

Simulation of Energy Used for Vehicle Interior Climate

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ABSTRACT

In recent years fuel consumption of passenger vehicles has received increasing attention by customers, the automotive industry, regulatory agencies and academia. However, some areas which affect the fuel consumption have received relatively small interest. One of these areas is the total energy used for vehicle interior climate which can have a large effect on real-world fuel consumption. Although there are several methods described in the literature for analyzing fuel consumption for parts of the climate control system, especially the Air-Condition (AC) system, the total fuel consumption including the vehicle interior climate has often been ignored, both in complete vehicle testing and simulation. The purpose of this research was to develop a model that predicts the total energy use for the vehicle interior climate. To predict the total energy use the model included sub models of the passenger compartment, the air-handling unit, the AC, the engine cooling system and the engine. Verification of the model was carried out against several complete vehicle tests using the new European driving cycle (NEDC) with different ambient temperatures ranging from -18°C to 43°C , different thermal states such as heat up or steady state and different sun loads. The agreement between simulation and measurement was demonstrated to be good for all compared properties except the compressor mechanical load. This research shows that it is possible to create a model that predicts the total energy use for vehicle interior climate for a wide range of different conditions.

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INTRODUCTION

Fuel consumption of passenger vehicles has received more and more attention in recent years. The main reasons for this are increased customer interest due to growing concern over environmental issues and rising fuel prices. Governments in many countries have created, implemented, or tightened, regulations on fuel consumption and emissions, especially regarding greenhouse gases [1, 2]. This emphasis on fuel consumption has led to several improvements, mainly in engine efficiency, aerodynamics and alternative power trains. These areas have large effects on certification cycles, but other areas which are excluded or negligible in the cycles can also have a large effect on real-world fuel consumption. One of these neglected areas is the total energy used for interior climate [1]. For example, one part of the climate system, the Air-Conditioning (AC) system is the largest auxiliary load on a vehicle. According to [3], 6% of the domestic petroleum consumption in the US is used for vehicle AC-systems.

There are several methods described in the literature for analyzing fuel consumption, energy consumption or power usage for parts of the climate system in vehicles. For example, test benches for AC-systems are common and used extensively by both car manufacturers, subcontractors and suppliers. Gaveau and Clodic [7] carried out extensive tests on an AC-system by varying different

parameters and measuring, among other things, compressor power and fan power. Gado et al. [8] carried out several tests on an AC-system, focusing on transient states.

Another approach for investigating the amount of energy used for interior climate is CO_2 emission tests on a complete vehicle, focusing on the AC-system, see [4,5]. This type of test measures the impact of AC-systems on emissions; it does not explain the mechanisms behind the result. Further, it does not look at the complete energy usage of the climate system, and it is powertrain dependent. de Moura and Tribess [6] used complete vehicle testing to compare how different compressors affected fuel consumption. A recent study, by Nielsen et al. [9], investigated both heating and cooling sources using complete vehicle testing. In addition, the electrical and mechanical power used for the interior climate was included. The investigation covered several conditions, such as different temperatures, sun load and thermal states of the vehicle.

There are many drawbacks to testing, especially with complete vehicle testing. For example, a vehicle, measurement equipment, test facilities and personnel are needed. In general testing is both time and resource consuming, and early in projects there are often no physical test object available. The main answer to avoid or reduce testing has usually been simulation of the climate control system. One early documented climate control system simulation was done by Davin et

al. [10] in 1972. It included both an AC-system model with evaporator, thermostatic expansion valve (TXV), condenser, compressor and a passenger compartment model. Subsequent models found in the literature added more details such as pressure drop, humidity, simplified air-handling unit and thermal storage, see e.g., Huang [11].

In recent years many different climate control system simulation models have been generated using various approaches. For example, several researchers have created three dimensional (3D) computational fluid dynamics (CFD) models of the passenger compartment [12] and combined them with comfort models to evaluate the sensitivity of the thermal load on passenger comfort [13]. Furthermore, Rugh [14] used many different linked simulation tools to show the effect of solar glazing on fuel consumption.

However, most models, to our knowledge, cannot predict the total energy used for the interior climate because many systems or components are not included. One example of a more detailed model without all systems is that of Kiss et al. [15]. The model focused on the AC-system, and did not, for example, include a cooling fan or a system for heating. Pitchaikani et al. [16] included all the major systems but used fairly simplified models due to real-time simulation. Another neglected area in all these papers is verification of models; seldom more than one specific case is verified.

In this paper we create a one dimensional (1D) model that includes all the important systems for the interior climate which can predict the total energy used. Further, the model can handle a wide range of different conditions, such as ambient temperatures, sun loads, velocities, humidity levels and thermal states. The focus is on the air-handling unit, the passenger compartment, the AC-system and the cooling fan for the radiator/condenser. Note that the engine model and engine cooling systems will only briefly be explained. A modular approach is used; consequently it is easy to change systems and/or components. The model is verified against the tests presented in Nielsen et al. [9].

METHOD

An extensive model consisting of relevant sub models of the different systems that affect energy use for the interior climate was created in the commercial software GT-SUITE by Gamma Technologies. The focus of the model was the energy used for the interior climate. Consequently the sub models with greatest impact on this attribute are described in more detail in the following sections.

Modeling Software

The modeling software solves the Navier-Stokes equations, more specifically conservation of mass, conservation of energy, and the conservation of momentum. It solves the equations in one dimension, which means that the quantities are averaged over a volume. A system is discretized into one or more volumes connected by boundaries, see Figure 1. Scalar quantities are assumed to be uniform in the volume, and the vector quantities are uniform for each boundary. Examples of scalar quantities are temperature, pressure and

enthalpy, and examples of vector quantities are mass flow and velocity. This setup is referred to as a staggered grid [17]. For more information regarding the software see [18].

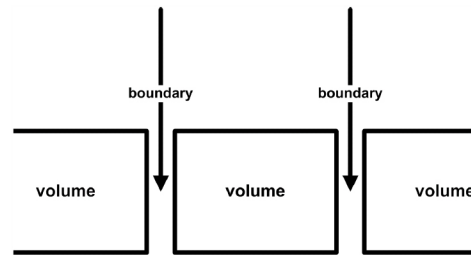


Figure 1. Volumes and boundaries

Modelled Vehicle

The modelled vehicle was a Volvo S60, D5, model year 2012, see Figure 2. It had a 2.4 L diesel engine, Electronic Climate Control (ECC), AC-system with a variable displacement compressor and a Fuel Operated Heater (FOH).



Figure 2. Modelled vehicle, Volvo S60.

Model

The model is divided into several sub models; see Figure 3 for an overview of the modelled systems.

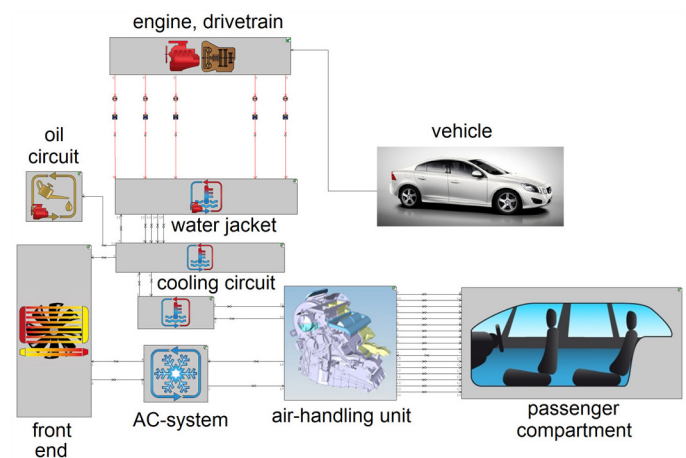


Figure 3. Model overview

The front end sub model consists of the twin electric cooling fans, fan shroud, radiator, the grille and grille shutter and a lumped pressure drop. The condenser is located in the AC-system model.

In general, the data used for calibration of different sub models came from independent component measurements, simulations or specifications. For some areas this was not possible and therefore systems or complete vehicle measurements were used. Relevant sources of data are mentioned in the model sections.

Air-Handling Unit

The air-handling unit is in the center of the climate control system, it conditions and distributes air into the passenger compartment through the outlets. The air-handling unit model consists of the air intake, recirculation-flap, air distribution-flaps, and temperature-flaps as well as blower, blower motor, blower control, filter, ducts and nozzles. For the evaporator and compartment heater see the AC-system model section and the section on the coolant circuit model. The geometry of the air-handling unit is complex; the main reasons for this are the many requirements on, for example, airflow, temperature distribution and size. To reduce model complexity the geometry is not modelled directly, instead pressure drops were created at defined points. One limitation of the air-handling sub model was that no left and right temperature differentiation was included. For an overview of the air-handling unit see Figure 4.

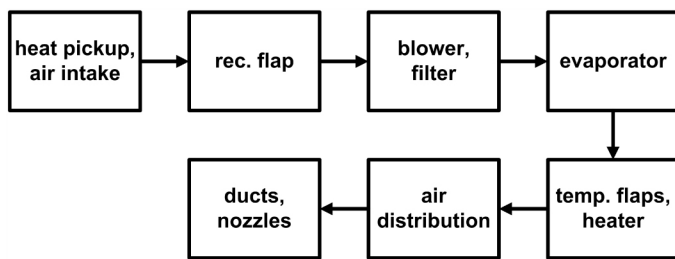


Figure 4. Overview air-handling unit

Ducts and Nozzles

In the air-handling sub model there were fourteen different ducts, subsequently fourteen outlets into the compartment, each with a nozzle: four defroster ducts, four front vent ducts, two front floor ducts, two rear floor ducts and two rear vent ducts. All ducts were modelled as circular ducts with equal constant diameter. The duct pressure drops were modelled with pressure loss coefficients calibrated using CFD simulation data. Each duct used one pressure loss coefficient, Eq. (1), calculated from the volume flow and the mass average pressure at the inlet and outlet of the duct, just before the nozzle.

$$\zeta = \frac{\Delta p}{\rho v^2 / 2}, \quad (1)$$

where ζ is the pressure loss coefficient, Δp is the pressure drop and v is the average duct velocity. The pressure loss due to the expansion from the nozzle to the interior of the passenger compartment was modelled with an orifice of a smaller diameter than the duct. Data for this pressure loss also came from CFD simulation, where the volume flow and the mass averaged pressure from the end of the duct to the outlet of the passenger compartment were used. In other words the

nozzle included the pressure drop of the passenger compartment although this was assessed to be negligible in comparison to other pressure drops.

Heat loss or heat pickup in the ducts was modelled with simple thermal masses connected through convection to ten of the fourteen ducts. The center vent and front floor ducts were too short to show any heat loss or heat pickup. One major simplification was that the thermal masses were only connected to the ducts. That is, there was no heat transfer to the interior of the passenger compartment or the outside. In other words the masses represented the duct mass and physically connected masses although isolated from the rest of the vehicle. The data used to calibrate the convective heat transfer coefficients and thermal masses came from complete vehicle tests, the same tests that were used for the calibration of the passenger compartment model, see the passenger compartment model section. Note that in the results the heat pickup of the ducts are not included in the heat pickup part, it is included in the thermal storage part.

Air Distribution

The air distribution part of the air-handling model controls how the air is distributed between the different modes. The geometry of the air-handling unit, especially between the evaporator and the inlet of the ducts is very complex. In the model this geometry was simplified. The flow was divided into different branches and each branch was composed of at least one pressure loss coefficient and a flap if it was present. The rear vent ducts were always open, therefore, no flap was present in that branch. See Figure 5 for an overview of the different branches.

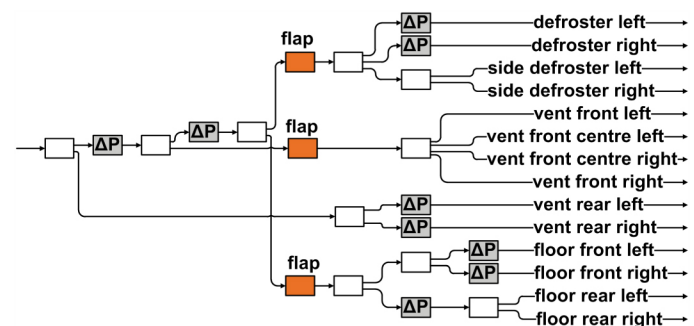


Figure 5. Overview of distribution model.

Four different modes, see Table 1, were simulated with CFD using a model that included the complete geometry from the evaporator to the passenger compartment outlet. The data used from the simulation were volume flow and mass-averaged pressure loss between the evaporator and the inlet of each duct. To find the pressure coefficients, an error function containing the distribution error and the total pressure error was created, Equation 2. This error function was minimized by optimizing the different pressure coefficients. In total nine different pressure coefficients were needed to model the different modes, see Figure 5 for the locations. The pressure drops were modelled in the same way as for the ducts, see Eq. (1).

Table 1. Simulated modes with CFD.

Mode	All	Defrost	Floor	Vent
Airflow (L/s)	120	120	120	120

$$\min(e(\zeta)) = \min \left(\sum_{mode} \left(a \sum_{each outlet} (\dot{V}_{1D}(\zeta) - \dot{V}_{CFD})^2 + b(\Delta p_{1D}(\zeta) - \Delta p_{CFD})^2 \right) \right). \quad (2)$$

In Eq. (2), e is the total error, $\dot{V}$ is the volume flow for each outlet, Δp is the total pressure drop for the mode and a and b are weighting factors.

Flaps

The air distribution flaps controlling defroster, vent, and floor airflow were modelled as simple orifices with a linear relationship between control signal and discharge coefficient. For the temperature-flap, which controls the airflow over the heater, and the recirculation-flap, this linear relationship did not hold, and tables for the relationship between control signal and discharge coefficient were used instead. The data for these relationships came from separate measurements made in a complete vehicle.

Filter

For the filter, a table with pressure drop as a function of volume flow from the supplier was used.

Blower, Blower Motor and Blower Control

A measurement of the blower was made in an airflow rig according to ASHRAE standard [19]. The controlled variables were voltage to the blower motor and the supply airflow. Outputs from the measurement were speed, pressure increase and current. Using the output data, a blower performance map was created.

The electrical motor was modelled as a simple permanent magnet D.C. motor [20]. The relationship between fan speed ω , voltage E , current I , magnetic constant K_m , armature resistance R_a is described by Eq. (3). Eq. (4) describes the relationship between torque τ , magnetic constant, current and magnetic drag torque I_0 ,

$$\omega = \frac{E}{K_m} - \frac{R_a I}{K_m}, \quad (3)$$

$$\tau = K_m(I - I_0). \quad (4)$$

Both K_m and R_a were calibrated from the blower measurements. These factors are temperature dependent although this aspect was neglected in the model.

The blower had a linear control, which controls the blower motor voltage independently of the supply voltage. One major drawback is that the control creates quite a lot of heat which needs to be dissipated. This heat, $\dot{Q}_{con.}$, was calculated according to Eq. (5) and

was added to the flow behind the blower, which corresponds to the physical location of the heat sink. P is electrical power and E_{sup} is supply voltage,

$$\dot{Q}_{con.} = P = (E_{sup} - E)I. \quad (5)$$

Total Pressure Drop, Air Intake, Heat Mode and Leakage

The air path through the air-handling unit is determined by the level of heating, i.e. the fraction of air that passes the heater. Different levels of heating result in different pressure drops, even if the heater pressure drop is accounted for. This pressure drop, the intake pressure drop, the total pressure drop and the leakage characteristics were calibrated with data from a complete vehicle measurement. Two cases were used: one with full heating and one without heating.

The leakage characteristics were unknown and difficult to measure, but a fairly good estimation could be made through an energy balance calculation based on vehicle measurement. Input to the energy balance were mass flow into the air distribution unit, temperature after the evaporator, heat transfer rate from the coolant, temperature after the heater and temperature at the duct inlets. Balance could be obtained by changing the mass flow, i.e. introducing a leakage. Based on experience a leakage was also introduced for the duct connections. In summary two leakages were introduced: one of approximately 5% just in front of the evaporator and one of approximately 10% just in front of the ducts. Note that leaked air still entered the passenger compartment although not through the outlets.

Heat Pickup

In the results, the heat pickup is defined as the temperature increase of the air before it reaches the evaporator. That is, the increase over the hood, at the intake, and in the air-handling unit. The heat pickup in ducts for example is not included in the presented results.

The main causes for heat pickup are the warm engine and the sun load. A fairly simple model was created to simulate these effects. The engine block, assumed to have the coolant temperature, was connected to a thermal mass, representing the hood, which in turn was connected to an air volume. Both the air flow over the hood and the hood-to-air heat transfer coefficient were dependent on the vehicle speed. The sun affected the hood thermal mass directly. All coefficients: the thermal mass, sun effect, heat transfer coefficient for engine to hood and hood to air were experimentally derived. The final air temperature over the hood was used as input to the next step.

The heat pickup in front of the windshield, before the air intake, was modelled in a similar way. The engine block was connected to a thermal mass, which in turn was connected to an air volume using the same inlet temperature as the hood air outlet temperature. The airflow was the same as the climate system airflow except that a minimum airflow was always maintained. The two heat transfer coefficients, engine to thermal mass and thermal mass to air were identical, and they were also experimentally derived.

The main calibration data came from complete vehicle tests carried out in a climatic wind tunnel. However, some minor manual adjustments were made from the main model verification tests presented in the Verification section. The main tests were used because there was a lack of tests with varying speed and idling, which were important for the heat pickup model.

Climate Control Software

In these tests the climate control software that controls the blower, flap actuators, evaporator set temperature and cooling fan level was not implemented in the model. All control signals were from the verification tests. That is, exactly the same inputs were used in both the measurement and simulation.

Passenger Compartment

The compartment model was constructed of 28 volumes of air, the windshield, the rear window, the four side windows, the four doors, the roof and interior masses spread in the air volumes. Two major structures were not modelled: the floor and the engine firewall. Parts of their masses were included in the interior masses, but there was no heat transfer from the engine compartment or between the floor and the ambient. Further, the air leakage was not included. That is, all air exits the model either through the evacuation or through recirculation. For an overview of the model see [Figure 6](#). Note that all connections have been removed to reduce clutter.

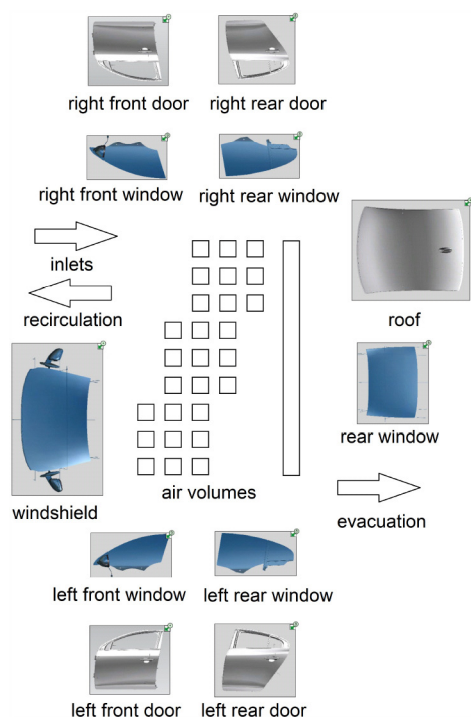


Figure 6. Passenger compartment model overview

Air Volumes

The compartment was divided into 28 separate air volumes: three in height, three in the length, three in width and one rear window volume. Each volume had unique orifices to other volumes depending on location and size. For example the upper front right volume was

connected to the defroster and side defroster, the air volume beneath, on the left and behind. It also had a thermal connection to the windscreen, the right front door, right front side window and the roof. All air volumes had similar connections to relevant outlets, air volumes, windows, doors and the roof.

Each air volume was also connected to an interior mass, which corresponded roughly to the mass in that location of the vehicle. All interior material was modelled with the same thermal properties and these properties were not temperature dependent. See [Table 3](#) for an overview of all thermal properties of the passenger compartment materials. The modelled conductive distance to each mass center was very small, which is admittedly a substantial simplification. However, the reason was to move the conductivity of the mass to the convective heat transfer because the real distance to the mass center would have been difficult to calculate. Furthermore, the combination also reduces variables. The cross-section area approximated the surface area of the materials in the volume. Note that there was no conduction between the masses in the different volumes.

Convective Heat Transfer Coefficients

In general, the convective heat transfer coefficient was modelled according to [Eq. \(6\)](#), see [\[21\]](#). The heat transfer coefficient is h , v is the air velocity over the surface, c , d , and k are constants.

$$h = c(d + kv^{0.5}) \left(\frac{W}{m^2K} \right) \quad (6)$$

This correlation was used on all external surfaces except the windshield. On the windshield the coefficient was increased by 20% due to higher velocities over the surface which decreased the boundary layer. The vehicle velocity was used for all air velocities.

In the compartment the same correlation was used, but with some modifications. The air velocity in each volume was used instead of the vehicle velocity. In addition, the heat transfer coefficients were divided into two groups: the first group consisted of all connections between the air volumes and the shell, i.e. windshield, rear and side windows, doors and roof, the second group was composed of all connections between air volumes and the interior masses. Each one of these groups had one common multiplier on the heat transfer coefficient. This multiplier depended on total air flow in the compartment, recirculation degree and mode. The correlations between the multiplier and the input factors were calibrated with data from complete vehicle tests. See [Table 2](#) for a complete list of these eleven calibration tests made in a climate chamber.

Table 2. Passenger compartment calibration tests

Temperature (°C)	Mode	Blower voltage (V)
35	Vent, Floor, Defrost, Allmode	8
35	Vent	5, 10, 12
35	Vent, 25, 50, 75 and 100% recirculation	8

There were some exceptions to the presented convective heat transfer coefficients. For example, during sun soak, natural convection was used on external surfaces. Additionally two different constant heat transfer coefficients were used inside the compartment, one of 1 W/m² K for all heat transfer from interior masses and 4 W/m² K for all heat transfer to the shell. The natural convection was calculated for a flat plate with either higher or lower temperature than the ambient temperature, for correlations used see [22]. Other exceptions were that the nozzle air velocities affected the coefficient for the inside of the windscreen and front side windows. That is, accounting for the effect of more air hitting the surfaces directly.

Windows

The windows were modelled as thermal masses with the thermal properties presented in Table 3. Both the exterior and interior heat transfer used the correlation presented in the earlier section Convective Heat Transfer Coefficients. The radiant flux on and through the window is described in section Radiation.

Table 3. Passenger compartment materials

material	Thermal conductivity	Specific heat	Emissivity	Absorptivity	Reflectivity
	W/m K	J/kg K			
ABS Plastic	0.17	1400	0.9	-	-
Carbon Steel	~60.50*	~434*	0.9	0.75	-
Cabin material	2	1100	-	1	-
Glass	0.9	840	0.8	.48	0.06

*temperature dependent

Doors

The door model consisted of an outside surface thermal mass, an air cavity, an intermediate thermal mass, another air cavity and the interior surface thermal mass. Consequently there were two isolated vertical air volumes in which the heat transfer was only by natural convection and radiation. The natural convection was calculated for an enclosed vertical cavity, for the correlation see [22]. The outer surface material and the intermediate mass material was carbon steel, and the inner surface was plastic, see Table 3 for thermal properties.

Roof

The roof was modelled with one outer surface thermal mass, one enclosed horizontal air cavity and an interior surface mass. Similar to the doors, the only heat transfer through the air cavity was from natural convection and radiation. The outer surface material was carbon steel, and the interior surface material was plastic. Note that the temperature used for the heat transfer to the interior air volumes used the average for all adjacent air volumes, i.e. the same surface temperature for the whole inside of the roof.

Radiation

The radiation heat transfer was simplified in many different ways, the largest being: First, there was no interior radiation, i.e. no masses in the compartment exchanged energy with another through radiation. Second, the transmitted solar radiation only affected the interior masses adjacent to the windows, i.e. some masses never receive any radiation despite being affected by solar radiation in real cases. Third, no indirect/diffuse radiation was included except for the outer surfaces. Despite these simplifications many effects were included in the model such as solar view factor of all exterior components.

The solar radiation could either be reflected from the window, absorbed in the window or transmitted through it. The incident angle affected the distance the radiation travelled through the windows, i.e. the absorptivity. All transmitted radiation was added to the masses adjacent to the windows, i.e. none of the incoming solar flux was reflected inside the compartment. For the roof and the doors the solar radiation could either be reflected or absorbed. When absorbed the radiation increased the temperature of surface mass.

AC-System

The AC-system included the compressor, condenser, evaporator, TXV, and pipes. Each part was calibrated from component measurement data, and no system calibration was performed.

Compressor

The compressor was a variable displacement compressor with a volume of 167 cm³. Unfortunately there was no calibration data available for the actual compressor in the vehicle; instead the model used data for an updated version of the same compressor. There were only minor differences between the two compressor variants.

The compressor model used a performance map created from a measurement with different compressor speeds and inlet/outlet pressure ratios. The input for the map was: compressor speed, suction and discharge pressure, suction and discharge temperature and volumetric efficiency. However, only one map was used, the one for 100% displacement. Instead of decreasing the displacement the model decreased the mass flow with a multiplier to decrease cooling power. This was done due to lack of knowledge concerning the displacement during the measurement, consequently creating good maps for displacement levels other than 100%. Although assumptions about the displacement were made and a better fit of calibration data were achieved the drawbacks were large compared with the mass flow approach. For example, irregularities in the maps due to varying displacement generated convergence issues for the simulation. A more thorough discussion of these assumptions can be found in the discussion section.

To control the compressor mass flow multiplier, two serial PI-controls were used. The first controller used the measured air temperature after the evaporator as well as the evaporator target temperature to generate a suction pressure set point through a pulse width modulation (PWM) signal. The second controller generated a mass flow multiplier using the set point and measured suction pressure.