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Multi-Variable Optimization of Electrically-Driven Vehicle Air Conditioning Systems Using Transient Performance Analysis

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ABSTRACT

The National Renewable Energy Laboratory (NREL) and U.S. Department of Energy (DOE) are interested in developing more efficient vehicle air conditioning (A/C) systems to reduce fuel consumption in advanced vehicle designs. Vehicle A/C systems utilizing electrically-driven compressors are one possible system design approach to increasing A/C system performance over various drive cycle conditions. NREL's transient A/C system model was used to perform multi-variable design optimization of electrically-driven compressor A/C systems, in which five to seven system design variables were simultaneously optimized to maximize A/C system performance. Design optimization results demonstrate that significant improvements in system COP are possible, particularly system COP > 3, in a properly optimized system design with dynamically-controlled operation. System optimization analyses investigated dynamic A/C system design strategies employing dual-compressor-speeds in electrically-driven systems to evaluate their effects on system performance. A system optimization methodology was developed which can systematically quantify impacts on A/C system design and performance resulting from varying degrees of design influence being given to widely different design objectives. The technique is based upon formulating optimization objective functions from linear combinations of critical design performance parameters that characterize independent design goals. It was demonstrated here by giving varying degrees of design influence to maximizing system COP and maximizing evaporator cooling capacity over SC03 and US06 drive cycles.
KEYWORDS: System Optimization Air Conditioning Transient Performance

NOMENCLATURE

English
A - Expansion Device Flow Area [cm^2]
D – Diameter [cm]
COP – System Coefficient of Performance
f_{obj} - Objective Function
N_c – COP Normalization factor
N_q – Q_c Normalization factor
P(t) – Transient Power [W]
p(t) - Transient Pressure Profile [Pa]
$\overline{P}$ - Average A/C System Power t_i to t_f [W]
$Q_c(t)$ – Transient Evaporator Load [W]
$\overline{Q}_c$ - Average Evaporator Thermal Load Over t_i to t_f [W]
S_{com} – Compressor Speed [rpm]
t – Time [seconds]
T - Temperature [°C or K]
V_{dis} – Compressor Displacement [cm^3]

Greek
α - COP influence coefficient
β - Q_c influence coefficient
η - energy conversion efficiency

Subscripts
ambient – ambient temperature
com - compressor
cond - condenser tube
f - final
eng – engine
exp - expansion device

gen –	generator/alternator	mot –	electric motor
i –	initial	tot –	total system power (compressor + blower)
h -	high pressure side of A/C loop	trans -	transfer line
l -	low pressure side of A/C loop		

1 INTRODUCTION

NREL and DOE develop innovative transportation technologies and systems that decrease vehicle fuel consumption and emissions across the nation, thereby reducing the nation's reliance on foreign oil consumption. Vehicle air conditioning (A/C) systems represent the major auxiliary load on the engines of light-duty passenger vehicles, sport-utility vehicles (SUV), and heavy-duty vehicles and have a dramatic effect on fuel consumption and exhaust emissions in conventional vehicles and hybrid electric vehicles (HEV). Recent studies [1] have shown that, during the SC03 drive cycle, the average impact of the A/C system over a range of light-duty vehicles was to increase 1) fuel consumption by 28%, 2) carbon monoxide emissions by 71%, 3) nitrogen oxide emissions by 81%, and 4) non-methane hydrocarbons by 30%. Recent tests on hybrid electric vehicles (i.e., Toyota Prius / Honda Insight) at NREL [2] have shown that HEV fuel economy decreases by 30%-35% when the A/C operates. The A/C system experiences transient conditions throughout standard drive cycles and during typical city/highway driving patterns around the country. In particular, the evaporator load, compressor speed, refrigerant flow rate, and heat exchanger airflow rates can be variable. Knowledge of transient A/C system behavior is critical to understanding A/C performance requirements, optimizing the A/C system design, and minimizing its effects on vehicle fuel consumption and emissions throughout a drive cycle.

There has recently been increased attention and research into understanding various aspects of vehicle A/C system transient behavior [3-7]. In order to more completely understand transient A/C system performance and its impact on vehicle fuel consumption and emissions, NREL has developed a transient A/C model within the SINDA/FLUNT analysis software environment and has integrated it with the ADVISOR vehicle systems analysis software [8,9]. This transient model captures all the relevant physics of transient A/C system performance, including two-phase flow effects in the evaporator and condenser, system mass effects, air side heat transfer on the condenser/evaporator, vehicle speed effects, temperature-dependent properties, and integration with a simplified cabin thermal model. Integration of the transient A/C system model into ADVISOR represents a subset of NREL's Digital Functional Vehicle project that intends to virtually co-simulate the entire vehicle design process. DFV creates a virtual vehicle design environment that can shorten the vehicle design cycle times, reduce the number of required test prototypes, and produce more optimized vehicle designs. SINDA/FLUINT analysis software and ADVISOR vehicle system analysis software employ built-in optimization capabilities that are used to optimize the vehicle A/C system within the overall vehicle design optimization process. The transient A/C model has been used, along with multi-variable optimization techniques, to optimize vehicle A/C system designs to reduce fuel consumption and exhaust emissions over the various federal drive cycles [9,10]. In particular, this work has now expanded into optimizing an electrically-driven compressor (EDC) A/C system using multi-variable optimization techniques to quantify the potential A/C system and vehicle-level benefits over SC03 and US06 drive cycles.

2 TRANSIENT AIR CONDITIONING SYSTEM MODEL

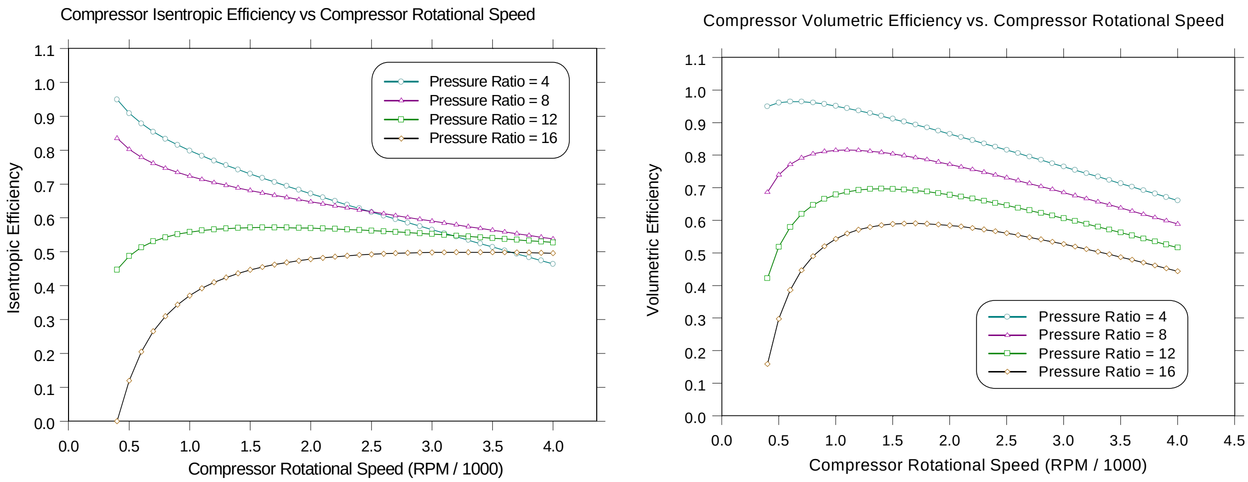
NREL's transient A/C model is a fundamental physics approach to transient A/C system performance analysis and includes dynamic two-phase-flow analysis in the condenser and evaporator. This one-dimensional, thermal-hydraulic model contains generic component sub-models for the fixed-displacement compressor, condenser, evaporator and expansion device, and generic representations for the system piping network and simulation of the system operational control strategy. It has been described in detail by Cullimore and Hendricks [8] and Hendricks [9,10]. The model uses fixed-displacement compressor sub-models that have characteristic isentropic efficiency and volumetric efficiency curves shown in Figure 1. These compressor curves are similar to, although not exact duplicates of, compressor efficiency curves of standard industry air conditioning compressors. Compressor sub-models do include high- pressure ratio regimes that most standard compressors do not operate in under normal conditions, but the sub-models are intended to portray how these compressors would operate under such extreme pressure ratios. The working fluid is R-134a, but could easily be any air conditioning refrigerant including carbon dioxide (CO_2). The fundamental physics approach allows us to truly optimize the A/C system performance without restrictions associated with specific supplier components.

References [8, 9, 10] discuss the A/C model configuration, system transient variables, dynamic flow conditions throughout the A/C loop, initial system performance results, and initial system optimization results. Some new capabilities were added since those documents were published. Specifically, evaporator blower energy consumption has been added to system energy calculations and a new cabin air flow-through sub-model has been added to the cabin thermal/fluid model. The cabin thermal/fluid model now has an option to use either a cabin air re-circulation sub-model or a cabin air flow-through model incorporating body leakage effects. These additions allow us to analyze more A/C system performance conditions in evaluating potential fuel economy improvements and emissions reductions.

NREL has used the transient A/C system model to investigate electric-driven A/C systems to identify and quantify potential A/C system performance improvements and their impact on vehicle fuel economy. Initial multi-variable system optimization analyses focused on simultaneously optimizing fixed-compressor displacement, expansion device diameter, condenser tube diameter, and transfer line diameter to maximize system COP over SC03 and US06 drive cycles [8,9]. Optimization results demonstrated that an A/C system optimized with respect to these four design parameters yields significantly higher system COPs over the SC03 and US06 drive cycles. The dynamic flow conditions (i.e., pressure, temperature, and quality) around the A/C loop were also very important, particularly the two-phase flow conditions in the condenser and evaporator, in optimizing system COP. Optimum pressure profiles and optimum flow quality profiles in time were identified and associated with optimum system COP. Optimum time-dependent (i.e., dynamic) pressure profiles were generally found that minimized the pressure spikes and variations throughout a given drive cycle [10]. The relationship between compressor displacement and expansion device diameter is particularly important in achieving higher system COP. First-order mass continuity analysis implies a relationship between the pressure ratio across the system compressor and the (compressor-displacement / expansion-area) ratio. Therefore, optimum time-dependent pressure profiles that maximize system COP imply that there also exists an optimum ratio between compressor displacement and expansion device flow area that maximizes system COP, which is in turn related to the transient system pressure ratio:

$$\left(\frac{V_{dis}}{A_{\exp}}\right)_{opt} \propto g\left[\left(p_{h,opt}(t)/p_l(t)\right)\right] \qquad [1]$$

where g is a generalized, analytically- or experimentally-determined function [10]. Optimum flow quality profiles suggest that optimum heat transfer and fluid flow conditions must exist in the condenser and evaporator to maximize system COP. These initial system optimization results strongly suggested that A/C system designs employing electric-driven and variable displacement compressors, and variable orifice valves could dramatically increase system performance.



(a) (b)

Figure 1 – Compressor Isentropic Efficiency and Volumetric Efficiency Models

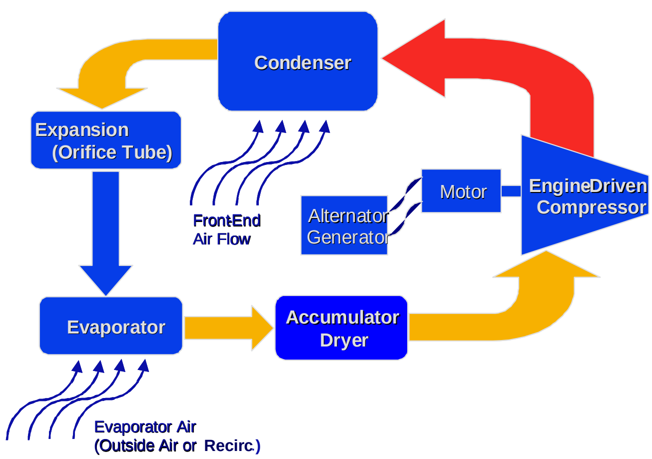


Figure 2 – Typical Vehicle Air Conditioning System With Electric-Driven Compressor

Figure 2 shows one configuration of a currently envisioned electrically-driven A/C system analyzed in this work. The compressor is typically driven by an electrical motor powered from an alternator/generator, which is turn is driven by a high-efficiency belt drive off the engine. One key to feasible electrically-driven systems is the efficiency of energy conversion from the belt drive through to the compressor. In these optimization studies, the belt drive is assumed to operate at 95% efficiency, the alternator is assumed to operate at 85% mechanical-electrical efficiency, and the motor is assumed to operate at 85% electrical-mechanical efficiency. These assumptions are intended to be within reasonably expected ranges; neither too conservative nor too optimistic based on expected near- and far-term component developments. The transient A/C model was modified to accurately calculate the engine energy requirement according to:

$$P_{eng}(t) = \frac{P_{tot}(t)}{\eta_{belt} \cdot \eta_{gen} \cdot \eta_{mot}} \qquad [2]$$

Energy conversion from the belt drive to the compressor is obviously less than ideal. Consequently, one goal was to determine if optimizing the A/C system design, in conjunction with an electrically-driven compressor, can increase the A/C system COP enough to more than compensate for energy conversion losses incurred in mechanical-electrical conversion equipment.

3 SYSTEM OPTIMIZATION OBJECTIVES

The EDC A/C system analyses focused on simultaneous optimization of five A/C system design parameters: 1) electric compressor speed (RPM), 2) compressor displacement, 3) expansion device diameter, 4) condenser tube diameter, and 5) transfer line diameter. The system design objectives concentrated on optimizing system COP and/or evaporator cooling capacity. Optimizing both system COP and evaporator cooling capacity is challenging because they are generally conflicting system design goals. Optimizing system COP reduces system power requirements and directly reduces vehicle fuel consumption, a primary DOE objective. On the other hand, optimizing evaporator cooling capacity increases cabin cool-down performance, which is often a major system design requirement in the automobile industry. However, optimizing evaporator cooling capacity is typically at the direct expense of higher A/C system power requirements. Consequently, the goal here was to define a system design methodology and a system operation strategy that could potentially satisfy both requirements simultaneously. The transient A/C model, with its integrated cabin thermal/fluid model and multi-variable optimization capability, provided a unique, powerful analytic tool for such a system optimization.

Initial system optimization focused on optimizing (i.e., maximizing) system COP within a set of design parameter constraints. The design goal was solely to minimize system energy consumption and the objective function was then:

$$f_{obj} = COP = \frac{\overline{Q_c}}{\overline{P}} \qquad [3]$$

where:

$$\overline{Q_c} = \frac{1}{t_f - t_i}\int_{t_i}^{t_f} Q_c(t)dt \qquad [4]$$

$$\overline{P} = \frac{1}{t_f - t_i}\int_{t_i}^{t_f} P_{tot}(t)dt \qquad [5]$$

as defined in references 9 and 10. Additional system optimizations were performed to optimize evaporator cooling capacity, Q_c, within a set of design parameter constraints, with the objective function set as:

$$f_{obj} = \overline{Q_c} \qquad [6]$$

The design goal in this case was to maximize cooling capacity and therefore cabin cool-down speed. These two system optimization functions led to much different optimum system design results that will be discussed in the following section.

The ultimate goal was to define a system design methodology and a system operation strategy, which could attempt to satisfy both design objectives simultaneously, or at least develop a system design compromise that simultaneously maximizes COP and Q_c to the extent possible. Final system optimizations therefore focused on the objective function:

$$f_{obj} = \alpha \cdot N_c \cdot COP + \beta \cdot N_q \cdot \overline{Q_c} \qquad [7]$$

where N_c and N_q are normalizing factors, and α and β are viewed as design influence factors. Values of N_c and N_q were set to normalize COP and Q_c to approximately 1, respectively. Influence factors were set such that $0<\alpha<2$ and $0<\beta<2$ subject to the constraint $\alpha + \beta = 2$. In this case, the value 2 was chosen because there were two independent objectives, COP and Q_c, that the method was trying to optimize simultaneously.

For system optimizations focusing on the objective function in Eq. 7, a particular dual-speed compressor operation strategy was employed. Figure 3 shows the basic compressor speed strategy based on cabin air temperature, in which one compressor speed is used until cabin air temperature drops to $T_{ambient}$, and a second compressor speed is used during cabin cool down. The initial compressor speed would be relatively high, allowing a short period during the cool down where evaporator cooling capacity would be allowed to be high to provide adequate cabin cool down speed. The second compressor speed would be relatively low, allowing system COP to increase and providing an energy-saving operational mode. In this optimization process, both compressor speeds were system optimization variables, thereby creating six variables simultaneously optimized in the multi-variable optimization.

4 SYSTEM OPTIMIZATION RESULTS

4.1 COP Maximization

This investigation first focused on simultaneously optimizing five independent design parameters to maximize system COP over an SC03 drive cycle using the objective function in Eq. 3. The operational strategy employed a single compressor speed over the entire SC03 drive cycle. Design parameters optimized were the single compressor speed, compressor displacement, expansion device diameter, transfer line diameter, and condenser tube diameter. Each design variable was allowed to vary within the ranges given in Table 1 during the system optimization.

Table 2 shows the results of the system optimization, the optimum values for the 5 design variables, and the resulting system COP, evaporator cooling capacity, and compressor and total system power. The total system power is the sum of the compressor power and the evaporator blower power. After 83 separate analyses, the optimum system design was determined to have a system COP = 3.42 at a compressor speed of 700 rpm, compressor displacement of 120 cm^3, and expansion device diameter of 0.191 cm. The noteworthy result is that a system COP > 3 is possible with a properly optimized system design, one in which the interdependent, coupled effects of multiple system design variables are simultaneously optimized. The COP result in Table 2 is approximately a factor of 2 higher than the best COP result obtained with this model using a mechanically-driven compressor over the SC03 drive cycle [10]. In that case the best COP realized in an optimized system over the SC03 drive cycle was COP=1.6 [10]. The compressor speed optimized at the relatively low value of 700 rpm. This is a much slower speed than current mechanically-driven A/C systems typically operate at over a drive cycle. However, it is a reasonable result because at this speed the compressor operates in a regime with much higher isentropic efficiency and volumetric efficiency, thereby yielding much higher system performance. These results clearly show that, given the equipment performance assumptions, a completely optimized, electrically-driven A/C system can indeed achieve high enough system COP (> 3) to more than compensate for the energy conversion losses associated with mechanical-electrical conversion equipment. Also noteworthy is that the optimum relationship between compressor displacement and expansion device diameter, defined in Eq. 1, is simultaneously found in this multi-variable optimization. This relationship and its important influence on the dynamic compressor pressure ratio are discussed by Hendricks [10]. It is also important to keep in mind that this multi-variable system optimization completely accounts for the dynamic two-phase flow conditions in both the condenser and evaporator.

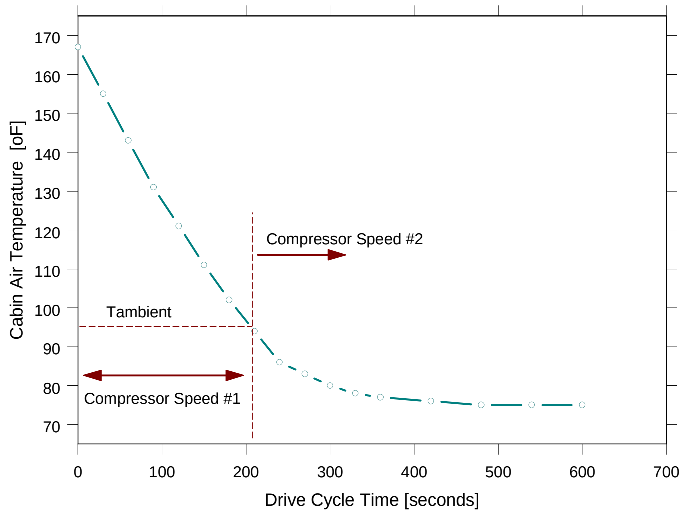


Figure 3 – Dual-Speed Compressor Operation Strategy During System Optimization on Eq.7

Table 1 – Design Parameter Constraints

	S_{com} [rpm]	V_{com} [cc]	D_{exp} [cm]	D_{trans} [cm]	D_{cond} [cm]
High	3000	300	2.41	2.54	0.889
Low	700	120	0.147	0.152	0.152

A system COP > 3 may seem quite optimistic for vehicle A/C systems, which typically operate at system COPs of about 1.5 to 1.7. However, vapor compression A/C systems are fully capable of operating at COPs of 4 or higher. Residential vapor-compression systems are certainly capable of this performance. Vehicle A/C system designs and their performance have simply been compromised too far to satisfy vehicle design requirements that don't include emphasizing overall vehicle energy management. A/C system design and its requirements are only accounted for late in a vehicle design cycle, and not enough design emphasis is placed on efficient energy management. This work shows that through a complete A/C systems design approach and system optimization, and an optimized operational strategy, vehicle A/C systems can operate at much higher system COPs than currently realized. Electrically-driven A/C systems represent a potentially important design approach to such higher-performance vehicle A/C systems.

4.2 Evaporator Cooling Capacity Maximization

The investigation then concentrated on optimizing the system design to maximize the evaporator cooling capacity over the SC03 drive cycle. The objective function was Eq. 6 in this case. The same single speed compressor strategy was used and the same five design variables were optimized as in the COP maximization. The five design variables again were allowed to vary in the ranges given in Table 1. Table 3 gives the results of this systems optimization, the optimum values for the five design variables, and the resulting system COP, evaporator cooling capacity, and compressor and total system power. The optimum system design to maximize cooling capacity is quite different than that for maximizing COP. The optimum compressor speed of 2066 rpm and optimum compressor displacement of 276 cm^3 is much higher than that for maximizing COP (i.e., 700 rpm, 120 cm^3, respectively). Although the evaporator cooling capacity has been maximized by this optimum system design, the COP of this design (i.e., 0.856) is far below that of the maximum COP system design in Table 2. The optimum expansion device diameter is also somewhat lower than that of the maximum COP system design in Table 2.

This highlights the system design differences required for achieving DOE goals of reducing A/C system power requirements, in order to reduce vehicle fuel consumption, and the automotive industry requirements to maximize cooling capacity and cabin cool down performance. In reality, these two design goals fundamentally conflict with one another, not only in vapor compression cooling system design, but for most other advanced cooling and heat pump systems one might consider. Consequently, a systematic design methodology that deals with this design incompatibility, and creates overall system optimization across the bounds of both design objectives, would be very useful and powerful. It could provide avenues to compromise optimum designs that attempt to satisfy both objectives to the extent physically possible. The next section presents results demonstrating one possible approach.

5 COMBINED OBJECTIVE DYNAMIC OPERATION

It is clear that a systematic methodology that addresses both maximizing COP and evaporator cooling capacity is going to require dynamic system operation. Figure 3 shows one example of an approach that couples the A/C system operation to the cabin air temperature; in this case a dual compressor speed was used and based on the cabin air temperature. Other design variables also could be altered and based on differing control variables and strategies. A/C system design optimizations were investigated using the dynamic control strategy in Figure 3 with the objective function in Eq. 7 in order to quantify the optimized A/C system performance possible, and evaluate compromise A/C system designs that could partially, or simultaneously, satisfy two diverse design objectives. A six-variable optimization was performed to optimize both compressor speeds and the four other system design variables to maximize Eq. 7.

Differing emphasis, or influence, on system design for COP maximization and cooling capacity maximization was accomplished simply by modifying the influence coefficients, α and β, subject to $\alpha + \beta = 2$. In the analysis presented, α and β values were adjusted to vary the (α/β) ratio from approximately 0.1 to 1.0 and evaluate the effect on the A/C system optimization. This created a range of optimized A/C system designs from those with a high degree of influence on maximizing system cooling capacity (i.e., typical automobile industry approach) to those with a high degree of influence on maximizing system COP (i.e., typical DOE / NREL objective). One goal was to establish whether it is possible to identify higher COP, energy-savings A/C system designs (i.e., DOE / NREL objective), while maintaining a relatively high A/C system cooling capacity (i.e., automobile industry objective).

The vehicle drive cycle can affect A/C system performance through airflow impact on the condenser heat transfer. Therefore, system optimization analyses were performed for both SC03 and US06 drive cycles to quantify this drive cycle effect on system design optimizations. Figures 4 and 5 show the results of the multiple A/C system design optimizations for various (α/β) ratios. Figure 4 shows the A/C system COP and the A/C system power requirement on the vehicle engine derived in the multi-objective system optimization for the SC03 and US06 drive cycle cases. Figure 5 shows the evaporator cooling capacity and compressor speeds derived from the multi-objective system optimizations for the SC03 and US06 drive cycle.

In Figure 4, the system COP (red dot dash line) for optimized A/C system designs on the SC03 drive cycle is seen to start at fairly low values (~0.9) for a low (α/β) ratios, where relatively low or no design influence is given to maximizing system COP. As the (α/β) ratio was increased to $\geq$ 0.21, thereby giving at least a reasonable design influence to maximizing system COP, the optimized A/C system COP increased dramatically to values of about 3.4. Simultaneously, the A/C system power demand on the engine (green solid line in Figure 4) decreases sharply as the

A/C system COP increases sharply for (α/β) ratios $\geq$ 0.2. Figure 5 shows that the optimum compressor speed (cool-down and steady-state) in such designs was again approximately 700 rpm, as found in the original COP maximization studies. Remarkably, the design influence functional relationship is a step- function, rather than a smooth continuous function, strongly suggesting that it may be very difficult to find a compromise A/C system design that simultaneously, or partially, satisfies to the extent possible, both design objectives depicted in Eq. 7. The optimized A/C system design jumps quite dramatically from one optimum design performance regime (system COP and power level) to another as the design influence (i.e., α/β ratio) changes. Similar system COP and engine power requirement results also are shown for the US06 drive cycle (red crosses and Xs) in Figure 4. The same conclusions on system performance are apparent from this US06 data as in the SC03 drive cycle case. Consequently, although these two drive cycles are quite different, this does not change the basic conclusions on optimum system performance for the two design objectives. This demonstrates that there indeed may be a very sharp demarcation between designs satisfying DOE / NREL design objectives, and those satisfying or focusing on automotive industry design objectives for electric-driven A/C systems.

Figure 5 shows that the evaporator cooling capacity also shows a sharp demarcation in optimized A/C system designs that incorporate a increasing emphasis on optimizing the design objective of maximizing system COP. The evaporator cooling capacity for optimized designs decreases by approximately 25% in such designs. However, what is most interesting about this is that the system cooling capacity does not fall to unreasonably low levels. Figure 5 shows that in this particular system design optimization there can still be 3600 watts of cooling capacity available in an electric-driven A/C system optimized for maximum COP. This demonstrates that if the auxiliary loads were appropriately reduced in the vehicle cabin environment, then there is a great opportunity to integrate such an optimized electric-driven A/C system to produce energy-efficient, thermally comfortable vehicle design solutions. In addition, there are tremendous benefits that an optimized electric-driven A/C system, which reduces power loads on the engine, could subsequently create by reducing harmful vehicle emissions (i.e., NO_x CO, and hydrocarbons). Consequently, there is great opportunity and need to integrate A/C system optimization programs with vehicle auxiliary load reduction programs within the U.S. DOE, Environmental Protection Agency, Department of Transportation and the U.S. and foreign automobile industries. A coordinated and integrated vehicle climate control system design philosophy can and must be implemented, for a variety of economic and national security reasons, to assist in reducing our nation's need for and addiction to imported foreign oil.

This study demonstrates the role that influence coefficients, α and β, play in the system optimization. The influence coefficients not only play a role in defining the optimum system design, but also help in quantifying how close one is to optimizing to a given system design objective versus another, and in evaluating different design emphases and philosophies during system design optimization. Consequently, this system optimization approach yields system designs that begin to accomplish the objective of optimizing both system design goals to the extent possible in a dynamic, dual-compressor-speed, electric-driven system. Current vehicle A/C system design concentrates too heavily on maximizing evaporator cooling capacity and cabin cool-down, with little emphasis placed on achieving higher system COP. The approach presented here provides a design methodology for better achieving both objectives and creating a better overall system design that reduces A/C system energy usage.

Variable displacement compressors or variable orifice valves also provide potentially beneficial system design approaches to improving system performance. Future research with transient A/C system optimization within ADVISOR will study 1) benefits of these components and other dynamic A/C system design approaches to improve energy management performance, and 2) A/C system optimization using this optimization methodology with other drive cycles.

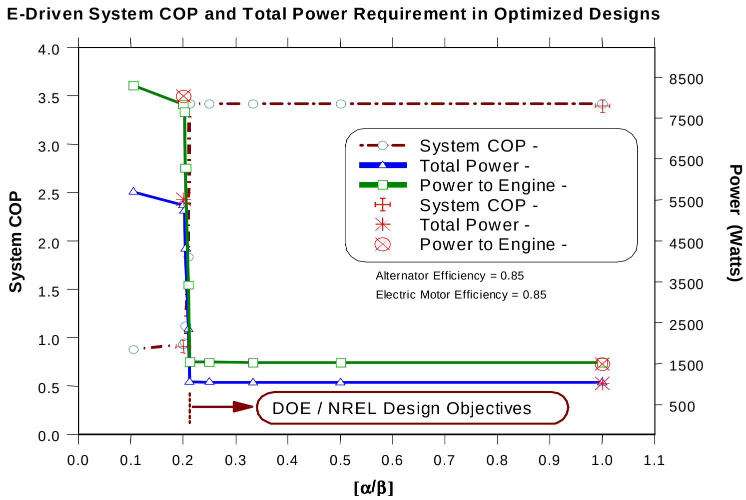


Figure 4 – Electric-Driven System COP and Total Power Requirement in Optimized A/C System Designs

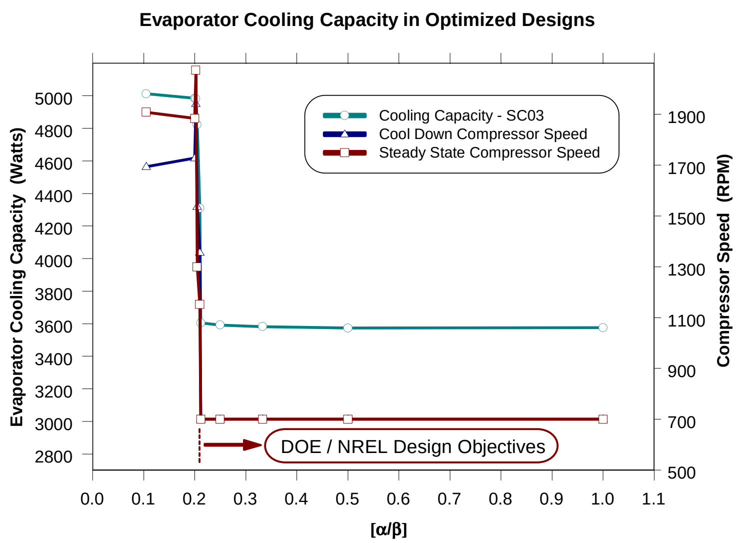


Figure 5 – Evaporator Cooling Capacity in Optimized A/C System Designs

6 CONCLUSION

Vehicle electrically-driven A/C systems have been modeled using NREL's transient A/C system model and optimized through multi-variable design optimization techniques within the SINDA/FLUINT thermal-hydraulic analysis package. Electrically-driven compressor A/C systems represent a potentially important design approach to higher-performance vehicle A/C systems. Design optimization results demonstrate significant improvements in A/C system COP are possible, particularly system COP > 3, in a properly optimized, electrically-driven A/C system with a dynamic control strategy. The system optimization work investigated various dynamic A/C system design strategies employing dual-compressor-speeds to evaluate their effects on system performance. A dual-compressor-speed strategy, coupled with the use of an electrically-driven compressor, was beneficial in developing system designs that improves system COP while maintaining a reasonable evaporator cooling capacity. System COP > 3 was possible under SC03 and US06 drive cycle conditions in a properly optimized A/C system design, approximately a factor of 2 higher than typical system COPs obtained using mechanically-driven compressors over the SC03 drive cycle [9]. The optimum compressor speeds were discovered to be much slower (i.e., 700 rpm) than typical mechanically-driven compressor systems, allowing the compressor to operate at higher efficiency regimes. Even more importantly, the improved COP A/C systems with reasonable evaporator cooling capacity create a tremendous opportunity to integrate vehicle auxiliary loads reduction in the vehicle passenger cabin with optimized electric-driven A/C systems to produce energy-efficient, thermally comfortable vehicle design solutions. U.S. Departments of Energy and Transportation, and the Environmental Protection Agency should collaborate with industry to make such coordinated, integrated programs a reality.

A new system optimization methodology also has been developed which can systematically quantify the impact on A/C system design and performance resulting from varying degrees of design influence being given to widely different design objectives. The technique is based upon formulating optimization objective functions from linear combinations of critical design performance parameters that characterize independent design goals. The technique has been demonstrated by giving varying degrees of design influence to maximizing system COP and maximizing evaporator cooling capacity over given drive cycles, such as the SC03 and US06. Combinations of design influence coefficients were discovered ($\alpha/\beta > 0.21$) that produced optimum A/C system designs with system COP > 3, while simultaneously maintaining reasonably high evaporator cooling capacity. The design influence coefficients not only play a role in defining the optimum system design, but also can help in quantifying how close one is to optimizing to a given system design objective, and in evaluating different design emphases and philosophies during system design optimization. A continuous design spectrum of varying degrees of design influence between two or more diverse and competing design objectives (e.g., maximizing system COP or evaporator cooling capacity) can now be systematically studied, rather than arbitrarily selecting one design emphasis or another. Other design objectives, such as system cost and weight, are possible and will be investigated in future work.

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REFERENCES

1. Bevilacqua, O.M., “Effect of Air Conditioning on Regulated Emissions for In-Use Vehicles”, Clean Air Vehicle Technology Center, Oakland, CA, Phase I Final Report Prepared for Coordinating Research Council, Inc., Atlanta, GA, CRC Project E-37, October 1999.
2. Kelly, K.J. and Rajagopalan, A., “Benchmarking of OEM Hybrid Electric Vehicles at NREL,” Milestone Report #NREL/TP-540-31086 to the U.S. Department of Energy, National Renewable Energy Laboratory, Golden, CO, August 2001.
3. Nadamoto, H. and Kubota, A., “Power Saving With the Use of Variable Displacement Compressors”, *Proceedings of SAE International Congress and Exposition*, SAE Technical Paper #1999-01-0875, Detroit, MI, March 1999.
4. Wang, M., Farley, D.L., Leitzel, L.L., “Air Conditioning System Head Pressure Spike During Vehicle Acceleration”, *Proceedings of SAE 2000 World Congress*, SAE Technical Paper #2000-01-0973, Detroit, MI, March 2000.
5. Khamsi, Y., and Petitjean, C., “Validation of Automotive Passenger Compartment and Its Air Conditioning System Modeling”, *Proceedings of SAE 2000 World Congress*, SAE Technical Paper #2000-01-0982, Detroit, MI, March 2000.

6. Hyun, L.K. and Wong, J.-F., "Thermal Design Study of a High Performance Evaporator for the Automotive Air Conditioner", *Proceedings of SAE International Congress and Exposition*, SAE Technical Paper #1999-01-1191, Detroit, MI, March 1999.
7. Kelemen, K., Tarunraj, S., and Mayne, R., "Modeling of an Automotive Air-Conditioning System", *Proceedings of SAE 2000 World Congress*, SAE Technical Paper #2000-01-1269, Detroit, MI, March 2000.
8. Cullimore, B.A. and Hendricks, T.J., "Design and Transient Simulation of Vehicle Air Conditioning Systems," *Proceedings of Vehicle Thermal Management Systems Conference 5*, SAE International, Paper #2001-01-1692, May 2001.
9. Hendricks, T.J., "Optimization of Vehicle Air Conditioning Systems Using Transient Air Conditioning Performance Analysis," *Proceedings of Vehicle Thermal Management Systems Conference 5*, SAE International, Paper #2001-01-1734, May 2001.
10. Hendricks, T.J., "Vehicle Transient Air Conditioning Analysis: Model Development & System Optimization Investigations," NREL Report #NREL/TP-540-30715, June 2001.

Table 2 – System COP Optimization – Single Speed Compressor, SC03 Drive Cycle
(Optimum System Design in Bold)

Solution Loop Count	Expansion Device Diameter [cm]	Transfer Line Diameter [cm]	Compressor Displace-Ment [cm^3]	Condenser Tube Diameter [cm]	Electric Motor Speed [rpm]	Evaporator Cooling Capacity Average [watts]	Compressor Power Average [watts]	Total Power Average [watts]	Power To Engine [watts]	COP Average
1	0.18288	1.14300	200.00	0.56059	1500	4512.714	2946.16	3135.95	4568.86	1.4390
6	0.18288	1.14300	200.00	0.57180	1500	4512.289	2943.46	3133.33	4565.05	1.4401
12	0.18523	1.15952	159.99	0.58138	1200	3999.874	1832.48	2021.51	2945.19	1.9787
16	0.19068	1.17449	127.99	0.58308	960	3778.004	1231.10	1419.77	2068.51	2.6610
22	0.19309	1.19771	120.01	0.59829	768	3592.934	930.73	1119.11	1630.46	3.2105
28	0.18904	1.18518	120.01	0.58375	700	3582.216	860.74	1049.09	1528.46	3.4146
51	**0.19074**	**1.18144**	**120.01**	**0.58427**	**700**	**3582.052**	**859.78**	**1048.13**	**1527.05**	**3.4176**
68	0.19513	1.16482	120.01	0.58561	700	3583.168	860.61	1048.96	1528.26	3.4159
76	0.19513	1.16482	120.01	0.58561	700	3583.145	860.59	1048.94	1528.24	3.4160
83	0.19416	1.14419	120.01	0.57427	700	3583.145	860.69	1049.04	1528.38	3.4156

Table 3 – Evaporator Cooling Capacity Optimization – Single Speed Compressor, SC03 Drive Cycle (Optimum System Design in Bold)

Solution Loop Count	Expansion Device Diameter [cm]	Transfer Line Diameter [cm]	Compressor Displace-ment [cm^3]	Condenser Tube Diameter [cm]	Electric Motor Speed [rpm]	Evaporator Cooling Capacity Average [watts]	Compressor Power Average [watts]	Total Power Average [watts]	Power To Engine [watts]	COP Average
1	0.182880	1.143000	200.0019	0.560588	1500.00	4512.71	2946.16	3135.95	4568.86	1.4390
7	0.180807	1.132515	230.2443	0.557936	1721.80	4766.13	3906.52	4096.82	5968.78	1.1634
10	0.180807	1.132515	234.8599	0.557936	1721.80	4782.72	3991.37	4181.70	6092.44	1.1437
14	**0.172395**	**1.013643**	**276.2875**	**0.52767**	**2066.16**	**5009.21**	**5662.60**	**5853.32**	**8527.87**	**0.8558**
16	0.175839	1.013643	276.2875	0.52767	2066.16	4996.42	5687.33	5878.05	8563.90	0.8500
18	0.172395	1.013643	276.2875	0.52767	2107.49	5007.78	5995.89	6186.70	9013.58	0.8094
19	0.172395	1.033943	276.2875	0.52767	2066.16	5001.48	5688.07	5878.75	8564.93	0.8508
20	0.172395	1.013643	276.2875	0.538216	2066.16	4986.70	5693.40	5884.11	8572.73	0.8475