SEER_x$, shall be calculated using equations 4.2, 4.3, and 4.4 (derivation of these equations is presented in [2]).

$$SEER_x = SEER_m \; [Q_x/Q_m]_{82} \,/\, [P_x/P_m]_{82} \bullet F_{TXV} \qquad (4.2)$$
$$[Q_x/Q_m]_{82} = [1+ (3.25 \bullet P_{F,m}/Q_m)] \bullet F_c^{0.35} \bullet F_e^{\alpha} - 3.25 \bullet P_{F,x}/Q_m \qquad (4.3)$$
$$[P_x/P_m]_{82} = 0.8 \bullet F_c^{.14} \bullet F_e^{\beta} + 0.1 \bullet P_{F,x} \,/\, P_{F,m} + 0.1 \qquad (4.4)$$

Symbols used in equation 4.1, 4.2, 4.3 and 4.4 are explained below.

Exponents

$\alpha = -0.15$ for $F_e \geq 1$
$\alpha = 0$ for $Fe < 1$
$\beta = 0$ for $F_e \geq 1$
$\beta = -0.2$ for $Fe < 1$

Other Symbols

F_c = indoor coil scaling factor calculated as explained in section 4.2.1

F_e = expansion device scaling factor calculated as explained in section 4.3.1

F_{rxv} = thermostatic expansion valve factor. (Shall be evaluated as shown in Table 1).

$P_{F,m}$ = power input to the indoor fan of a matched system as defined in section 4.4.1 (watt).

$P_{F,x}$ = power input to the indoor fan of the mixed system as defined in section 4.4.2 (watt).

Q_m = capacity of the matched system at Test A as certified by its manufacturer (Btu/h).

Q_x = capacity of a mixed system at Test At as calculated by equation 4.1 (Btu/h).

$[Q_x/Q_m]_{82}$ = ratio of capacities at Test B conditions of the mixed and matched system.
$[P_x/P_m]_{82}$ = ratio of power inputs at Test B conditions of the mixed and matched system.
$SEER_m$ = seasonal energy efficiency ratio of the matched system (Btu/(h- watt) as certified by its manufacturer.
$SEER_x$ = seasonal energy efficiency ratio of the mixed system (Btu/(h-watt) as calculated by equation 4.2.

4.2 Indoor Coil Scaling Factor

4.2.1 Determination of the Indoor Coil Scaling Factor

The indoor coil scaling factor, F_c is defined by the following equation:

$$F_c = Q_{c,x} / Q_{c,m} \qquad (4.5)$$

where:

$Q_{c,x}$ = cooling capacity of a mixed coil at the air mass flow rate specified for the mixed system. The air mass flow rate specified for the mixed system shall satisfy conditions of Appendix M to Subpart B of [6].
$Q_{c,m}$ = cooling capacity of a matched coil at the indoor-air volumetric flow rate, CFM_m (ft^3/min), at which matched system capacity, Q_m, was measured. If CFM_m information is not available, the value for the indoor air volumetric flow rate shall be calculated as follows:

$$CFM_m = Q_m \bullet 425/\ 12000 \ \ (ft^3/min)\ . \qquad (4.6)$$

Capacities of matched and mixed coils shall be obtained using the same verified method (see Section 4.2.3).

Coil capacities shall be obtained at the following conditions:
- inlet air conditions - 80°F dry bulb/67°F wet bulb
- refrigerant saturation temperature at the evaporator outlet - 45°F
- identical refrigerant superheat at the evaporator outlet.

If coil capacities are obtained by means of a catalog or computer simulation, the same catalog or computer simulation shall be used for both coils. Coil material and geometry (e.g., inside tube diameter, tube staggering, fin spacing, fin thickness, fin shape, and number of tube rows) shall be accounted for by the method used. That is, the methodology used must have these parameters as independent variables.

4.2.2 Restrictions

The acceptable range of values for the indoor coil scaling factor, Fc, is from 0.8 to 1.3. This rating procedure shall not be used if the indoor coil scaling factor is smaller than 0.8. If the ratio $Q_{c,x}/Q_{c,m}$ results in a value greater than 1.3, the value of the indoor coil scaling factor, Fc, shall be 1.3.

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4.4 Power Input to the Indoor Fan

4.4.1 Power Input to the Indoor Fan of the Matched System

Power input to the indoor fan, $P_{f.m}$ shall be measured in accordance with Appendix M to Subpart B of [6], at the indoor-air volumetric flow rate, CFM_m at which capacity of the matched system, Q_m, was measured.

If CFM_m information is not available, the value for the indoor-air volumetric flow rate shall be calculated by equation (4.6). If the indoor fan is not supplied with the system, $P_{f.m}$ shall be evaluated by the equation:

$$P_{f.m} = 0.365 \bullet CFM_m \qquad (4.12)$$

where CFM_m (ft^3/min) is a volumetric flow of air through the matched indoor coil at which system capacity, Q_m, was measured.

4.4.2 Power Input to the Indoor Fan of the Mixed System

Power input to the indoor fan, $P_{f,x}$ shall be measured in accordance with Appendix M to Subpart B of [6], at the indoor volumetric air flow rate, CFM_x, at which capacity of the mixed system, Qx, is evaluated.

If the indoor fan is not supplied with the system, $P_{f,x}$ shall be evaluated by the equation:

$$P_{f,x} = 0.365 \bullet CFM_x \qquad (4.13)$$

where CFM_x (ft^3/min) is the volumetric flow of air through the mixed indoor coil at which the capacity of the mixed system, Q_x, is to be evaluated."