A/C Model Development and Validation



P.I.: Jason A. Lustbader
National Renewable Energy Laboratory

Team: Tibor Kiss and Larry Chaney

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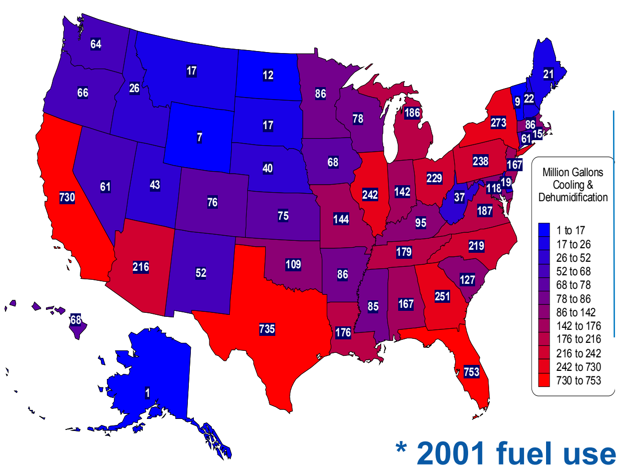
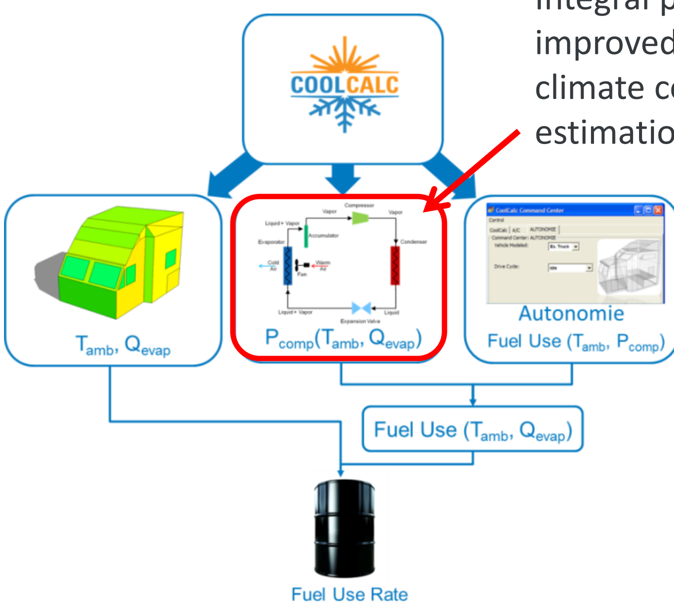
Background

- **When operated, the air conditioning (A/C) system is the largest auxiliary load**
- **A/C loads account for more than 5% of the fuel used annually for light-duty vehicles (LDVs) in the United States[1]**
- **A/C load can have a significant impact on electric vehicle (EV), plug-in hybrid electric vehicle, and hybrid electric vehicle performance**
  - Mitsubishi reports that the range of the i-MiEV can be reduced by as much as 50% on the Japan 10–15 cycle when the A/C is operating[2]
  - Hybrid vehicles have 22% lower fuel economy with the A/C on[3]
- **Increased cooling demands by an EV may impact the A/C system**
- **A/C contributes to heavy-duty vehicle idle and down-the-road fuel use**



*** 2001 fuel use data**

Integral part of improved high-level climate control estimation process



1. Rugh et al., 2004, Earth Technologies Forum/Mobile Air Conditioning Summit
2. Umezu et al., 2010, SAE Automotive Refrigerant & System Efficiency Symposium
3. Idaho National Laboratory, Vehicle Technologies Program 2007 annual report, p145.

Overview

Timeline

Project Start Date: FY11
Project End Date: FY13
Percent Complete: 80%

Budget

Total Project Funding:

DOE Share: $900K
Contractor Share: $0k

Funding Received in FY12: $300K

Funding for FY13: $300K

Barriers

- Cost – *Timely evaluation of HVAC systems to assist with R&D*
- Computational models, design and simulation methodologies – *Develop tool to help with optimization of future HVAC designs and prediction of impacts on fuel economy*
- Constant advances in technology – *Assist industry advance technology with improved tools*

Partners

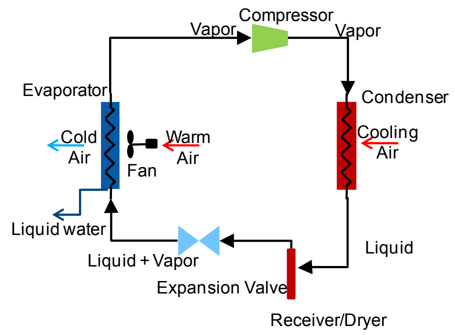
- Collaborations
  - Halla Visteon Climate Control (Visteon)
  - Argonne National Laboratory (ANL)
  - Daimler Trucks
- Project lead: NREL

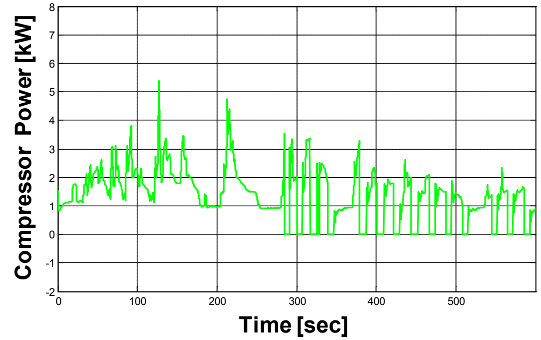
Relevance/Objectives

- Overall Objectives
  - Develop analysis tools to assess the impact of technologies that reduce the thermal load, improve the climate control efficiency, and reduce vehicle fuel consumption
  - Develop an open source, accurate, and transient A/C model using the Matlab/Simulink environment for co-simulation with Autonomie
  - Connect climate control, cabin thermal, and vehicle-level models to assess the impacts of advanced thermal management technologies on fuel use and range
- FY12/13 Objectives
  - Improve mechanical LDV A/C model and validate
  - Add electrical compressor capability and associated controls
  - Develop simplified model options for more rapid, less detailed analysis, with a focus on vehicle co-simulation with Autonomie
  - Demonstrate co-simulation of A/C system with Autonomie
  - Develop heavy-duty vehicle sleeper and cab A/C system models
  - Release A/C model plug-in for Autonomie

Milestones, FY12-FY13

Date	Milestone or Go/No-Go Decision
04/01/2012	Delivered stand-alone model to Visteon
06/14/2012	Delivered electric A/C model to ANL
06/01/2012	Completed initial validation
09/30/2012	Completed summary report and first release of the A/C model
04/15/2013	Autonomie integrated model released
04/16/2013	SAE World Congress paper "A New Automotive Air Conditioning System Simulation Tool Developed in MATLAB/Simulink," SAE 2013-01-0850
09/30/2013	Summary report and second release of the A/C model





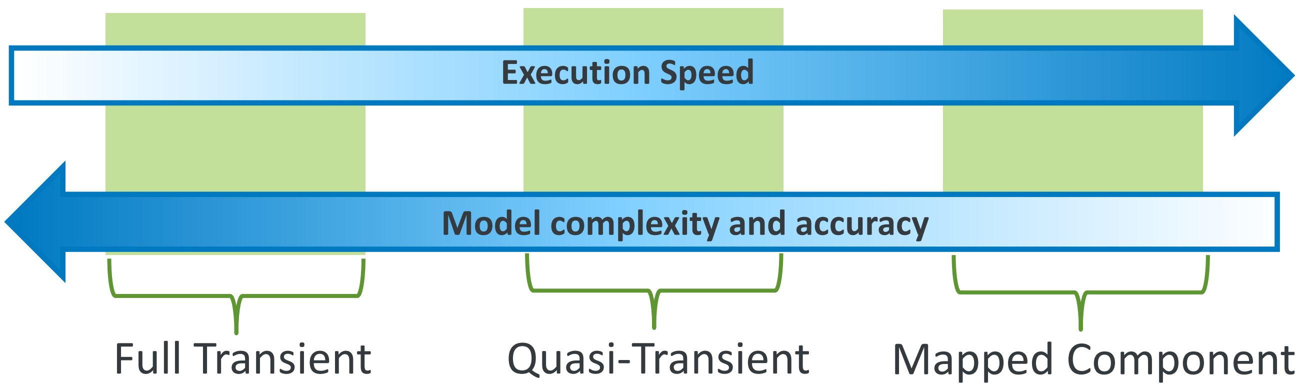
Approach – Matlab/Simulink-Based Tool

- Base a simulation tool on first principles; conservation of mass, momentum, and energy are solved in 1-D finite volume formulation
- Create open source software tools and make them available to the public
- Easily interface to Autonomie vehicle simulation tool
- Develop flexible software platform, capable of modeling vapor compression refrigeration cycle
- Model refrigerant lines and the heat exchangers as 1-D finite volumes, accounting for the lengthwise distribution of refrigerant and flow properties
- Include all major components: compressor, condenser, expansion device, evaporator, and accumulator/dryer (receiver/dryer)
- Provide model options with a range of run times while minimizing the impact of increasing speed on accuracy to meet a range of analysis needs

Approach: Three Model Versions

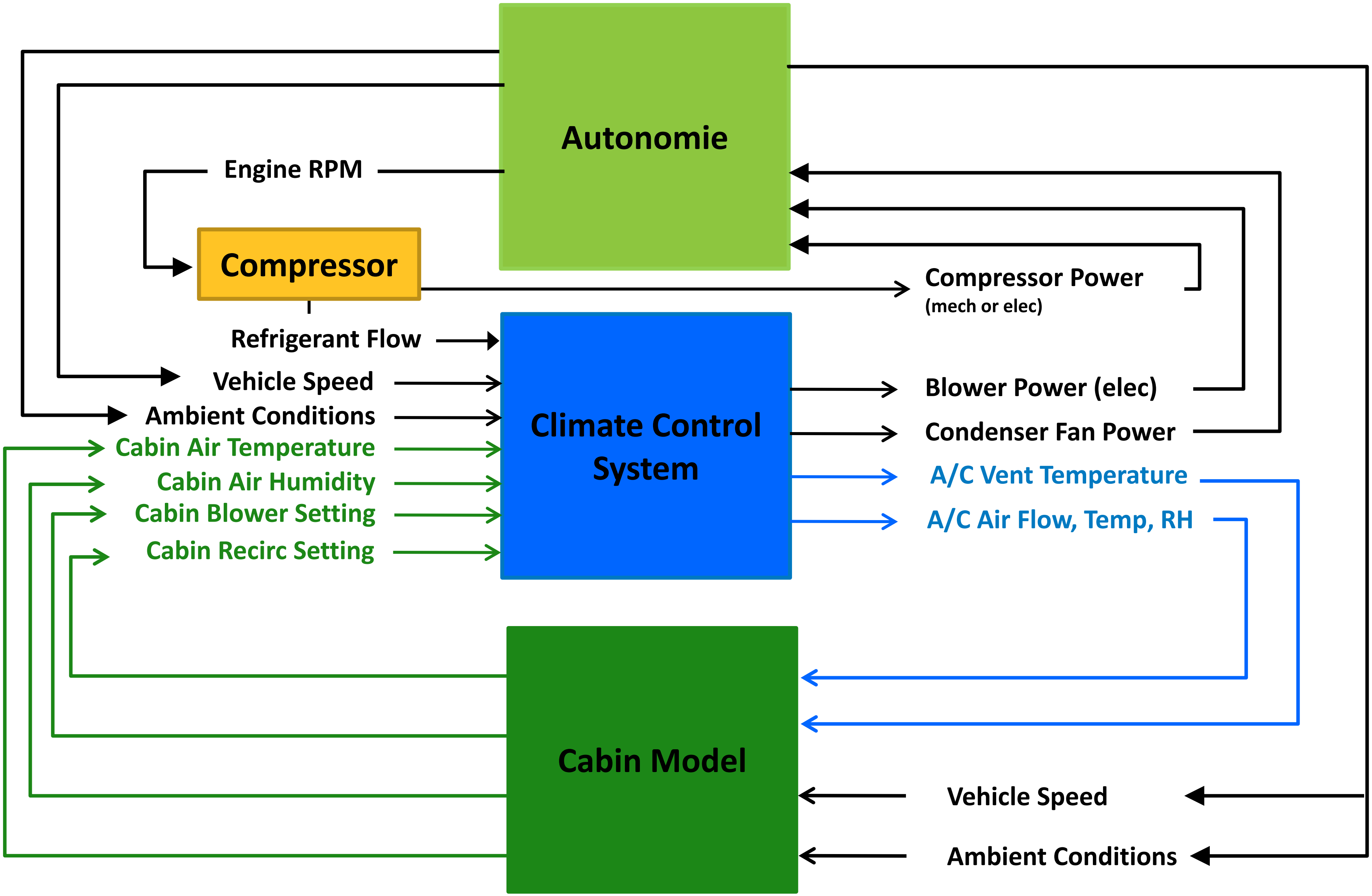
Serving Different Customer Needs

Model Type	Application	Speed	Accuracy
Full Transient *(finite volume, fully conservative)*	Detailed A/C models for design and control	1/12th of real time	Highest, time-resolved
Quasi-Transient *(simplified refrigerant volumes)*	Detailed vehicle co-simulation and created mapped components	Real time	Moderate
Mapped Component *(simplified refrigerant volumes and heat exchangers)*	High level co-simulation with a vehicle focus	10 X real time (estimated)	Lowest



Approach: Climate Control System Integration with Autonomie

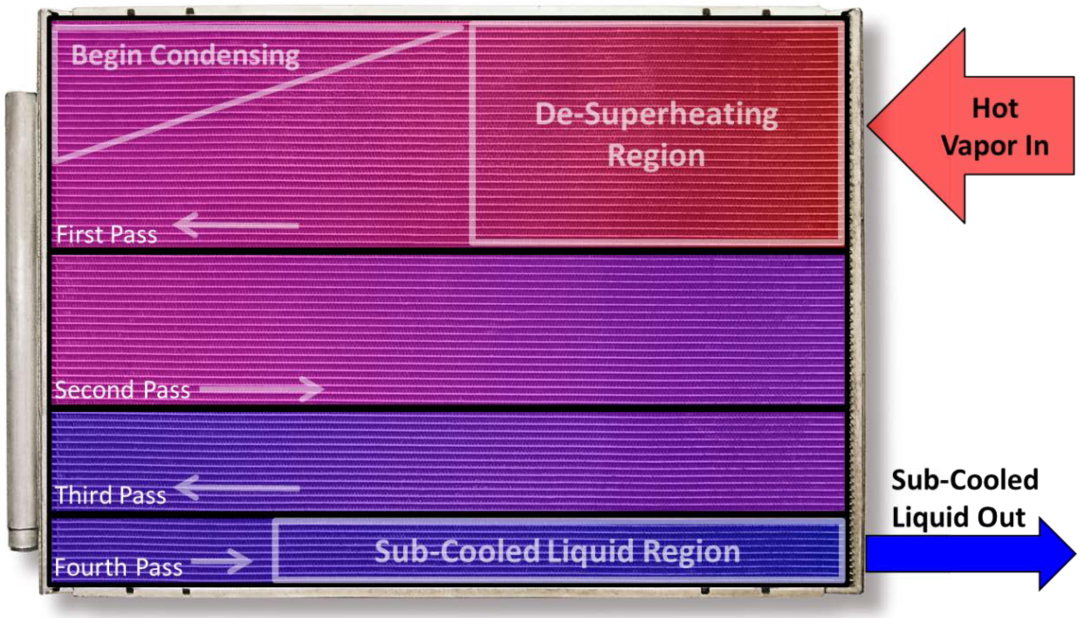
Enables co-simulation with vehicle models



A/C Model Development

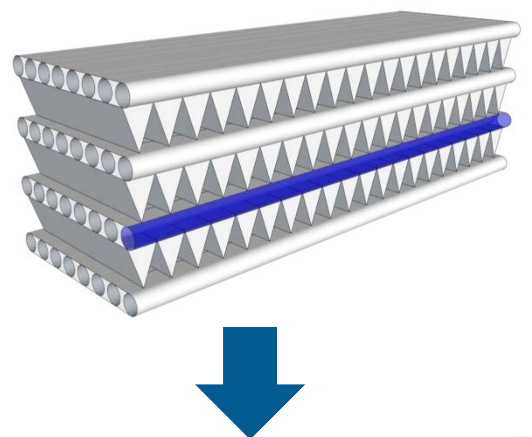
Development of Component Models, Heat Exchanger

Four refrigerant passes in this example

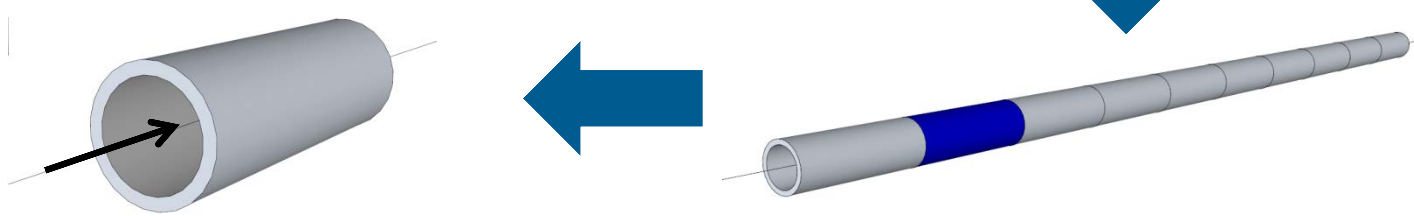


Complex heat exchanger

- Multiple passes
- Multi-channel tubes
- Micro channels
- Multiple refrigerant phases



Conservation Equations Solved in Refrigerant Lines



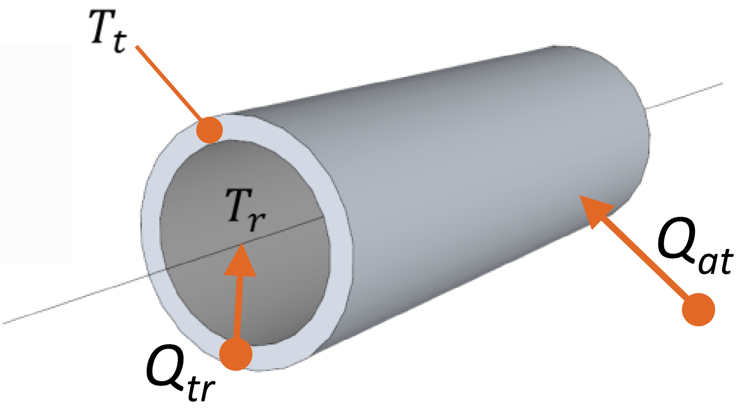
- Four refrigerant passes become four flow paths in this example
- Each flow path is divided into many segments, or finite volumes
- The 1-D finite volumes account for the lengthwise distribution of refrigerant and flow properties

Accomplishments: Heat Transfer

Improved heat transfer and fin heat loss calculations

Pipe wall to refrigerant

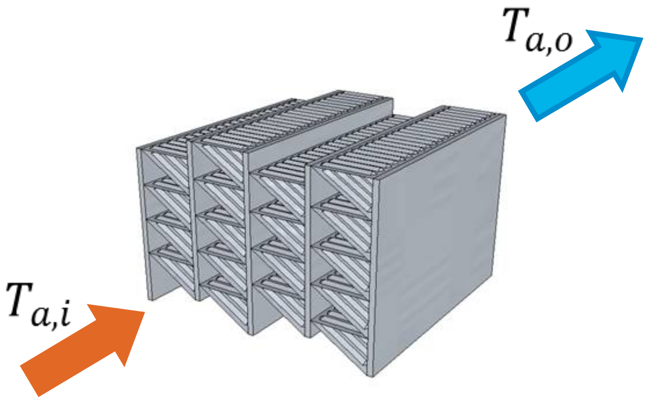
$$Q_{tr} = \bar{h}_{tr} A_t (T_t - T_r)$$



Heat transfer from air to pipe wall[3]

$$Q_{at} = (\dot{m}_a \cdot C_{p,adry} + \dot{m}_w \cdot C_{p,w}) \cdot (T_{a,o} - T_{a,i})$$

$$T_{a,o} = T_{a,i} + (T_t - T_{a,i}) \cdot \left[1 - \exp\left(\frac{-\bar{h}_a\, A}{\dot{m}_a \cdot (C_{p,adry} + \omega\, C_{p,w})}\right)\right]$$



Calculation assumptions:

- h_{wr} obtained from Dittus-Boelter equation and Chen correlation
- *h_a obtained through correlations for louver fin compact heat exchangers[1,2]*
- Fin effectiveness calculated using Number of Transfer Units (NTU) method
- Pipe modeled as radially isothermal, contains thermal mass
- Saturated mixture refrigerant properties are quality averaged values of sat. liquid and sat. vapor
- System accounts for possible water condensation in the air stream

1. Chang, Y.J., and Wang, C.C., "A Generalized Heat Transfer Correlation for Louver Fin Geometry," *Int. J. Heat Mass Transfer,* Vol. 40, No. 3, pp. 533-544, 1997
2. Chen, J.C. (1966). "A Correlation for Boiling Heat Transfer of Saturated Fluids in Convective Flow," *Ind. Eng. Chem. Process Ses. Dev.,* Vol. 5, No. 3, pp. 322-329.
3. See nomenclature slide at end of presentation

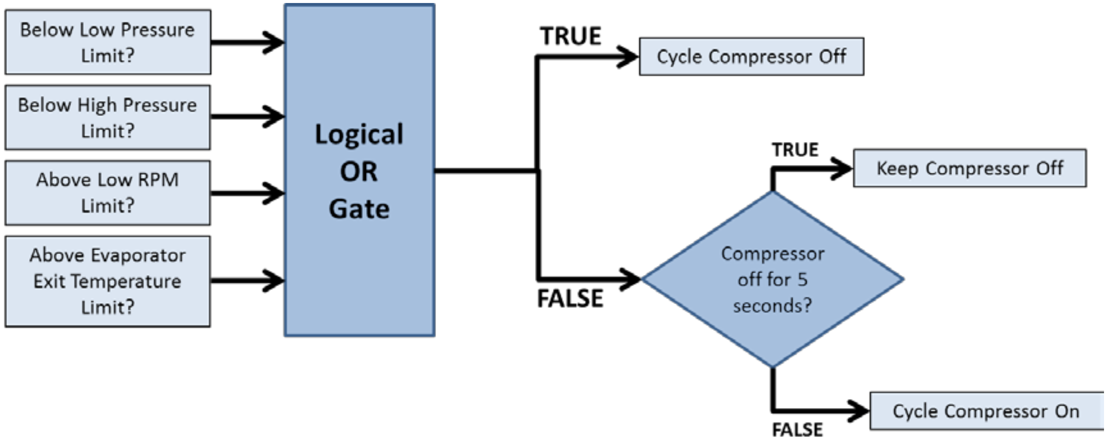
Accomplishments: Compressor

Added electric compressor and associated controls

- **Compressor, general**
  - Mechanical (piston) or electrical (scroll), electrical added this year
  - Volumetric efficiency
  - Discharge enthalpy found using isentropic efficiency
- **Electric compressor**
  - RPM controlled by $T_{wall,evap,exit}$ (metal T)
  - Blower air mass flow rate controlled by $T_{air,cabin}$
  - No windup PI controllers implemented
  - If compressor RPM command goes below limit, compressor cycles off. When compressor comes back, it starts up near this limit

[1]

$$\dot{m} = \rho_u \cdot \eta_{vol} \frac{dV}{rev} \cdot RPM/60$$

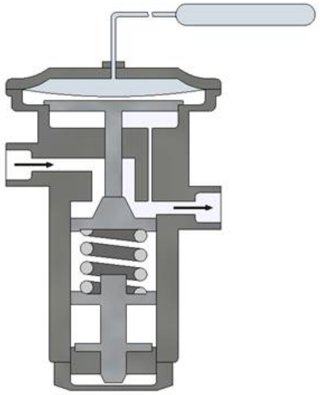


[1] Compressor photograph, NREL, John Rugh & Jason Lustbader

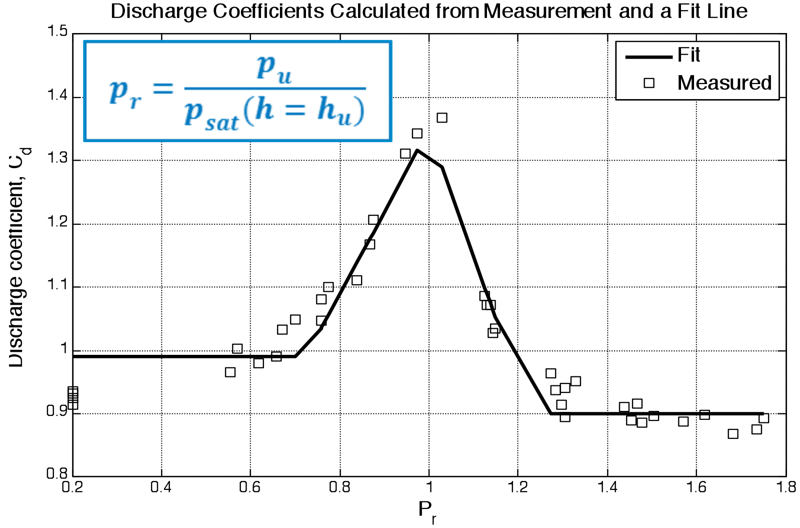
Accomplishments: Thermal Expansion Device (TXV)

Semi-dynamic model improves TXV accuracy

- **Thermal Expansion Device (TXV)**
  - Two-phase equilibrium orifice flow model
  - Capturing flow area dependence on evaporator-out superheat
- **Semi-dynamic[1] model addresses response time issues**
  - Valve ball position determined from static force balance
  - One dynamic factor – bulb temperature response to evaporator exit temperature
  - Response is fast to pressure differences but slow to temperature changes – just like in a real TXV



$$\dot{m} = C_d(\mathrm{d}P_e) \cdot \rho_{throat} \cdot v_{throat} \cdot A_{orif}$$



Discharge coefficient from experimental data accounts for non-equilibrium effects

[1] *This was found to have superior performance to a full dynamic model, which was also developed*

Accomplishments: Component Validation

Validation data cover wide range of operating conditions

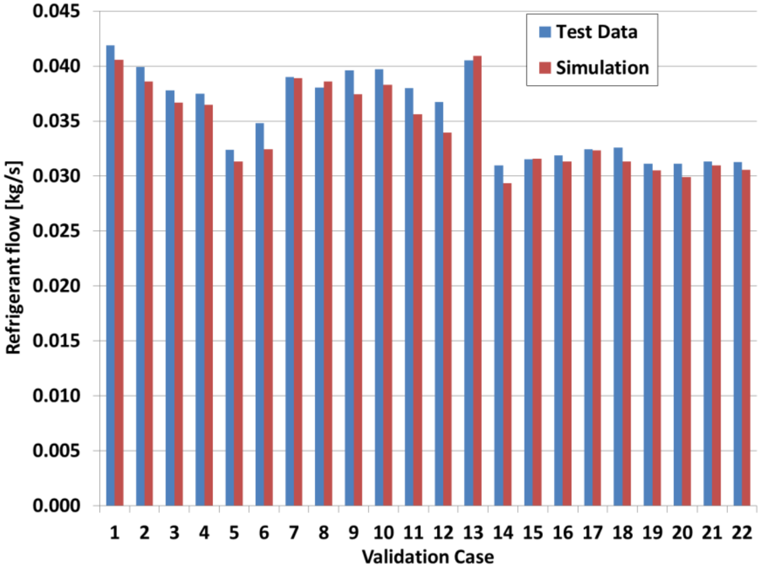
- Model results compared to 22 steady-state experimental bench data points provided by Visteon
- Test points cover a wide range of operating conditions

Range of Bench Test Data			
	Low	High	Units
Vehicle speed	0	112	km/h
Ambient air temperature	21	43	°C
Relative humidity	25	40	%
Evaporator air inlet temperature	10	43	°C
Evaporator air flow	0.042	0.137	m^3/s

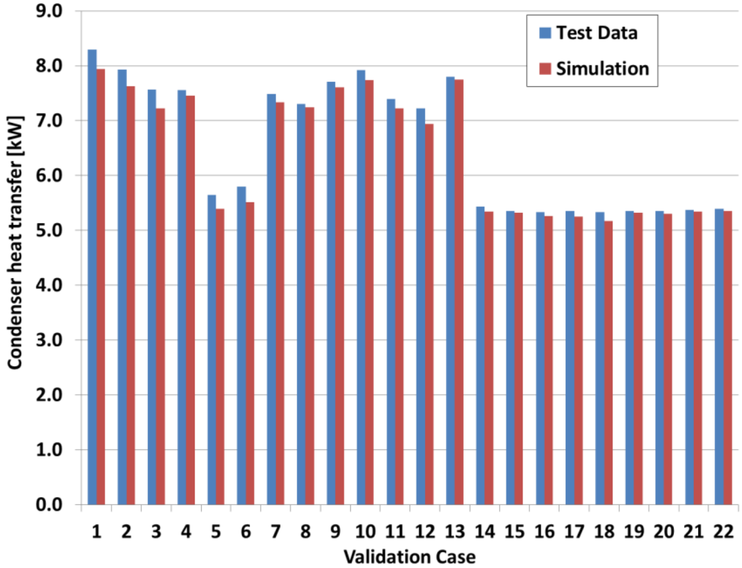
Accomplishment – Component Validation

Improvements to model resulted in better agreement with data

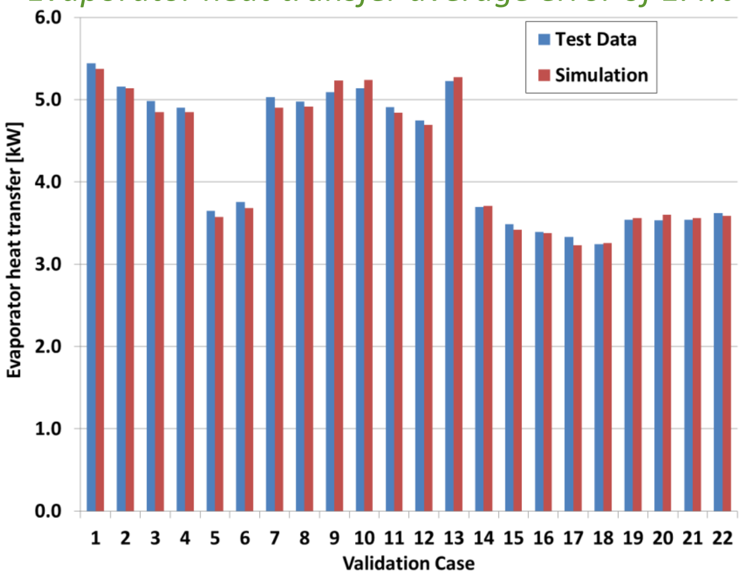
Refrigerant flow rate average error of 3.1%



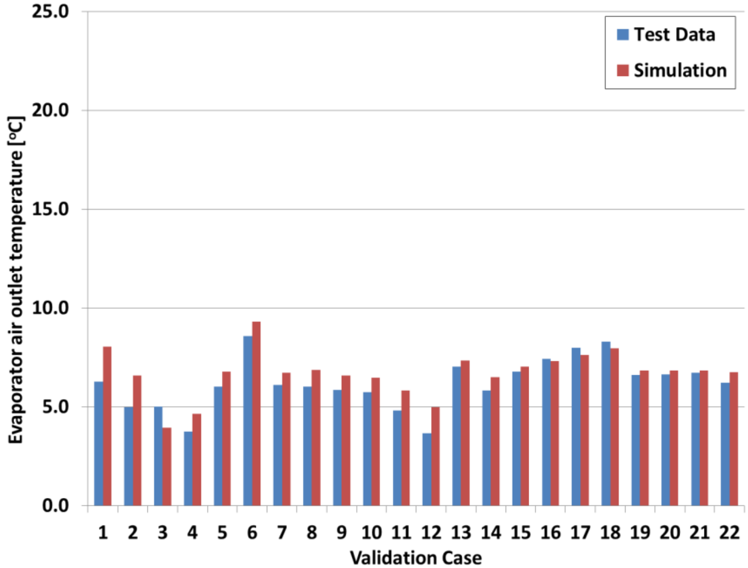
Condenser heat transfer average error of 2.2%



Evaporator heat transfer average error of 1.4%



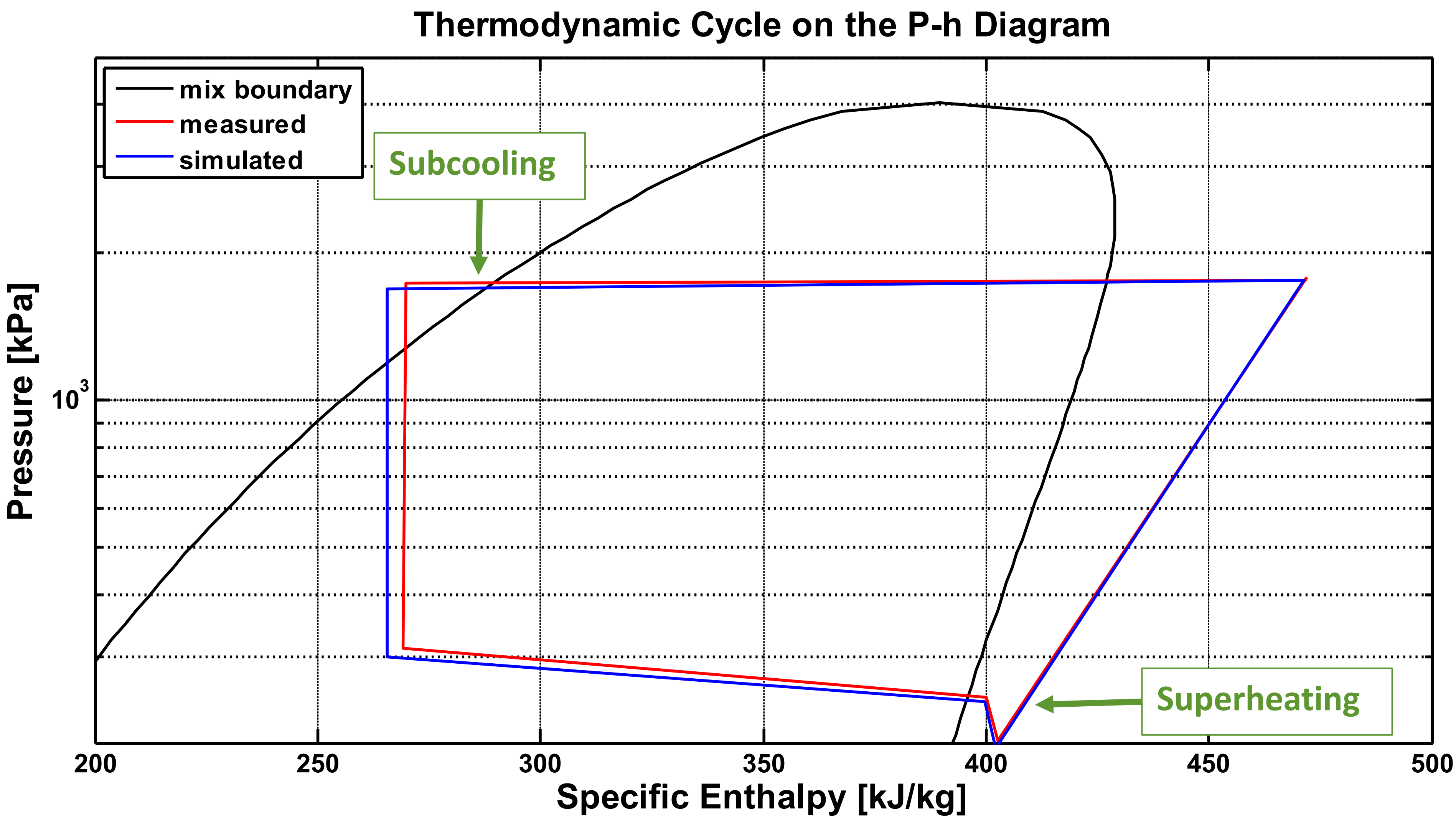
Evaporator air outlet temperature error of 2.9%



Accomplishment – System Validation, Typical Point

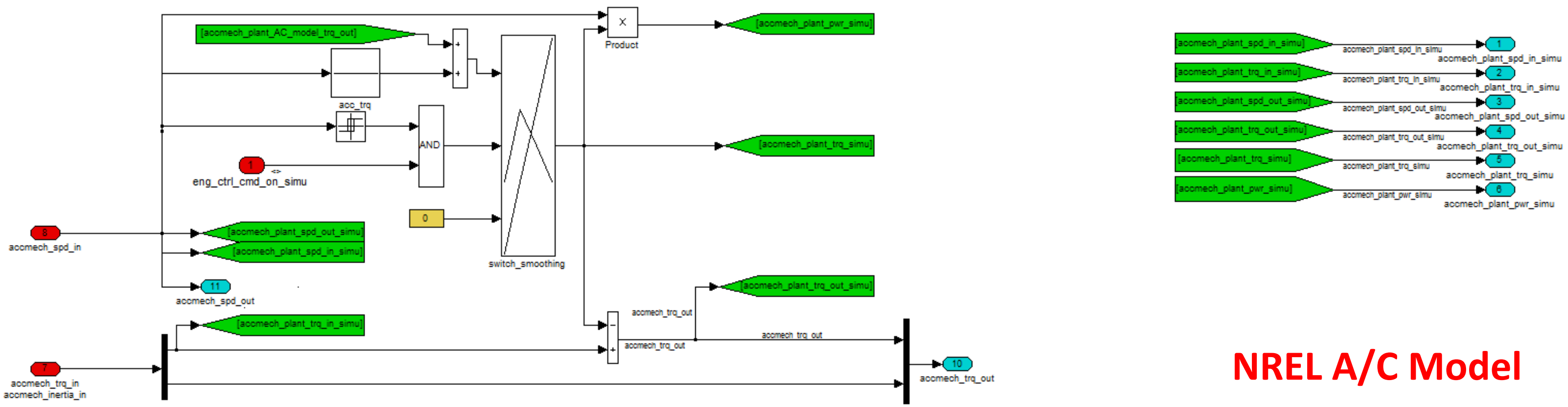
Good Agreement for System Thermodynamic Cycle

Full transient model

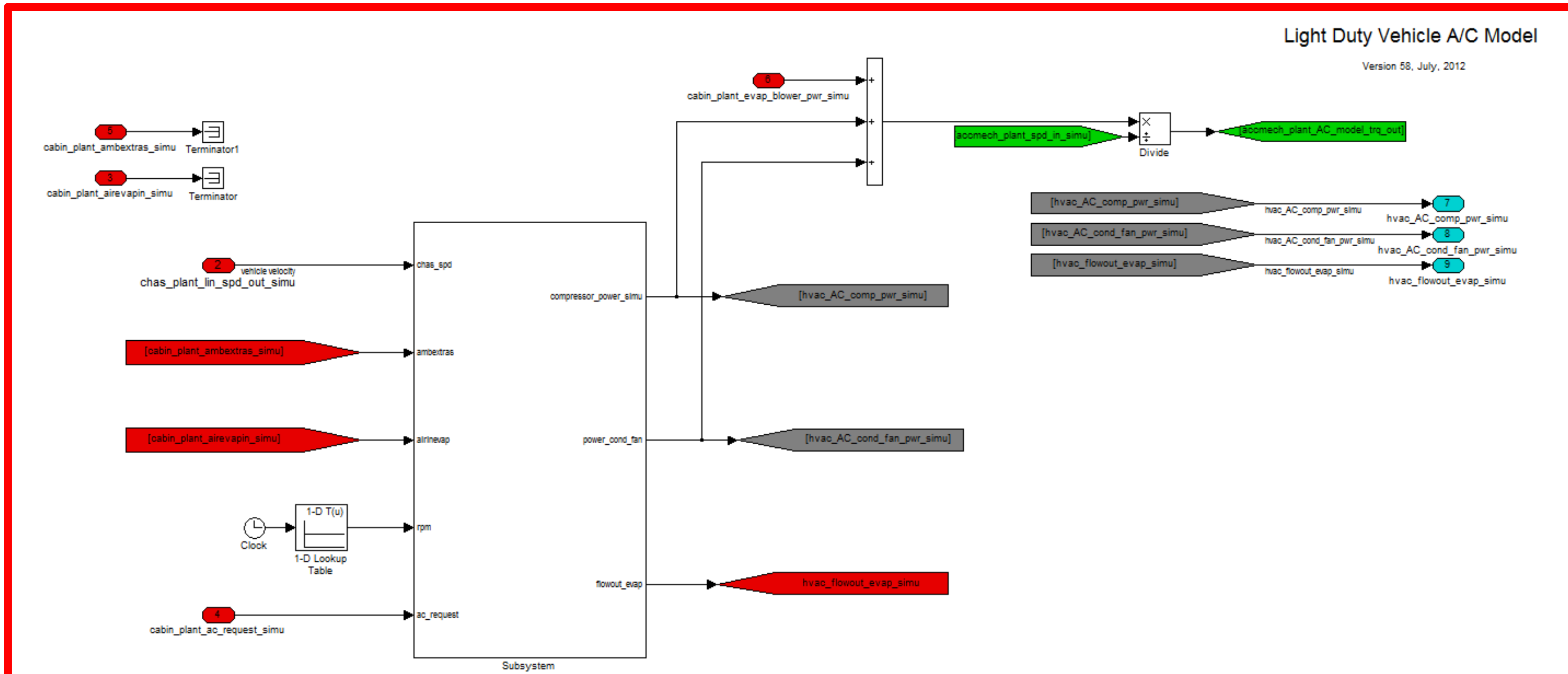


Accomplishments – Autonomie Integration

Top-level model, adjusted code for better integration with next Autonomie release

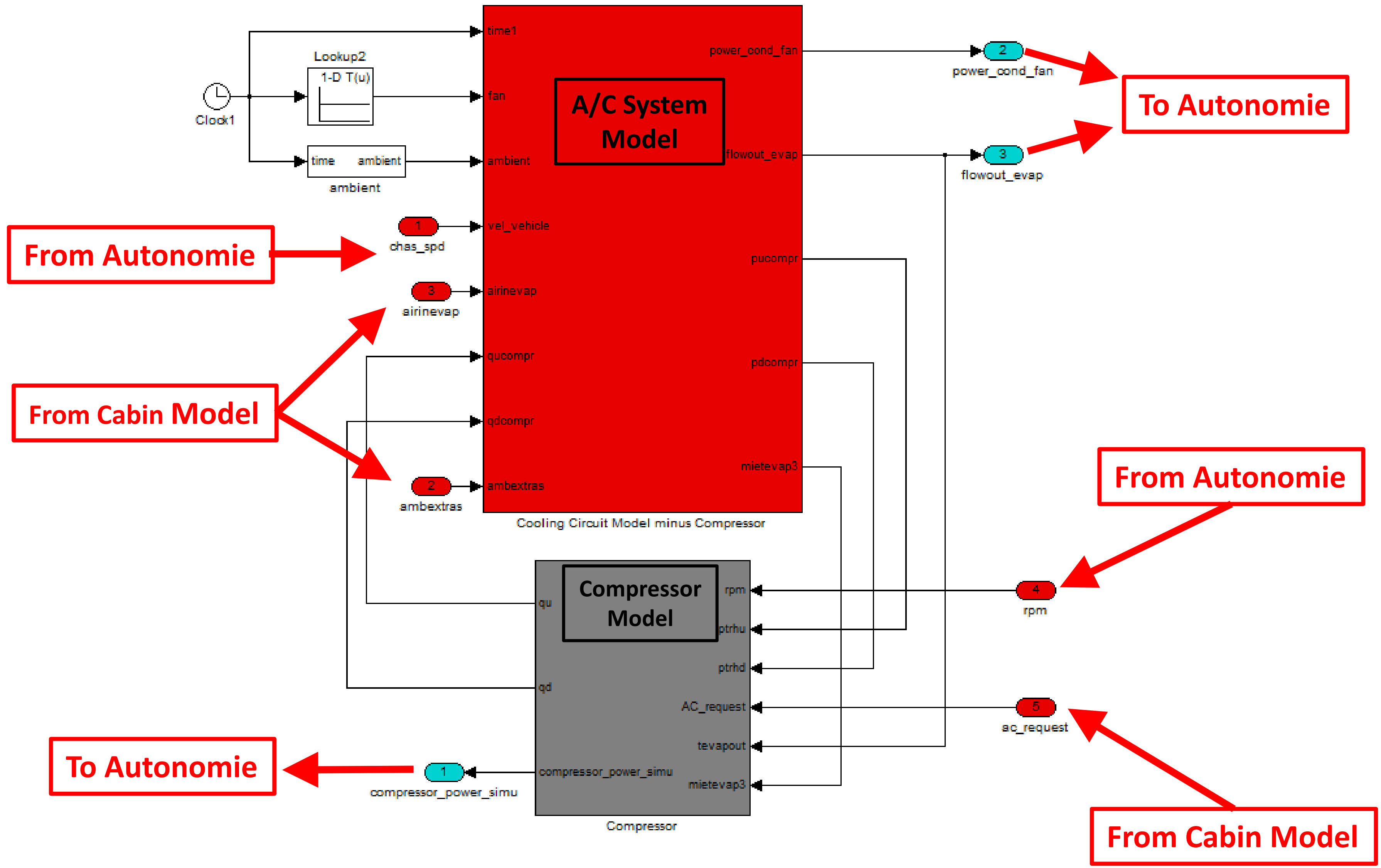


NREL A/C Model



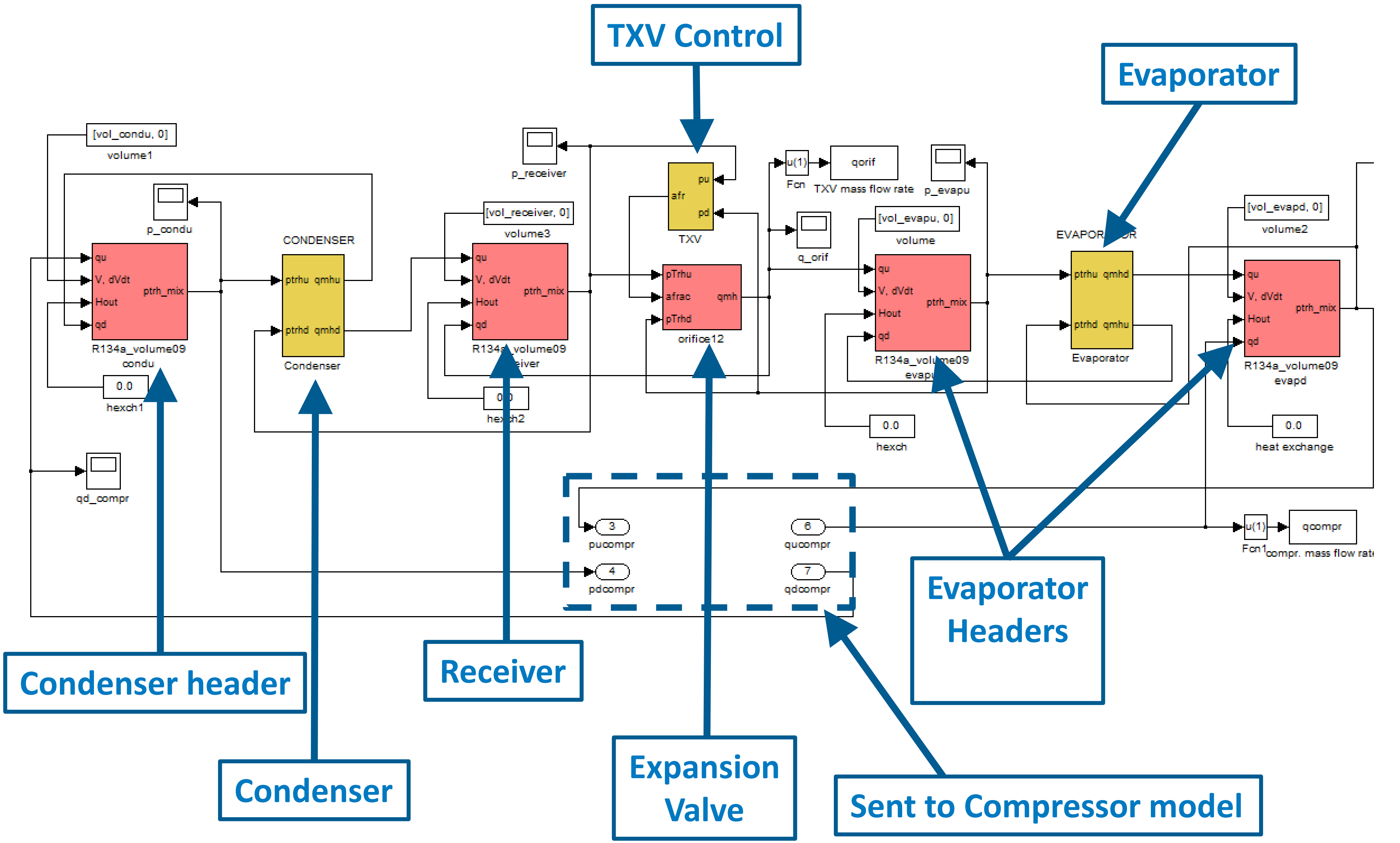
Accomplishments – Autonomie Integration

Second-level model: Compressor made separate and cabin moved to chassis



Accomplishments – Autonomie Integration

Third-level A/C model: Components, compressor separated



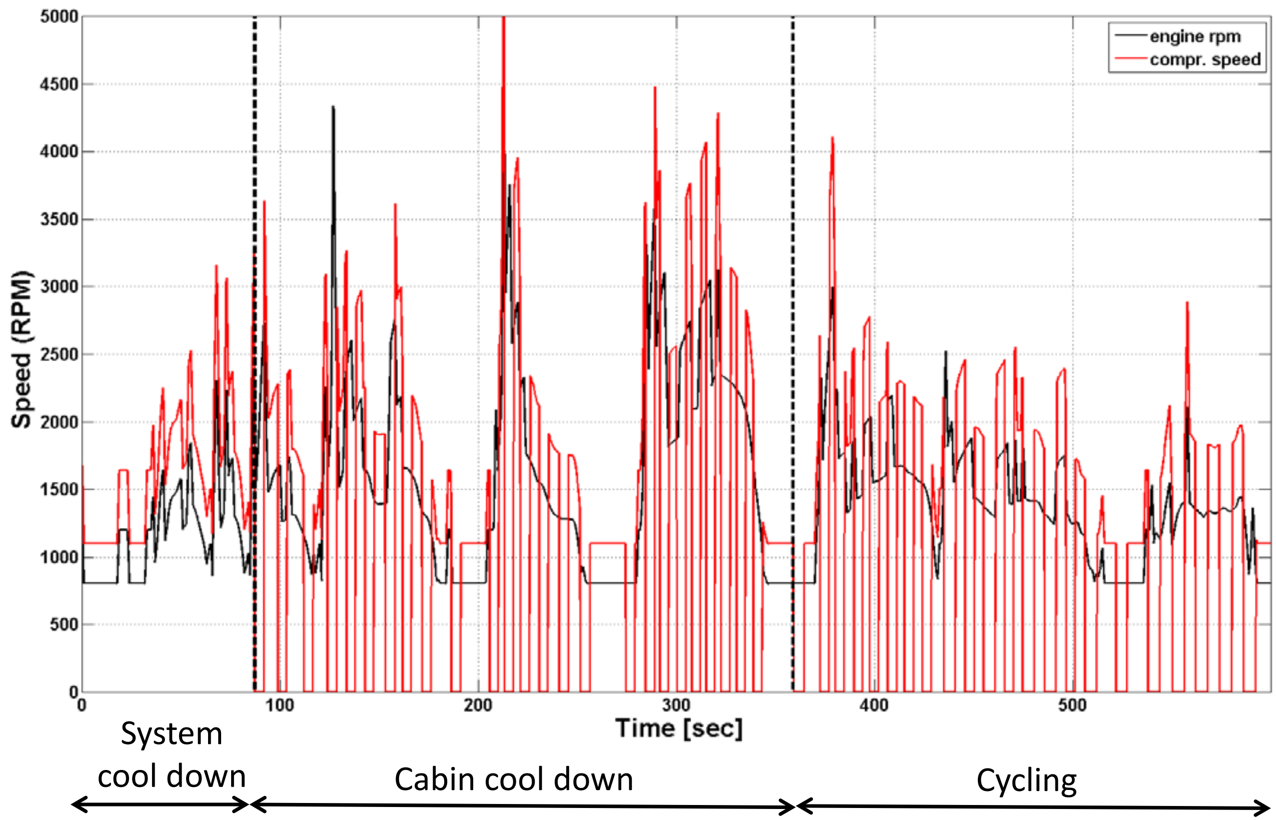
Accomplishments – SC03 Cycle

System model SC03 example

- **Simulated the A/C system over drive cycle**
  - Used SC03 drive cycle
  - Conventional 2wd Midsize Auto Default in Autonomie
  - Demonstrated robust system performance and cabin cooldown

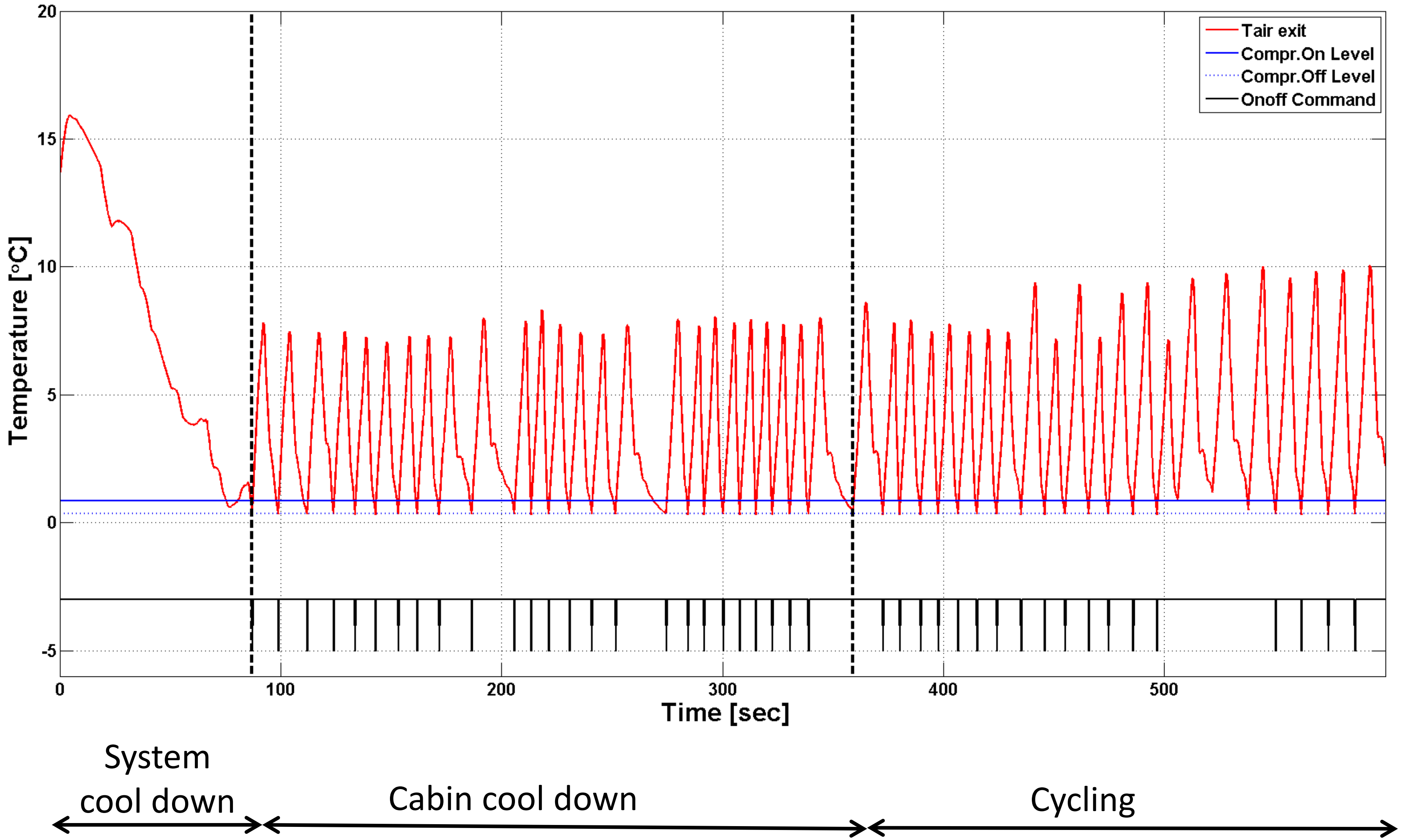
Conditions and Controls Settings

Variable	Value	Units
Ambient Temperature	30	°C
Cabin initial relative humidity	40	%
Solar load	1000	W
Cabin target temperature	20	°C
Air recirculation	90	%



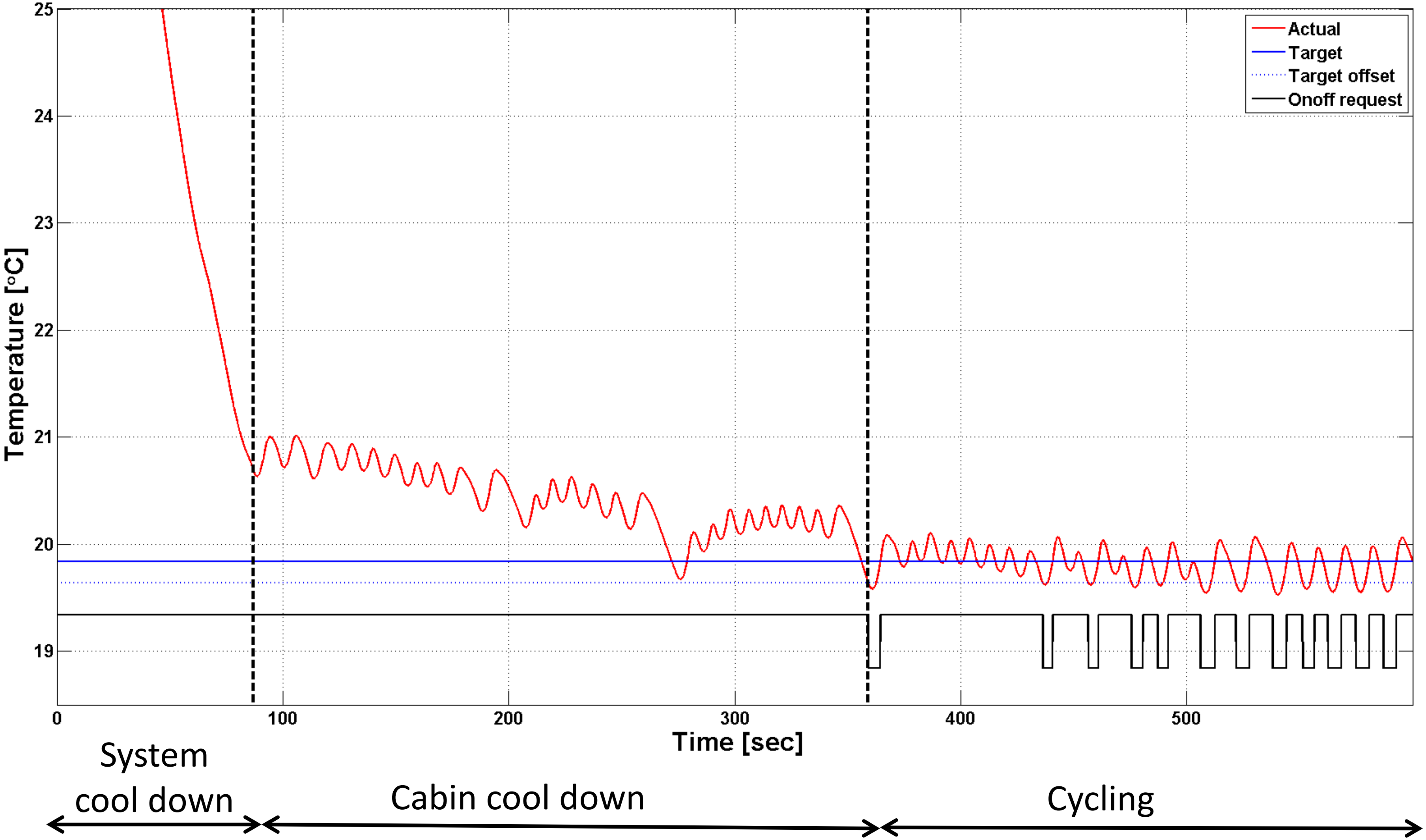
Accomplishments – SC03 Cycle Evaporator Temperature Control

Evaporator freeze protection control reached in 87 sec



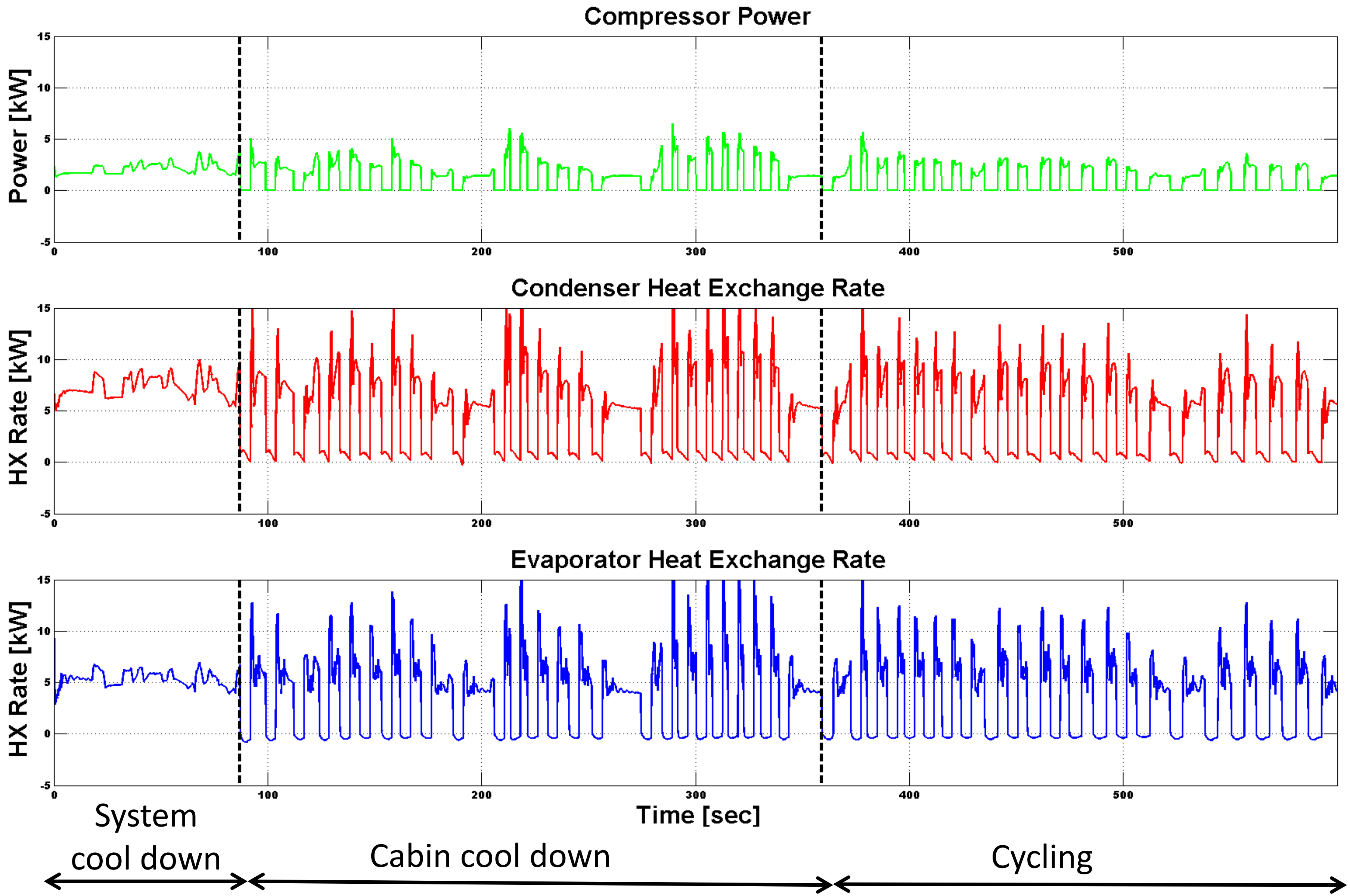
Accomplishments – SC03 Cycle Cabin Temperature Control

Cabin temperature control reaches set point in 359 sec



Accomplishments – SC03 Cycle Heat and Compressor Power

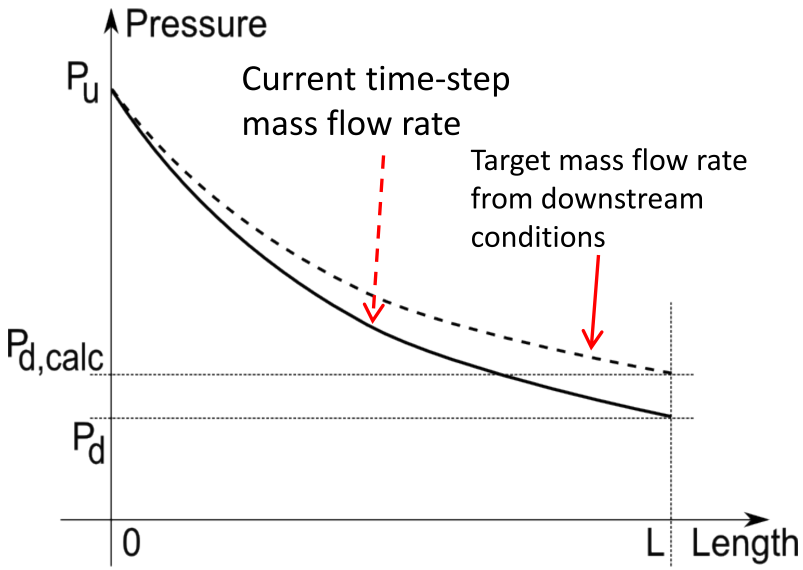
Dynamic thermal and mechanical power captured



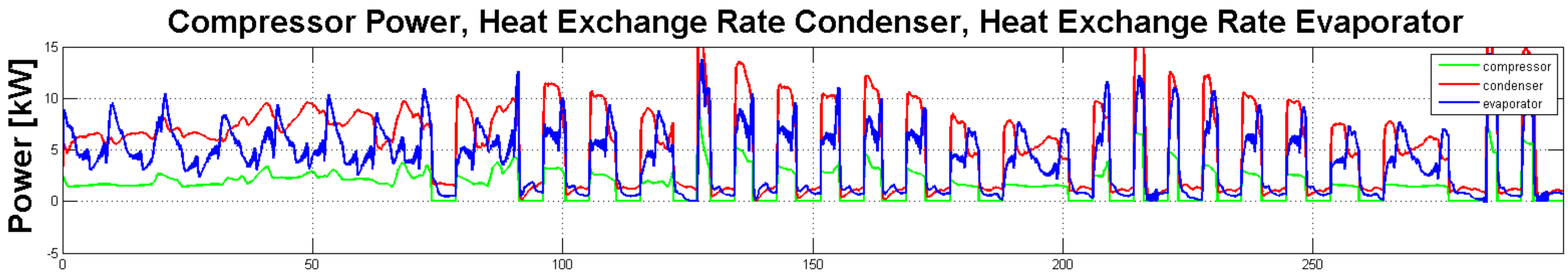
Accomplishments – Quasi-Transient Model

Simplifications to increase maximum time step and thus speed by 12X

- **Only refrigerant line and 0-D volume simulation blocks modified**
- **Modifications allow larger simulation time steps and thus faster execution speed**
- **Changes to refrigerant line blocks**
  - Refrigerant side formulation no longer finite volume, algebraic marching scheme used
  - Mass flow rate
    - Same in all the segments of the line
    - Only state variable (calculated from its time derivative through an integration step)
  - Allows larger simulation time step
- **Changes to 0-D volume blocks**
  - Mass and energy are preserved
  - A modified bulk modulus is used (compressibility adjusted) to calculate the pressure in the volume
  - Allows for a larger time step

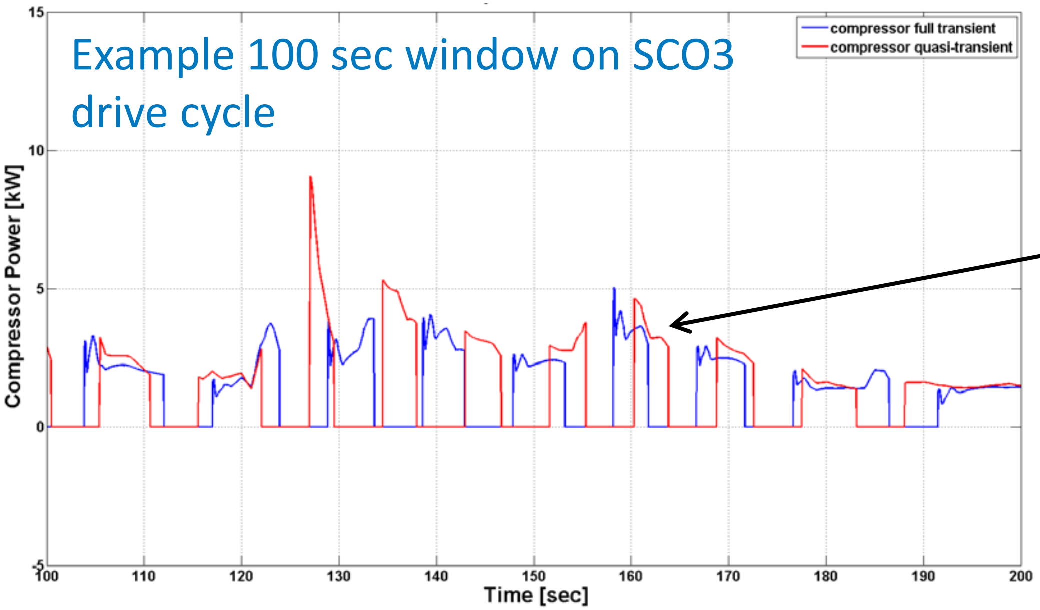


$$\dot{m} = C \cdot \frac{(P_{d,calc} - P_d)}{(P_u - P_d)} \cdot \dot{m}$$



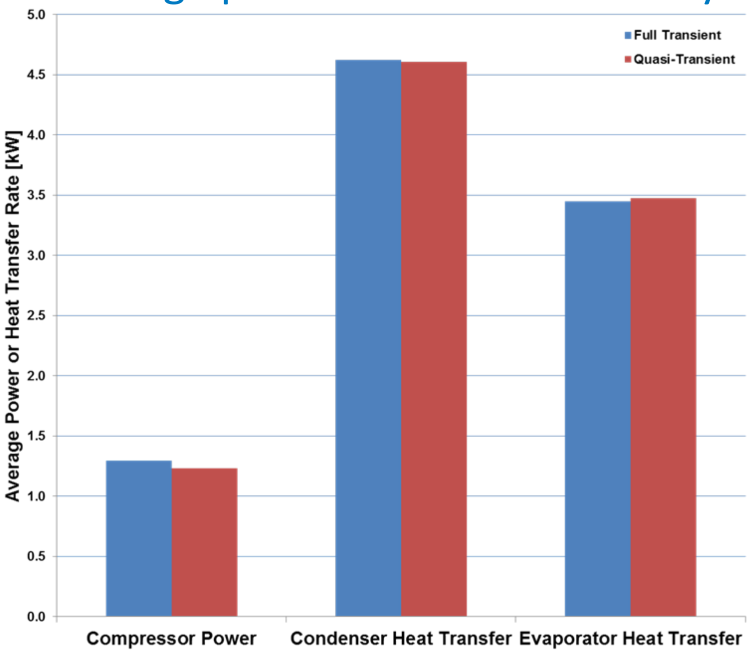
Accomplishment – Quasi-Transient Compared to Full Transient

Good agreement between models over full cycle, quasi-transient 12 times faster



Some loss in transient accuracy and time resolution

Average power over SCO3 drive cycle



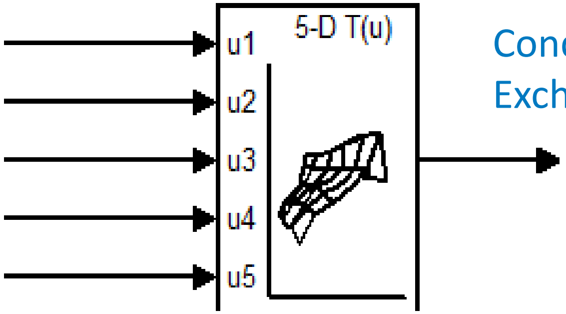
Errors offset overtime and integration provides similar results with much faster simulation

Accomplishments – Mapped A/C Model development

Faster execution time, ~10X real time (120 X Full Transient model)

- **Heat exchanger calculations replaced by performance maps**
- **Quasi-transient model used to create lookup tables for the condenser and evaporator**
  - 5- and 6-dimensional lookup tables are the best compromise between speed and accuracy, respectively
- **Several thousand steady-state simulations were conducted for both condenser and evaporator to create the lookup tables**
- **Working on improving the model further**

Upstream Pressure
Upstream Enthalpy
Refrigerant Pressure Drop
Air Mass Flow Rate
Air-In Temperature

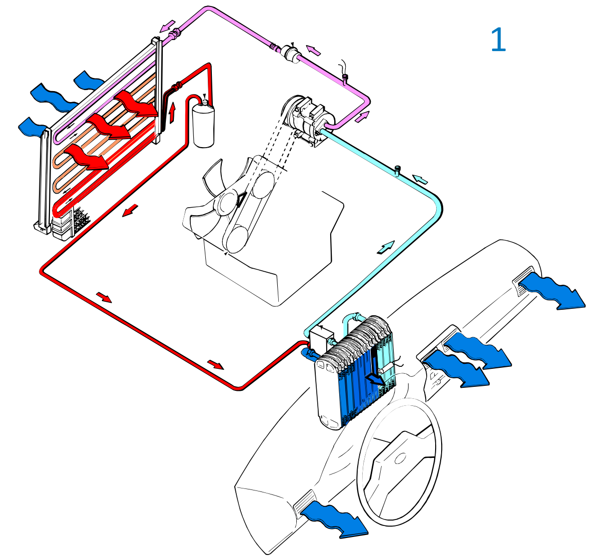


Condenser Heat Exchange Rate

Lookup n-D Heat Exchange Rate

Collaboration

- Halla Visteon Climate Control
  - Technical advice
  - A/C system and component test data
  - Co-authored paper for SAE World Congress
- Argonne National Laboratory
  - Integration of A/C model into Autonomie
  - Vehicle test data
- Daimler Trucks
  - Support Super Truck work



1

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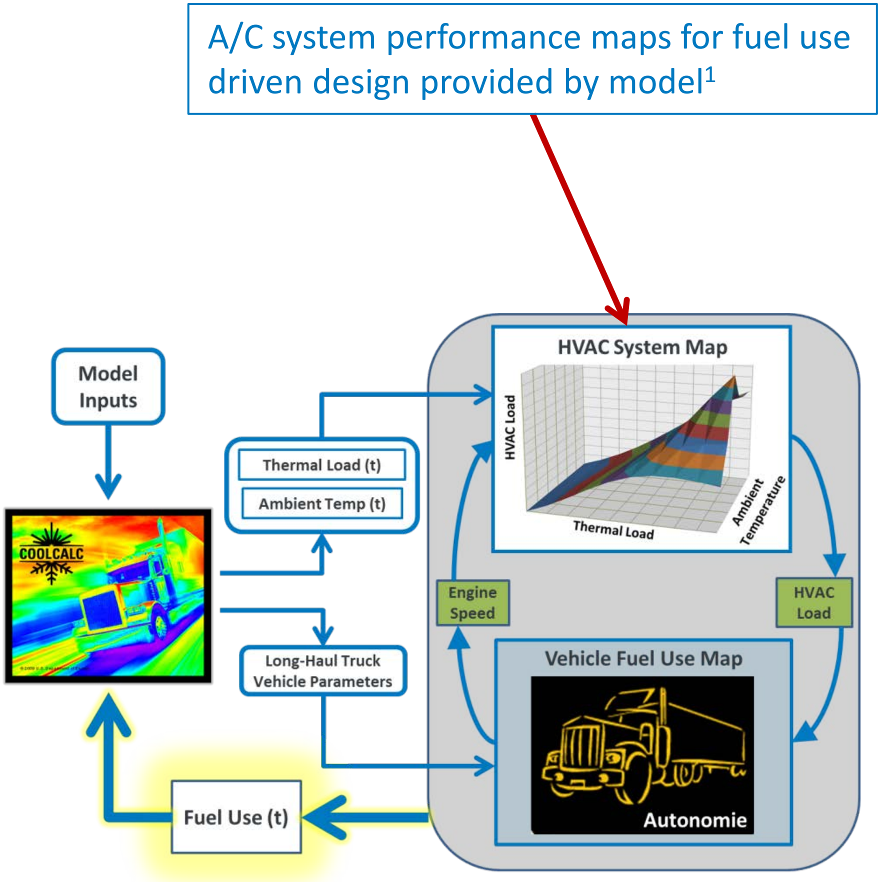
1. Diagram courtesy of Visteon Corporation

2. Daimler Super Truck Logo, Courtesy of Daimler Trucks, 2011

Future Work

FY13

- Complete long-haul truck sleeper A/C system model for use with CoolCalc
- Validate model with ANL's Advanced Powertrain Research Facility (APRF) data
- Develop and release mapped component models (will run 10X real time) for co-simulation with Autonomie
- Release Autonomie A/C plug-in and updated standalone model



1. See VSS075, CoolCab Test and Evaluation & CoolCalc HVAC Tool Development presentation for more information

Summary

DOE Mission Support

- A/C use can account for significant portion of the energy used by light-duty and heavy-duty vehicles.
- Reducing A/C energy use is essential to achieving the President's goal of 1 million electric drive vehicles by 2015.

Approach

- Develop a transient open source Matlab/Simulink-based HVAC model that is both flexible and accurate. Base model on first principles and do not rely on component flow and heat transfer data as input.
- Interface HVAC model with Autonomie vehicle simulation tool to simulate effects of HVAC use on vehicle efficiency and range.

Summary

Technical Accomplishments

- Improved a Matlab/Simulink model of light-duty vehicle A/C system and showed close agreement with experimental data over a wide range of operating conditions
- Added electrical compressor capability and associated controls
- Improved model for co-simulation with Autonomie
- Developed simplified model options for more rapid, less detailed analysis, with a focus on vehicle co-simulation with Autonomie
- Developed an initial heavy-duty vehicle sleeper system model
- Presented "A new Automotive Air Conditioning System Simulation Tool Developed in MATLAB/Simulink" at SAE world congress.

Collaboration

- Halla Visteon Climate Control
- Argonne National Laboratory
- Daimler Trucks

Summary – Acknowledgments

- **U.S. Department of Energy**
  - Lee Slezak, Vehicle Technologies Program
  - David Anderson, Vehicle Technologies Program
- **Halla Visteon Climate Control**
  - John Meyer
- **Argonne National Laboratory**
  - Aymeric Rousseau

References

nomenclature

Q_{tr} is heat transfer from pipe wall to refrigerant

h_{tr} is the heat transfer coefficient from pipe wall to refrigerant

A_t is the area of inner pipe surface

T_t is the pipe wall temperature
T_r is the refrigerant temperature

Q_{at} is heat transfer from air to pipe wall

m_a is mass flow of air

$C_{p, adry}$ is constant pressure specific heat of dry air

m_w is the mass flow of water

$C_{p,w}$ is constant pressure specific heat of water vapor

$T_{a,o}$ is air temperature out, or leaving

$T_{a,i}$ is air temperature in, or entering

h_a is the heat transfer coefficient from air to pipe wall

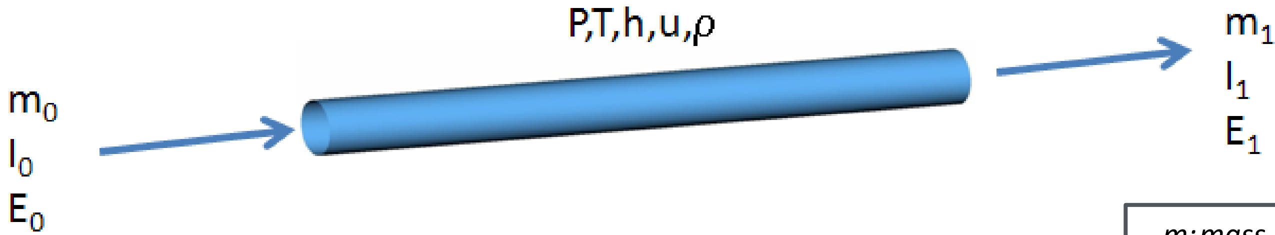
A is the total heat transfer area

ω is absolute humidity

Technical Back-Up Slides

Conservation Equations Solved in Refrigerant Lines

(One-dimensional Finite Volume Formulation)



m:	*mass*
I :	*momentum*
E:	*energy*
P:	*pressure*
T:	*temperature*
h:	*enthalpy*
u:	*specific internal energy*
ρ:	*density*
v:	*velocity*

Continuity:

$$\frac{dm}{dt} = \rho_{in} A v_{in} - \rho_{out} A v_{out}$$

Momentum Equation:

$$\frac{dI}{dt} = \rho_{in} A v_{in}^2 - \rho_{out} A v_{out}^2 + (p_{in} - p_{out})A + F_{wf}$$

Energy Equation:

$$\frac{dE}{dt} = A v_{in}\left(p_{in} + u_{in}\rho_{in} + \rho_{in}\frac{v_{in}^2}{2}\right) - A v_{out}\left(p_{out} + u_{out}\rho_{out} + \rho_{out}\frac{v_{out}^2}{2}\right) + Q_{tr}$$

where '*in*' and '*out*' subscripts mean inlet boundary and outlet boundary of finite volume, respectively

(F_{wf} is wall friction and Q_{tr} is heat addition rate)

Heat Transfer Correlations Used in Model

Condenser wall to refrigerant: $Q_{tr} = \bar{h} A_i (T_t - T)$

where the film coefficient is calculated with the Dittus-Boelter equation:

$$\left(\overline{Nu}_D \equiv\right) \frac{\bar{h} D}{k} = 0.023 Re_D^{4/5} Pr^n$$

The coefficient ***n*** can be modified for a particular geometry.

Heat Transfer Correlations Used in Model

Evaporator wall to refrigerant: $Q_{tr} = h_{tp} A_i (T_t - T)$

where the film coefficient is calculated with the Chen correlation:

$$h_{tp} = h_{FZ} S + h_L F$$

(composed of the sum of boiling and convective contribution)

h_{FZ} is the Forster-Zuber correlation for nucleate boiling

$$h_{FZ} = 0.00122 \left[\frac{k_L^{0.79} c_{pL}^{0.45} \rho_L^{0.49}}{\sigma^{0.5} \mu_L^{0.29} h_{LG}^{0.24} \rho_G^{0.24}} \right] \Delta T_{sat}^{0.24} \Delta P_{sat}^{0.75}$$

(h_{LG} is the latent heat of vaporization, subscript L is liquid phase, subscript G is vapor phase, ΔT_{sat} is the temperature difference between the inner tube wall [T_{wall}] and local saturation temperature [T_{sat}])

h_L is the liquid phase heat transfer coefficient given by the Dittus-Boelter correlation

$$h_L = 0.023 Re_L^{0.8} Pr_L^{0.4} \left(\frac{k_L}{d_i} \right) \qquad Re_L = \frac{\dot{m}(1-x) d_i}{\mu_L} \qquad Pr_L = \frac{c_{pL}\, \mu_L}{k_L}$$

Heat Transfer Correlations Used in Model

Evaporator wall to refrigerant (continued):

F is Chen's two-phase multiplier, and X_{tt} is the Martinelli parameter, which accounts for the two-phase effect on convection

$$F = \left(\frac{1}{X_{tt}} + 0.213\right)^{0.736} \qquad X_{tt} = \left(\frac{1-x}{x}\right)^{0.9} \left(\frac{\rho_G}{\rho_L}\right)^{0.5} \left(\frac{\mu_L}{\mu_G}\right)^{0.1}$$

S is the Chen boiling suppression factor:

$$S = \frac{1}{\left(1 + 0.00000253 Re_{tp}^{1.17}\right)} \qquad Re_{tp} = Re_L F^{1.25}$$

Chen, J.C. (1966). "A correlation for Boiling heat Transfer of Saturated Fluids in Convective Flow," *Ind. Eng. Chem. Process Ses. Dev.,* Vol. 5, No. 3, pp. 322-329.

Heat Transfer Correlations Used in Model

Heat transfer from air to pipe wall:

$$Q_{at} = \bar{h}_a A_o (T_a - T_t)$$

$j = 0.425 * Re_{Lp}^{-0.496}$ where j is the Colburn factor

$j = St * Pr^{0.666}$ and $St = \dfrac{h_a}{c_p \rho V}$

and Re_{Lp} is the Reynolds number based on the louver pitch.

Or the more general correlation by Chang and Wang

$$j = Re_{Lp}^{-0.49} \left(\frac{\theta}{90}\right)^{0.27} \left(\frac{F_p}{L_p}\right)^{-0.14} \left(\frac{F_l}{L_p}\right)^{-0.29} \left(\frac{T_d}{T_p}\right)^{-0.23} \left(\frac{l}{L_p}\right)^{0.68} \left(\frac{T_p}{L_p}\right)^{-0.28} \left(\frac{\delta_f}{L_p}\right)^{-0.05}$$

Where Θ is the louver angle, F_p is the fin pitch, L_p is the louver pitch, F_l is the fin length, L_l is the louver length, T_d is the tube depth, T_p is the tube pitch, and δ_f is the fin thickness.

Chang, Y.J., and Wang, C.C., "A Generalized Heat Transfer Correlation for Louver Fin Geometry," *Int. J. Heat Mass Transfer,* Vol. 40, No. 3, pp. 533-544, 1997.

A/C Model Development

Compressor Model

- Subscripts u and d are for upstream and downstream, respectively
- Mass flow rate:

$$\dot{m} = \rho_u \cdot \eta_{vol} \frac{dV}{rev} \cdot RPM/60$$

where $\eta_{vol} = \eta_{vol}(\frac{p_d}{p_u}, RPM)$ and dV/rev is the displacement per revolution

- Downstream enthalpy ($h_{d,actual}$) calculated using isentropic efficiency:

$$h_{d,actual} = h_u + \frac{h_{d,isentropic} - h_u}{\eta_{isentropic}}$$

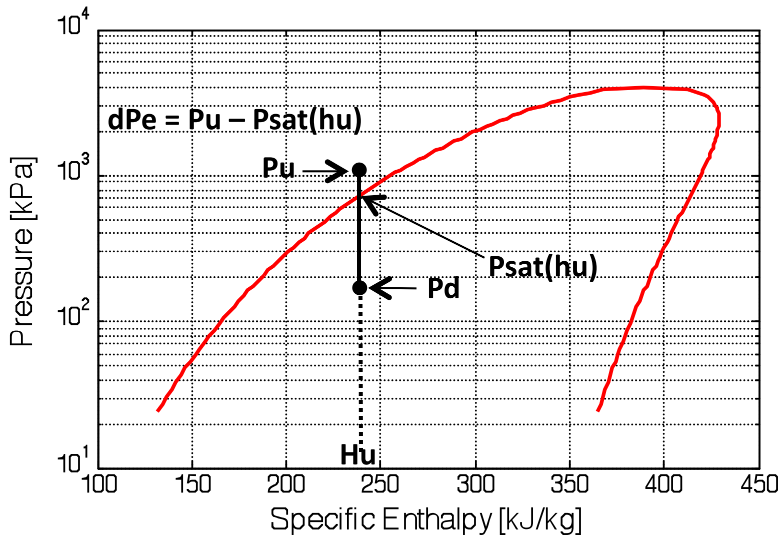
- where $h_{d,isentropic} = h(s_u, p_d)$ and $\eta_{isentropic} = \eta_{isentropic}(\frac{p_d}{p_u}, RPM)$

A/C Model Development

Thermal Expansion Valve (TXV) Model

- Two-phase equilibrium orifice flow model with feedback control on orifice flow area based on Evaporator-out superheat ('SH')
- Orifice flow model calibrated to measured data using a discharge coefficient that is dependent on dP_e

$$\dot{m} = C_d(\mathrm{d}P_e) \cdot \rho_{throat} \cdot v_{throat} \cdot A_{orif}$$



- Feedback control:

$$\frac{dA_{orif}}{dt} = -C \cdot (T_{SHtarget} - T_{SH})$$

- Large C results in quick convergence but may lead to hunting
- Small C results in slow convergence but avoids hunting