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Optimization of Vehicle Air Conditioning Systems Using Transient Air Conditioning Performance Analysis

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ABSTRACT

The National Renewable Energy Laboratory (NREL) has developed a transient air conditioning (A/C) system model using SINDA/FLUINT analysis software. It 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. It has demonstrated robust and powerful system design optimization capabilities. Single-variable and multiple variable design optimizations have been performed and are presented. Various system performance parameters can be optimized, including system COP, cabin cool-down time, and system heat load capacity. This work presents this new transient A/C system analysis and optimization tool and shows some high-level system design conclusions reached to date. The work focuses on R-134a A/C systems, but future efforts will modify the model to investigate the transient performance of alternative refrigerant systems such as carbon dioxide systems. NREL is integrating its transient air conditioning model into NREL's ADVISOR vehicle system analysis software, with the objective of simultaneously optimizing A/C system designs within the overall vehicle design optimization.

INTRODUCTION

Major Department of Energy (DoE) objectives include developing 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 engine of light-duty passenger vehicles, sport-utility vehicles (SUV), and heavy-duty vehicles. The A/C system performance has a dramatic effect on fuel consumption and exhaust emissions. 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%. 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 the 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. Consequently, the National Renewable Energy Laboratory (NREL) and other researchers [2-6] are giving increased attention to analyzing and modeling steady-state and transient A/C system performance.

TRANSIENT AIR CONDITIONING SYSTEM MODEL

In order to more completely understand and quantify transient A/C system performance and its impact on vehicle fuel consumption and emissions, NREL has developed a transient A/C model using SINDA/FLUINT analysis software and is integrating it with the ADVISOR vehicle systems analysis software. The transient, one-dimensional, thermal-hydraulic model was developed using a nominal representative A/C system that was identified in discussions with NREL's automotive industry partners. 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. This model can predict typical transient A/C compressor power requirements, system pressures and temperatures, system mass flow rates, and two-phase/single-phase flow conditions throughout the A/C system flow circuit.

SINDA/FLUINT is capable of correctly analyzing the various two-phase flow regimes, such as bubbly flow, slug flow, annular flow, as well as the heat transfer and pressure drop conditions in both the evaporator and condenser. It contains several built-in heat transfer coefficient and friction factor correlations that are used to automatically evaluate heat transfer, pressure drop, and flow quality conditions within the A/C system components during its system computations. SINDA/FLUINT also has built-in correlations for determining transitions between different two-phase flow regimes in the condenser and evaporator, and can easily analyze slip flow conditions that may occur during two-phase flow in these components. The **Component Effects** section presents flow quality and flow regime results, and discusses the influence of system components on flow quality and flow regimes in the condenser and evaporator.

The simplified cabin thermal model predicts cabin and panel outlet temperatures during transient cool-down periods and during steady state operational periods. The combined model predicts A/C system and cabin thermal conditions during various drive cycles, including vehicle idle, SC03, US06 or other typical federal test and passenger-induced drive cycles. The SC03 and US06 federal drive cycles, shown in the **APPENDIX**, are currently incorporated within the transient A/C model so that transient performance and optimization results can be tailored to each unique set of driving conditions. With its current integration to a simplified cabin model, the transient A/C system model thereby provides the system link connecting cabin thermal comfort requirements back to vehicle fuel consumption and emissions. Future work will continue to integrate the transient A/C model to higher fidelity, 3-dimensional cabin thermal / fluid models based on finite element computational fluid dynamics formulations.

Figure 1 shows a schematic diagram of a typical vehicle vapor-compression air conditioning system being modeled in this work. Figure 2 shows a schematic diagram of the transient SINDA/FLUINT model of a nominal representative A/C system represented in Figure 1. The model consists of a nominal compressor, condenser design (heat exchanger HX 3000), orifice tube expansion device, and evaporator design (heat exchanger HX 6000). The model includes thermal regeneration between the orifice tube and the suction line. The compressor is characterized by a compressor displacement of 0.0002 m^3 and representative isentropic and volumetric efficiencies. The compressor isentropic efficiency and volumetric efficiency are characterized by the following relationships respectively:

$$\beta_i = 1. - \left[\frac{A_o}{P_r R_r} + \frac{A_1}{P_r} + \frac{A_2}{R_r} + \frac{A_3}{P_r} \cdot R_r + A_4 + \frac{A_5}{R_r} \cdot P_r \right] \quad [1]$$

$$\beta_v = 1. - \left[\frac{B_o}{R_r} + B_1 + \frac{B_2}{R_r} \cdot P_r + B_3 \cdot R_r + B_4 \cdot P_r \right] \quad [2]$$

The condenser heat exchanger is a serpentine-type design with 6 serpentine passes, 10 parallel channels, and a weight of 11 lbms. The tube diameter was an optimization variable and determined as discussed in the following section on system optimization. The evaporator heat exchanger is also a serpentine-type design with 12 serpentine passes, a tube diameter of 0.0625 inch, and a weight of 6.6 lbms. The heat exchangers were typical of designs shown in Kargilis [7].

The transport lines shown in Figure 1 between the compressor and condenser and between the condenser and the expansion device are critical components in the A/C system design. Their diameter and length impact system performance. Compressor characteristics and orifice diameter are other key system parameters that impact transient system performance. The **System Optimization Studies** section will demonstrate how this component design is important to optimizing system COP and inter-dependent on other important system components, particularly the condenser.

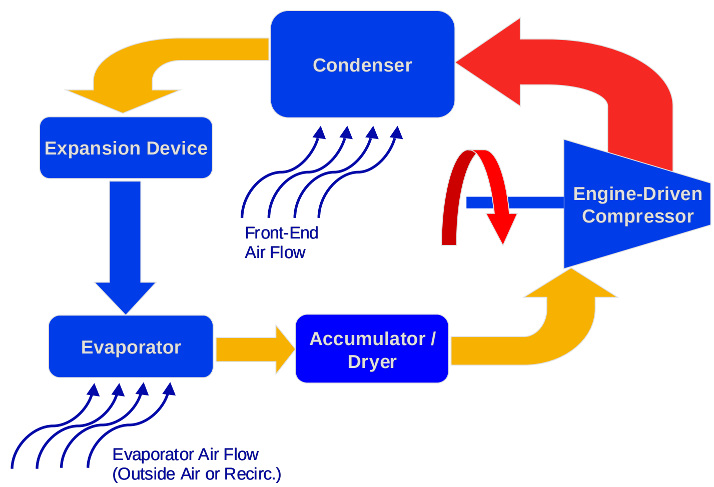

Figure 1 – Schematic Diagram of a Typical Vehicle Air Conditioning System

The transient A/C model also contains a simplified cabin thermal model that is a two-node model consisting of the cabin air thermal mass and interior cabin hardware mass representing seats, instrument panel, consoles, and various other cabin hardware. The cabin air volume is 102 ft^3 and the interior cabin hardware mass is 1000 lbms in this model. The cabin is modeled to absorb passenger thermal energy dissipation and solar energy across the entire wavelength spectrum. The cabin model also incorporates conductive thermal energy exchange between the cabin internal air and the external

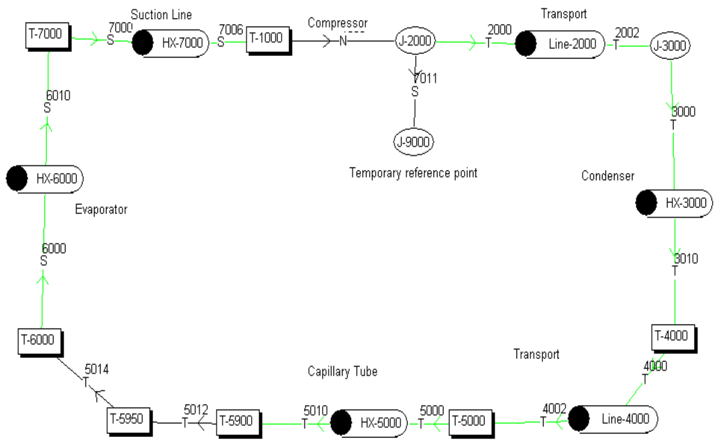

Figure 2 – Schematic Diagram of SINDA/FLUINT Transient Air Conditioning System Model

ambient environment and convective thermal energy exchange between the cabin internal air and the internal cabin mass. Humidity effects, and corresponding latent thermal energy in the cabin air, also are accounted for during cabin cool-down periods.

Figure 3 shows a typical transient compressor power prediction from the transient A/C-cabin thermal model during the 10-minute SC03 drive cycle after extreme hot soak conditions to 167 ^{o}F. The compressor power has been normalized by the average compressor power over the SC03 drive cycle, but the variation in compressor power is quite substantial. A pressure spike in the compressor outlet and condenser inlet pressures occurs with each compressor power spike in Figure 3. These pressure spikes are associated with vehicle accelerations in the SC03 drive cycle and are similar to those discussed by Wang et al. [3]. Figure 4 displays the corresponding typical cabin temperature cool-down prediction from the transient A/C-cabin thermal model during the same 10-minute SC03 drive cycle after the same extreme hot soak conditions to 167 ^{o}F. No system performance optimization has been done as part of these results.

COMPONENT EFFECTS

Figure 5 shows a typical condenser flow quality profile predicted during an SC03 drive cycle. In this figure the condenser is mathematically represented by 20 discrete sections (i.e., flow lumps), and red indicates full vapor conditions, blue represents full liquid conditions, and intermediate colors represent various stages of two-phase flow conditions. Several flow quality / regime profiles at various points in time in the SC03 drive cycle are shown in Figure 5. During the drive cycle the condensation front (red) shifts position in the condenser, which causes flow and heat transfer conditions to vary dynamically. A similar front-movement occurs with the evaporation front in the evaporator. Figure 6 displays a typical evaporator flow quality profile prediction during an SC03 drive cycle. The evaporator here is represented by

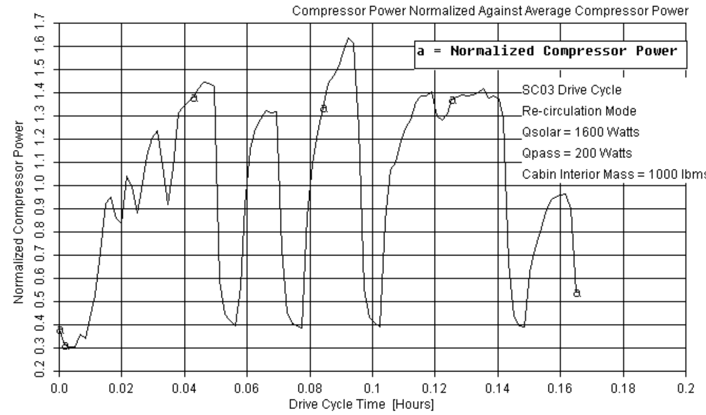

Figure 3 – Typical Compressor Power Prediction for an SC03 Drive Cycle after Hot Soak Conditions to 167 ^{o}F.

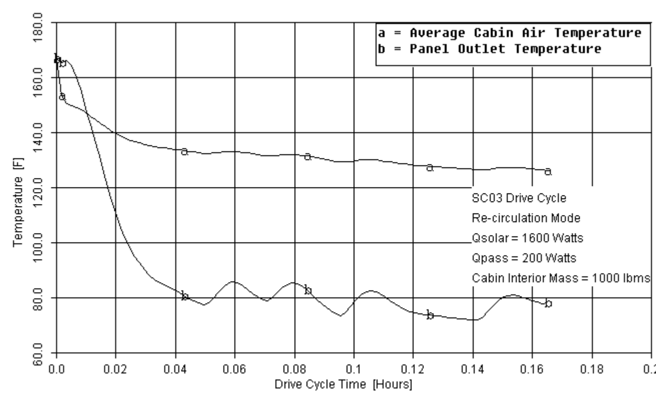

Figure 4 – Typical Cabin Temperature Cool-Down Prediction for an SC03 Drive Cycle after Hot Soak Conditions to 167 ^{o}F.

10 discrete sections (i.e., flow lumps), and red again indicates full vapor conditions, but here blue represents varying degrees of two-phase flow conditions. It is critical to account for and understand this behavior in developing optimized systems. This behavior certainly complicates the design and optimization of vehicle A/C systems.

A further complication is that the compressor and orifice or thermal expansion valve (TXV) can have major effects on the flow quality and flow regimes within both the evaporator and condenser during a typical drive cycle. The compressor performance can change the condensing front movement and the evaporation front movement in the condenser and evaporator, respectively. Similarly, the orifice or TXV can change liquid front positions (blue in Figure 5) in the condenser and the evaporation front movement (red in Figure 6) within the evaporator. In the past, comprehensive analytic tools have not been available to quantify this behavior and determine its impact on the system design and optimization. However, the current SINDA/FLUINT transient A/C model provides a powerful tool to evaluate and understand these component-level effects on system designs prior to system fabrication and testing.

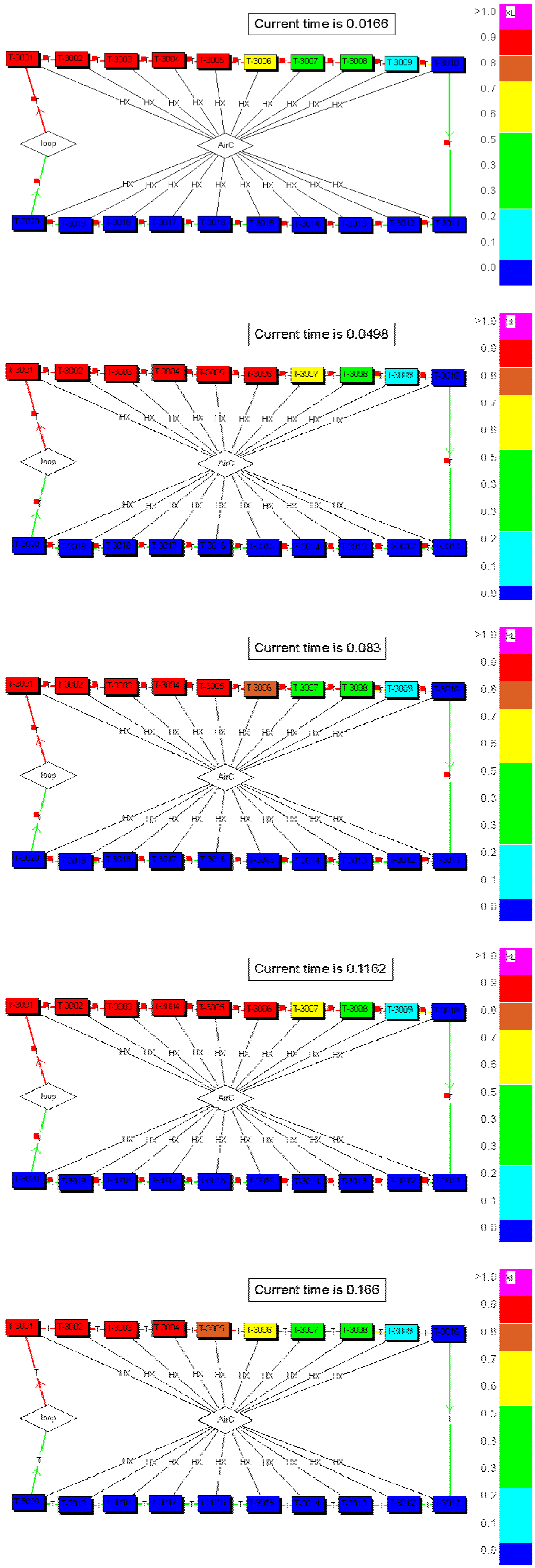

Figure 5 – Transient Flow Quality Profiles in Condenser for an SC03 Drive Cycle (Current Time in Hours)

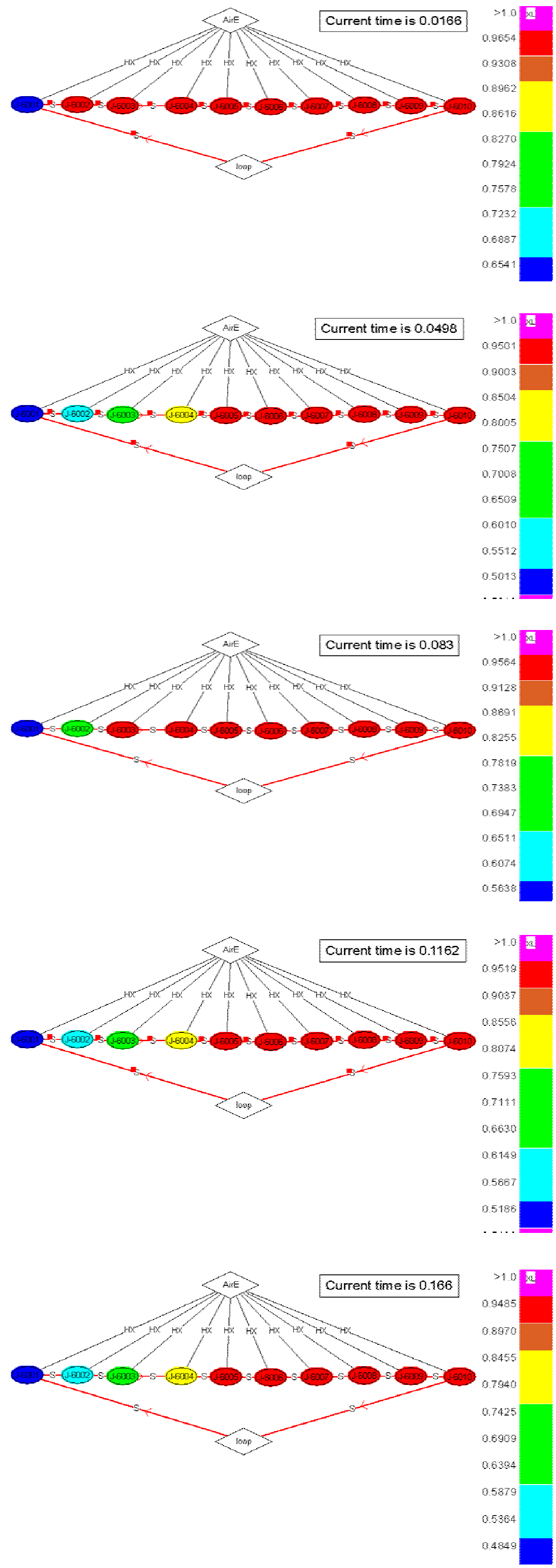

Figure 6 – Transient Flow Quality Profiles in Evaporator for an SC03 Drive Cycle (Current Time in Hours)

SYSTEM OPTIMIZATION STUDIES

Optimizing the A/C system design can significantly reduce the mass and volume of current A/C systems in various vehicles, and impact the A/C system control strategy, in satisfying cabin thermal comfort requirements. However, truly optimizing the A/C system design is complicated by the transient nature of the system performance and its inherent coupling to cabin thermal conditions (i.e., heat loads, air temperatures, vent configurations, etc.). The A/C system pressures, temperatures, and condenser and evaporator flow conditions are all time-dependent. They are controlled by transient compressor speed, refrigerant and air flow rates, and vehicle thermal loads. Refrigerant and airflow rates are, in turn, determined by vehicle velocity and compressor speed. Compressor power, being typically proportional to compressor speed cubed, can be highly variable during the SC03 or any other drive cycle, creating highly variable torque requirements on the engine. Consequently, the important thermal design parameters, such as the evaporator heat load, compressor power, and COP, become time-averaged design parameters given by equations:

$$\hat{q} = \frac{1}{t_2 - t_1} \int_{t_1}^{t_2} q(t)dt \qquad [3]$$

$$\hat{P} = \frac{1}{t_2 - t_1} \int_{t_1}^{t_2} P(t)dt \qquad [4]$$

$$\hat{\eta} = \frac{\hat{q}}{\hat{P}} \qquad [5]$$

These time-averaged quantities are what one typically measures in A/C system bench-top or vehicle-level testing and can therefore be correlated with transient model predictions.

In the most rigorous sense, another complicating factor in the transient performance of a vehicle A/C system is the thermal mass, and therefore thermal energy storage potential, associated with each system component. One then has to be concerned with time-average thermal energy storage for each ith component given by:

$$\frac{1}{t_2 - t_1} \int_{t_1}^{t_2} m_i \cdot \overline{C}_{p,i} \cdot \left(\frac{dT}{dt}\right)_i dt \qquad [6]$$

which complicates system and component energy balances. Accurately accounting for each of the time-dependent quantities in a full A/C system transient analysis makes it imperative that a high-level system analysis tool, such as SINDA/FLUINT, be utilized in any A/C system design optimization studies based on transient performance.

Additional challenges exist because A/C system design optimizations can be performed using various system-level assumptions. Among these one can assume:

1) Constant system refrigerant mass,

2) Constant system initial pressure,

3) Various vehicle drive cycle conditions, or

4) Constant/variable vehicle solar thermal loads.

Care must be taken in A/C system optimization work because which assumption is made and what specific value is assumed, for initial system pressure or solar thermal load for example, can affect the final solution.

SC03 DRIVE CYCLE RESULTS - Optimization studies were initiated by performing a single-variable optimization maximizing system COP, defined in Eq. 5, with varying condenser tube diameter. The integrated A/C system / cabin thermal model was used to investigate a range of condenser tube diameters for constant system refrigerant mass (2.15 lbms) assuming an SC03 drive cycle, a cabin external solar load of 1600 Watts, a cabin passenger heat load of 200 Watts, and an initial internal cabin temperature of 75 ^{o}C (167 ^{o}F). Figure 7 shows the results of this system optimization for SC03 drive cycle conditions. The system COP maximizes at a condenser tube diameter of approximately 0.222 inch for the above assumptions. It is important to note that this optimization was performed within SINDA/FLUINT accounting for all the two-phase heat transfer / pressure drop effects and flow transition effects in the condenser and evaporator. There is a small computational uncertainty, noted in Figure 7, associated with this system optimization procedure as SINDA/FLUINT searches the domain space and verifies the final optimal solution. Tightening up solution convergence tolerances within SINDA/FLUINT could easily reduce this computational uncertainty. The optimal condenser tube diameter is still clear after accounting for the computational uncertainty shown.

Additional dual-variable system optimization studies were performed to determine the combined effect of transport line diameter and condenser tube diameter on the system design. The integrated A/C system / cabin thermal model was used in SINDA/FLUINT to investigate a range of transport line diameter/condenser tube diameter combinations for the same system and cabin thermal conditions given above, particularly the same constant refrigerant mass. Figure 8 shows the results of this simultaneous dual-variable optimization in a 3-dimensional plot of the COP vs. transport line diameter and condenser tube diameter as determined during several optimization searches performed by SINDA/FLUINT. In any system optimization it is important to perform a comprehensive search of the variable design space to locate the optimum solution.

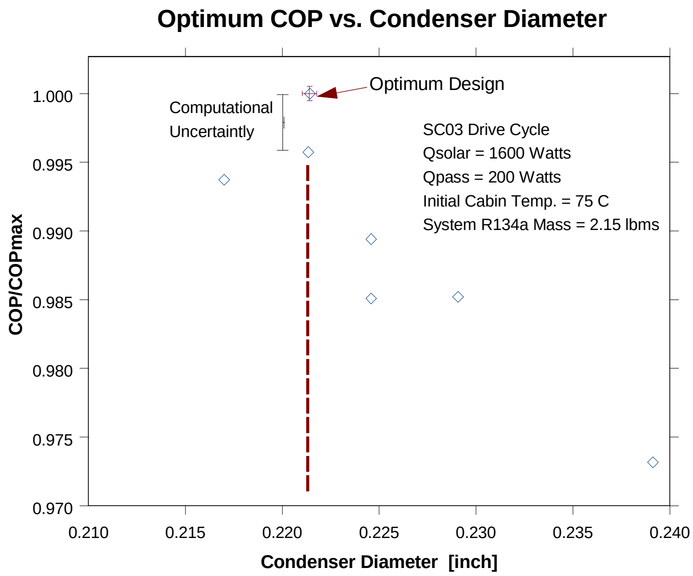

Figure 7 – Optimum Condenser Diameter Exists Maximizing System COP

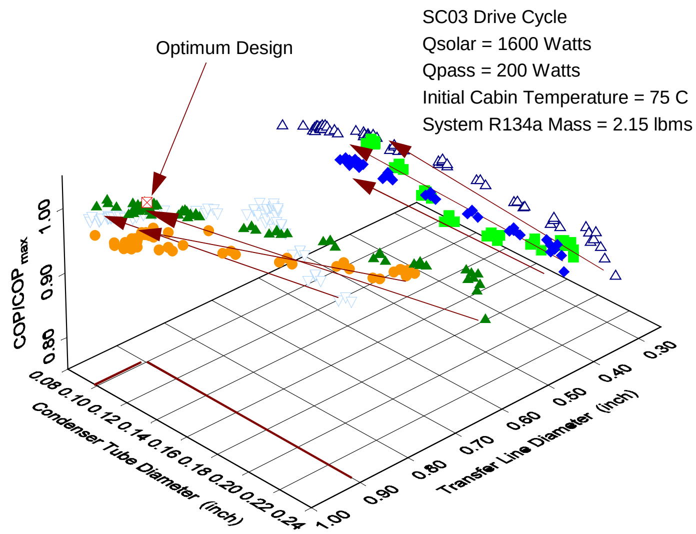

Figure 8 – Optimum Dual Variable Solution Exists to Maximize System COP – Optimum Transport Line Diameter & Condenser Tube Diameter

Figure 8 shows the results of several different runs starting at various initial guesses. In general, the arrows in the Figure 8 plot show the various initial guesses for these parameters and general direction of solution as the optimization progresses to a solution. The optimal solution maximizing system COP was found to be a transport line diameter of approximately 0.90 inch and a condenser tube diameter of approximately 0.10 inch. The optimal solution shown in Figure 8 demonstrates the design performance benefit of expending available pressure drop in the condenser, and thereby enhancing heat transfer in the condenser, and minimizing pressure drop in the transport lines between the system components. These optimum results are quite different from current vehicle A/C systems, transport line diameters are generally about 0.35 inch and condenser tube diameters of 0.2 inch are common. It is important to again note that this optimization was performed with SINDA/FLUINT simulating all the two-phase heat transfer / pressure drop effects and flow transition effects in the condenser and evaporator.

One important point here is that the optimal selection for condenser tube diameter is strongly affected by the dual-variable optimization in Figure 8 vs. the single-variable optimization in Figure 7. A much different optimal condenser tube diameter was discovered when the transport line diameter optimization was performed simultaneously. This demonstrates the strong inter-relationships and effects between various system design parameters in the A/C system. A full system, multiple-variable optimization is required to accurately optimize system performance, whether maximizing system COP or optimizing other system performance parameters.

Additional optimization studies were performed assuming a somewhat more realistic constant initial pressure at the start of the drive cycle and slightly different compressor characteristics. In these studies the initial system pressure was 245 psi. Figure 9 displays the to-date results of this optimization study, with three different runs performed starting at three separate initial starting points. Arrows in Figure 9 again show the initial guesses for the design parameters and general direction of solution during the optimization. A different optimal solution was found as a result of changing the system assumption to constant initial pressure, rather than constant refrigerant mass, and using slightly different compressor characteristics. This solution is still tending toward larger transfer line diameters (approximately 0.51 inch), but it is also indicating larger condenser tube diameters (approximately 0.24 inch) than the previous optimization using constant refrigerant mass. This work emphasizes the need to optimize the A/C system as a system, even slight changes in one critical component like the compressor can lead to a much different optimum system design. Certainly taking one off-the-shelf component and combining it with other off-the-shelf components, without a true system design approach and optimization, will not produce the optimum system performance.

Multiple variable optimization studies were also extended to simultaneous optimizations of three design parameters, orifice tube diameter, transfer line diameter, and condenser tube diameter, over the SC03 drive cycle. The initial system pressure assumption of 245 psi was maintained, along with all the other system conditions depicted in Figures 8 and 9. Previous dual- variable optimization work discussed above was problematic and tedious because, during the optimization process, it is actually quite easy to define and attempt to analyze a non-realistic design. This quickly leads to computational stability problems because it is just as impossible to analyze a non-realistic system design as it is to fabricate and test such a design. Although the optimization problem becomes more complex and computationally

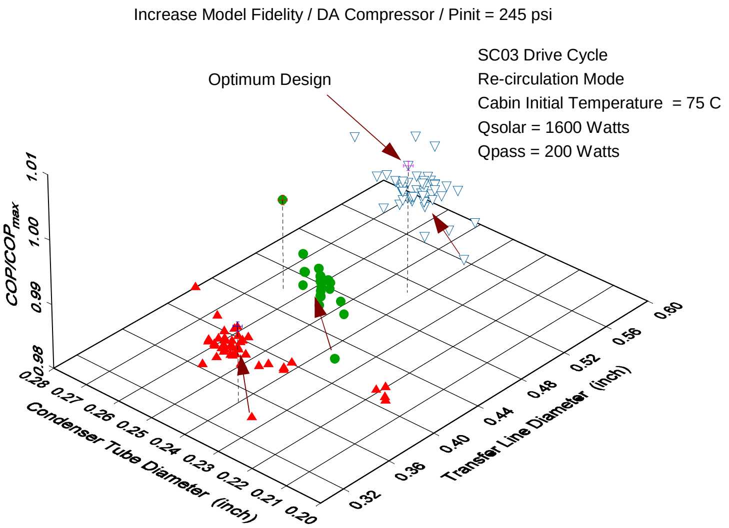

Figure 9 – Optimum Transfer Line Diameter & Condenser Tube Diameter to Maximize System COP

intense with triple-variable or quadruple-variable optimization, this work found that this does not necessarily lead to more computational instabilities. In fact, simultaneously optimizing the orifice tube diameter, transfer line diameter, and condenser tube diameter was surprisingly stable, most likely due to the added degree of freedom in the system.

As before, the system optimization runs started from multiple initial starting points with a constant initial pressure of 245 psi. Table 1 displays the results of 5 separate optimization runs. The first column is the optimization loop count and tells how many solution iterations were required to achieve a solution and check for convergence. The next three columns show the design parameter values at each search point. The next column shows the value of the objective function, in this case the average COP over the SC03 driving cycle. The last column is simply the COP ratio relative to the maximum.

Amongst the 5 runs in Table 1, Runs 2 and 5 determined two potential solutions that maximized COP within this group. In one case (Run 2), a relatively large orifice diameter (0.0869 inch), a relatively large transfer diameter (0.795 inch), and a condenser tube diameter of 0.231 inch produced the optimal performance. In Run 5 a much smaller orifice diameter and smaller condenser tube diameter, with roughly the same transfer line diameter (0.752 inch), produced the optimal result. Runs 1, 3 and 4 simply achieved intermediate solutions representing local maximums that could go no further in the optimization process because of convergence criteria limitations. It is quite common in multi-variable optimization of non-linear problems to discover multiple solutions that satisfy, either maximizing or minimizing, the objective function goal. The solutions in Runs 2 and 5 correspond to quite different optimum system solutions that one would evaluate further. At this point, additional design information would have to be considered (possibly system cost, weight, etc.) to differentiate the two to one system-level solution. One could develop a different objective function involving additional design objectives and re-optimize the system design to quantifiably distinguish between the two solutions.

US06 DRIVE CYCLE RESULTS - Multiple variable optimization studies were also extended to simultaneous optimizations of three design parameters, orifice tube diameter, transfer line diameter, and condenser tube diameter, over the US06 drive cycle. All system assumptions used in the SC03 optimization studies were maintained. As before system optimization runs started from multiple initial conditions with the constant initial pressure of 245 psi. Table 2 displays the results of the 5 separate US06-drive-cycle optimization runs using the different initial starting conditions.

Table 2 shows that this US06 design optimization produced a much different optimum system solution than that for the SC03 drive cycle. The optimum system performance for US06 drive cycle conditions was discovered at an orifice diameter of 0.058 inch, a transfer line diameter of 0.813 inch, and a condenser tube diameter of 0.235 inch. This demonstrates that optimum system designs can vary depending on the drive cycle because of the different compressor and system flow dynamics. Therefore, a sophisticated optimization strategy simultaneously incorporating the impact of various drive cycles is required in future system design optimizations. The fact that the optimum A/C system design can vary with drive cycle conditions also creates the potential opportunity for a dynamically variable A/C system design based on drive cycle conditions. Some design work is being done in this area, but this work can greatly benefit from the dynamic A/C modeling capability presented here.

A final note is required on this system optimization work. It is clear that the optimization results shown above do not show large improvements in COP. This is largely due to the particular choices of design variables selected to optimize, the convergence criteria that limited the global search in favor of maintaining computational stability, and the results to-date only represent a work-in-progress. The intent of this work was to present this new transient A/C system analysis and optimization tool, and show some high-level system design conclusions reached to date. Future work will simultaneously optimize more system variables with more sophisticated objective functions, involving system weight, cost, cabin cool-down time, and others, in identifying optimum vehicle A/C system designs.

ADVISOR INTEGRATION

NREL is now integrating the transient A/C system model described above into its ADVISOR vehicle system analysis software. Integration of the transient A/C system model into ADVISOR represents a subset of NREL's Digital Functional Vehicle (DFV) project that

Table 1 – Optimum Orifice Diameter, Transfer Line Diameter, & Condenser Line Diameter Maximizing COP During the SC03 Drive Cycle (Optimums in Red)

Run #1 Loop #	Orifice Diameter (Inch)	Transfer Line Diameter (Inch)	Condenser Line Diameter (Inch)	Ave. COP	COP Ratio
1	0.066	0.5000	0.212796	1.087	0.9811
67	0.08623	0.78877	0.27054	1.108	1
70	0.09950	0.85126	0.274356	1.107	0.9998

Run #2 Loop #	Orifice Diameter (Inch)	Transfer Line Diameter (Inch)	Condenser Line Diameter (Inch)	Ave. COP	COP Ratio
1	0.07000	0.4500	0.220704	1.144	0.9825
99	0.08689	0.7953	0.230556	1.164	1

Run #3** Loop #	Orifice Diameter (Inch)	Transfer Line Diameter (Inch)	Condenser Line Diameter (Inch)	Ave. COP	COP Ratio
1	0.07000	0.4500	0.220704	1.141	0.9934
26	0.07386	0.46978	0.22908	1.149	1
38	0.07384	0.46109	0.228924	1.149	0.9998

** Different Solution Convergence Criteria Used In This Run

Run #4 Loop #	Orifice Diameter (Inch)	Transfer Line Diameter (Inch)	Condenser Line Diameter (Inch)	Ave. COP	COP Ratio
1	0.07000	0.4500	0.220704	1.143	0.9949
20	0.07138	0.46853	0.235308	1.149	0.9998
92	0.07037	0.49045	0.211104	1.149	1

Run #5 Loop #	Orifice Diameter (Inch)	Transfer Line Diameter (Inch)	Condenser Line Diameter (Inch)	Ave. COP	COP Ratio
1	0.07000	0.4800	0.219	1.143	0.982
86	0.05800	0.78593	0.1119	1.164	0.9998
137	0.05863	0.75242	0.149436	1.164	1

Table 2 – Optimum Orifice Diameter, Transfer Line Diameter, & Condenser Line Diameter Maximizing COP During the US06 Drive Cycle (Optimum in Red)

Run #1 Loop #	Orifice Diameter (Inch)	Transfer Line Diameter (Inch)	Condenser Line Diameter (Inch)	Ave. COP	COP Ratio
1	0.066	0.5000	0.212796	0.984	0.982
86	0.10927	0.81491	0.18648	1.002	1

Run #2 Loop #	Orifice Diameter (Inch)	Transfer Line Diameter (Inch)	Condenser Line Diameter (Inch)	Ave. COP	COP Ratio
1	0.06800	0.5000	0.2148	0.993	0.9971
18	0.07220	0.53995	0.22806	0.996	1

Run #3 Loop #	Orifice Diameter (Inch)	Transfer Line Diameter (Inch)	Condenser Line Diameter (Inch)	Ave. COP	COP Ratio
1	0.07000	0.4500	0.22070	0.982	0.9697
99	0.05800	0.81282	0.23498	1.013	1

Run #4 Loop #	Orifice Diameter (Inch)	Transfer Line Diameter (Inch)	Condenser Line Diameter (Inch)	Ave. COP	COP Ratio
1	0.07000	0.4500	0.220704	0.982	0.9741
74	0.09392	0.65906	0.25176	1.008	1
81	0.08072	0.62189	0.2415	1.007	0.9996

Run #5 Loop #	Orifice Diameter (Inch)	Transfer Line Diameter (Inch)	Condenser Line Diameter (Inch)	Ave. COP	COP Ratio
1	0.07000	0.4800	0.2193	0.988	0.9798
100	0.10098	0.66446	0.23984	1.008	1

intends to virtually simulate many of 1st-order energy-management and emissions-producing mechanisms in the vehicle design process. DFV creates a virtual vehicle design environment that can shorten vehicle design cycle times, reduce the number of required test prototypes, and produce more optimized vehicle designs. The SINDA/FLUINT analysis software and the ADVISOR vehicle system analysis software contain built-in optimization capabilities that will optimize the vehicle A/C system within the overall vehicle design optimization process. This will allow NREL to simultaneously optimize the A/C system with other vehicle systems, such as passenger cabin systems, engine coolant systems, energy recovery systems, and other dynamic systems.

This transient A/C system model has also been integrated with NREL's newly-developed vehicle solar thermal load simulator program that can predict transient vehicle solar thermal loads for a variety of vehicle configurations. The solar load simulator can predict the transient solar thermal loads in the vehicle cabin for any vehicle drive direction, varying geographic locations, and a number of different vehicle glass packages. It writes out the predicted solar thermal load as a function of time into a file which the transient A/C system model automatically reads real-time during an analysis run.

NREL can now incorporate vehicle dynamic solar loads into any transient A/C system simulation or optimization.

CONCLUSIONS

NREL has developed a robust and flexible transient vehicle air conditioning model, integrated with a simplified cabin thermal environment model, in the SINDA/FLUINT analysis software environment. It simultaneously models the entire A/C system and its components for various external vehicle environments and drive cycle conditions. It has demonstrated capability to predict transient system pressures and temperatures, mass flow rates, flow quality and flow regime conditions, transient compressor power, transient evaporator and condenser heat loads, and transient cabin thermal conditions during any user-specified vehicle drive cycle conditions. The model currently focuses on R-134a A/C systems, but future work is planned to investigate advanced alternative refrigerant systems such as carbon dioxide systems.

The transient AC/cabin model has been used to perform single-variable and multiple variable optimization of A/C system performance during SC03 and US06 drive cycles. Because of the strong inter-dependencies amongst A/C system and cabin design variable impacts, the results have demonstrated that multiple variable optimization is critical to truly optimizing A/C system performance for various drive cycles. Since various drive cycles can produce different optimum system designs, sophisticated optimization strategies must be developed simultaneously incorporating the effects of different drive cycles in the system optimization process.

Model development continues to increase its analytic power and flexibility while the SINDA/FLUINT model is integrated within NREL's ADVISOR vehicle system analysis software. Future plans include performing A/C system design optimization within overall vehicle design optimizations using ADVISOR and expanding the model's use to heavy vehicle applications. Future efforts also will investigate additional system/vehicle design optimizations using other combinations of design variable objective functions, which might involve design parameters such as system cost, system weight, and transient cabin cool-down speed. The transient A/C model is now also integrated with NREL's vehicle solar thermal load simulator that predicts transient solar thermal loads on a vehicle for various drive directions, geographic locations, vehicle glass packages, and vehicle configurations.

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NOMENCLATURE

English

A_n - Coefficients of Compressor Isentropic Efficiency
B_n - Coefficients of Compressor Volumetric Efficiency
$\overline{C}_{p,i}$ - Average Specific Heat of i^{th} System Component
COP - System Coefficient of Performance
COP_{max} - Maximum System Coefficient of Performance
m_i - Mass of i^{th} System Component [lbm or kg]
P - Compressor Power (time-dependent) [Btu/hr or Watts]
P_r - Compressor Pressure Ratio
q - Evaporator Heat Load (time-dependent) [Btu/hr or Watts]
R_r - Reduced Compressor RPM (RPM/1000) [rev/min]
t - Time [seconds or hours]
T - Temperature [^oC]

Greek

β_i - Compressor Isentropic Efficiency
β_v - Compressor Volumetric Efficiency
η - System Coefficient of Performance

Supercripts

$\wedge$ - Time-averaged quantity

APPENDIX – SUPPLEMENTAL FEDERAL TEST PROCEDURE DRIVE CYCLES