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ABSTRACT

e-Thermal is a vehicle level thermal analysis tool developed by General Motors to simulate the transient performance of the entire vehicle HVAC and Powertrain cooling system. It is currently in widespread (global) use across GM. This paper discusses the details of the air-conditioning module of e-Thermal. Most of the literature available on transient modeling of the air conditioning systems is based on finite difference approach that require large simulation times. This has been overcome by appropriately modeling the components using Sinda/Fluint. The basic components of automotive air conditioning system, evaporator, condenser, compressor and expansion valve, are parametrically modeled in Sinda/Fluint. For each component, physical characteristics and performance data is collected in form of component data standards. This performance data is used to curve fit parameters that then reproduce the component performance. These components are then integrated together to form various A/C system configurations including orifice tube systems, txv systems and dual evaporator systems. The A/C subsystem uses airflow rates, temperatures, humidity's and compressor speed as inputs. The outputs include overall system energy balance, system COP, refrigerant flow rates and system pressures. The A/C simulation runs about three times faster to three times slower than real time. The modeling technique used is also capable of tracking the effect of system charge on the overall system performance. A database of automotive air conditioning components accompanies the simulation tool. This database is then integrated in e-Thermal to provide the component data for modeling. Validation results for component level models are demonstrated. They form the basis of system level models. System level validation is also demonstrated. The simulation times vary from 3 times faster than real time to 5 times slower than real time depending on the nature of the simulation.

INTRODUCTION

e-Thermal is a transient, vehicle level, PC (Windows) based simulation tool developed by General Motors that focuses on evaluating thermal HVAC/PTC performance. It consists of the following different modules: Front end flow module, Air-Handling module, Air-Conditioning module, Passenger Compartment module, Cooling System module, Vehicle module, Transmission module, Engine module and Load Case (Driving Scenario) module. It is a transient performance simulation tool. Each of these modules can be configured to simulate a specific system in the vehicle. Depending on requirement, a number of modules can be selected to run simultaneously, interacting with each other to predict the system level thermal performance. If a module is not chosen, appropriate boundary conditions need to be supplied to the model. This paper focuses in detail on the Air-Conditioning module of e-Thermal.

Designing of Air-Conditioning systems has been for a large part driven by testing, empirical rules and experiences. With the push in the auto industry for shorter vehicle development cycles, fewer resources available in the initial stages of vehicle development, and to enable quick decisions on changes to a vehicle during its lifecycle, a tool is needed that facilitates in making quick decisions about HVAC/PTC performance and sizing issues. There is a need to quickly study the impact of A/C component changes on system performance, passenger comfort and fuel economy.

Existing models that try to simulate air-conditioning systems have largely depended on computational fluid dynamics analysis. Because of the two-phase nature of the system these CFD models typically have very large simulation times. These models are also created in anticipation of obtaining accurate results at the cost of increased number of modeling inputs that includes detailed component geometry, additional computing resources and longer simulation time. These models typically have predictive capabilities on component level designs and are more suited for optimization of component level performance by studying in greater detail the physical parameters of a component.

The objective of the A/C module is to simulate system level transient as well as steady state performance and is developed with the ability to easily swap components.

Instead of using CFD to model the system, a one-dimensional lump based modeling approach is taken that uses a finite-difference modeling scheme. A commercial finite difference thermal/fluid solver, Sinda/Fluint, is used. The drawback of many commercially available system level thermal tools is the inability to simulate transient A/C systems because of their limited two-phase capability. Sinda/Fluint is a fast and stable Thermal and Fluid solver, and is well suited for this purpose.

The refrigerant properties that are used by Sinda/Fluint are generated from REFPROP [1] routines developed and maintained by National Institute of Standards and Technology.

In this paper, the parametric modeling of A/C components is described. The parameters are then optimized for each A/C component model based on the data collected from the manufactures in the form of component data standards.

The component data standard is a standardized format for providing component characteristic information including geometry and performance data from test/math that manufacturers need to provide. The performance data cover the range of operating refrigerant and airflow conditions.

The component models are then validated individually at component level. These models are then integrated together to form a system level model. There exists a database for each component. The user chooses the components from the drop down list in the e-Thermal Graphical User Interface as shown in Figure 1. Depending on the users selection of components, e-Thermal creates an appropriate system level model by combining the selected component models and their performance characteristics obtained from a database. The transient results from the simulation can also be viewed in e-Thermal.

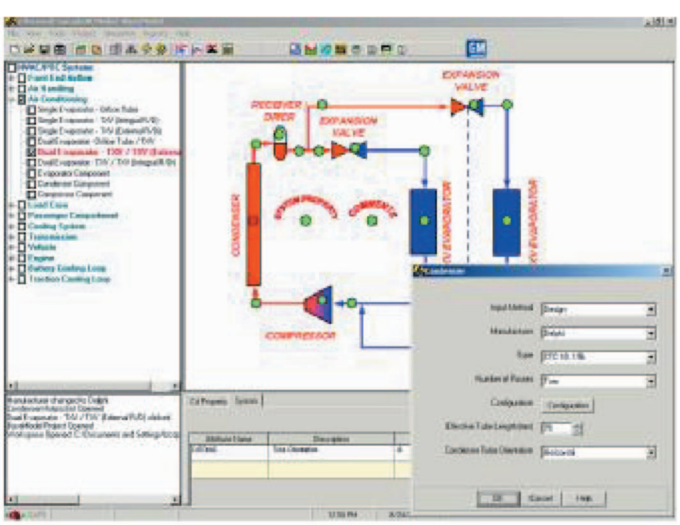

Figure 1 e-Thermal Graphic User Interface

A user can run a complete thermal simulation like soaking a vehicle in solar radiation till the interior gets hot followed by a cool down, for the purpose of evaluating the A/C system performance. The model tracks the refrigerant charge in each component of the system. The user can vary the charge in the system to perform a charge determination procedure and determine the optimum system charge. The component models are flexible enough to simulate variable displacement compressors, integral R/D condensers, suction line heat exchangers and several other emerging automotive technologies. Depending on the simulation being carried out, the simulation times vary from being 3 times faster than real time to 5 times slower than real time.

MODELING OF A/C SYSTEM

In e-Thermal, the boundary conditions for the A/C system are provided by the air-handling module, the front-end module and the engine module as shown in Figure 2. If any of the above-mentioned modules are not chosen, e-Thermal prompts the user to provide the required boundary conditions.

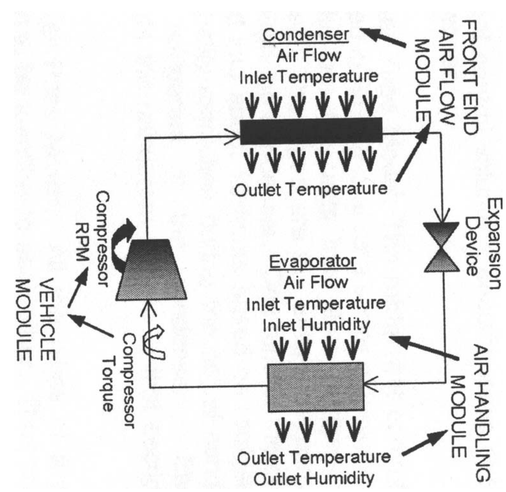

Figure 2 Boundary Conditions for A/C system

There are six different refrigerant circuit models available in e-Thermal. These include,

- Single evaporator orifice tube system
- Single evaporator, external R/D, TXV system
- Single evaporator, internal (sub-cooled) R/D, TXV system
- Dual evaporator, Orifice tube-TXV system
- Dual evaporator, external R/D TXV-TXV system
- Dual evaporator, internal (sub-cooled) R/D TXV-TXV system

In addition, the user may also simulate the evaporator, condenser, and compressor component models individually.

A typical automotive A/C system may include a compressor, condenser, orifice tube / TXV, accumulator dryer, receiver dryer, evaporator, and connecting pipes and hoses. All components are modeled independently. The modeling techniques are described in the following section.

All the models are based on the three basic thermodynamic equations - the continuity equation, the momentum equation and the energy equation. These equations are reduced to the one-dimensional form in Sinda/Fluint.

CONDENSER

Condenser, as the name suggests, condenses the high temperature and high pressure refrigerant that exits the compressor. Refrigerant exiting the condenser is typically sub-cooled, though under some extreme conditions, two-phase refrigerant could exit the condenser. In automotive A/C systems, the condenser is typically a cross flow heat exchanger that uses air on the outside.

<u>Background on Condenser Models</u>

The complicated geometry of the condensers makes detailed modeling more complex. Domanski [2, 3] developed a tube-by-tube analysis for the heat exchanger. It assumed that the air passes directly from one tube to the next. The heat and mass transfer of each tube was evaluated based on its own inlet and outlet states. Conde and Sutter [4] developed a similar simulation with some modifications; the important one being the ability to determine the charge inventory inside the system. Sami and Zhou [5] describe the closure equations for the modeling of the heat exchangers. These include the heat transfer and pressure drop correlations for single phase and two-phase flows. Jacobsen [6] also describes the equations used for modeling the heat exchangers as a bulk entity.

Willatzen et al. [7] and Pettit et al. [8] give a complete formulation of two-phase flows heat exchanger that includes the transient phenomena. This is based on one-dimensional partial difference equations representing mass and energy conservation (leaving out the momentum equations assuming the pressure drops to be negligible) and the tube-wall energy equation. For modeling purposes it takes into consideration the three refrigerant zones: liquid, two-phase and vapor.

<u>Condenser Model</u>

The condenser is modeled from its smallest representative element – the tube. Each tube of the condenser is divided into three control volumes (Figure 3). The first control volume is the refrigerant control volume. The second control volume is the wall control volume and the third control volume is the air control volume.

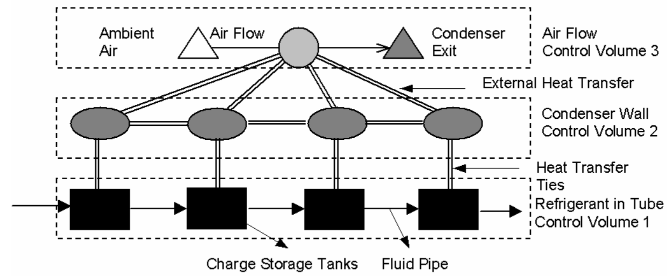

Figure 3 Condenser control volumes

Control Volume 1 (Refrigerant Control Volume)

Refrigerant control volume 1 encompasses one tube of the condenser.

Condenser Tube Model: The refrigerant control volume is modeled in Sinda/Fluint as a series of tanks and pipes. The tanks are the units that store refrigerant in the condenser. The pipes are the units that represent the flow through the condenser. One tube of the condenser is divided into about 10 tanks based on refrigerant two-phase quality resolution during the actual condensation of the refrigerant in the condenser. Each tank represents the refrigerant volume in that section of the tube.

Condenser Pass Model: All the tubes of a pass are assumed to be identical to each other. Thus one tube is modeled and duplicated the required number of times to produce one pass of the condenser. This pass is in turn duplicated with different number of tubes for each additional pass required in the condenser. Thus condensers with different pass configurations are created. In this control volume, the hydraulic diameter of the tube and the tube flow area are the only parameters that vary from one type of condenser construction to another.

Control Volume 2 (Wall control volume)

Wall portion of each tube is divided into a number of capacitances. Each refrigerant tank created has a corresponding thermal capacitance in the system. This is important to reproduce the thermal inertia that the condenser would see during transients in the system. This thermal inertia is well defined after knowing the basic geometry of the condenser along with its fin geometry. Thermal Conductance between each thermal capacitance unit of the tube is also accounted for.

Control Volume 3 (Air flow path)

This control volume represents the air flow over one tube of the condenser. The air inlet to the condenser is currently simulated with uniform air flow and temperature profile across the face of the condenser. If needed this can be easily modified to simulate a user provided temperature and airflow distribution across the face of the condenser. This air flows over the condenser and picks up heat depending on the lump over which it is flowing. The air then mixes to give the mixed condenser air out temperature.

Heat Transfer between Control Volume 1 and 2

The heat transfer from the refrigerant control volume to the wall control volume take place by defining Sinda/Fluint element called Ties. These elements are set to use Sinda/Fluint's in-build heat transfer correlations

Two-phase heat transfer is a complex phenomenon and various correlations have been developed to simulate it. SINDA/FLUINT has the ability to use the appropriate heat transfer coefficients by taking into account the state of refrigerant and the type of flow regime [9]. For single-phase laminar flow, a constant Nusselt number is used. For single-phase turbulent flow, Dittus-Boelter equation is used and for single-phase transition flow Hausen's equation is used. For two-phase condensation heat transfer Rohsenow's equation is used with four different flow regimes for frictional pressure drop that include bubbly, slug, annular and stratified regimes. The various pressure drop correlations used based on the flow regimes are: Darcy's friction factor correlation, McAdams homogenous correlation, Lockhart & Martinelli correlation, Baroczy correlation, and Friedel correlation.

Heat Transfer between Control Volume 2 and 3

The NTU - effectiveness method is used to calculate the heat transfer between the condenser wall/fins and the air. To calculate this heat transfer coefficient, Stanton number, Prandtl number and Reynolds numbers are correlated as

Stanton number = f(Reynolds number, Prandtl number)

The above correlation uses two additional constant parameters to characterize the condenser heat transfer across various construction styles. This variation is due to different fin geometries and development of different air velocity profiles across the condenser.

Parametric Model

Thus a generic parametric refrigerant side condenser model is created. From the description above, it is noted that the complete condenser model is defined based on just four parameters - the hydraulic diameter, tube flow area and two air side parameters mentioned above. Using the raw data provided by the suppliers through the component data standards, these parameters are optimized to reproduce the heat transfer and the pressure drop characteristics for the condenser. Typical heat transfer correlation errors are less than 3% and refrigerant pressure drop correlation errors are less than 20%. Figure 4 and Figure 5 show the heat transfer and pressure drop correlations respectively, for one set of condenser data.

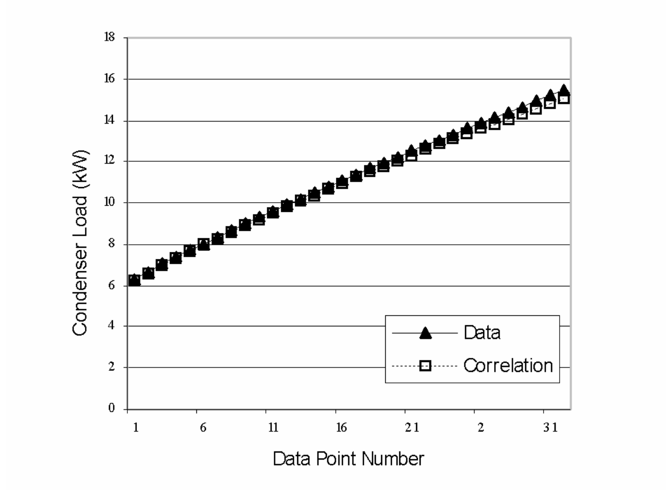

Figure 4 Condenser Heat Transfer Correlation

Air Side Pressure Drop Coefficients

Air Side Pressure drop per unit thickness of the heat-exchanger core is plotted against the face velocity. For doing so, both pressure drop and velocity are translated to normal conditions (20 deg C and 1 atm) by accounting for both density and viscosity effects. The condenser is simulated as a porous media. Hence a quadratic curve is fitted through this plot; the quadratic coefficients represent the air side pressure drop parameters. The above two parameters coupled with physical information like face area and core thickness are required inputs from the user for pressure-drop calculation purposes. Figure 6 shows the Air side pressure drop correlation for the same condenser along with the pressure drops from the test data.

About thirty different condenser constructions from various automotive condenser manufacturers all over the world have been correlated to this model and added to the e-Thermal database.

User can vary the following properties of the condenser through the e-Thermal user interface:

1. The number of passes
2. The number of tubes of each pass
3. The length of the condenser

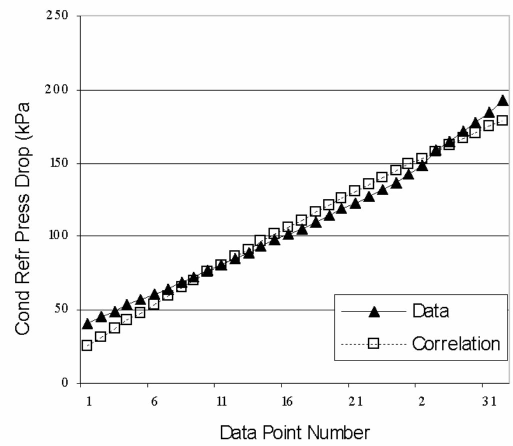

Figure 5 Condenser refrigerant pressure drop correlation

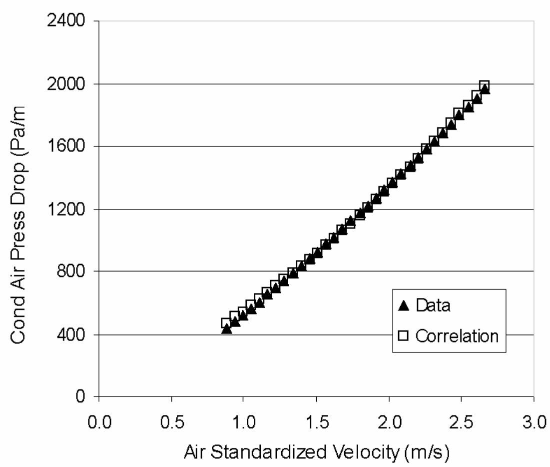

Figure 6 Condenser air pressure drop correlation

This capability provided in e-Thermal for changing the condenser parameters has been validated for various condenser styles using available data.

Header and tube condensers, tube and fin condensers and the serpentine condensers are various different kinds of condensers that are modeled using this method.

It should be noted that the effect of oil concentration is taken into consideration while fitting the data. The amount of oil that is provided by the suppliers in the data will remain consistent with that component. Currently, changing the oil circulation rate in the system is not simulated due to insufficient data.

EVAPORATOR

Evaporator cools the air flowing over it while evaporating the two-phase refrigerant that enters it from the expansion device. Refrigerant exiting the evaporator is typically saturated vapor for orifice tube systems and superheated vapor for TXV systems.

Background on Evaporator Models

The history of evaporator modeling is similar to that of the condenser models. The additional requirements that the evaporator poses are condensation and frosting. Oskarsson et al. [10] developed models that take into consideration the three cases of dry evaporator, wet evaporator and frosted evaporators.

Evaporator Model

This model is similar to the condenser model. It has an additional factor to be taken into consideration: humidity of the air entering the evaporator. The control volume 3 (Air control volume) of the condenser model is the only aspect of the model that is modified for evaporators. This control volume is divided into many lumps along the direction of the air flow to take into account the humidity change that takes place in the air. The width of the evaporator is divided into as many as ten lumps on the air side to get the desired heat transfer accuracy and resolution on the air side. Evaporator out air temperature is assumed to be saturated with moisture whenever the air is cooled below its dew point. All the condensate is assumed to flow out from the evaporator core. The U-channel evaporators are also modeled along with the conventional header and tube evaporators. In the U-channel evaporators, the air temperature resolution is high enough to take into account the temperatures and humidity conditions after the front half along the direction of the air flow and before entering the rear half of the evaporator.

Psychrometric equations from ASHRAE handbook are used as the standard for calculating air-moisture properties.

Figure 7, Figure 8, and Figure 9 show the heat transfer, refrigerant pressure drop and air side pressure drop correlation for one set of evaporator data.

About forty different evaporator constructions from various evaporator core manufacturers have been correlated and added to the e-Thermal database. Typical heat transfer correlation errors are less than 3% and refrigerant pressure drop correlation errors are less than 20%.

Similar to the condenser, oil is taken into consideration while fitting the data but cannot be modified.

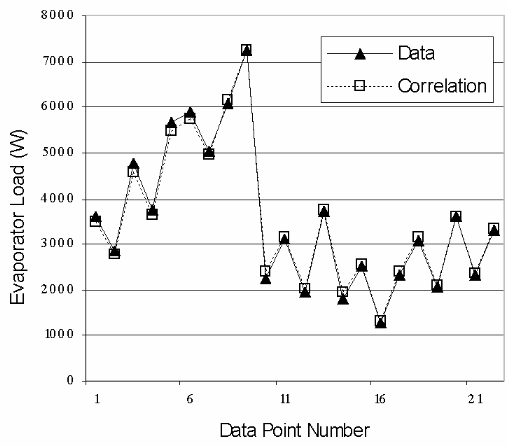

Figure 7 Evaporator heat transfer correlation

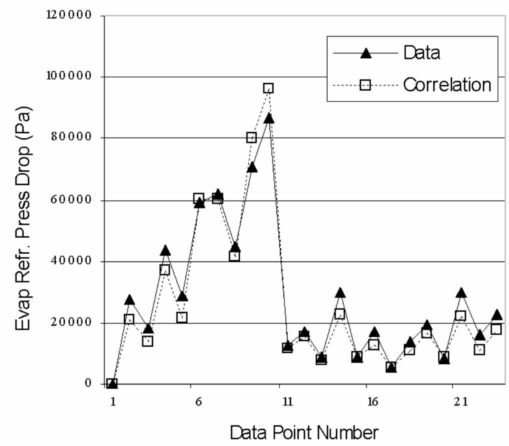

Figure 8 Evaporator refrigerant pressure drop correlation

COMPRESSOR

Compressor takes the vapor from the outlet of the evaporator/accumulator and compresses it till it reaches discharge pressures in the condenser. Over a typical range of operating conditions, only vapor enters the compressor. Under extreme cases two-phase refrigerant may enter the compressor.

<u>Background on Compressor Models</u>

Various degrees of complexities of compressor models are available in the literature.

For a generalized method for describing the compressor, using the geometry and bulk parameters, a complex set of mass and energy equations are used [5]. These equations take into consideration the leakage flow, back flow and the flow at pipe boundaries while designing the compressors. Some models also take into consideration the oil dissolved in the refrigerant in the compressor casing for energy balance [5,11]. These models take into consideration the heat and mass transfer between the refrigerant and the oil. For mass flow rates, these models use some assumed correlations.

Another method to compute mass flow rates is to use theoretical correlations available for the compressors volumetric flow rate [12]. Yet another strategy [7] involves divides the compressor into two control volumes - the suction chamber and the discharge chamber, for computing equilibrium. These control volumes along with vapor compression equations complete the model.

A few other variations of compressor models have been described in literature. These models include those that divide the compressor into various different regions ranging from three zones to nineteen zones for energy, momentum and mass conversations. Some models also implement the dynamics for each of the valves as a

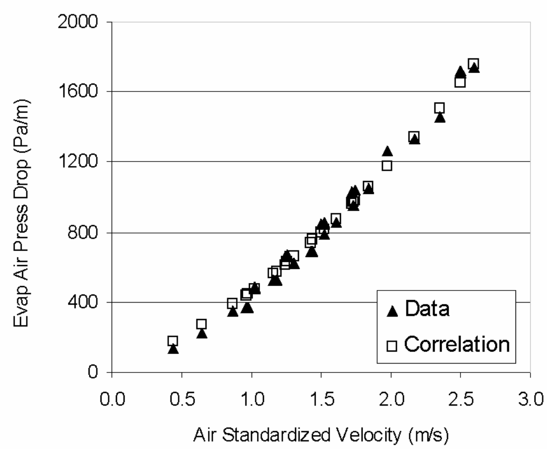

Figure 9 Evaporator air side pressure drop correlation

spring, mass and damper system. One model also models the valves using a two dimensional finite element approach to study in great detail the valve dynamics.

Other modeling techniques [13,14,15] use curve fitted correlations for simulating compressor efficiencies and flow rates.

All the models have their advantages and their disadvantages varying from computing resources to complexity and input information required to simplicity and generalization for different variations of compressors.

Compressor Model

Various kinds of compressors are used in the automotive industry. These include reciprocating simple piston compressors, swash plate compressor, scroll compressors, etc. A generic method of curve fitting was used to model the various types of compressors. The compressor is modeled as a mass flow rate device, with inputs being the evaporator out pressure, temperature, quality, condenser in pressure, and engine rpm. Based on this information, compressor performance is characterized by the three basic efficiencies.

- Isentropic efficiency
- Volumetric efficiency
- Mechanical efficiency

All the efficiencies are a function of pressure ratio of high side pressure and low side pressure and the compressor RPM.

Knowing the compressor displacement and these efficiencies; the compressor refrigerant mass flow rate, outlet temperature, and compressor torque can be calculated.

Figure 10, Figure 11, and Figure 12 show the compressor efficiencies from a typical compressor that is fitted using manufacturer supplied raw data.

About thirty different compressors have been fitted using this methodology. The maximum errors observed in the curve fits for the efficiency are less than 5%. Some of the data fits with as small as 1% error.

Variable displacement compressors have also been modeled in e-Thermal. Both, the internal valve controlled variable displacement compressors, and external evaporator air out temperature based and suction pressure based variable compressors can be simulated. The internal valve controlled compressors are simulated based on the valve characteristics that are provided in the model.

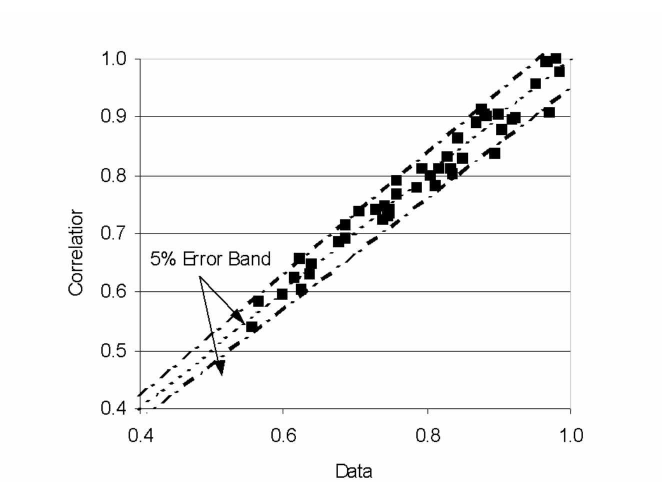

Figure 10 Compressor isentropic efficiency correlation

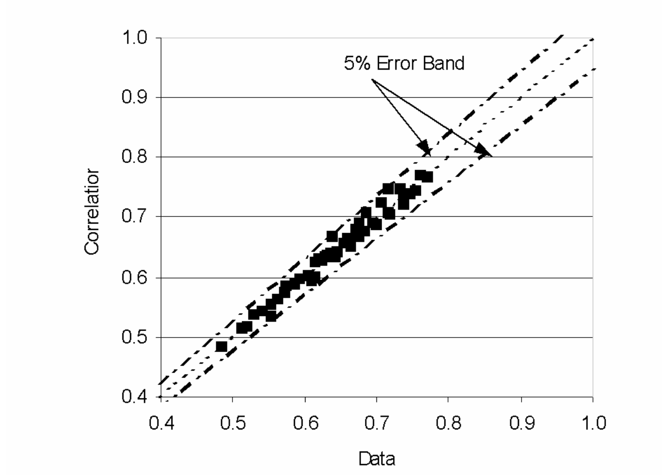

Figure 11 Compressor volumetric efficiency correlation

Compressor clutch controls are also modeled to take into account the cycling of the compressors to prevent icing of the evaporator. This is based on compressor suction pressure. High pressure compressor clutch cutoff is also modeled to make sure that the system does not operate above the high pressure cutoff value specified by the user.

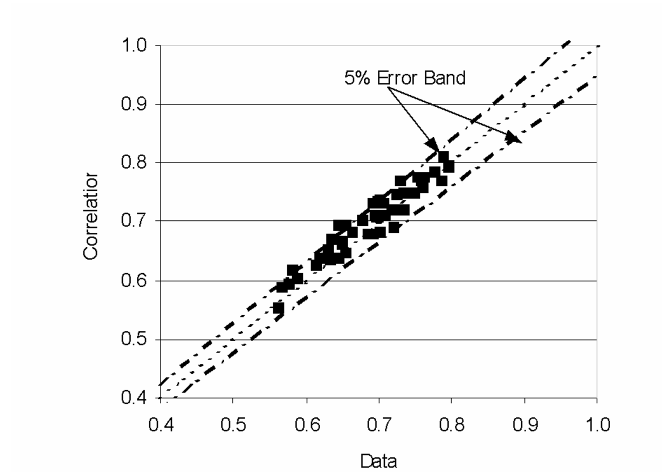

Figure 12 Compressor mechanical efficiency correlation

ORIFICE TUBE

Orifice tubes are simple devices that act as throttling valves between the condenser and the evaporator. This device limits the flow between the two components.

Background on Orifice tubes

Various studies [16] have been conducted with various different refrigerants to evaluate performance of the orifice tubes. In one study [17] the orifice tubes are correlated as a function of inlet and outlet refrigerant conditions, orifice physical dimensions of diameter and length.

Orifice Tube Model

Very often it is assumed that these are pressure drop devices and are modeled as such. These devices, during typical operation, have choked flow conditions. During choked flow it is not possible to calculate the downstream pressure. The Orifice Tube is modeled as a mass flow device. This implies that given the upstream temperature, pressure and quality and downstream pressure, the mass flow rate is determined.

An Orifice tube correlation was developed internally in General Motors for a 1.8mm diameter orifice tube that was tested. For this correlation the mass flow rate was correlated as a function of inlet pressure, inlet sub-cooling, inlet quality and outlet pressure.

Subsequently, it was observed that the correlation developed by University of Illionis, Urbana Champaign [17] was comparable to our model in terms of accuracy. This is demonstrated in Figure 13. This correlation had the advantage that it could also model orifice tubes of various different diameters and lengths. Hence this correlation was used to model the orifice tubes.

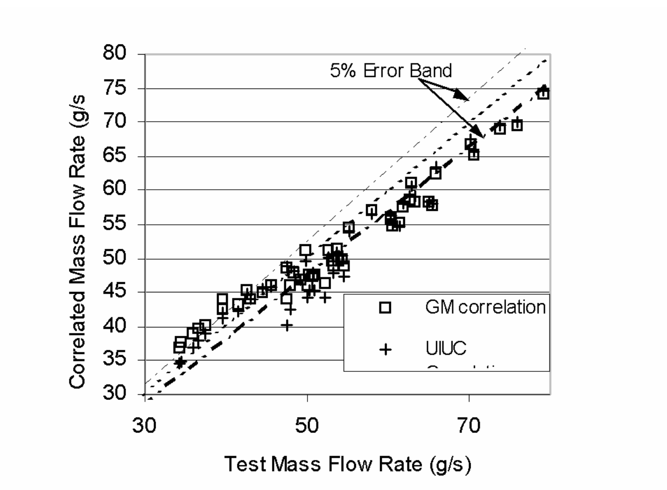

Figure 13 Orifice tube mass flow rate correlation

THERMOSTATIC EXPANSION VALVE (TXV)

TXV is a control valve that varies the valve opening depending on the operating conditions. It maintains superheat at the evaporator outlet over a wide range of operating conditions.

Background on TXV models

A very detailed model of the TXV is described by University of Illinois [18]. This model goes into details about the construction of a particular TXV and involves a significant amount of data for correlation of different parameters of the TXV including thermal lags, force balance inside the TXV for valve displacement. There are other TXV models [19, 20] that are very comprehensive, but these need exhaustive TXV property data.

TXV Model

There are two TXV models that are available in e-Thermal. The first one is a generic TXV. In this model, the user specifies the evaporator out superheat value. To maintain this superheat value over different operating conditions, the model transiently modulates the flow rates through the system. If the superheat values are higher than set superheat value the model increases the flow rate through the TXV and vice versa, until the desired superheat value is achieved.

The second TXV model is based on the physical working of a TXV.

The TXV mass flow rate at a given instant depends on three basic parameters:

- the pressure differential across it
  - determined by knowing the outlet pressure and the inlet pressure
- the effective TXV port area which depends on
  - the TXV port geometry
  - the valve stroke which in turn depends on
    - Evaporator outlet temperature
    - TXV inlet temperature, pressure, quality
    - TXV outlet pressure
    - TXV physical properties like spring constant, bulb pressure etc.
- the coefficient of discharge which depends on
  - the flow rate/valve stroke
  - TXV geometry

To calculate the mass flow rate of the TXV the inlet and outlet temperature and pressure conditions are used. The valve stroke is calculated using a correlation developed based on evaporator outlet temperature, TXV inlet temperature, pressure and quality. This correlation tries to simulate the overall effect of TXV parameters like the bulb pressure, spring forces and other mechanical interactions in the TXV. The coefficient of discharge is also correlated as a function of valve displacement. The lag in the bulb temperature due to convection to the TXV is also taken into account. This lag is modeled as a thermal capacitance. TXV models have been created for both front and rear TXV's. Figure 14 shows the correlated TXV stroke length with that of test data. Figure 15 shows the corresponding correlation for TXV mass flow rate.

About fifteen different TXV's from various manufacturers have been modeled using this method. The accuracy, stability of these models and the ability to do a charge determination are some of the advantages of the more physical TXV model.

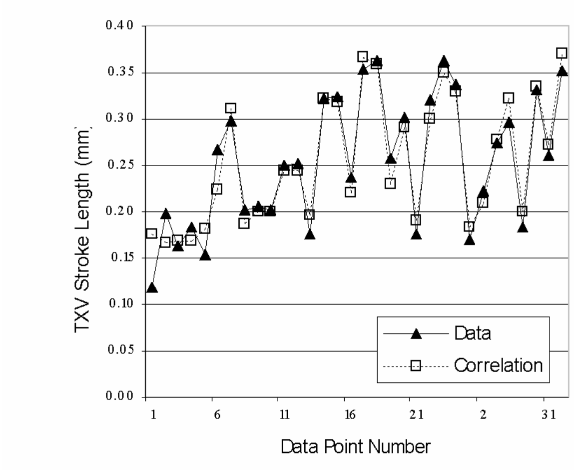

Figure 14 TXV stroke length correlation

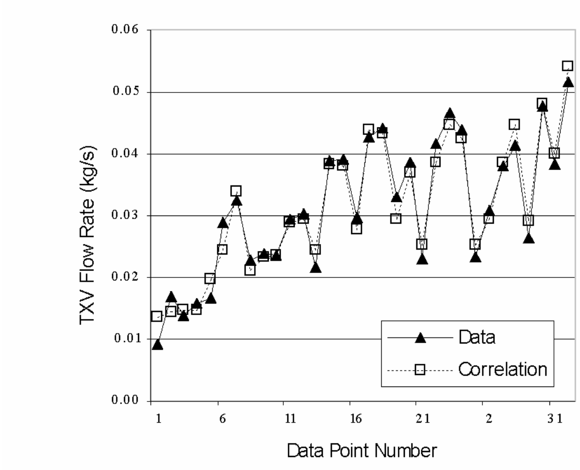

Figure 15 TXV mass flow rate correlation

ACCUMULATOR-DRYER

Accumulator is a device used in the A/C circuit to prevent liquid from entering the compressor. These are typically used with orifice-tube systems in which control over the evaporator outlet conditions is limited.

Background on Accumulator models

Very few studies have been conducted to simulate accumulators in great detail. One investigation by University of Maryland, College Park [21] takes into account the basic internal geometry of the accumulator and tries to predict accumulation of oil. There are some simplified models of the accumulators [22] that model accumulators as simple filters that do not allow liquid refrigerant droplets to exit the accumulator.Accumulator model

Accumulator is a reservoir at the exit of the evaporator. Its primary purpose is to safeguard compressors from liquid refrigerant.

Accumulator is modeled as a generic device. It is characterized by a volume. This volume is defined as the theoretical volume of refrigerant that the device can store. It is calculated by subtracting the dryer volume, the internal pipe volume and the volume above the exit of the accumulator from its actual physical volume. It is modeled such that vapor refrigerant continues to exit the accumulator as long as the void fraction inside the accumulator is greater than 0.1. A mixture of vapor and liquid refrigerant exits the accumulator when the void fraction is between 0.05 and 0.1. Pressure drop across the accumulator is calculated based on refrigerant expansion and contraction and frictional pressure drops.

This model neglects the oil return capability of accumulators. It also assumes ideal mixing of the liquid and vapor inside it. An internal report suggests that the accumulator has a very turbulent flow in its interior. Few accumulators that have been studied showed that the storage capacity of accumulators is very sensitive to the internal geometry of the accumulators. Changing the orientation of the exit pipe can significantly change the storage capacity of the accumulator. It can store from 30-80% of liquid refrigerant by volume before starting to leak liquid droplets to the compressor. Getting accumulator performance data from manufacturers will improve the system performance analysis capability.

RECEIVER- DRYER

Receiver-Dryers (R/D) are added into the high side of the system and are typically used in TXV systems. The primary aim of R/D is to act as a reservoir while allowing only liquid refrigerant to flow into the TXV's.

Receiver Dryer model

R/D is modeled similar to an accumulator. The flow inside the R/D is assumed to be homogenous. The R/D is also characterized by a volume. This volume is obtained after subtracting the dryer volume and the pipe volume from the physical volume of the R/D. The R/D is modeled such that when the void fraction of R/D is less than 0.6, only liquid exits the R/D. When the void fraction is between 0.6 and 0.85 a mixture of liquid and vapor exit the R/D. As the void fraction goes higher than 0.85 the quality exiting the R/D is the same as the quality of the R/D. This control curve is obtained after correlation to a few R/D Txv systems. It is critical that this control of R/D exit quality is a continuous curve to ensure stability of the model. The same model is used to simulate both the external R/D and the R/D in sub-cooled condenser designs.

PIPES AND HOSES

All the components are connected together to form the system using pipes and hoses.

Pipe and Hose Model

Pipes are physically modeled with their characteristic properties of length, inner diameter and number of different kind of bends: < 45°, > 45° and < 90° and >90°. They are modeled in Sinda/Fluint as Sinda/Fluint pipes. This model takes into account the pressure drop depending on the length, diameter and that caused due to various number of bends. Volume of these pipes is also modeled to get an accurate internal system volume for tracking refrigerant charge. These pipes also have heat transfer with the air temperatures that they see, be it under-hood or underbody. For insulated pipes the pipe heat transfer can be further reduced.

SYSTEM LEVEL MODEL INTEGRATION

The component models described are combined to form a system level model as shown in Figure 16. It should be noted that the compressor and the expansion valve determine the flow rates through the system depending on the pressures and the condenser and the evaporators determine the transient and steady state pressures in the system depending on the refrigerant flow rates.

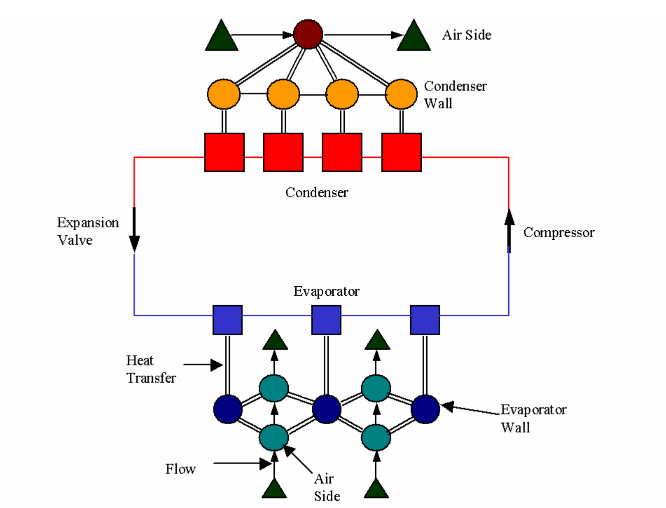

Figure 16 System level model

These components have been connected together in various configurations that are used in automotive systems. These configurations include: orifice tube systems and their variations, internal and external R/D TXV systems, dual evaporator orifice tube-TXV systems and dual evaporator dual TXV systems.

VEHICLE LEVEL VALIDATION

A couple of different validation cases are shown including a charge determination and a soak and cool down using a variable displacement compressor.

For a mid size vehicle, a charge determination procedure was carried out at GM's Climatic Wind Tunnel to determine the critical charge for an integral R/D TXV system. This procedure is simulated transiently in e-Thermal using the Front End flow module to determine the front-end flow characteristics, the Air Handling module for air-handling characteristics, the Air Conditioning module with all the vehicle specific components and the Load Case module to specify the test procedure. Similar to the actual vehicle test, an incremental charge is added to the A/C system on the suction side at regular intervals and the system is given 900 seconds to stabilize.

Figure 17 shows the variation of the condenser head pressure and evaporator discharge air temperatures with refrigerant charge. As charge is added, the condenser pressure rises until it reaches the critical point where the condenser sub-cooling is just above zero. This critical charge is 0.5 kg in both the test and the simulation. As charge is increased further, the pressure stabilizes and forms a plateau. This happens when the additional refrigerant starts accumulating in the R/D. As charge is increased further, the R/D starts filling up. After it is almost full, liquid refrigerant starts backing up in the condenser. This is when the condenser pressures start to shoot up again. The discharge temperatures are very high at very low charges and start decreasing as charge is added to the system. After the critical charge is reached, the discharge temperatures stabilize indicating that increasing the charge further has no effect on the discharge temperatures.

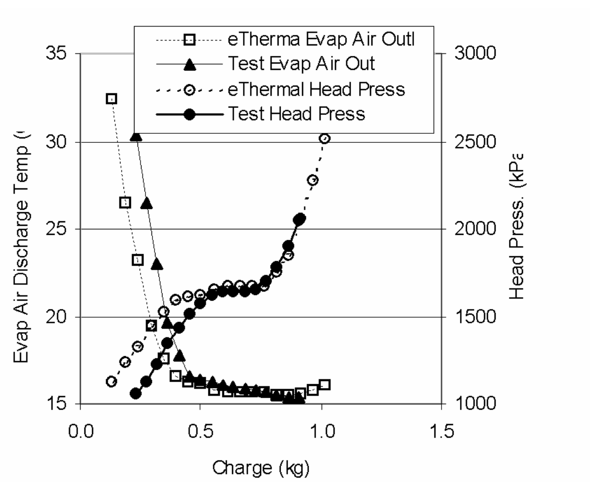

Figure 17 Variation of head pressure and evaporator air discharge temperature with system charge

Figure 18 shows the condenser out sub-cooling and evaporator out superheat in the system. Superheat is very high when the system is starved of refrigerant. During this time the TXV is fully open but because of two-phase inlet, there is not enough flow across it. When the inlet density increases, the flow increases thus decreasing the superheat. The superheat stabilizes when the system reaches the critical charge point. The critical charge the system has zero sub-cooling. Below the critical charge condenser out has two phase refrigerant exiting as charge is added the sub-cooling value increases. This stabilizes when the R/D starts accumulating liquid refrigerant. With further increase in charge, the condenser starts filling up with liquid and the pressures rise and hence the sub-cooling value increases.

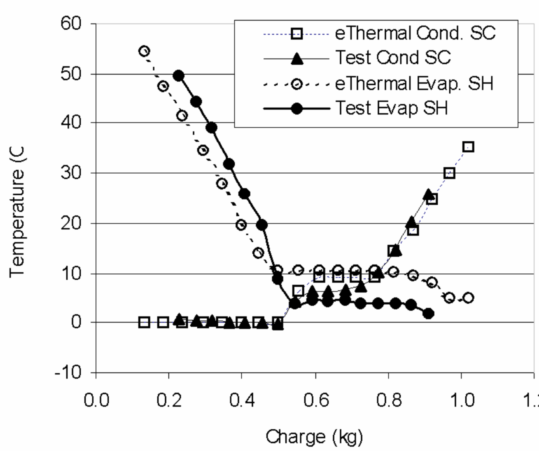

Figure 18 Variation of evaporator outlet superheat and condenser outlet sub cooling with system charge

Figure 19, demonstrates the refrigerant distribution in the various components as computed by e-Thermal. Initially most of the additional charge is stored in the condenser with corresponding increase in the condenser pressure. As charge is increased, the condenser out refrigerant quality decreases. This results in an increase in the charge stored in the condenser out pipe. Charge stored in the condenser increases till the R/D starts storing refrigerant. This happens when the inlet to the R/D is saturated liquid. After the R/D has accumulated all the charge it can hold, any addition of charge in the system is stored in the condenser and as a consequence condenser pressure starts rising again.

A soak and cool down test was run for a current production small sedan. The A/C circuit was a single TXV, integral R/D system with an internally controlled variable displacement compressor. The test scenario was a soak into a 50kph production re-circulation (P/R) condition. This was followed by 80kph stabilized p/r condition, which is followed by 50kph and 80kph outside air points (OSA). The test was run at a high ambient condition with high solar. In addition to the Air Handling module, Front End airflow module, Air Conditioning module and the Load case module this model also had the Passenger Compartment module to predict the soak temperatures and the cool down rate.

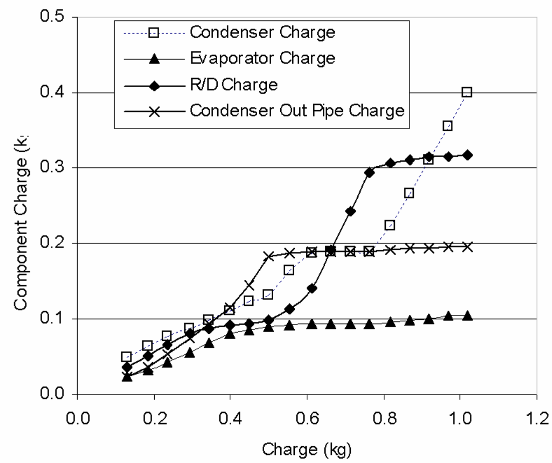

Figure 19 Refrigerant distribution in various components as a function of system charge

Figure 20 shows the comparisons of A/C outlet discharge temperatures and the upper interior air temperatures. The duct out temperatures show good correlation over the entire test span. However, the passenger compartment model under predicts the measured breath temperature by a consistent 4 to 6 degrees. Some of this error can be attributed to the fact that the measured temperature is a single point localized temperature while the predicted temperature is the bulk average temperature for the entire upper seat air volume.

Figure 21 shows the comparison of the head pressure and suction pressures. Both the head pressures and suction pressures show good correlation with the test data. The condenser out pressure starts out high and keeps falling down as the evaporator is in and the evaporator inlet temperature keeps falling. During the OSA it is observed that the pressures are higher.

This system uses a variable displacement compressor. Figure 22 shows how the compressor displacement varies along the test. In the initial part of the test when the system is running at 50kph P/R point, the load on the system is lower. The compressor de-strokes during this portion of the test. It keeps de-stroking depending on the system load. When the test moves on to the OSA point, the heat loads on the system increase and the compressor displacement then goes to full stroke.

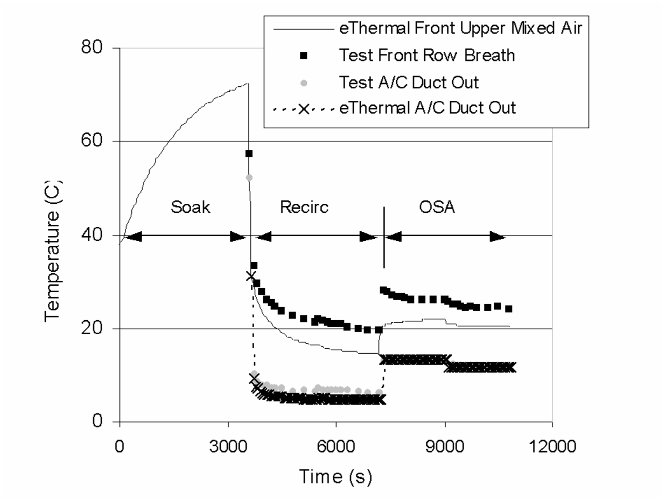

Figure 20 Soak and cool down: Temperatures

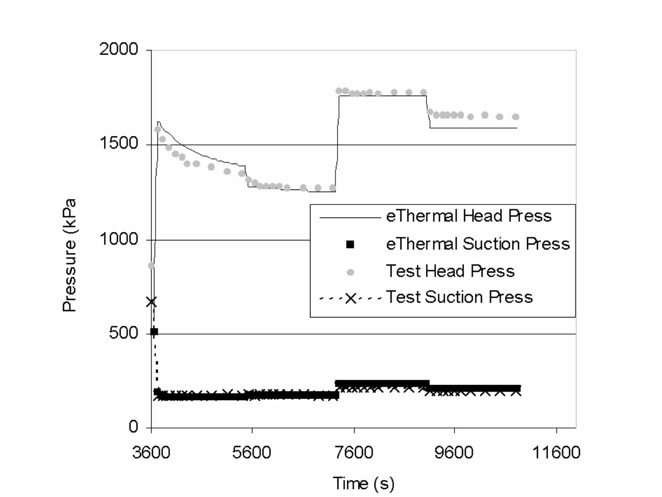

Figure 21 Soak and cool down: head pressures and suction pressures

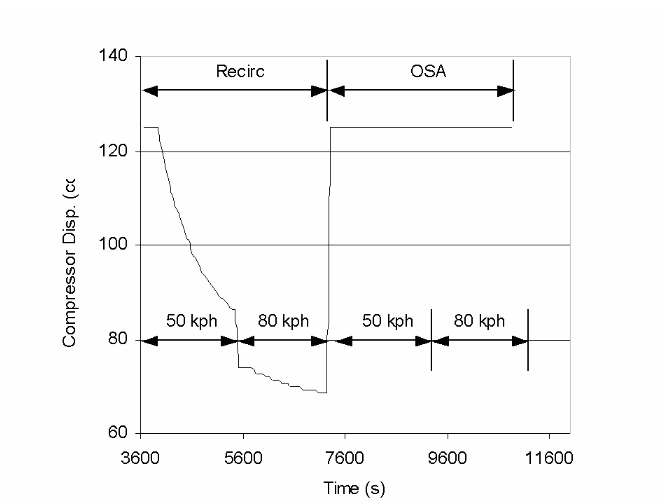

Figure 22 Soak and cool down: Compressor displacement

CONCLUSION

Simulation of the vehicle A/C performance using Sinda/Fluint - a one dimensional Thermal-Fluid solver is demonstrated. Details of component level models are discussed and correlation results demonstrated. These correlations for heat exchangers were applied to various different designs from many manufacturers. The error in correlation was found to be with 3% for heat transfer, 20% for refrigerant pressure drop and 2% for air side pressure drop. Correlations for compressors were found to predict refrigerant mass flow rates within an accuracy of 5%. TXV correlations were developed and found to be accurate within 5%. Additional data is needed to establish validity of this correlation over the entire operating range.

System level modeling capability by assembling validated component models from a database is also demonstrated and validated with two different test cases. Capability to track A/C system charge is shown. The soak and cool down test shows the applicability of the tool for system performance testing. e-Thermal's ability to track the refrigerant system charge and its comprehension of various control strategies increases the fidelity of the A/C model.

The transient nature of the tool and its real time simulation speed along with an easy to use interface makes it possible for both development and analysis engineers to simulate most of General Motors test procedures enabling General Motors to fulfill its goal of minimizing hardware tests.

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