THESIS FOR THE DEGREE OF DOCTOR OF PHILOSOPHY

AUTOMOTIVE CLIMATE SYSTEMS
Investigation of Current Energy Use and Future Energy Saving Measures

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Building Services Engineering
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CHALMERS UNIVERSITY OF TECHNOLOGY
Göteborg, Sweden 2016

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ISBN 978-91-7597-399-9

Doktorsavhandlingar vid Chalmers tekniska högskola
Ny serie nr 4080
ISSN 0346-718X

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Printed by
Chalmers Reproservice
Göteborg, Sweden 2016

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ABSTRACT

Automotive climate systems use energy to achieve thermal comfort for the vehicle passengers. This energy use affects vehicle fuel consumption. The objectives of this thesis are to understand the energy use of automotive climate systems and investigate the effect of different energy saving measures.

The research was conducted in several steps. First, comprehensive laboratory measurements of a complete vehicle, a Volvo S60. The main focus of the measurement was on heat flows and electrical and mechanical work of the climate system. Second, the most important climate systems were modelled with a one dimensional commercial software. The modelled systems were the passenger compartment, air-handling unit and air conditioning system, although engine, water jacket, cooling circuit, oil circuit and drivetrain were also included. Third, development of a test cycle representative for real-world conditions. The test cycle was based on hourly ambient conditions around the world weighted with sales distribution of Volvo Cars and departure time. In the last step, the model and developed test cycle were used to investigate different energy saving measures.

The measurement demonstrated that the energy use can be reported individually for better understanding of the system. That is, the different heat flows from sources to sinks and the electrical and mechanical work can and should be presented separately. Furthermore, the developed test cycle showed that intermediate conditions, ambient temperatures from 5 to 22°C, were by far the most common. Combining the simulation model and test cycle provided an estimation of current energy use of automotive climate systems. In average the system used 180 W of electrical power, 475 W of mechanical power, a total of 1820 W for heating and 1030 W for cooling. The average heat flow into the passenger compartment was 1190 W for heating and 280 W for cooling. 26 energy saving measures were investigated. Few single energy saving measures could decrease the energy use significantly, however, combinations of measures had a large potential. A reduction of the electrical power with 50% and the mechanical power with 44% were possible with realistic measures. Further, the heat flows into the passenger compartment could be reduced with roughly 20% for both heating and cooling. Measures on the source side, how the heating and cooling was generated, showed most potential.

The results show that how the system operates in intermediate conditions determines the energy use. The interaction between the automatic climate control system, the air conditioning system and the requirement of de-humidification have a large influence on the operation of the climate system in these conditions.

Keywords: Air conditioning, fuel consumption, automotive, HVAC, simulation and modeling, thermal management, passenger compartment, energy efficiency, energy consumption, testing

This study has been funded by Volvo Cars and Fordonsstrategisk Forskning och Innovation (FFI), Energi & Miljö.

Apart from the unknowns,
everything is obvious.

James P. Hogan

Acknowledgement

This thesis could not be done without the work of others and this is my opportunity to thank all the people that made this research possible.

I would like to thank my main supervisor Jan-Olof Dalenbäck, mainly for letting me do this research my way and immediately support me when I required. My industrial supervisor Åsa Uddheim deserves my appreciation for protecting me from all kind of administrative work and for always having insightful comments on my writing. I now see challenges instead of problems. Furthermore, I would like to thank my supervisor Torbjörn Lindholm for all the help with the thesis.

Many thanks to my closest colleague at Volvo Cars, Sam Gullman, whose excellent ideas are the basis for many of the simulation models and whose opinion I value highly. Further, he has also endured countless lunch discussions and my endless complaining over the years. Fredrik Wallin should also be mentioned for his outstanding modelling contribution. I also express thanks to my manager at Volvo Cars, Thomas Landelius and technical expert Ronnie Hansson for always showing great interest in all my results. Further, I would like to thank all my colleagues at the Climate Control System department, the climate wind tunnel staff and the excellent mechanics at Volvo Cars.

All past and present people in the Division of Building Services Engineering at Chalmers that I have meet should also be acknowledged for accepting me as one of them, despite that my building had wheels.

Volvo Cars should be acknowledged for not only claim that research is important but also is willing to fund it.

Finally I would like to thank my family, especially my wife and children; Emma, Assar and Ingrid. Without you this research would have been finished a long time ago, but what would have been the point of that?

Although I really want to blame others for mistakes made in this work, it is not possible. This is my thesis, faults and all, and only I can be blamed for it.

Jag vill ju bestämma som jag vill!

Göteborg, May 2016

Filip Nielsen

LIST OF PUBLICATIONS

This thesis is based on three peer-reviewed journal articles and one peer-reviewed conference article that are appended and cited according to the following order:

I. Nielsen, F., Uddheim, Å., and Dalenbäck, J., "Measurements of Energy Used for Vehicle Interior Climate," *SAE Int. J. Passeng. Cars - Mech. Syst.* 7(4):1404-1416, 2014, doi:10.4271/2014-01-9129.

II. Nielsen, F., Gullman, S., Wallin, F., Uddheim, Å. et al., "Simulation of Energy Used for Vehicle Interior Climate," *SAE Int. J. Passeng. Cars - Mech. Syst.* 8(4):1218-1234, 2015, doi:10.4271/2015-01-9116.

III. Nielsen, F., Uddheim, Å., and Dalenbäck, J., "Reduction of Energy Used for Vehicle Interior Climate," SAE Technical Paper 2016-01-0250, 2016, doi:10.4271/2016-01-0250.

IV. Nielsen, F., Uddheim, Å., and Dalenbäck, J., “Potential Energy Consumption Reduction of Automotive Climate Control Systems,” submitted to *Applied Thermal Engineering*.

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Symbols, Abbreviations and Definitions

Symbols

E	Energy, [J]
I	Current, [A]
P	Power, [W]
$\dot{Q}$	Heat flow, [W]
$\dot{Q}_{ducts}$	Heat flow, to duct masses, [W]
$\dot{Q}_{engine}$	Heat flow, engine waste heat, [W]
$\dot{Q}_{evacuation}$	Heat flow, out from passenger compartment, air evacuation, [W]
$\dot{Q}_{FOH}$	Heat flow, fuel operated heater, [W]
$\dot{Q}_{heat\ pickup}$	Heat flow to the air in the air handling unit up to the evaporator, [W]
$\dot{Q}_{lat.\ cool}$	Heat flow, latent cooling evaporator, [W]
$\dot{Q}_{leakage}$	Heat flow, out from passenger compartment, air leakage, [W]
$\dot{Q}_{mass}$	Heat flow, to interior masses, [W]
$\dot{Q}_{other}$	Heat flow, non-air to and from the passenger compartment air, [W]
$\dot{Q}_{outlet}$	Heat flow, out from passenger compartment by air, [W]
$\dot{Q}_{pass.\ comp.}$	Heat flow into passenger compartment, [W]
$\dot{Q}_{person}$	Heat flow, sensible, from one person, [W]
$\dot{Q}_{rec}$	Heat flow, recirculation, [W]
$\dot{Q}_{sen.\ cool}$	Heat flow, sensible cooling evaporator, [W]
$\dot{Q}_{shell}$	Heat flow, out from passenger compartment through the shell, i.e. doors and roof, [W]
$\dot{Q}_{windows}$	Heat flow, out from passenger compartment through windows, [W]
U	Voltage, [V]
$\boldsymbol{V}$	Velocity, [m/s]
$\dot{V}$	Volume flow, [m^3/s]
$\dot{W}_{blower}$	Rate of work, blower electrical, [W]
$\dot{W}_{comp}$	Rate of work, compressor mechanical, [W]
$\dot{W}_{electrical}$	Rate of work, electrical, [W]
$\dot{W}_{mech}$	Rate of work, mechanical, [W]
c_p	Specific heat capacity, constant pressure, [J/kg K]
gZ	Potential energy, [J]
h	Enthalpy, [J]
k_{ratio}	Compressor axle gearing, [-]
m	Mass, [kg]
$\dot{m}$	Mass flow, [kg/s]
ΔP	Pressure difference, [Pa]
ΔT	Temperature difference, [K]
$\eta_{alternator}$	Efficiency, alternator, [-]
τ_{comp}	Compressor torque, [Nm]
$\omega_{eng\ rpm}$	Angular speed of engine, [RPM]

Subscripts

C.V.	Control volume
e	Control volume, exit
i	Control volume, inlet
osa	Outside air
rec	Recirculated air

Abbreviations

1D	One Dimensional
3D	Three Dimensional
AC	Air Conditioning
CAD	Computer-Aided Design
CFD	Computational Fluid Dynamics
CLTD	Cooling Load Temperature Differential
ECC	Electronic Climate Control
EV	Electric Vehicle
FOH	Fuel Operated Heater
HVAC	Heating, Ventilation and Air Conditioning
HWFET	HighWay Fuel Economy Test
IAQS	Interior Air Quality System
ICE	Internal Combustion Engine
MAC	Mobile Air Conditioning
NEDC	New European Driving Cycle
NREL	National Renewable Energy Laboratory
OSA	OutSide Air
PMV	Predicted Mean Vote
PPD	Predicted Percentage of Dissatisfied
PWM	Pulse-Width Modulated
REC	RECirculation
TC1	Test Cycle 1
TC3	Test Cycle 3
TC5	Test Cycle 5
TXV	Thermal eXpansion Valve
WLTP	Worldwide harmonized Light vehicles Test Procedures

Definitions

Air-handling unit

The *air-handling unit* includes air intake, recirculation-flap, air distribution-flaps, and temperature-flaps as well as blower, blower motor, blower control, filter, evaporator, heater, ducts and nozzles. Often the designation HVAC is used for the same unit, however, the *air-handling unit* does not include the AC-system except the evaporator. Furthermore, only the heater from the engine cooling system is included. That is, the *air-handling unit* include the path of the incoming air.

Air distribution

The *air distribution* is the part of the air-handling unit which distributes the air to the different ducts, i.e. outlets. Sometimes the ducts can be included in the designation. There are many different combinations of modes, i.e. distribution between different ducts. Pure modes are defrost, when air is directed to the windshield and front side windows. Vent, when air is directed through the instrument panel outlets and floor, when air is directed to the front and rear floor. All-mode releases air in all outlets, floor-defrost releases air in both floor and defroster outlet and floor-vent in both floor and panel vent outlets.

Climate system/Climate control system

Climate system and *climate control system* are used interchangeably throughout the thesis and they denote the same thing. The *climate system* includes all sub systems devoted to the interior climate; the air-handling unit, AC-system, automatic climate control and fuel operated heater.

Cool down

Cool down is defined as a process where the end temperature is significantly lower than the start temperature, i.e. where heat have to be removed from the passenger compartment. This process can also be called pull down.

Front end

Front end is defined as the front of the vehicle where the radiator, twin electric cooling fans, AC-system condenser and various other heat exchangers are located.

Heat pickup

Heat pickup refers to the temperature and energy increase of the incoming air up to the evaporator due to hot surfaces. The hot surfaces are mainly due to engine waste heat and sun load, however, *heat pickup* can also come from blower inefficiencies and blower control. Sometimes this can also be referred to as *heat addition*.

Heat up

Heat up is defined as a process where the end temperature is significantly higher than the start temperature, i.e. where heat have to be supplied to the passenger compartment.

Heater

The *heater* is defined as the heat exchanger transferring heat from the coolant circuit to the incoming air.

Reheat

Reheat is defined as the mode of operation where the air is initially cooled, mainly for dehumidification, and then reheated to achieve the required temperature.

Soak

Soak is defined as the effect of ambient conditions on the passenger compartment with the climate system disengaged. Mainly used in combination with sun load.

Steady state

Steady State is defined as both the engine and the compartment have their normal operating temperature for those conditions. That is, no large temperature changes of the passenger compartment, the engine or any other components during the test. Note that other parameters, such as velocity, does not need to be constant.

Transient state

A *transient state* is defined as a process where the initial and end states are not comparable. For example, in a test in cold conditions both the engine and the passenger compartment have significantly higher temperatures at the end compared to the start condition.

1 INTRODUCTION

This chapter provides background, objectives, method, thesis structure and limitations.

1.1 Background

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, see for example Burgdorf, Johnson and Johnson, [1–3]. This emphasis on fuel consumption has led to several improvements of passenger vehicles, mainly in engine efficiency, aerodynamics and alternative powertrains. These areas have large effects on current certification cycles, but other areas which are excluded or negligible in the cycles can also have a large effect on real-world fuel consumption. In general auxiliary systems are not engaged in certifications cycles and one of these systems is the climate system. The climate system use energy for heating and cooling the passenger compartment, maintaining comfort, de-icing and de-misting windows and providing good air quality. One sub system of the climate system, the Air Conditioning (AC) system is the largest auxiliary load on a vehicle. According to Johnson [4], 6% of the domestic petroleum consumption in the US is used for vehicle AC-systems. Another source present a figure of 3.9%, see [5].

The effect of the climate system on fuel consumption has received increasing attention by research communities, industry and regulatory agencies. In the US a newly-developed cycle, AC17, focused on the AC-system is being introduced [5]. This cycle will include four elements; a pre-conditioning cycle, a 30-minute solar soak period, a SC03 and a “HighWay Fuel Economy driving schedule” (HWFET). These steps are done with and without the AC-system engaged. The measured fuel consumption difference is attributed to the AC-system. The new test procedure “Worldwide harmonized Light vehicles Test Procedure”, WLTP, developed by European Union, Japan and India intend to take the AC-system into account in phase 2 [6], however, the time and format of this is still undecided.

One area that has received special focus is the effect of climate systems on the range of Electric Vehicles (EV). Due to waste heat shortage the effect on an EV is much larger compared to a vehicle with an Internal Combustion Engine (ICE). Several authors have published papers in this area, see for example Farrington and Rugh, Lohse-Busch et al., Kambley and Bradley, [7–9]. However, even for vehicles with an ICE there are increasing challenges due to improved engine efficiency. Less waste heat available for heating the passenger compartment is a current issue.

In summary, the number of requirements on the climate system are increasing. Not only are there requirements on heat up and cool down performance, cost, size, automatic climate control, air quality among others but there are also future requirements on energy efficiency and heating without waste heat surplus.

1.2 Previous Research

Automotive climate systems have been a research area for quite a long time, however, the amount of research has recently increased significantly. The stated causes in the previous section might be the reason for this. That is, the energy consumption of the climate system in EVs have a much larger impact compared to vehicles with combustion engines. Furthermore, the increased focus on fuel consumption and emissions from vehicles due to environmental issues are other reasons. In this chapter a review of the research in three different, although related, categories is presented. The first category includes analysis of fuel consumption of the climate system, the second focus on modelling and simulation of climate system and the third category contains energy saving measures.

1.2.1 Fuel consumption analysis

Generally, there have been four different methods to analyse fuel consumption, CO_2-emissions, energy consumption or power use of the climate system: Complete vehicle testing, test benches for AC-systems, simulations and overall fleet consumption calculations. The focus has primarily been on the AC-system, heating has regularly been neglected due to the abundance of waste heat available in internal combustion engines and its low impact on fuel consumption. Other energy users in the climate system have rarely been investigated.

Complete vehicle testing

Complete vehicle testing is used by automotive industry, see for example de Moura and Tribess [10], but there are also examples from other organisations. Weilenmann et al. [11], Weilenmann et al. [12] have tested AC-system fuel consumption for six gasoline vehicles in 2005 and six diesel vehicles in 2010. Some notable results were that the fuel consumption could be as high as 82.7 g/km for the AC-system in some cases and that some vehicles used twice as much fuel as other for the AC-system for a comparable conditions. However, there are many drawbacks with complete vehicle testing, for example, it is expensive and time consuming and this limits the tested conditions and vehicle variants. Despite these disadvantages the method is commonly used. Complete vehicle testing is also going to be used in the upcoming regulations regarding fuel consumption of the AC-system, see [5, 6]. For an example of a test procedure measuring air conditioning fuel use, see Rugh [13].

Test benches

Test benches for AC-systems are rather common and has been used extensively by both vehicle manufacturers and suppliers. Gaveau and Clodic [14] tested six parameters, evaporator and condenser airflows, initial compartment temperature, evaporator and condenser air temperatures and compartment set point temperature. The output was, among other, compressor and fan power levels. Gado et al. [15]

also made a series of AC-system tests in a test bench with a simulated passenger compartment focusing on transient states. The results included total energy used during the test cycle.

Simulations

There are many simulation models available that calculates fuel consumption for vehicles. Silva et al. [16] analysed three: EcoGest, developed at Instituto Superior Técnico, Comprehensive modal emission model developed by the University of California at Riverside and the University of Michigan and ADVISOR developed at the National Renewable Energy Laboratory (NREL). However, these models were focused on the total vehicle fuel consumption and the climate system energy use was only an adjustable constant. Therefore, the models could only analyse the effect of climate systems on the total fuel consumption when the energy used for vehicle interior climate already was known. The actual energy use of the climate system required another source.

Brizard [17] presented a model more focused on the actual fuel consumption of the climate system. Using the commercial software AMESim a heat up of hybrid vehicle was analysed for different electrical heater settings and thermal storage configurations. Cool downs were also simulated with different settings on the AC-system and recirculation degree. A different approach is presented by Rugh [18], many different software's were used to evaluate techniques of reducing soak temperatures and then calculate the decrease of fuel consumption. In the first step a Computer-Aided Design (CAD) model of the vehicle was created, then Vehicle Solar Load Estimator was used together with NREL's solar radiation model to calculate solar loads for the passenger compartment thermal model. Computational Fluid Dynamics (CFD) was used to calculate soak temperatures, a human thermal comfort model and a transient AC-system model was combined to compute required compressor power. The compressor power was used as an input to ADVISOR which calculated the fuel consumption. In general, the combination of fuel consumption models and advanced climate system models are unusual. See chapter 1.2.2 for more examples of climate system models.

Overall fleet consumption calculation

In a paper by Johnson [4] the total energy used for the air conditioning in all cars and trucks in USA for one year was calculated. First a mean radiant temperature was computed for cars in different cities using weather data. Second, the Predicted Percent Dissatisfied (PPD) was calculated. All dissatisfied, due to high temperatures, was assumed to engage the AC-system and combining this with driving data the fuel consumption could be estimated.

1.2.2 Modelling and simulation of climate system

Climate systems have been modelled for a long time, Davis et al. [19] made simulations on a system with evaporator, condenser, compressor, Thermal eXpansion Valve (TXV) and passenger compartment. The evaporator, condenser and compressor were quite thoroughly modelled whereas the passenger compartment and air-handling unit were less detailed.

Cherng and Wu [20] used almost the same models as Davis et al. [19] but improved the passenger compartment to three dimensions, added heating and used a more user friendly interface on a personal computer. However, the passenger compartment model was not explained well in the paper and no verification of the models was presented.

In two papers presented by Ingersoll et al. [21] and Ingersoll et al. [22], a passenger compartment model derived from a more advanced three dimensional (3D) CFD-model was used. The passenger compartment consisted of 82 surfaces and 36 volume elements and the implementation was focused on solar radiation. For example, single rays were traced through windscreens and randomly determined, according to the material properties, if reflected, transmitted or absorbed by different shapes in the cabin. The passenger compartment model was not verified with real data, instead with a CFD-model. In the second paper a human thermal comfort model was combined with the passenger compartment model.

In the nineties several papers concerning simulation of the AC-system and components were published although no model combined the passenger compartment with the rest of the system before Selow et al., Khamsi et al. and Huang [23–25]. The model made by Huang was the most complete of the three models and it included many components, pressure drop, air-handling unit, transient effects on components and humidity. A lumped system model with one air zone was used for the passenger compartment. The model was further explained in Arici et al., Huang et al., Huang et al. and Huang et al. [26–29]. This model was verified with real tests. Khamsi and Petitjean [30] included a validation, with good results, of the model presented earlier, [24].

Ding and Zito [31] summarized previous work and derived relationships for the cabin from basic equations. This approach could not simulate all effects on the passenger compartment, just show relationships and no verification with real tests was presented.

Another common approach to climate system modelling is to use CFD combined with a thermal comfort model and one of the earliest papers on this subject was Lin et al. [32]. A simplified passenger compartment was simulated with different air-flows, modified windows, different location of outlets and evacuations. The resulting temperatures and air velocities were combined with the predicted mean vote (PMV) model and the PPD was calculated for each cell in modelled volume. This was further developed in Han et al. [33], by using a more advanced thermal comfort model. The CFD-calculations of the passenger compartment was verified in Huang and Han [34]. In Han and Chen [35] and Han et al. [36] the method was used for analysing the effect of insulation, thermal mass reduction and solar reflecting windows on passenger compartment soak time and temperatures and the cool down afterwards. Wolfahrt et al. [37] combined many different levels of modelling to evaluate de-icing, de-misting and thermal comfort for different test cases. Zero dimensional simulation methods for the drivetrain, one dimensional (1D) models for the cooling system and 3D models for under hood and passenger compartment flows.

Pitchaikani et al. [38] developed a complete system with simplified engine and transmission model, a somewhat more advanced Heating, Ventilation, Air Conditioning (HVAC) system and passenger compartment model using Modelica language in a Dymola environment. The aim was real-time testing of a climate control software which was successful according to the article but no verification with real tests was presented.

A completely different approach was used by Zheng et al. [39]. The paper explains how Cooling Load Temperature Differential (CLTD) was used for estimating cooling heat loads of a truck cabin. CLTD is a method used for approximation of heat gain through walls in buildings.

Kossel et al. [40] investigated a simulated trip with a coach. The AC-system, the heating circuit, the passenger compartment, longitudinal dynamics, driving profile and ambient conditions were each modelled with different complexity. They presented, among others, the passenger compartment temperature, different energy flow rates and the coefficient of performance for the AC-system during the trip.

Ghebru et al. [41] presented results of heat-ups modelled in Matlab/Simulink. It was possible to simulate recirculation, electric heater and optimised airflow and distribution control with this approach, the presented results included temperatures and heat flows. The model was developed and validated with real tests.

The number of simulation models focused on climate systems have increased significantly during the last years, see for example [42–48]. However, most of these models have focused on the AC-system and passenger compartment. Many models have not included other systems and the climate control is frequently very basic. In other words, many real world effects and limitations are excluded and this affects the results. However, there are some exceptions of research which includes more systems, see for example Gravelle et al. [49].

In summary, there are a lot of simulation models focused on climate systems, ranging from 1D system models to 3D CFD models. Unfortunately, many models exclude important sub systems, such as condenser cooling fan, air-handling unit and automatic climate control systems.

1.2.3 Energy saving measures

There are many proposed energy saving measures in the literature and in this chapter some of them are reviewed.

One of the most researched areas is reducing the thermal load on the passenger compartment. Levinson et al. [50] investigated reduction of solar load by using reflecting colours on exterior surfaces. Akyol and Kilic [51] made a similar investigation on exterior colours but also included reflecting and absorbing glass. Moreover, there are many studies of glazing technologies, see for example Rugh et al. [52] and Türler et al. [53], the latter also included insulation investigation. Further studies of glass are included in Gravelle et al., Bridge et al., Bharathan et al. and Jeffers et al. [49, 54–56]. All these four papers also includes many more measures to reduce the passenger compartment load, for example pre-ventilation, insulation, ventilated seats, zonal cooling, blinds, reflecting exterior and interior colours. Zhang et al. [57] simulated different energy saving ideas such as insulation and reducing the transmitted radiation into the passenger compartment.

Another method that could reduce the energy use is localized cooling and heating, i.e. the passenger compartment is only heated or cooled where it’s needed. This method can also be called zonal cooling or heating. Besides the previously mentioned research, Huang et al., Oh et al., Kwon et al. and Wang et al. [58–61] also presented research in this area. Tabei [62] included, in addition to zonal heating and cooling, a split HVAC which could recirculated air in the floor region without increasing the humidity in the defroster air. This reduced the energy needed for heating.

There is also research on the efficiency of the AC-system. For example, Lin et al. [63] describes a desiccant wheel assisted AC-system which could decrease the total power input with 20% by decreasing latent cooling. Subiantoro et al. [64] also investigated measures aimed at reducing the effect of humidity as well as increased passenger compartment temperature and improved condenser heat exchange. Ünal and Yilmaz [65] describe how a two phase ejector increases the efficiency of the AC-system and Fritz et al. [66] investigated how different storage technologies could reduce the energy use of the AC-system. Khayyam et al. [67] reported that a coordinated energy management system could reduce energy consumption of the AC-system and still maintain comfort. The impact on fuel consumption when preventing the AC-system to be used at lower ambient temperatures was investigated by Monforte and Mandrile [68]. According to Roscher et al. [69] energy could be saved for an EV, by limiting the maximum HVAC power depending on the power demand of the drivetrain, thanks to reduced losses.

One relatively new system in the research area of climate systems is heat pumps. Ahn et al. [70] compares single source heat pumps with dual source using both air and waste heat as heat source. Hosoz et al. [71] compares a heat pump using different heat sources with a heat pump using waste heat directly from a diesel engine. Fleming et al. [72] investigated the effect of thermal batteries added to the heat pump setup. Another approach for improving the efficiency of heating is done by Diehl et al. [73] and Chiew et al. [74]. Waste heat in the exhaust is used to heat the passenger compartment. Waste heat have also been considered for cooling the passenger compartment with an absorption or adsorption cycle, see for example Talom and Beyene [75] and Verde et al. [76].

In summary, there are many interesting energy saving measures, however, it is very difficult to estimate their effect on the total energy use for vehicle interior climate, because they are all evaluated during different conditions and many sub systems are not included in the analysis.

1.3 Objectives and Method

The objectives of the thesis are to understand the processes involved in climate systems, state which factors are important for the energy use for interior vehicle climate and investigate energy saving measures. Furthermore, if required investigated and developed different methods for evaluating the energy. The objective can be summarized as "*How should future vehicles be designed with the aim of using much less energy for the interior climate compared to vehicles designed today*?"

The method used in this thesis consisted of five major parts, the first part was a literature study focused on fuel consumption attributed to the automotive climate system, modelling and simulation of climate systems and energy saving measures. The second part was a thorough measurement and analysis of the energy used for interior climate of a current vehicle in a climatic wind tunnel. The third part was 1D modelling of the important climate systems for the energy use, mainly the passenger compartment, air handling unit and AC-system. Furthermore, a comprehensive validation of the simulation model using the measurements obtained in the second part was also included. The fourth part included the development of a real-world representative test cycle which incorporated all different modes that the climate system operates in. The last part was a study of the effects of many different energy saving measures using the simulation model and the developed test cycle.

1.4 Disposition

The thesis is divided into nine chapters with the full versions of the four journal and conference papers included in the appendix. Chapter 2 includes a general description of the climate system, consisting of the passenger compartment, the air-handling unit, AC-system, engine cooling system and control system. Many properties are compared in the thesis and papers, a more thorough review of these are included in chapter 3. In the next chapter, 4, paper I, regarding the complete vehicle measurement, is summarized. Furthermore, some supplementary results concerning steady state heat flows and the airflow estimation is also found in this chapter. The journal paper on the model, paper II, is discussed in chapter 5, together with an additional compressor power comparison. The next chapter, 6, includes a summary of the test cycle, an investigation of the energy use of the base case and some comparison with other test cycles. The results from paper III and IV on the reduction of energy use can be found in chapter 7. Furthermore, results from other investigations completed during the project are also included. In the next chapter, 8, discussion of the strengths and limitations of the thesis and some additional aspects on the total energy use, can be found. The chapter also includes the main conclusions of the thesis. The last chapter, 9, deals with future research in the area of energy use for vehicle interior climate.

1.5 Limitations

The area of automotive climate system is large with many different aspects, everything from phase changing refrigerants to subjective evaluation of thermal comfort. In order to make a suitable investigation of the energy use for vehicle interior climate the investigation needs therefore to be limited. This section includes the main limitations of the thesis.

Vehicles with an ICE will probably retain its dominant market position for the foreseeable future, see for example [77]. For that reason the main focus of the thesis is on these types of vehicles. Furthermore, the legislation regarding ICE vehicles and climate system are also increasing, see [3]. Moreover, including several other types of drivetrains, for example plug-in hybrid electric vehicles, would expand the research excessively. However, many of the results are applicable on other types of drivetrains despite the focus on ICE vehicles.

An in depth investigation of the energy use for vehicle interior climate requires a narrow approach on which system to look into. Therefore, one limitation was to only include one main type of climate system as this will enable a more thorough investigation.

Another area which is deliberately omitted from the thesis is the evaluation of thermal comfort. Including thermal comfort would have increased the complexity significantly. Instead it is assumed that the tested vehicle fulfil the requirements on thermal comfort and that this is true for similar conditions in the passenger compartment. However, this limits the possible energy saving measures that can be investigated, for example, zonal heating and cooling cannot be evaluated.

2 AUTOMOTIVE CLIMATE SYSTEM

Automotive climate system has been a part of the vehicle for a long time and the first vehicles used the same solutions that were available for horse and carriage. For example charcoal foot heaters, robes, gloves, coats and hats, see [78] . Over the years the system has had many different formats, but, during the last years the basic system seems to conform to more or less the same structure, independent of vehicle type or manufacturer. However, even if the basic system have remained the same more and more features have been added, especially during the last twenty years. For instance, AC-systems have become standard, not only in the US but also in Europe. Furthermore, automatic climate systems are quite common and more emphasis are placed on air quality, especially by premium brands. Another feature that is increasing is additional heaters. Because less waste heat is available due to more efficient combustion engines, cold markets now requires additional heaters such as Fuel Operated Heater (FOH) and electrical heaters to fulfil comfort requirements. Furthermore, electrical powertrains have even less available waste heat which increases the need for additional systems even further. However, the basic system has not changed significantly and will probably be used for a significant time.

In this chapter the different parts of the climate system of the investigated vehicle are described in detail.

2.1 Overview

Throughout this work the same vehicle have been used for both measurement and simulations. The vehicle was a Volvo S60, D5 of model year 2012, see Figure 2.1. It was equipped with a 2.4 l diesel engine, Electronic Climate Control (ECC), AC-system with a variable displacement compressor and a FOH. The described system in this chapter applies for this vehicle although many systems are similar throughout the automotive industry. Moreover, the vehicle model, S60, was launched in 2010 and the platform which the model is based on was launched in 2006.

Figure 2.1 Volvo S60

The climate system consists of many different parts, see Figure 2.2 for an overview. In general, air is supplied from the outside, the passenger compartment or a combination. Then the air is filtered, cooled, possibly dehumidified, heated and distributed into the passenger compartment in different outlets depending on ambient conditions. The area inside the dotted line in Figure 2.2 consists of the parts that are denoted HVAC in the automotive industry. However, most of the cooling and heating systems are not included. Therefore the more general term air-handling unit is used in this work for the HVAC and the ducts.

There are five different objectives for the automotive climate system: Heat up, cool down, maintain comfort, de-frost and de-mist and maintain good air quality. The two first objectives are easily understood, heat or cool the passenger compartment as fast as possible until comfort temperature is reached. The purpose of the third objective, maintain comfort, is that the comfort temperature should be maintained for the entire driving time, even if the conditions changed dramatically. Most people driving vehicles believes that it is important to see out of the passenger compartment. Hence, the fourth objective, keep the windows clear of ice and mist is safety related. The last objective is maintaining good air quality, i.e. keep the levels of pollutants low, both from the inside of the compartment and from the outside. Furthermore, the climate system must also manage individual customer settings of different types as thermal comfort to a large degree is subjective.

2.2 System Components

2.2.1 Air intake

The air intake is located between the windshield and engine hood, the plenum area, in a location that is protected both from rain and snow intrusion. Furthermore, it is also protected from the engine emissions. However, some heat pickup from the engine is present due to the vicinity of warm surfaces. The sun can also heat surfaces surrounding the air intake, for example engine hood and this also increases the heat pickup. For these reasons the intake air is regularly warmer than the ambient air.

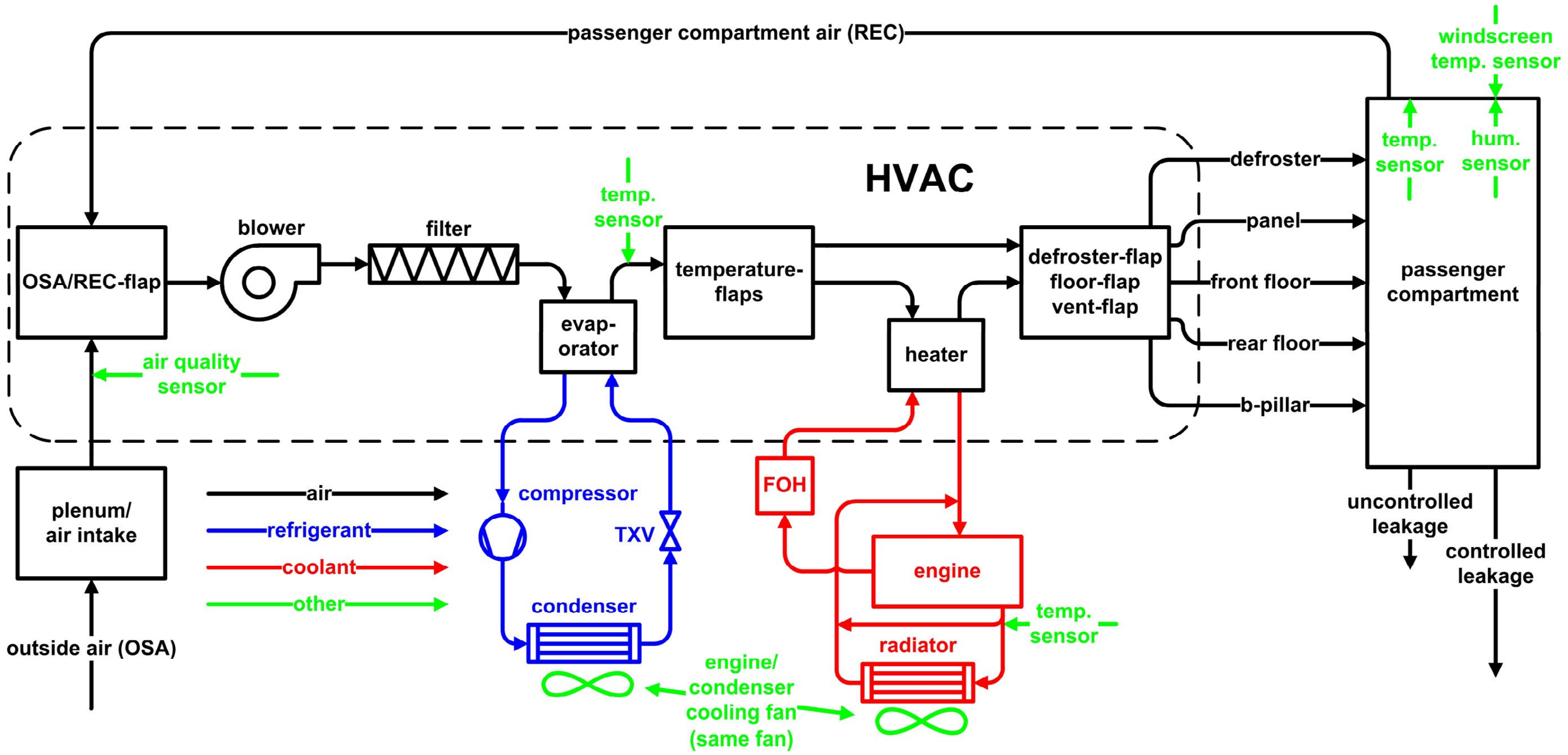

Figure 2.2 Climate system overview

2.2.2 Outside air flap and recirculation flap

Two flaps, controlled by one actuator, controls the amount of OutSide Air (OSA) and RECirculated air (REC). Outside air can also be called fresh air, however the outside air is not always fresh. For high velocities the ram pressure can increase the total airflow into the passenger compartment. In normal conditions the airflow can be kept constant by decreasing the blower level. However, for some conditions this will not be enough and the OSA flap will throttle the outside air, i.e. decrease the airflow.

The most common cause for recirculating air is to reduce the cooling load in warm conditions, especially during cool downs. Full recirculation, that is 100% recirculated air, is usually only used for extreme cool downs because the air quality will decrease through human induced pollutants. Additionally, the recirculation inlet is located in the front of the passenger compartment and for a low airflow, essentially low air velocity, the air will not reach the second row, i.e. decreasing the thermal comfort significantly for that location. For these reasons the recirculation degree is generally around 70%, that is, 70% recirculated air and 30% outside air, in warm conditions. In general, the automotive industry have avoided recirculation in cold climate. The main reason for this has been the risk of water condensation on the inside of the windows due to the humidity added by the passengers. Passenger added humidity is caused, for example, by snow and rain brought into the compartment and breathing. The risk of condensation in combination with abundant supply of waste heat have historically prevented the use of recirculation in cold conditions.

Another function that can recirculate the air, in this vehicle, is the interior air quality system (IAQS). A sensor located in the air intake measures air pollutants and can request recirculation if the outdoor pollution levels are increasing, for example, if the vehicle enters a tunnel or tailing a truck.

2.2.3 Blower, blower motor and blower control

The blower is a centrifugal blower, the motor is a standard permanent magnet DC motor and the control is a linear resistive control. The airflow is controlled by a pulse width modulated (PWM) signal from the climate control module to the blower control which regulates the voltage over the motor. This type of control is cheap and reliable, however, a lot of energy is dissipated because the voltage is reduced with resistive elements. The heat is dissipated into the airflow which increases the cooling load.

The blower is located after the OSA/REC flaps and before the filter in the air-handling unit. It is located inside the passenger compartment under the dashboard.

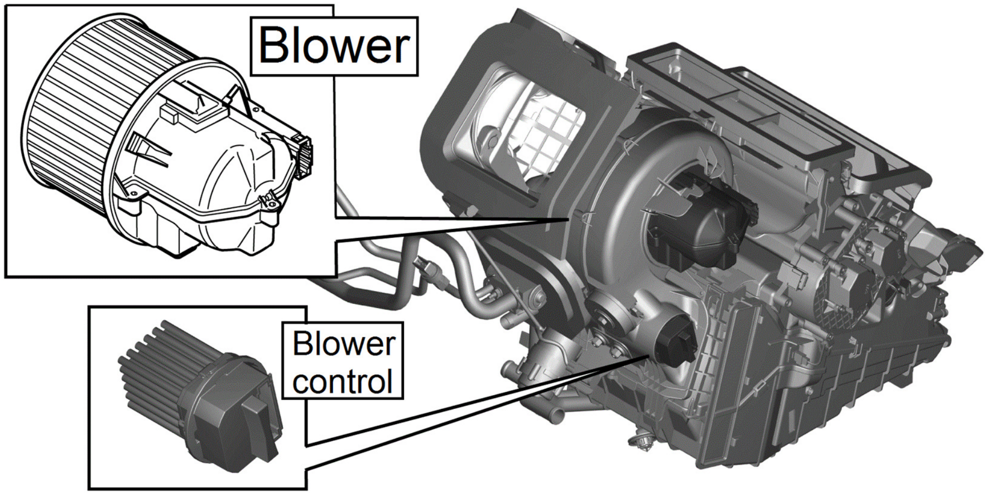

Figure 2.3 Blower and blower control

2.2.4 Filter

The filter is located after the blower. The equipped filter is a multi-filter, this means that in addition to filtering particles it contains active carbon which can reduce the concentration of some gases, i.e. increase the air quality. There is a significant pressure drop over the filter.

2.2.5 AC-system

The AC-system is a vapour compression system used for cooling and dehumidifying the air. It consists of many parts, the most important are the compressor, condenser, TXV and evaporator. For more information see for example [79]. An overview of the AC-system is presented in Figure 2.4.

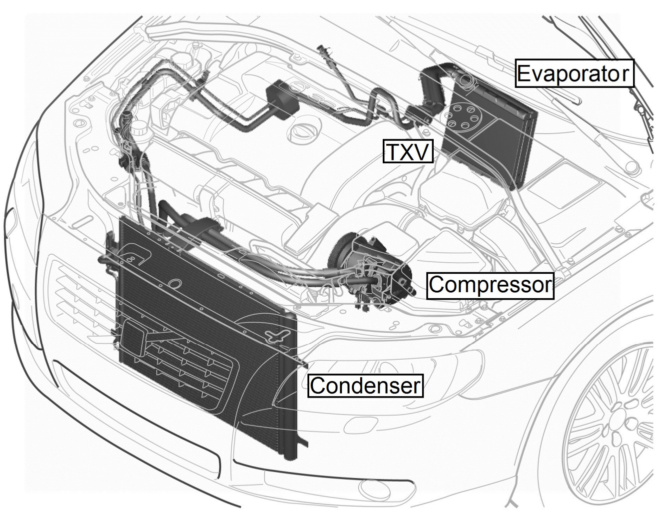

Figure 2.4 Overview of the AC-system

Compressor

The compressor, see Figure 2.5, in the investigated vehicle is a variable displacement compressor with a maximum displacement of 167 cm^3. The displacement is externally controlled by a control current and the current is controlled by the automatic climate control. It is powered by the engine through the crankshaft and the belt drive. Furthermore, the compressor can be disengaged by a magnetic clutch. The compressor control and magnetic clutch use electrical power when engaged.

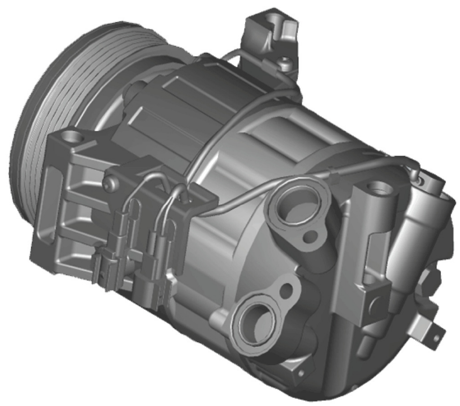

Figure 2.5 Compressor

Condenser

The condenser is a heat exchanger located in the front end of the vehicle; in front of the engine radiator. Its purpose is to remove heat from the system. The performance of the condenser is very dependent on the airflow. When the vehicle is moving ram airflow provides the required condenser cooling, but for low velocities or when idling the ram airflow is insufficient. In these cases the electric engine cooling fan can be engaged. However, at low vehicle velocities engine heat can affect the temperature of the air due to recirculation of hot engine compartment air. Furthermore, a lot of other heat exchangers, in addition to the condenser and radiator, are located in the same area, for example transmission oil cooler and charge air cooler. This reduces the active area of the condenser and increases the compressor load.

Thermal expansion valve

The purpose of the TXV is to regulate the refrigerant flow in the circuit. It is mounted before the evaporator and is controlled by the evaporator outlet temperature and pressure.

Evaporator

The evaporator is a heat exchanger that is mounted in the climate system air stream. When the cold low pressure refrigerant fluid passes the evaporator heat is absorbed and the refrigerant evaporates. The air is cooled and in many conditions water is condensed on the evaporator surface. A temperature sensor is located after the evaporator on the air side to give input to both the AC-system control and the general climate control software.

2.2.6 Temperature control, engine cooling system, FOH

The air temperature is controlled in two steps: First the air is cooled in the evaporator. Second, a specific rate of the airflow passes the heater, a heat exchanger in the engine cooling circuit, and the rest bypasses it, see Figure 2.2 and Figure 2.6. The rate of air is controlled by two flaps, one for each side. After the heat exchanger the flow is to some extent mixed, not completely because different temperatures are required in different outlets.

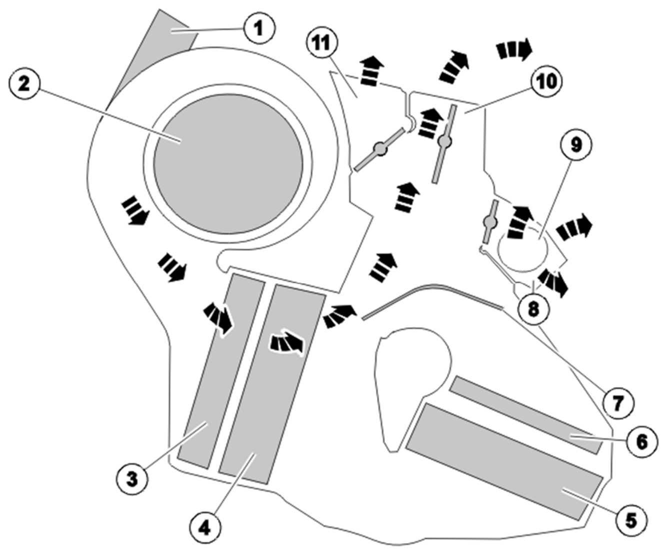

Figure 2.6 Air-handling unit, cross section

Table 2.1 Description of components in Figure 2.6

No	Component
1	Air intake
2	Blower
3	Filter
4	Evaporator
5	Heater
6	Electric heater (not mounted)
7	Temperature flaps
8	Duct inlet, rear floor
9	Duct inlet, front floor
10	Duct inlet, panel vents
11	Duct inlet, defroster

In general, the climate system use waste heat from the engine. A mechanical pump drives the coolant through the coolant circuit. For a simplified overview of the engine cooling system see Figure 2.7.

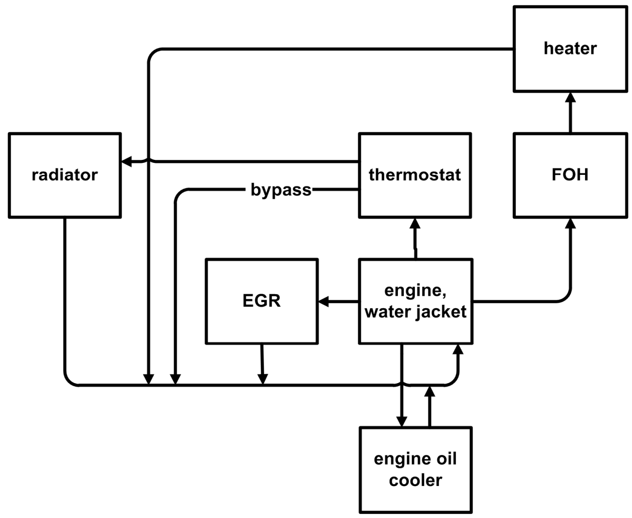

Figure 2.7 Cooling system

For some vehicles the waste heat from the ICE is not enough to heat the passenger compartment sufficiently fast or to the required comfort level. One solution to this problem is to mount a FOH in the engine compartment which supplies heat to the coolant circuit, see Figure 2.8. The FOH is supplied with fuel from the normal tank and uses roughly 0.6 l/h while providing approximately 5 kW of heat. For the start-up the FOH use an electrical element to ignite the fuel. The start-up can be relatively slow, it can take a couple of minutes before it supplies the rated heat. During the operation it uses electrical power for control, glow plug and air fan.

One major advantage with this type of heater is that it can be used for pre-conditioning of the passenger compartment.

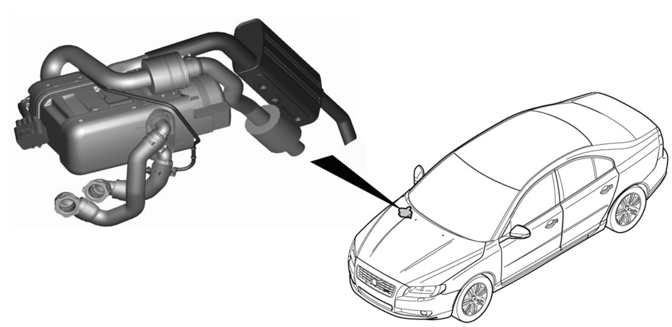

Figure 2.8 Fuel operated heater

2.2.8 Distribution and ducts

The air is directed through different outlets, this airflow distribution is controlled by the automatic climate control or customer settings. There are fourteen different outlets: two defroster outlets, two side defroster outlets, two front centre panel vents, two outer panel vents, two second row b-pillar vents, two front floor outlets and two second row floor outlets. See Figure 2.9 for an overview of the air distribution and ducts. Two actuators control the air distribution, one controls the defroster and the other the distribution between floor and panel vents. In many cases the required temperature in different outlets are not equal, warmer air is necessary on the floor and in the defroster compared to the panel vents. Internal geometries in the air distribution unit achieves this temperature stratification.

The ducts are of different length and size depending on outlets; for instance the second row ducts are much longer than the front floor and centre panel vent ducts. Heat can be added or lost in the ducts, however, most of this heat still enters the passenger compartment, although slower than if it would entered with the air. Furthermore, some heat can be lost to the outside and air can leak both inside the air distribution unit and in connections between the ducts.

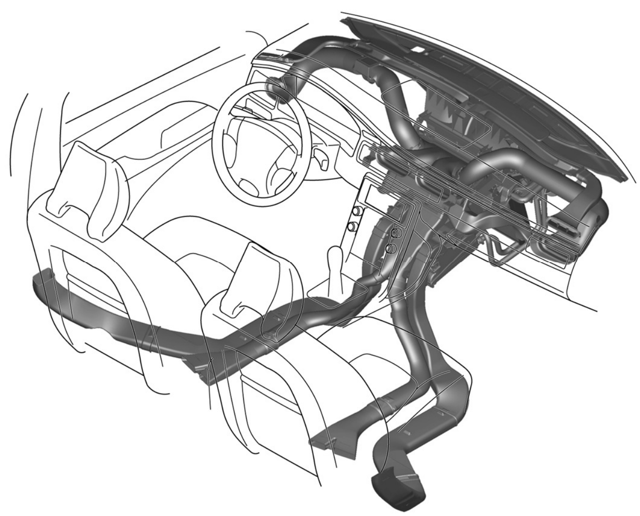

Figure 2.9 Air distribution and ducts

2.2.9 Passenger compartment

The air enters through different outlets depending on ambient conditions, passenger compartment thermal state and customer setting. The air in the compartment has three possible exits, through the recirculation outlet located in the front, through the evacuation located in the rear and through leakage through the vehicle body.

There are sensors located in the passenger compartment for the climate control; a temperature sensor for temperature control, a humidity sensor and a windscreen temperature sensor for determining windscreen mist risk.

2.4 System Control

In general the climate system operates in three different modes, heating in cold conditions, cooling in warm conditions and reheat in intermediate conditions. The sub systems of the climate system can operate differently in the various conditions.

2.4.1 Manual and automatic climate control

There are many possible settings on the climate system in the investigated vehicle: A fully automatic mode where airflow, distribution, temperature, AC-system and recirculation are controlled by the software or different levels of manual control. See Figure 2.10 for the user interface of the climate system.

Figure 2.10 Climate system interface

2.4.2 Heating, automatic mode

In pure heating mode, below approximately 0°C, the AC-system is not active. The two main reasons for this are: No need for cooling and dehumidification is impossible because the water vapour will create ice on the evaporator. Additional heaters, such as FOH, are engaged if needed.

During a heat up, i.e. cold start, the available heat is usually low in the beginning. The airflow is therefore limited until the temperature of the coolant is sufficiently high. That is, the airflow increases steadily with the coolant temperature rise. Initially all air is directed through the defroster. There are two reasons for this; the need to de-ice and de-mist the windows and avoid releasing cold air to the floor, as this is uncomfortable. When the temperature increases the airflow to the floor is increased. As the temperature in the passenger compartment approaches a comfortable temperature the airflow decrease until the steady state flow is reached. The level of heat, i.e. airflow passing the heater also decreases. That is, the temperature flaps bypass the heater with some of the air. In Figure 2.11 a heat up sequence in -18°C is presented.

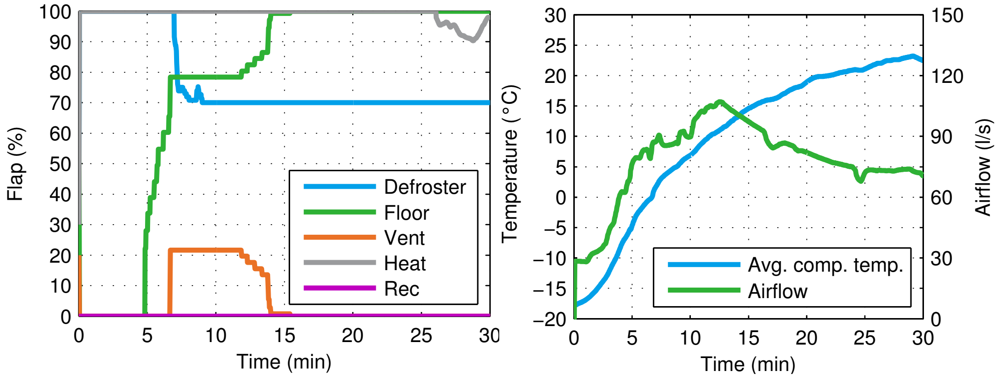

Figure 2.11 Flap positions, average compartment temperature and airflow during heat up in -18°C

2.4.3 Reheat, automatic mode

In intermediate conditions, approximately between 0° and 20°C, the AC-system is activated. The main reason for the activated AC-system is dehumidification of the air in order to keep the windows free of mist. To achieve the required inlet temperature the air is heated after the cooling and dehumidification, in other words reheat. Otherwise, the system operates similar as for heating except that more air and colder air is released through the panel vents, especially in steady state in the upper temperature range. In Figure 2.12 the reheat sequence in 15°C and 200 W/m² sun load are presented. The vehicle was soaked in the sun for one hour before the start.

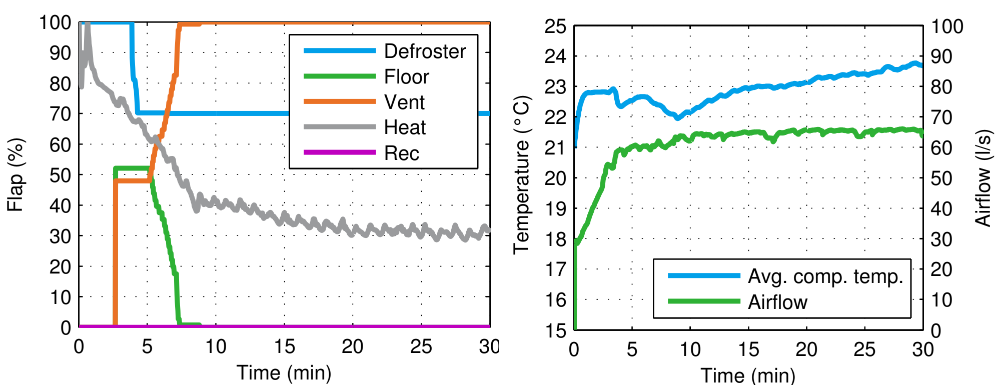

Figure 2.12 Flap positions, average compartment temperature and airflow in intermediate climate, 15°C and 200 W/m² sun load

2.4.5 Cooling, automatic mode

In warm conditions the climate system cools the passenger compartment, that is, the AC-system is activated for cooling. Unlike a heat up cooling is available practically from the start and for this reason the airflow starts at a relatively high level and decreases as the passenger compartment approaches comfort. The air is initially directed through the panel vents for maximum cooling, however, as the compartment approaches steady state temperature air is also released through the defroster outlet. This improves comfort by reducing drag and increasing homogeneity of the climate. In the beginning of a cool down the temperature of the passenger compartment determines if recirculation is used. If the passenger compartment temperature is lower than the ambient temperature all air is recirculated. When the compartment temperature nears the comfort temperature and the evaporator temperature set point is reached the level of recirculation is decreased. That is, in steady state in warm conditions part of the air is recirculated. This transient cool down sequence is presented in Figure 2.13 for a vehicle soaked for approximately one hour in 43°C with 1000 W/m² sun load.

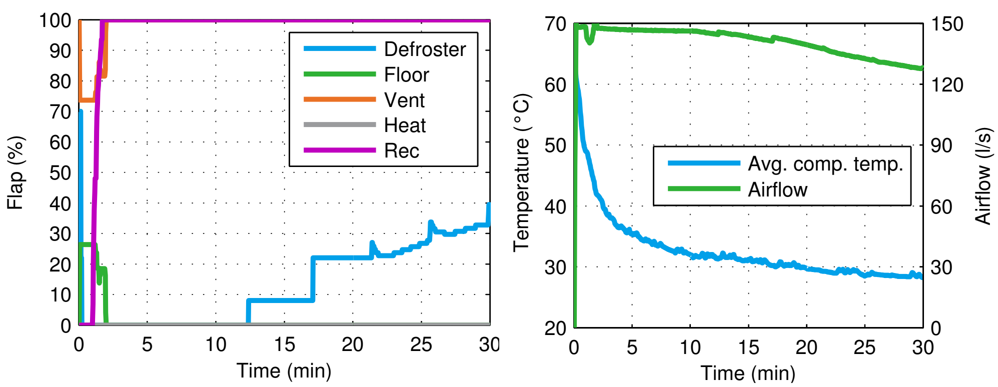

Figure 2.13 Flap positions, average compartment temperature and airflow during a cool down in 43°C and 1000 W/m² sun load

3 INVESTIGATED PROPERTIES

The energy use for vehicle interior climate has several different aspects, as a result, the energy use cannot easily be summarize in one quantity without losing the understanding of the process. For this reason many different properties and aspects are investigated. In this chapter these properties are explained in a more general way, that is, an overview of the investigated factors included in the energy use.

In most cases the property is measured or simulated at a rate of 1 Hz and afterwards an average is calculated, however, there are numerous exceptions. See Table 3.1 for a summary of the investigated properties.

Table 3.1 Investigated properties

Property	Description
Blower	Compartment blower power (including motor and control)
FOH electrical load	FOH electrical power
Pump	Climate circuit pump power
Cooling fan	The power for the twin cooling fan (including motor and control)
Comp. clutch & control	Compressor electrical power for clutch and control
Compressor	Compressor mechanical power
Total electrical power	The sum of blower, FOH electrical load, pump, cooling fans and Comp. clutch & control
Engine ($\dot{Q}_{engine}$)	Heat flow from the engine through the climate system heater
FOH ($\dot{Q}_{FOH}$)	Heat flow from the FOH through the climate system heater
Heat pickup ($\dot{Q}_{heat\ pickup}$)	Heat flow to the air up to the evaporator
Cooling, sensible ($\dot{Q}_{sen.\ cool}$)	Sensible cooling over the evaporator
Cooling, latent ($\dot{Q}_{lat.\ cool}$)	Latent cooling over the evaporator
Recirculation ($\dot{Q}_{rec}$)	Heat flow received through recirculation
Heat into compartment, air ($\dot{Q}_{pass.\ comp.}$)	Heat flow into the passenger compartment with the airflow
Heat out of compartment, air ($\dot{Q}_{outlet}$)	Heat flow out from the passenger compartment with the airflow
Conduction/radiation/thermal storage ($\dot{Q}_{other}$)	Non-air heat flows to and from the air in the passenger compartment
Average compartment temperature	Average air temperature in the passenger compartment

3.1 Electrical Loads

Various components use electrical energy for their operation. In the measurements the power was calculated from voltage and current measurements, see equation (3.1), or estimated according to specifications.

$$P = UI. \tag{3.1}$$

In equation (3.1) P is the power (W), U is the voltage (V) and I is the current (A). The blower power included two separate electrical components, the blower motor and the blower control. Each was measured individually, however, only the sum is presented in the thesis. The cooling fan power was comprised of the two cooling fans and the fan control. Both the fans power and the total power was measured, consequently, the control power was calculated. Note that for almost all measured cases and for all simulated cases the AC-system was the only requester of the cooling fan. The rest of the electrical loads, the pump, the compressor clutch and control and the FOH electrical load were not measured, instead the electrical loads were based on specifications and other measurements.

3.2 Mechanical Loads

The only mechanical load included was the compressor mechanical load. The torque was measured on the compressor axle inside the compressor and the engine speed was measured by the engines control module. The power was calculated as:

$$\dot{W}_{comp} = \frac{2\pi}{60}(\omega_{eng\,rpm} k_{ratio} \tau_{comp}) \tag{3.2}$$

where $\dot{W}_{comp}$ is the compressor power (W), $\omega_{eng\,rpm}$ is the engine speed (RPM), k_{ratio} is the compressor axle gearing and τ_{comp} is the torque (Nm). Other mechanical loads such as the engine coolant pump have not been included in the investigation.

3.3 Heat Flows

In this section the different heat flows are explained. The heat flows are divided into two categories for better understanding the energy use of the climate system. Sources of heating and cooling is the first category, it contains the main sources for the generation of heating and cooling. The second category is the sink category, it contains all the different sinks for heat in the passenger compartment. The source and the sink are connected through the heat flow into the passenger compartment, split into a heating and a cooling flow. Note that for the heat flow a uniform terminology was not used in the journal papers, however, in the thesis the denominations are consistent.

3.3.1 Sources of heating and cooling

The FOH heat flow, $\dot{Q}_{FOH}$, is the heat flow from the fuel operated heater, through the coolant, to the heater, i.e. it is not the total heat flow from the FOH because some of the heat is used for heating the coolant and for some cases even the engine. In other words, only the heat used for the interior climate is included. The heat flow is calculated as the added heat over the FOH in the coolant circuit, limited by the heat transfer rate in the heater, i.e. never larger than the heat flow from the heater to the air.

The engine heat flow, $\dot{Q}_{engine}$, is the heat flow from the engine and other components in the coolant circuit, such as engine oil cooler and exhaust gas recirculation cooler, to the heater. In other words not all waste heat generated by the engine, only the part that is used for the vehicle interior climate.

Heat pickup, $\dot{Q}_{heat\ pickup}$, is defined as the heat flow to the incoming air up to the evaporator. The temperature increase is due to heated surfaces on the outside, for example the hood, hot masses in the air-handling unit and waste heat from the blower and blower control. Note that the expression sometimes also can refer to losses in ducts, however, in the results in this thesis the expression never refers to this. The calculation of the heat pickup use the air temperature before the evaporator, the ambient temperature and recirculation temperature depending on degree of recirculation.

The sensible cooling, $\dot{Q}_{sen.\ cool}$, over the evaporator, is defined as the energy change as a result of temperature decrease of the air over the evaporator. Condensation of water on the evaporator, i.e. latent cooling of the air, $\dot{Q}_{lat.\ cool}$, is included as a separate load on the evaporator. It is measured and reported, however, it is not included in the calculated heat flow into the passenger compartment.

Recirculation, $\dot{Q}_{rec}$, is the heat flow recirculated back from the passenger compartment, it is calculated from the recirculation air temperature and ambient temperature. In other words it is a heat flow relative to the ambient temperature.

3.3.2 Heat flow into the passenger compartment

The heat flow into the passenger compartment, $\dot{Q}_{pass.\ comp.}$, is divided into two parts for transparency. Heating flow is the positive heat flow into the compartment compared to the ambient temperature. Similarly, cooling flow is the negative heat flow into the compartment compared to the ambient temperature. All these heat flows are calculated with the temperature difference between the air temperature before the duct inlets in the air-handling unit and the ambient temperature. Note that air humidity changes are ignored in these calculations, i.e. the focus is on temperature difference rather than enthalpy difference, see section 3.3.4.

3.3.3 Passenger compartment sinks

The different heat flows to and from the passenger compartment was divided into different sink-categories depending on the ability to evaluate them separately. In the measurement there were two categories: First the heat flow out from the compartment with the air, $\dot{Q}_{outlet}$. It was calculated as the sum of the three different exiting airflows, recirculation, $\dot{Q}_{rec}$, leakage, $\dot{Q}_{leakage}$, and evacuation, $\dot{Q}_{evacuation}$. Each of these heat flows used ambient temperature as reference. Note that $\dot{Q}_{rec}$ is the same heat flow as in the source. The second category was conduction/radiation/thermal storage, $\dot{Q}_{other}$, which accounted for all other modes of heat flows to and from the passenger compartment air. The second category was not measured directly, it was calculated as the difference between the incoming and outgoing air heat flows.

For the simulations the second category could be divided into separate sinks. These were the ducts, $\dot{Q}_{duct}$, the interior mass, $\dot{Q}_{mass}$, the shell, $\dot{Q}_{shell}$, and the windows, $\dot{Q}_{windows}$. Note that the actual energy needed for the temperature change of the passenger compartment air is, in general, not included in the results as it was very small compared to other sinks. Passengers could also add heat to the passenger compartment, $\dot{Q}_{person}$. The heat balance of the passenger compartment did not take humidity changes into account, i.e. only temperature affected the heat flows, see next section.

3.3.4 Humidity

Throughout this thesis the heat balance does not include humidity changes of the air. The most important effect off humidity is the condensation of water on the evaporator surface, i.e. latent cooling which increases the evaporator load. This latent cooling is included in the presented results but not in $\dot{Q}_{pass.\ comp.}$, or any sink heat flows. Furthermore, both in tests and simulations humidity is added in the passenger compartment by breathing. If the air is recirculated it can affect the evaporator load.

Are there any potential disadvantages as a result of excluding humidity changes from the heat balance? Recirculation heat flow is the single most affected flow when humidity changes are excluded. However, the effect is dependent both on mode of operation, ambient and passenger compartment conditions. For example, during warm humid conditions the recirculation heat flow would be even lower due to the decreased humidity compared to the ambient conditions, that is, the recirculated air is already cooled and dehumidified. The recirculation heat flow would better represent the actual effect and energy saving. However, for recirculation in cold conditions added humidity would increase the heat flow but actually be worse due to the increased risk of condensation. In that case the recirculated energy would not represent the actual effect and energy saving. In other words, the inclusion of humidity could both increase and decrease the understanding of the heat flows from the different sources to the different sinks and it is dependent on the conditions.

In summary, it was chosen to evaluate the heat flows from a sensible heat balance perspective and only include humidity as an additional load on the evaporator.

3.3.5 Heat balance

One method to describe the source and the sink is to divide them into two control volumes. The source control volume is an air volume that includes all air from the climate system intake to the end of the air distribution, at the inlet of the ducts, see Figure 3.1. The second control volume includes all air from the inlet of the ducts to the air evacuation of the passenger compartment, see Figure 3.2. Note that this is a simplification, in both the measurements and simulation there is leakage in different stages, i.e. there are several different airflows into the passenger compartment. Moreover, the actual calculations are somewhat more complicated, see mainly the paper I for details. In summary the heat flows were calculated from a sensible heat balance with latent cooling added as a load on the evaporator. These sensible heat balances are derived in the next two sections.

Heat balance, source control volume

For a general control volume the first law of thermodynamics can be formulated as

$$\frac{dE_{C.V.}}{dt} = \dot{Q}_{C.V.} + \dot{W}_{C.V.} + \sum \dot{m}_i \left(h_i + \frac{1}{2}\boldsymbol{V}_i^2 + gZ_i \right) - \sum \dot{m}_e \left(h_e + \frac{1}{2}\boldsymbol{V}_e^2 + gZ_e \right) \quad (3.3)$$

according to [80]. $E_{C.V.}$ is the energy of the control volume (J), $\dot{Q}_{C.V.}$ is the heat flow to the control volume (W) and $\dot{W}_{C.V.}$ is the work rate on the control volume (W). Furthermore, $\dot{m}$ is mass flow (kg/s), h is enthalpy (J), $\boldsymbol{V}$ is the velocity (m/s) and gZ the potential energy (J). The subscripts i and e denotes incoming and outgoing flow.

For the source control volume assume steady state and two inlets, OSA and REC, and one outlet, passenger compartment, see Figure 3.1. The mass flow of the inlets are denoted $\dot{m}_{osa}$ and $\dot{m}_{rec}$ and the single outlet $\dot{m}_{pass.\ comp.}$.

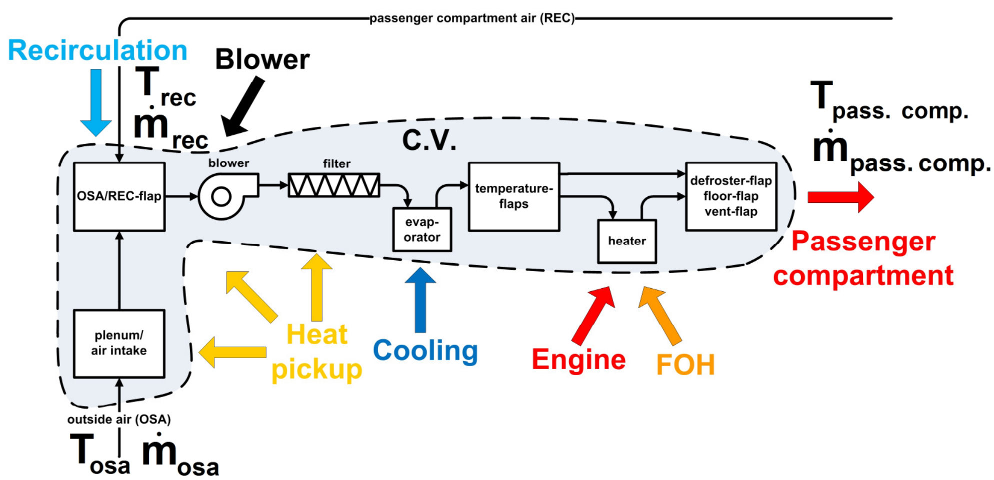

Figure 3.1 Source control volume, air-handling unit

Further, if we assume that changes in kinetic or potential energy are negligible equation (3.3) can be rewritten as:

$$0 = \dot{Q}_{C.V.} + \dot{W}_{C.V.} + \dot{m}_{osa}h_{osa} + \dot{m}_{rec}h_{rec} - \dot{m}_{pass.\ comp.}h_{pass.\ comp.} \quad (3.4)$$

Assume no humidity change, no work and ideal gas, then equation (3.4) is rewritten as:

$$0 = \dot{Q}_{C.V.} + \dot{m}_{osa} c_p T_{osa} + \dot{m}_{rec} c_p T_{rec} - \dot{m}_{pass.\ comp.} c_p T_{pass.\ comp.}. \tag{3.5}$$

where c_p is the specific heat capacity (J/kg K), T_{osa} is the ambient temperature (°C) and T_{rec} is the average recirculation air temperature (°C). Further, $T_{pass.\ comp.}$ is the average inlet temperature of the airflow into the passenger compartment, not the passenger compartment temperature. We assumed that there was no change in air humidity, however, condensation is included as an evaporator load.

By adding and subtracting $\dot{m}_{rec} c_p T_{osa}$, and rearranging, equation (3.5) can be written as

$$0 = \dot{Q}_{C.V.} + (\dot{m}_{osa} + \dot{m}_{rec}) c_p T_{osa} + \dot{m}_{rec} c_p (T_{rec} - T_{osa}) - \dot{m}_{pass.\ comp} c_p T_{pass.\ comp.}. \tag{3.6}$$

Conservation of mass:

$$\dot{m}_{osa} + \dot{m}_{rec} = \dot{m}_{pass.\ comp.}. \tag{3.7}$$

Further rearranging of equation (3.6) and using equation (3.7) yields:

$$\dot{m}_{pass.\ comp.} c_p (T_{pass.\ comp.} - T_{osa}) = \dot{Q}_{C.V.} + \dot{m}_{rec} c_p (T_{rec} - T_{osa}) \tag{3.8}$$

The most important heat flow for the investigation of energy use in this thesis is the heat flow into the passenger compartment. This heat flow is defined as:

$$\dot{Q}_{pass.\ comp.} = \dot{m}_{pass.\ comp.} c_p (T_{pass.\ comp.} - T_{osa}). \tag{3.9}$$

Equation (3.8) can then be rewritten with the sources in Figure 3.1 and equation (3.9) as

$$\dot{Q}_{pass.\ comp.} = \dot{Q}_{heat\ pickup} + \dot{Q}_{rec} + \dot{Q}_{sen.\ cool} + \dot{Q}_{engine} + \dot{Q}_{FOH} \tag{3.10}$$

where

$$\dot{Q}_{rec} = \dot{m}_{rec} c_p (T_{rec} - T_{osa}). \tag{3.11}$$

In summary, $\dot{Q}_{pass.\ comp.}$ is the sum of all other source heat flows and represents the heat flow into the passenger compartment.

The work on the control volume was assumed to be negligible, as the only work was done by the blower. Most of this work became heat due to friction and are included in the heat pickup. However, there is a pressure increase of the outgoing flow and the maximum rate of the flow work of the blower, $\dot{W}_{blower}$, can be calculated as

$$\dot{W}_{blower} = \Delta P\dot{V} = 200\ Pa \cdot 0.15\ m^3/s \approx 30\ \mathrm{W} \tag{3.12}$$

with pressure increase, ΔP, and volume flow, $\dot{V}$, from air-handling unit simulations. The average heat flow into the compartment for comparable conditions is of the order of 6 kW, see paper I. The blower work can be neglected in the energy balance.

For equation (3.4) steady state was assumed, however, for many cases the start and end temperatures are not equal, especially not for heat up cases. Consider a heat up case in -18°C, assume that the air temperature after the heater is 80° after 30 minutes and that half of the air in the system have this temperature. Further assume that the mass of the air in the air handling unit, i.e. source, m_{source}, is 0.1 kg at the start. The average heat flow for the temperature increase can be approximated with

$$m_{source}c_p\frac{dT_{source}}{dt} = \frac{0.1}{2} \cdot 1000 \cdot \frac{98}{1800} \approx 3\ \mathrm{W} \tag{3.13}$$

which is negligible compared to the other factors. The steady state assumption was valid.

Heat balance, sink control volume

The second control volume, the sink or passenger compartment, has many different heat flows and the heat balance can be expressed as:

$$\dot{Q}_{pass.\ comp.} + \dot{Q}_{person} = \dot{Q}_{outlet} + \dot{Q}_{shell} + \dot{Q}_{windows} + \dot{Q}_{mass} + \dot{Q}_{ducts}. \tag{3.14}$$

In Figure 3.2 an overview of the sink control volume, i.e. passenger compartment, is presented.

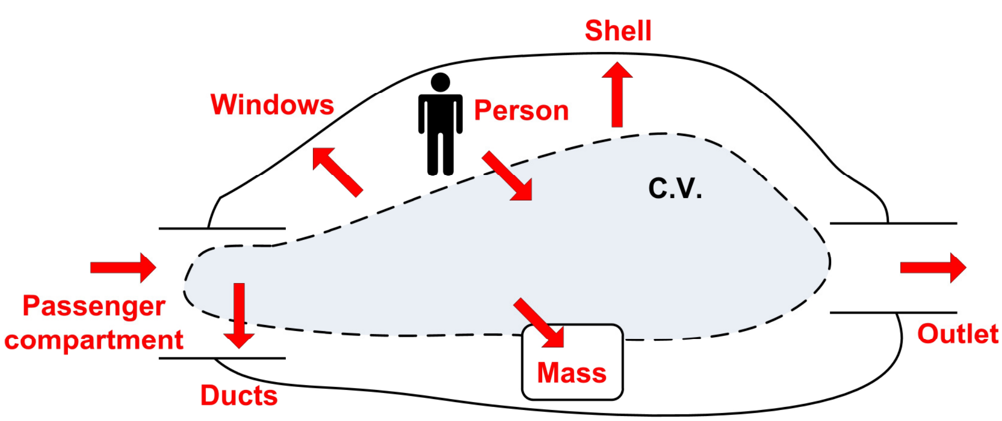

Figure 3.2 Sink control volume, passenger compartment

Note that the outlet air, $\dot{Q}_{outlet}$, was the sum of all exiting airflows,

$$\dot{Q}_{outlet} = \dot{Q}_{evacuation} + \dot{Q}_{rec} + \dot{Q}_{leakage} \tag{3.15}$$

where $\dot{Q}_{rec}$, is the same as in equation (3.10) and (3.11). Note also that for the measurements $\dot{Q}_{other}$ is the sum of all non-air heat flows,

$$\dot{Q}_{other} = \dot{Q}_{shell} + \dot{Q}_{windows} + \dot{Q}_{mass} + \dot{Q}_{ducts} - \dot{Q}_{person}\,. \tag{3.16}$$

For many cases the start and end temperatures are not equal and the air mass is much larger compared to the source control volume. However, the temperature difference is also much smaller, for a heat up in -18°C the average passenger compartment air temperature has increased to approximately 20°C after 30 minutes. Equation (3.13) for the sink control volume is:

$$m_{sink} c_p \frac{dT_{sink}}{dt} = 3 \cdot 1000 \cdot \frac{38}{1800} \approx 60 \text{ W}. \tag{3.17}$$

Event tough it is larger than the other neglected factors it is still small compared to the main heat flows.

3.4 Passenger Compartment Temperature

In the measurement, the average compartment temperature was calculated from temperature sensors located in eight different zones. Front head level for both left and right side, front floor level also for left and right side and the same setup in the rear seat. When the measurement temperature was compared to the simulation the same zones were used although instead of a point measurement the simulation used a volume average of each zone. This because the passenger compartment was not discretized enough to allow point measurements. The temperature average of the total air volume was used for comparisons between different simulation cases.

4 COMPLETE VEHICLE MEASUREMENTS

One method to better understand the current system, how it operates and how much energy it uses in different conditions is testing. For this investigation complete vehicle tests were done in a climatic wind tunnel for many different conditions. In this chapter the complete vehicle tests that were presented in paper I, “Measurements of Energy Used for Vehicle Interior Climate”, are summarized. Furthermore, some additional results are presented which were not included in the paper and an expanded review of the airflow estimation.

4.1 NEDC Tests and Temperature Sweep

The objective of the study presented in paper I, was to investigate the energy use for the vehicle interior climate. Emphasis was on the heat flows from the sources to the sinks. Additionally, the electrical and mechanical work were also measured. The measurements were done with a complete vehicle, Volvo S60, in a climatic wind tunnel. Three different types of tests were performed; 5 transient and 7 steady state tests using the New European Driving Cycle (NEDC) in temperatures ranging from -18°C to 43°C. The state regarded the thermal conditions affecting the interior climate. Furthermore, a temperature sweep from 43°C to -18°C in 50 km/h was also completed. The vehicle had measurement sensors installed, mostly temperature sensors but also voltage, current, pressure, torque and humidity sensors were used. Most temperature sensors were located in the air-handling unit, the engine cooling system and the passenger compartment. The voltage and current sensors were mounted, among others, on the main electrical loads such as the blower and engine cooling fan.

The results were divided into three parts: First, averages of each of the NEDC tests, second a time-resolved examination of one heat up test and one cool down test. Finally, the temperature sweep was presented. The averages of the 12 NEDC tests included the following electrical loads: blower, condenser cooling fan, FOH electrical load, coolant pump, compressor clutch and control. Furthermore, the mechanical compressor load was also measured. The results showed that the conditions affected the loads significantly, average electrical loads varied from 140 W to 850 W and mechanical loads from 0 to 3700 W. Average heat flows divided into sources of heating and cooling were also presented for the 12 NEDC tests. The sources were waste heat from the engine, heat pickup, heat from the FOH, cooling from recirculation, sensible and latent cooling over the evaporator. The average heat flow varied between 6 and -6 kW depending on conditions. The sinks of the passenger compartment were also investigated, however, only the heat flow of the air out from the passenger compartment could be measured. As a consequence the rest of the possible sinks were lumped together in one category, conduction/radiation/thermal storage.

Two tests were examined more closely, a heat up in -18°C and a cool down of a sun soaked vehicle in 43°C and 1000 W/m² solar load. The heat up showed that the peak heat flow into the passenger compartment was approximately 10 kW and more heat was transferred through conduction/radiation/thermal storage than by the airflow out from the passenger compartment. Moreover, the peak electrical load was 350 W. For the cool down test the peak heat flow into the passenger compartment was approximately -5 kW, the peak electrical load was 1100 W and maximum compressor load was 6 kW.

The most significant result from the temperature sweep, besides the actual size of the investigated properties, was the magnitude of the reheat. For the temperature of 24-25°C the heat flow into the passenger compartment was approximately 0 W. However, the sum of the sources of heating was 1.2 kW and the cooling source sum was -1.2 kW, i.e. a lot of energy was used to achieve the required heat flow into the compartment.

A thorough review of the measurement uncertainty was also performed. It showed that the weakest part of the measurements were the compressor load which varied significantly for comparable loads on the AC-system. Consequently, all compressor load results should be used with caution. Moreover, the airflow was only estimated during the tests, thus the accuracy of the heat flows was very dependent on the accuracy of the estimation.

In summary, the paper provides data of the energy used for the interior climate for a wide range of different conditions. Moreover, it suggests that the different heat flows from the sources to the sinks and the electrical and mechanical work should be presented separately for better understanding of the system.

4.2 Additional Steady State Test Results

During the complete vehicle tests additional conditions were tested. These results are presented here. For more information regarding the measurement setup see paper I.

Several steady state tests were performed. The definition of steady state is that both the engine and compartment reached its operation temperature, for the specific conditions, before the measurement started. That is, no significant temperature changes of the engine, passenger compartment or any other components during the test. However, in these tests the test time was between 15 and 20 minutes for each condition, this was not always enough to reach a steady state level for the 26/26 and 18/18 settings. These settings represent the approximate set temperature for each side in the passenger compartment. Note that the actual passenger compartment temperature can differ significantly from the set temperature as the set temperature only represents the equivalent uniform comfort temperature. The different test conditions are presented in Table 4.1. Note that only the heat balance of the passenger compartment air is presented.

Table 4.1 Steady state test conditions

Test name	Temp. setting (°C)	Vehicle velocity (km/h)	Airflow	Recirculation
25	22/22	25	auto	auto
50	22/22	50	auto	auto
150	22/22	150	auto	auto
rec.	22/22	50	auto	100%
26/26	26/26	50	auto	auto
18/18	18/18	50	auto	auto
red.	22/22	50	~50% of auto	auto

In Figure 4.1 the heat balance of the passenger compartment air for ambient temperature of -18°C can be found. The airflow was roughly 70 l/s and there was no recirculation for the auto setting. The conditions are different for each test and this affects the average passenger compartment temperature. This temperature is included in Figure 4.1 above each column.

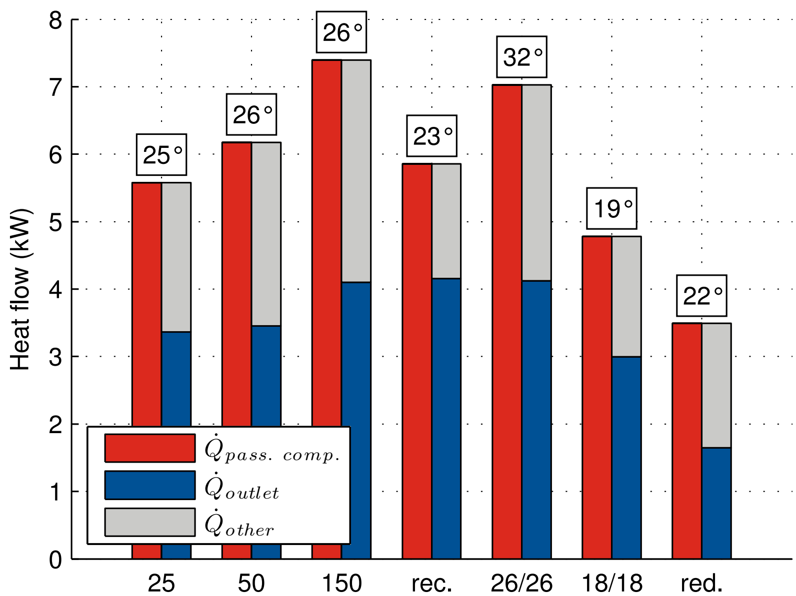

Figure 4.1 Heat flow into and out from passenger compartment, -18°C

The airflow has a large impact on the heat flow into the passenger compartment. Comparing 18/18 and the test with the reduced airflow shows that despite a flow of only 50% the reduced airflow can maintain higher temperature and still use less energy. However, with the reduced throw from, especially the defroster outlet, the de-frosting and de-misting capacity was probably degraded. Furthermore, the removal of humidity is also decreased with the reduced airflow.

In Figure 4.2 steady state tests in -10°C are presented. One interesting result was that the reduced airflow could keep the temperature in the passenger compartment at the same level as with the normal airflow but using roughly 25% less heat. There was a quite large influence on the heat flow from the vehicle velocity. With increased vehicle velocity the conduction, radiation and thermal storage part increased, most likely due to increased convection on the outside of the vehicle which decreased the surface temperature and increased the conduction. The radiation and thermal storage part were most likely vehicle velocity independent. The total energy use was also decreased with approximately 25% compared to the -18°C tests.

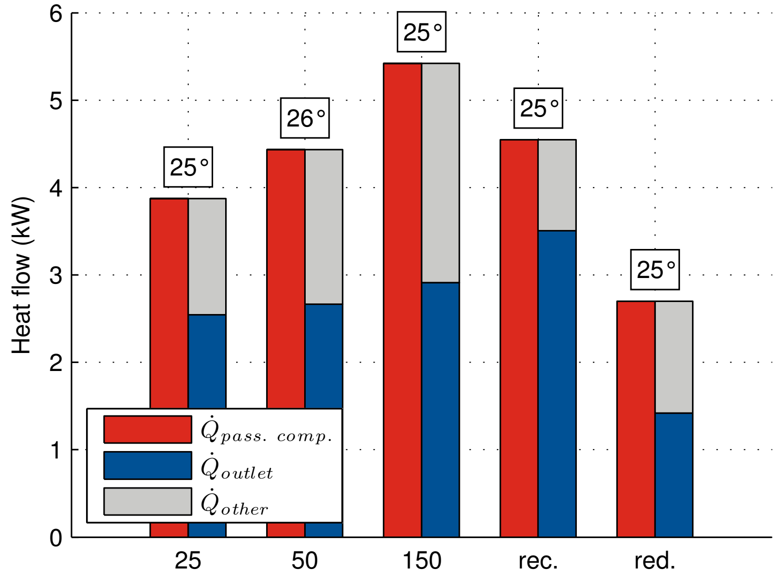

Figure 4.2 Heat flow into and out from passenger compartment, -10°C

In Figure 4.3 the results of the steady state tests in 2°C can be found. The most noticeable result was that the vehicle velocity dependence decreased significantly compared to colder temperatures. Another result, that was general for all temperatures, including 2°C, was that recirculation increased the heat flow out from the compartment by the air and significantly decreased the conduction and radiation share.

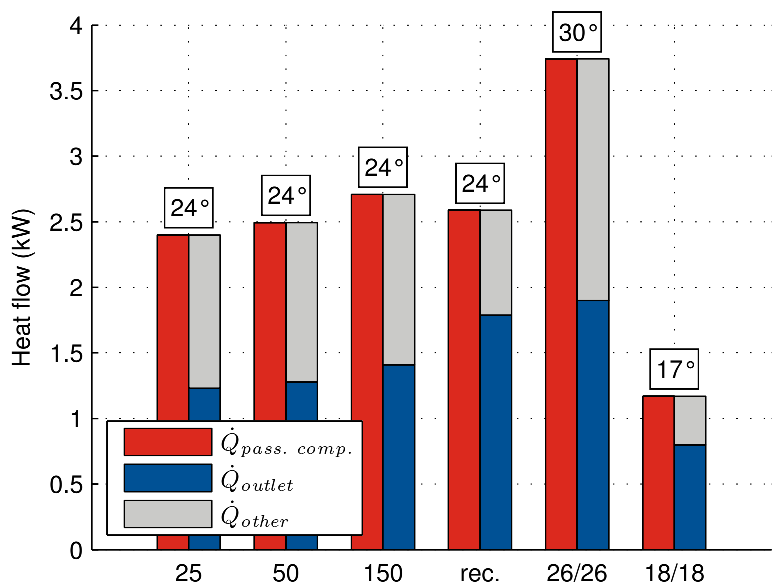

Figure 4.3 Heat flow into and out from passenger compartment, 2°C

Figure 4.4 presents the steady state tests in 10°C. Apart from generally lower heat flows the noteworthy result was that the 18/18 setting did not achieved steady state.

If no heat was added to the passenger compartment, as in the test, the temperature would eventually reach 10°C. However, the positive heat flow out with the air and the negative conduction, radiation and thermal storage part indicate that the interior masses are still being cooled.

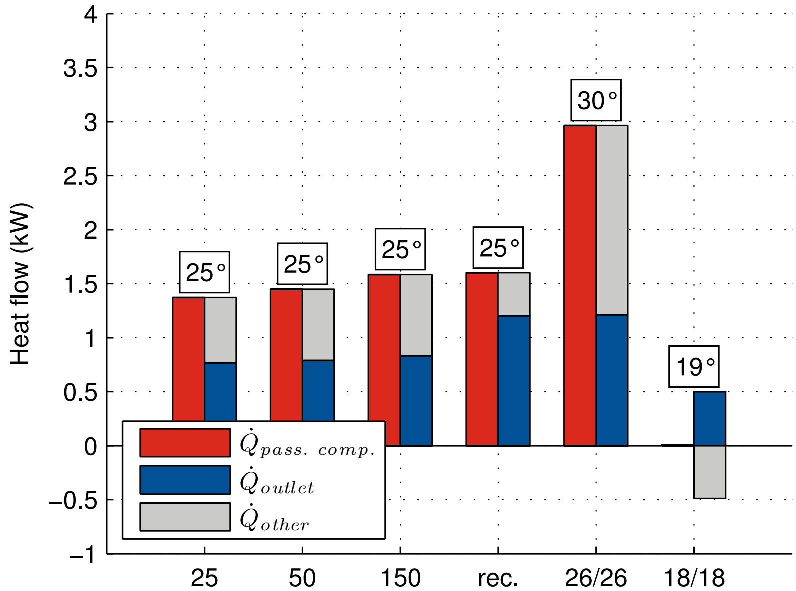

Figure 4.4 Heat flow into and out from passenger compartment, 10°C

With an ambient temperature of 24°C the difference between the passenger compartment temperature and the surrounding temperature was very small. Since the temperature difference was small the heat transfer was also small between the passenger compartment and the outside and the velocity had no effect. See Figure 4.5. Furthermore, the base recirculation level for these conditions were approximately 50%.

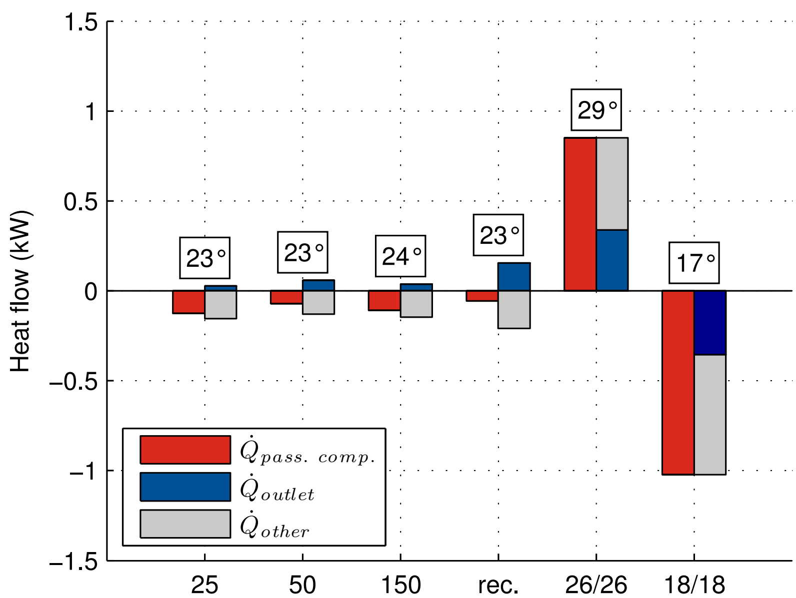

Figure 4.5 Heat flow into and out from passenger compartment, 24°C

The most demanding warm steady state test was performed in 43°C with a sun load of 1000 W/m², see Figure 4.6. In this case the vehicle velocity did not have a large effect on the required cooling. The average passenger compartment temperature of the 50 km/h test was also high, i.e. steady state was not achieved for this particular test. Approximately 3.5 kW of cooling was required in these conditions.

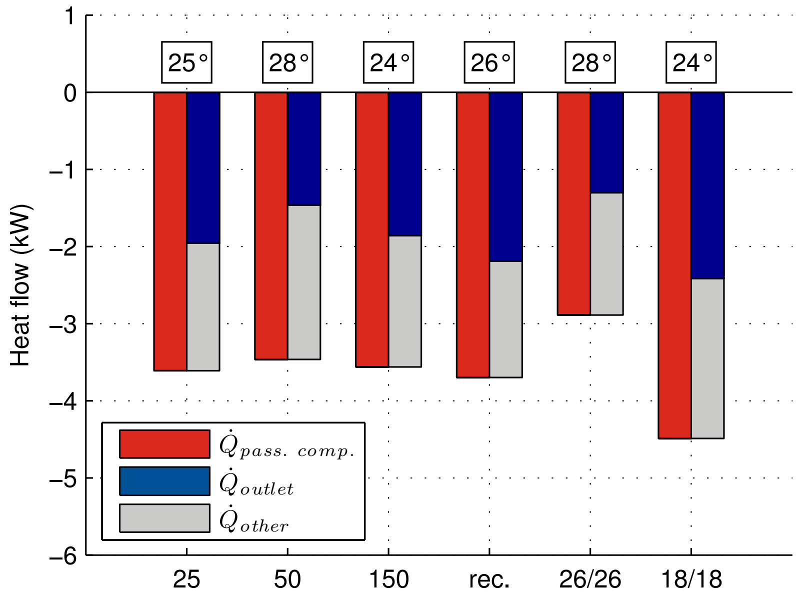

Figure 4.6 Heat flow into and out from passenger compartment 43°C and 1000 W/m² sun load

Overall the steady state tests showed that the vehicle velocity was only important for the required heat flow if there was a large temperature differences between the surroundings and the passenger compartment. Large temperature differences were only achieved in cold climates. The results also showed that it was possible to significantly reduce the airflow in cold climate and maintain a comfortable temperature. However, other important factors such as de-frosting and de-misting capacity, removal of humidity and operative temperature were not investigated and these factors could definitely influence the results.

4.3 Airflow Estimation

The airflow has a proportional effect on the heat flow and consequently the accuracy of the airflow is very important. However, there is no suitable method for measuring the airflow in the vehicle during operation and for that reason the airflow was separately measured and then estimated during the tests. The estimation was only briefly described in paper I and therefore a more detailed description is included here.

The total airflow into the passenger compartment was measured for several different distribution modes and blower voltages; in total 50 different combinations. For explanation of the different modes see air distribution in Definitions. The results are plotted in Figure 4.7 and two different levels can clearly be seen. These levels corresponds to maximum heat and minimum heat, i.e. maximum and minimum airflow through the heater. The total airflow could be approximated with two different 2nd degree polynomials depending on the location of the temperature flaps, see Figure 4.7. All different temperature flap positions were interpolated between these two functions. Floor and all-mode showed the largest differences with the approximation, still all results were within ±7%. The measurement equipment had an accuracy of 2.4% for 95% coverage probability. These figures were the basis for the estimated accuracy of the airflow presented in paper I.

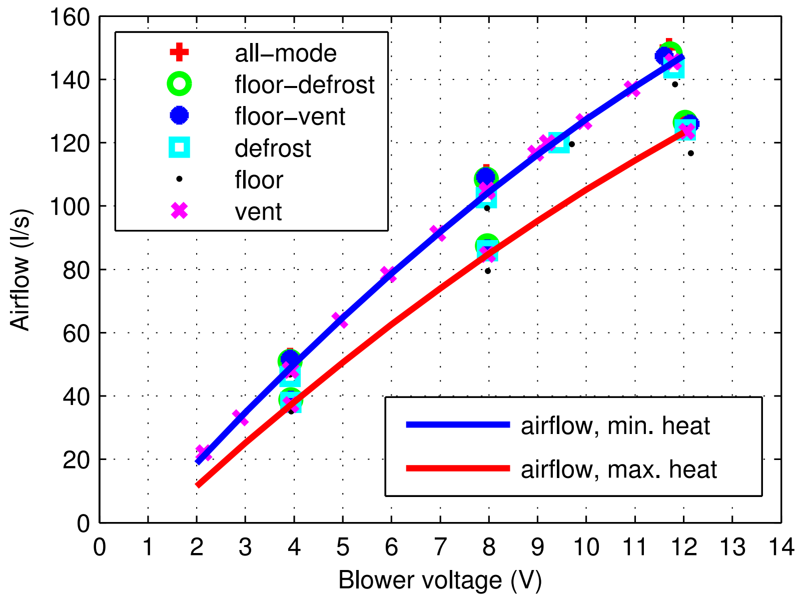

Figure 4.7 Airflow measurement and approximation

The measurement was performed with zero pressure in the passenger compartment, however, in real conditions the compartment introduces a pressure drop for the exiting air. This pressure drop was measured and the airflow estimation was decreased with up to 6% depending on recirculation degree. That is, with 100% recirculation the compartment pressure drop was 0 Pa. Furthermore, for part recirculation the outgoing airflow was measured and the rest was assumed to be recirculated. This gives an estimation of the amount of recirculated air during part recirculation in the measurement. Many other effects were left out of the airflow estimation. For example, pressure differences around the vehicle, at different velocities, could influence the airflow out from the passenger compartment, this was not included. Another ignored effect was increased pressure drop in the evaporator due to condensation. Furthermore, not all leakages in the air handling unit were known, this could affect the estimation.

However, the heat flow on the air side of the heater could be used as a verification of the airflow through the heater because the heat flow on the coolant side was known. These comparisons with different airflow levels and temperature flap positions were informative regarding airflow and leakage in the air-handling unit, for instance through flaps.

In summary, the airflow estimation appeared to be sufficiently good.

5 SIMULATION MODEL

Complete vehicle tests are difficult, time consuming and expensive. One alternative to tests are simulations, however, creating a model which includes all the important sub systems of the climate system is a challenge. Paper II, “Simulation of Energy Used for Interior Climate”, is summarized in this chapter. This paper focused on how the different sub systems, which are important for the climate system, were modelled. In addition, this chapter also includes a thorough investigation of the difference between measured and simulated compressor power.

5.1 Development and Verification

The objective of the paper was to explain how a simulation model of the climate system can be created, that is, how the different important sub systems can be and were modelled.

The model was created in the commercial software GT-SUITE by Gamma Technologies, which is a 1D simulation software. The modelled vehicle was a Volvo S60, that is, the same vehicle that was investigated in paper I. In Figure 5.1 the different sub models of the complete model are presented. The main emphasis of the model was on the air-handling unit, the passenger compartment, the AC-system and the cooling fan for the radiator and condenser. The model also included engine, water jacket, oil circuit, drivetrain and engine cooling circuit.

The air-handling unit consisted of the air intake, recirculation-flap, air distribution-flaps and temperature-flaps, blower, blower motor, blower control, filter, ducts and nozzles. The main objective of the air handling model was to estimate a correct total airflow, distribution between different outlets, air temperature, blower power and heat pickup. In general the correct distribution and pressure loss was modelled with pressure loss coefficients representing the different geometries and ducts. Many different data sources were used for the air handling unit, for instance CFD for the ducts and air distribution, component measurement for the blower, blower motor and control and complete vehicle tests for the total pressure drop and heat pickup.

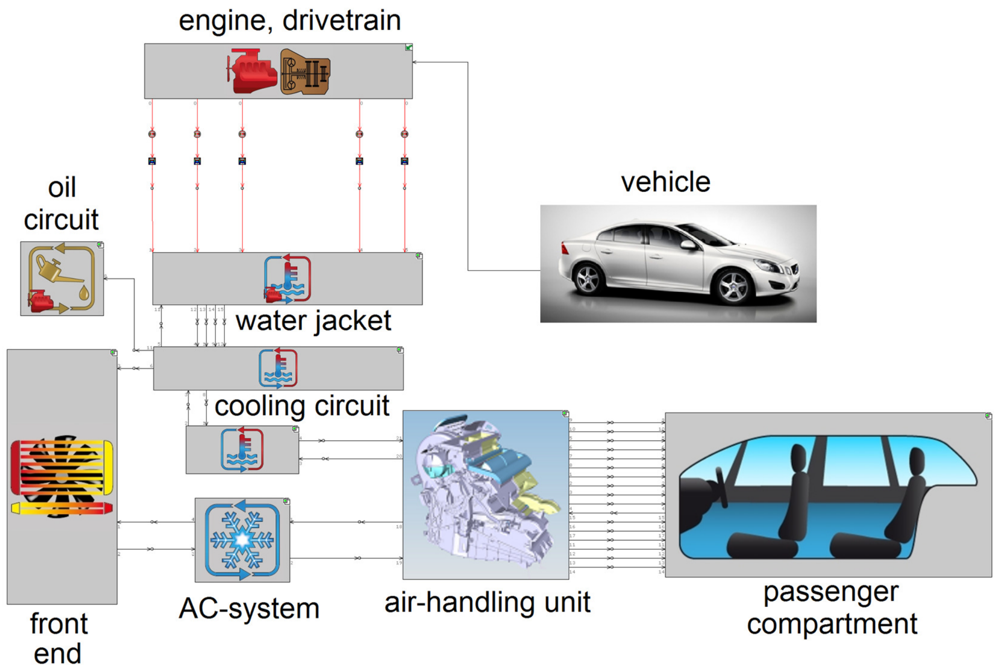

Figure 5.1 Overview of the model

The main objective of the passenger compartment model was to simulate a realistic average temperature for different combinations of airflow, inlet temperatures, ambient conditions and thermal state of the interior masses. Furthermore, the purpose was also to estimate the heat flow to different sinks, e.g. windows, shell, and exiting airflow. The model was divided into 28 air volumes, six windows, four doors, the roof and interior masses spread throughout the air volumes. Air flowed through the volumes and heat was transferred to the exterior components and interior masses through convection. The heat transfer coefficients from the air in the compartment to the interior masses and the exterior were multiplied by a function calibrated from complete vehicle tests. The function were dependent on total airflow, air distribution mode and recirculation. In total eleven different tests were made in a climate chamber for the calibration function.

The AC-system model consisted of a compressor, condenser, evaporator, TXV and pipes. All components were calibrated with either component measurement or specifications, that is, no complete system calibration was performed. The important output from the model was refrigerant mass flow, compressor power and the cooling capacity, i.e. evaporator temperature.

The main purpose of the sub models front end, engine, drivetrain, vehicle, water jacket and oil circuit, was to simulate a realistic heat flow from the engine to the coolant depending on road load and ambient conditions. Furthermore, the condenser airflow and engine cooling fan power were also important outputs from these models.

The model was extensively verified, all tests presented in the measurement paper, paper I, were used for verification. In general there was good agreement between the measurement and simulation, especially regarding test averages. For instance, the largest mean absolute error for the average compartment temperature was 2.4°C.

However, the compressor power differed significantly between the measurement and the simulation, for small loads up to 60% and 260 W and for large loads 40% and 1200 W. Several potential causes of the difference was discussed in the paper, although a definite answer to the problem could not be found. The most likely explanation was issues with the torque measurement.

In summary the paper showed how the important systems for the energy use for vehicle interior climate could be modelled. Furthermore, it presented extensive verification of many different conditions. In general the physical vehicle and model showed good agreement except for the compressor power.

5.2 Compressor Power Investigation

One of the weakest part of the complete vehicle thermal model was the relatively large difference between measured and simulated compressor power. Some details regarding measurement problems were included in the first paper, however, the verification presented in paper II showed errors larger than could be explained by the measurement uncertainty. The simulation paper highlighted some potential issues and concluded that the measurement was most likely the problem although this could not definitely be settled. In this section a comparison with another torque measurement in a complete vehicle is presented.

In this section measurements of the original S60, another S60 and a simulation are compared. The vehicles were quite similar, same model, essentially the same climate system although different engines. The test temperature was 43°C, the sun load 1000 W/m² and the test was 90 minutes long with three different vehicle velocities. First, 50 km/h for 30 minutes, then 100 km/h for 30 minutes followed by 30 minutes of idling. Before the start of the test the vehicle was soaked until the average compartment temperature was 60°C. The climate settings were full vent, i.e. all air directed through the panel vents, maximum airflow, maximum cooling and full recirculation. In the simulation the temperature before the evaporator was set to be exactly the same as in the measurement. The original vehicle is denoted "meas." and the compared is denoted "meas. other S60".

In Figure 5.2 the compressor power of the comparison is presented. The simulation clearly have the smallest power and the measured the largest. However, during the first 30 minutes the other S60 have higher power than the simulation, this changes during the 100 km/h phase where they are almost equal. The measurement also shows large stepwise changes.

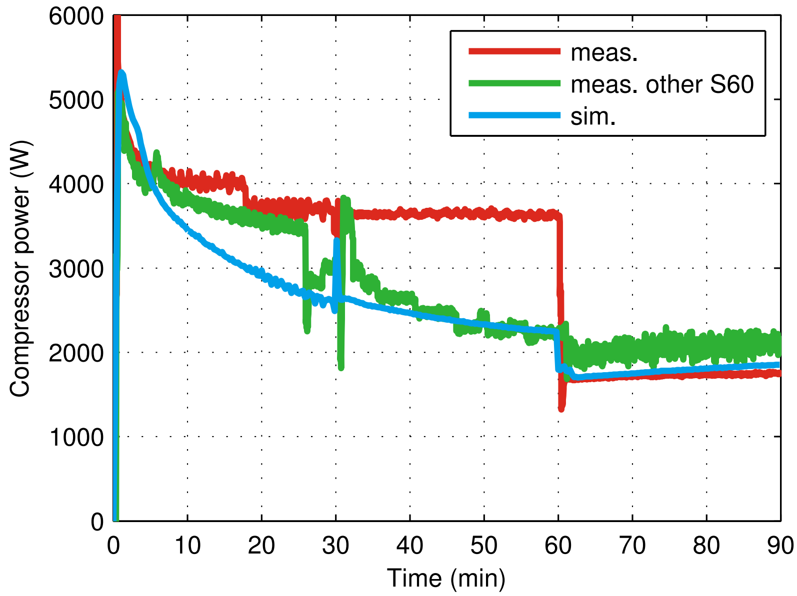

Figure 5.2 Compressor power comparison

The results are summarized in Table 5.1. The most striking difference between the different tests were during the 100-phase where the measurement had a much higher load which did not decrease.

Table 5.1 Average compressor power comparison

	"50 phase" (kW)	"100 phase" (kW)	Idle (kW)	Average (kW)
Meas.	3.9	3.6	1.7	3.1
Meas. other S60	3.6	2.5	2.1	2.7
Sim.	3.3	2.4	1.8	2.5

In the compared tests the airflow over the evaporator were similar, the measurement and simulation have been verified earlier and the other S60 uses an identical air-handling unit. A comparison of the temperatures before and after the evaporator is presented in Figure 5.3. The measurement and simulation use the exact same temperature before the evaporator and all the other temperatures were quite similar with the exception of the temperatures during idle. Humidity levels and condensation were similar in the measurement and simulation, the amount of condensation in the other S60 was unknown, however, it is likely that the load was similar because of the identical test procedure. In summary the evaporator load was comparable between the different tests.

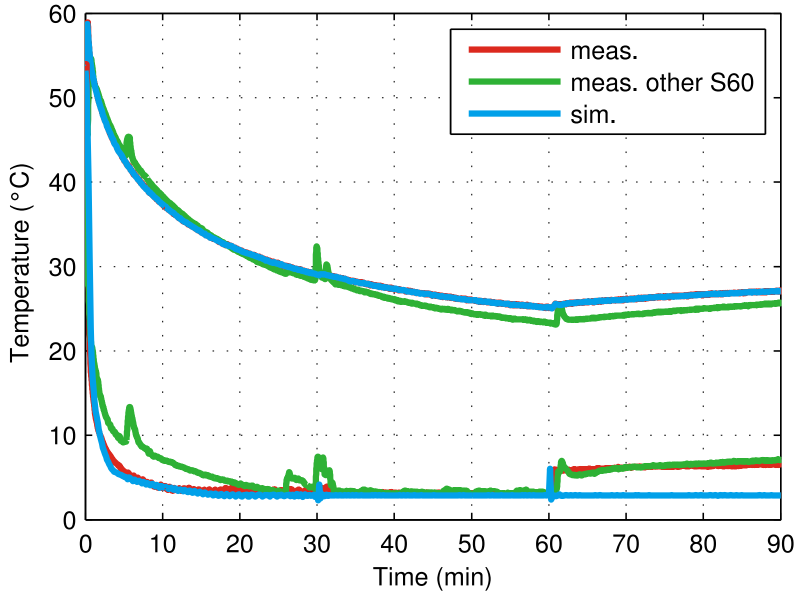

Figure 5.3 Evaporator temperature comparison

The original S60 and the compared S60 were equipped with different compressor although the specifications of the compressors were similar. Still, the compressor speed during the test differed significantly due to different engine speeds and different gearing, see Figure 5.4.

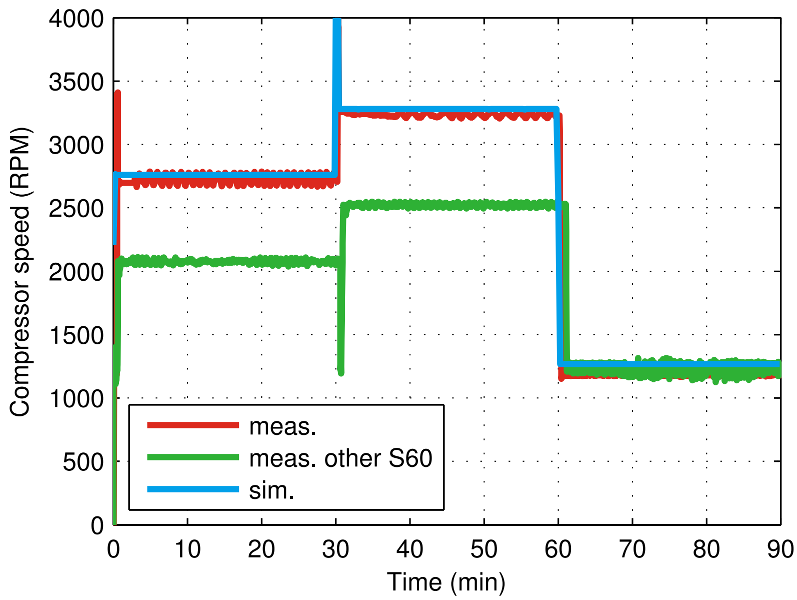

Figure 5.4 Compressor speed comparison

Another factor that differed were AC-system pressures, in Figure 5.5 both the high and low pressures of the systems are presented. The simulation had the lowest high pressure, though it was comparable to the other S60 in the 100-phase.

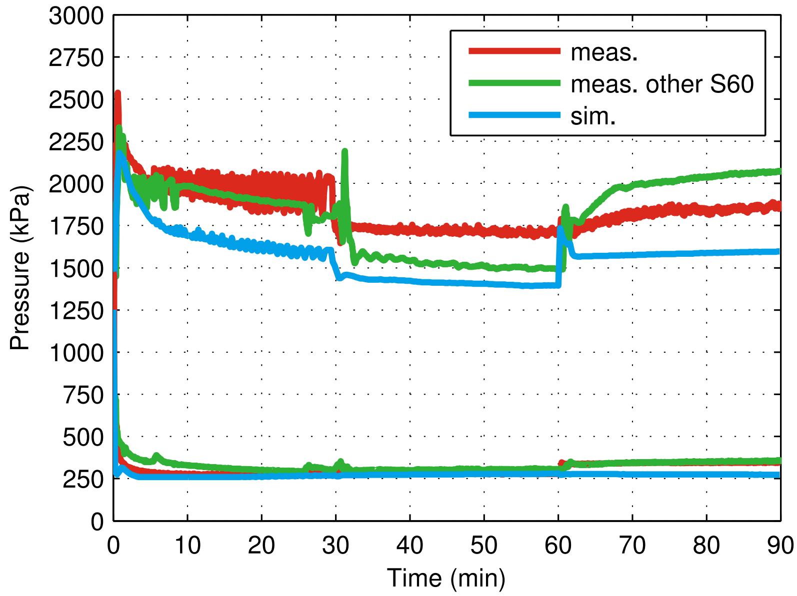

Figure 5.5 AC-system pressure comparison

The simulation model was not designed to handle idling, this explains the difference between the measurements and simulation in the last 30 minutes. The difference is mainly due to condenser air temperature, the real vehicle have recirculation of hot engine compartment air which is not implemented in the model. See Figure 5.6 for a comparison of the average condenser air temperatures.

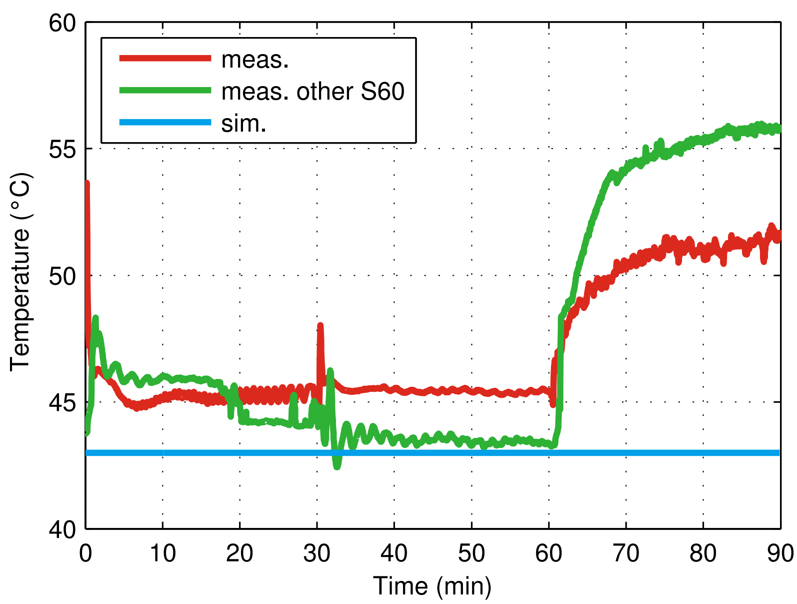

Figure 5.6 Condenser air temperature comparison

The condenser airflow is calibrated for the simulation from a combination of a CFD simulation and estimations from AC-system bench tests. It was possible to increase the high pressure and compressor power by decreasing the airflow. However, the decrease had to be very large to get any significant increase in the compressor power, for a 10% increase of compressor power the airflow had to decrease with 30%. The level of decrease that would be needed for comparable compressor power between the measurement and simulation would be much less than current levels used in these vehicles.

What does these results indicate? That the simulation behaves much more like another comparable system regarding pressure and compressor power. Moreover, the compressor power, i.e. torque measurement, for the measurement seems unreliable for the 100 km/h phase. It does not decrease despite that the load decreases significantly. However, the pressures differs between the measurement and simulation which indicate that the system does not operate exactly the same. The system boundary with the largest difference and most uncertain factors was the condenser cooling. The temperatures were not the same and the airflow was unknown in the measurement, however, the decrease required in the simulation for comparable compressor powers were unrealistic. One possible explanation for the difference is that the AC-system behaves very differently for high compressor speeds compared to test bench tests, i.e. a large decrease in efficiency. However, nothing except these measurements suggest that this could be the case. Problems with the measurement still seems like a reasonable explanation, however, the pressure difference is not explained by this.

The average compressor power for the three cases without the idling is 3.8 kW for the measurement, 3.1 kW for the other S60 and 2.9 kW for the simulation. The simulation is definitely comparable to the measurement in that case.

6 TEST CYCLE

Vehicles can be used in many different conditions, from arctic cold conditions to blazing hot desert conditions and everything between. However, the extreme conditions are generally only important for performance, not the average energy use of the vehicle. Therefore an investigation into what the most common conditions were was done with the objective of creating a test cycle representative of real-world conditions.

6.1 Development

Paper III, "Reduction of Energy Used for Vehicle Interior Climate", contained two parts, one part discussing a representative test cycle and one part using the test cycle to evaluate a specific set of combined energy saving measures. In this section the test cycle development is summarized, the evaluation is summarized in chapter 7.2.

One objective of the test cycle was to include the three different modes that the climate system operates in. These are heat up in cold conditions, reheat in intermediate conditions and cool down in warm conditions. Another objective was that the cycle should be representative for the customer experienced real-world energy consumption. As a consequence, three different tests were created and the energy use was weighted with vehicle use in each condition. The result would be a representative energy use.

The conditions for the test were based on a combination of the hourly ambient conditions for 32 different cities weighted with sales distribution of Volvo Cars and departure time. The sales were distributed around the world although a majority of vehicles were sold in Western Europe, see Table 6.1 for an overview. This combination provided a trip distribution as a function of ambient temperature, see Figure 6.1, and this was used to set the test temperatures. Other quantities followed from the trip distribution division. See Table 6.2.

Table 6.1 Sales, Volvo Cars 2014

Market	Sales	Share
Sweden	61 357	13%
USA	56 371	12%
China	81 574	18%
Western Europe	182 157	39%
Other markets	84 407	18%
Total sales	**465 866**	**100%**

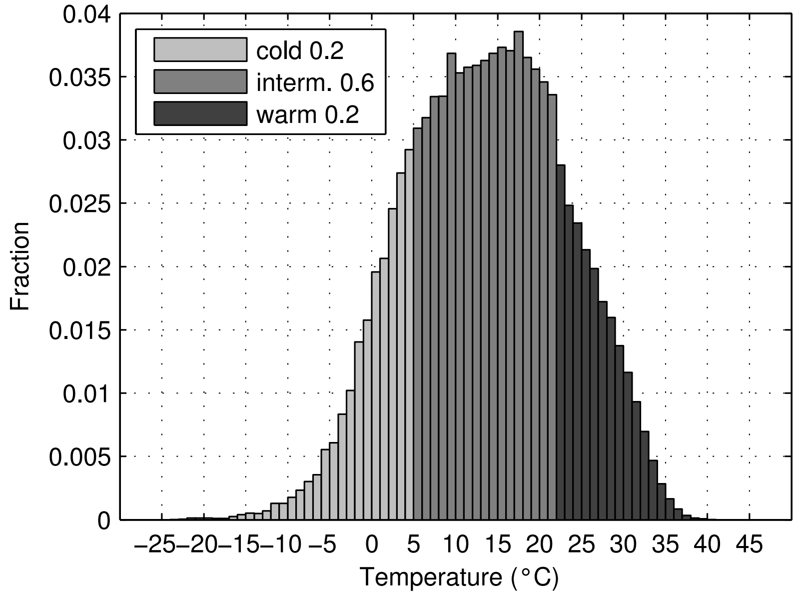

Figure 6.1 Trip distribution as a function of temperature, weighted with departure time and sales distribution

Table 6.2 Conditions for the tests included in the test cycle

	Weight	Temperature (°C)	Dewpoint (°C)	Sun load (W/m²)
Cold	0.2	0	-4	0
Intermediate	0.6	15	7	200
Warm	0.2	27	17	400
Weighted average		**14**	**7**	**200**

The sun height was 37° for the intermediate test and 50° for the warm test. The azimuth angle rotated 360° around the vehicle. Furthermore, the vehicle was sun soaked for one hour in the intermediate and warm tests. The parking time was not investigated further. Other properties such as velocities and climate settings were also not thoroughly investigated. For the velocity profile and test time the WLTP, see [6], was used, furthermore the climate control system was set to automatic mode with a temperature setting of 22/22°C.

In summary, a test cycle was developed which included the three distinct modes of operation of the climate system and weighted these with the estimated proportion of trips in that condition.

6.2 Base Case Results

The energy used for the interior climate was estimated by using the simulation model with the developed test cycle. In this section the results are presented from the three different tests, which were included in the test cycle, and the weighted averages. Furthermore, it also includes a more thorough review of the sinks during the tests.

6.2.1 Average energy use of the climate system

Averages of the important energy users for the interior climate is presented in this section. The presented case is the base case representing the average energy use for this vehicle.

In Figure 6.2 the average electrical loads are presented. The highest average power was in warm climate where the higher cooling fan level separated it from the other conditions. In average approximately 180 W of electrical power is used by the climate system, excluding seat heating.

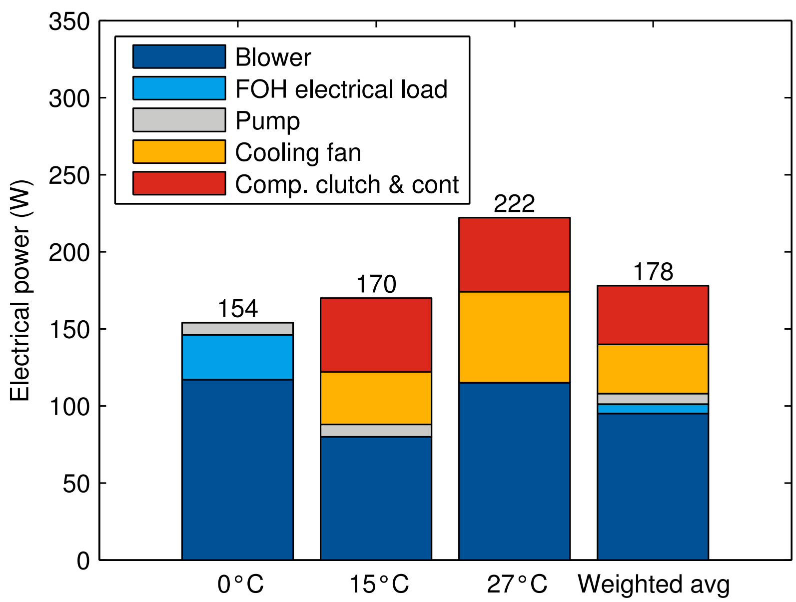

Figure 6.2 Average electrical loads, base case

The mechanical load, consisting of just the compressor, is larger than the electrical load, see Figure 6.3. In 0°C the AC-system is disengaged and the 15°C test represents temperatures from 5° to 23°C, this means that the cut-off temperature for the AC-system is 5°C in the simulation. This is somewhat higher compared to the real vehicle were the limit is approximately for 0°C ambient temperature. The average mechanical load was 475 W.

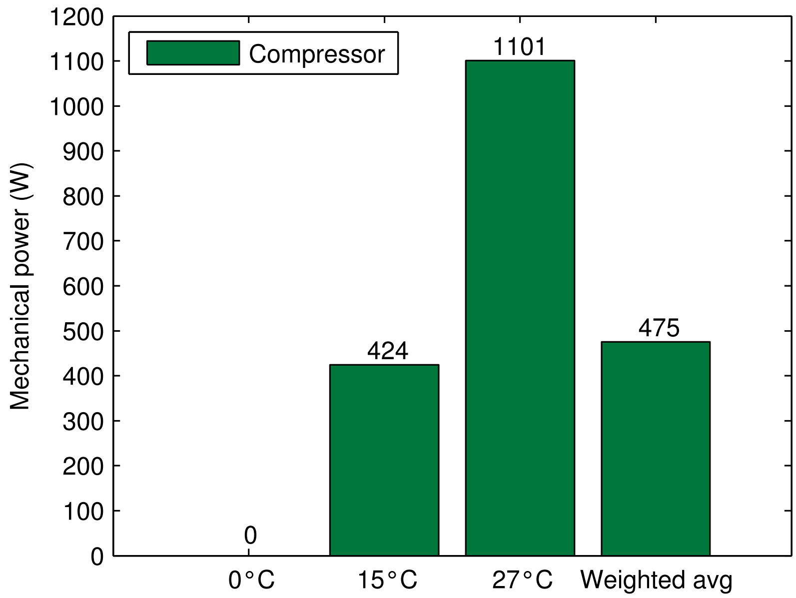

Figure 6.3 Average mechanical loads, base case

The fuel operated heater was only engaged in 0°C. However, the heater cooled the coolant flow somewhat in the other tests, decreasing the average heat received from the FOH, see Figure 6.4. This should not have been included in the average, however, for consistency this calculation has not been changed. The average FOH heat would have been approximately 25 W larger if the correct calculation was used.

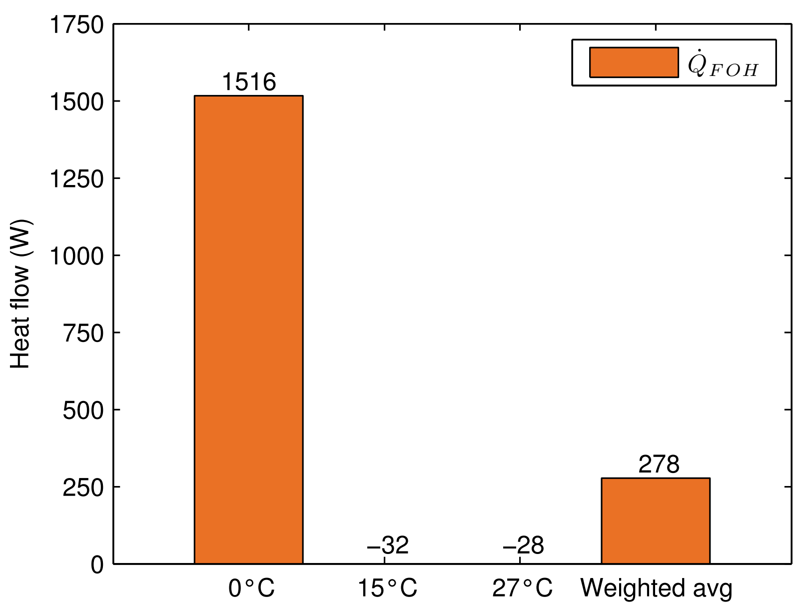

Figure 6.4 Average heat flow from the FOH, base case

There were many different sources for heating and cooling in the climate system. The heat sources were engine waste heat, heat from the FOH and heat added to the incoming air. Two sources of cooling were available, sensible cooling of the air over the AC-system evaporator and recirculation of passenger compartment air. To achieve the required sensible cooling there could also be latent cooling, i.e. condensation. In intermediate conditions this could be helpful by de-humidifying the air to avoid condensation on windows, however, in warm climate the de-humidification was mostly a negative side effect of the required sensible cooling. That is, more humid air would probably increase the comfort, i.e. reduce dry eyes and similar problems. See Figure 6.5 for the heat flows for the different tests. In 0°C only heating was required, however in 15°C the system cooled the air and then heated it, i.e. reheat. The purpose of the cooling was de-humidification, but in the average case the ambient humidity was too low, i.e. the dew point was lower than the evaporator temperature. In warm climate all heat pickup and most of the engine heat was unwanted and required extra cooling.

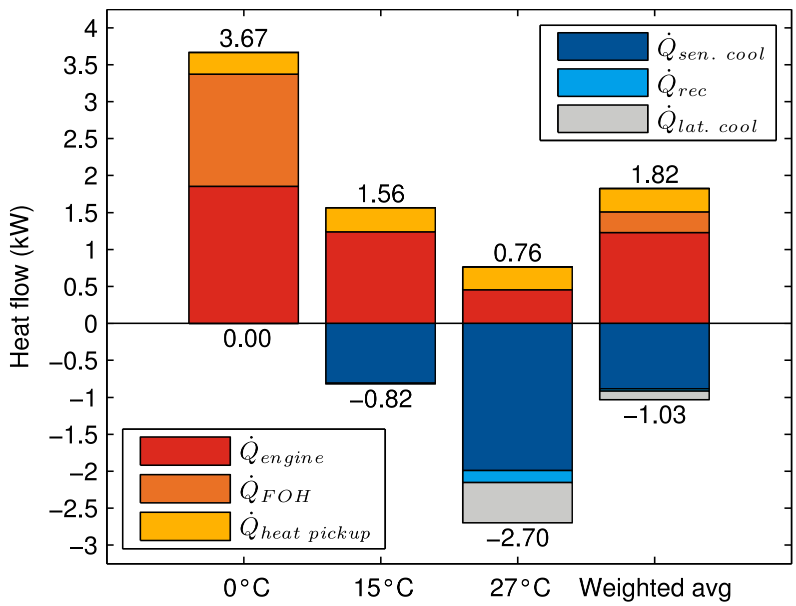

Figure 6.5 Heat flows, base case

The heat flow into the passenger compartment is presented in Figure 6.6. It shows clearly that the required average heat flow into the passenger compartment was smaller than the heat flow of the complete system. The heating flow was 35% smaller, $(1.19 - 1.82)/1.82 \approx -35\%$, and the cooling flow was 73% smaller, $(-0.28 - (-)1.03)/(-)1.03 \approx -73\%$. This indicates that the operation of the system is very important for the average energy use and not the thermal load on the passenger compartment.

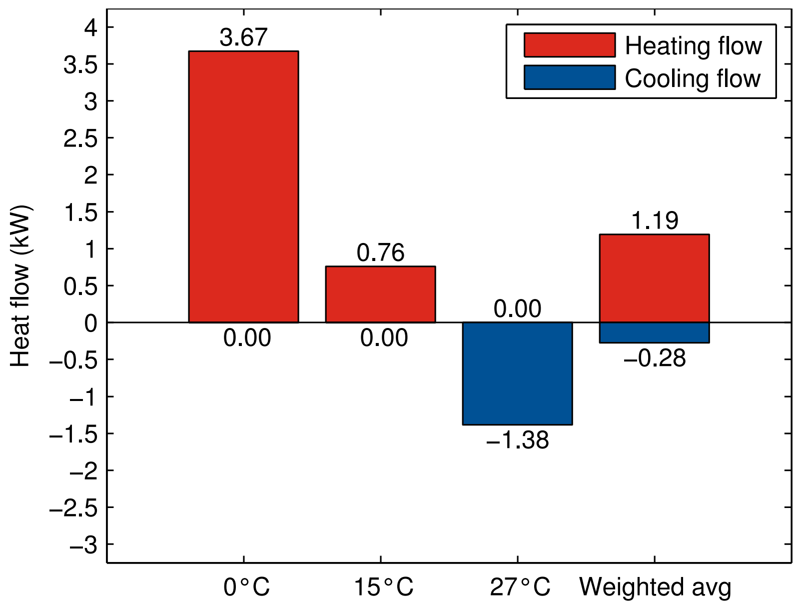

Figure 6.6 Heating and cooling flows into the passenger compartment, base case

The heat flow into the passenger compartment have several different sinks: the ducts, interior masses, shell, windows and the outgoing airflow. In Figure 6.7 the distribution between the different sinks are presented. The single largest sink was the outgoing air, in other words most of the lost heat exits the passenger compartment with the air.

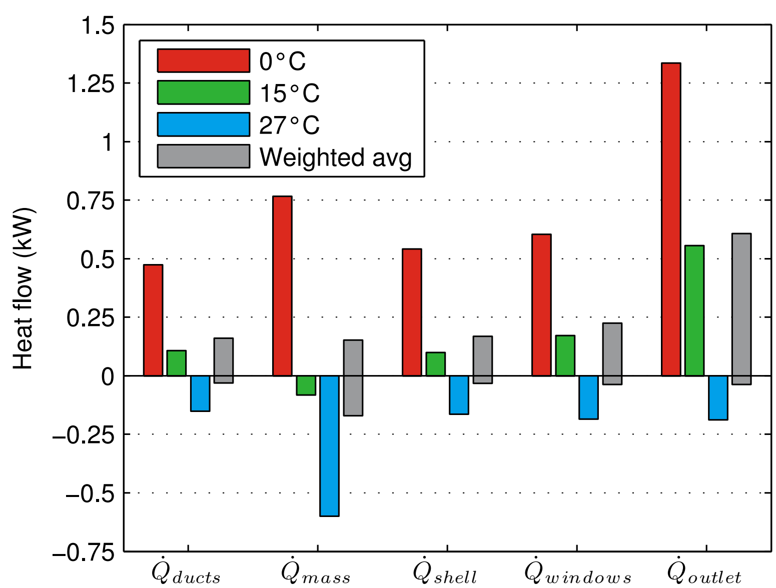

Figure 6.7 Average passenger compartment sinks, base case

6.2.2 Transient sink analysis

Depending on ambient conditions and elapsed time in the transient phase the sinks vary significantly. In Figure 6.8 the different sinks are presented during the heat up test in 0°C. See equation (3.14) for the energy balance and Figure 3.2 for the sinks. The sources of heat in this test were the heated inlet air and the person located in the passenger compartment. The difference between the heat flow of the inlet plus the person and the sum of the different sinks was the energy that increased the temperature of the passenger compartment air. In average only ~50 W was needed to increase the air temperature to comfort level, that is, the average heat required with a perfectly insulated passenger compartment and no airflow.

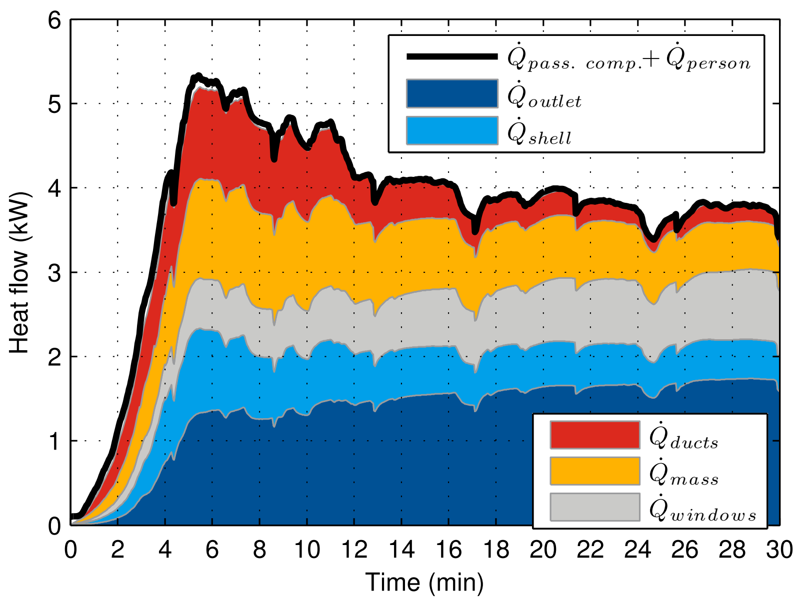

Figure 6.8 Sinks, transient state, 0°C, base case

In Figure 6.9 the different sinks in 15°C can be found. One major difference compared to the colder case was that the mass was now a source of heat. The reason for this was that all transmitted solar radiation was absorbed by the interior masses During the one hour soak enough energy was transferred to the masses to heat it higher than the steady state temperature of the air.

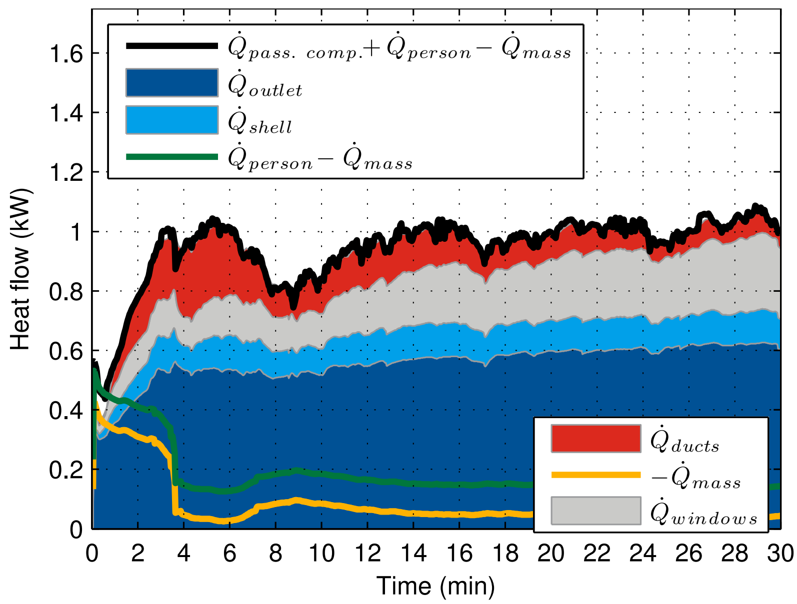

Figure 6.9 Sinks, transient state, 15°C, base case

For the test in 27°C the interior mass was the largest sink for the cooling, see Figure 6.10. A lot of energy was stored in the mass during the soak, furthermore solar radiation was constantly transmitted into the passenger compartment during the test. The temperature difference between the ambient surroundings and the passenger compartment air was also small, decreasing the effect of all other sinks.

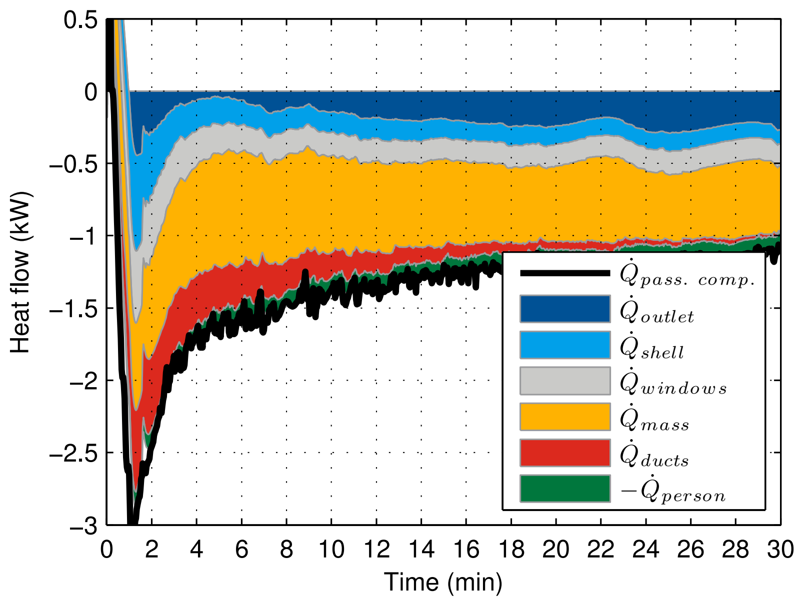

Figure 6.10 Sinks, transient state, 27°C, base case

In summary, for the cold and intermediate case the largest sink was the airflow out from the passenger compartment. In the warm case the sun load is by far the most important contribution, hence the mass becomes the largest sink for cooling.

6.3 Evaluation

The developed test cycle was composed of three different tests, this was described in chapter 6.1. However, it is not obvious that three different test is the best compromise between accurate representation of the different conditions and test cycle complexity. If one test could recreate the total energy use all investigations of energy saving measures would be much simpler. On the other hand one or three test might not capture all the effects on the climate system for different conditions. Therefore a comparison was made of test cycles including 1, 3 and 5 tests. The test cycles are denoted Test Cycle 1 (TC1), Test Cycle 3 (TC3) and Test Cycle 5 (TC5) in this chapter.

The different conditions for each test included in the test cycles are presented in Table 6.3-6.5. TC1 was an average of all conditions and this was represented by the intermediate test of the developed test cycle. The test cycle with three tests, TC3 were the same as explained in chapter 6.1, i.e. the developed test cycle. In order to have colder and warmer test conditions in the cycle the weight of the extreme conditions needed to decrease. Two new tests with 10% weight were therefore added, creating the test cycle with five tests, TC5. However, extreme conditions were unusual and the temperature did not decrease or increase significantly.

Table 6.3 Conditions, TC1

	Weight	Temperature (°C)	Dewpoint (°C)	Sun load (W/m²)
All conditions	1.0	15	7	200

Table 6.4 Conditions, TC3

	Weight	Temperature (°C)	Dewpoint (°C)	Sun load (W/m²)
Cold	0.2	0	-4	0
Intermediate	0.6	15	7	200
Warm	0.2	27	17	400
Weighted average		**14**	**7**	**200**

Table 6.5 Conditions, TC5

	Weight	Temperature (°C)	Dewpoint (°C)	Sun load (W/m²)
Very cold	0.1	-3	-7	0
Less Cold	0.2	5	-1	50
Intermediate	0.4	15	7	200
Less Warm	0.2	23	14	325
Very warm	0.1	30	19	450
Weighted average		**14**	**7**	**200**

The different fractions included in each condition TC5 can be seen in Figure 6.11. The test in the coldest conditions is now performed in -3°C instead of 0°C and the warmest in 30°C instead of 27°C. However, the largest difference between TC5 and TC3 is the increased sun load and the changed temperature range of the FOH and

AC-system. In TC3 the lower temperature limit of engaged AC-system was 5°C and for TC5 it was 1°C, this was more like the real vehicle. Furthermore, the upper temperature limit of activated FOH increased from 5°C to 8°C, which also represents the real vehicle slightly better.

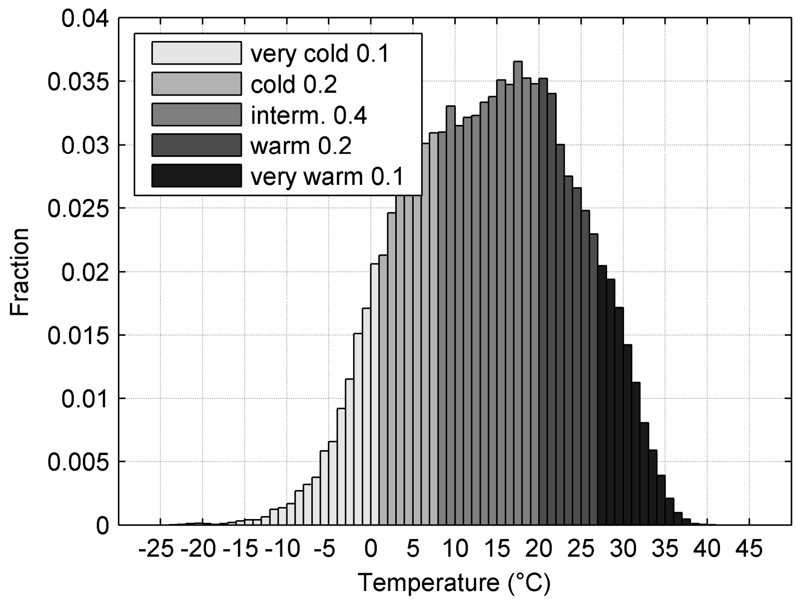

Figure 6.11 Trip distribution as function of temperature, weighted with departure time and sales distribution

The electrical loads for the different test cycles are presented in Figure 6.12. The difference between the cycles was small, i.e. from an electrical stand point the cycle did not influence the results.

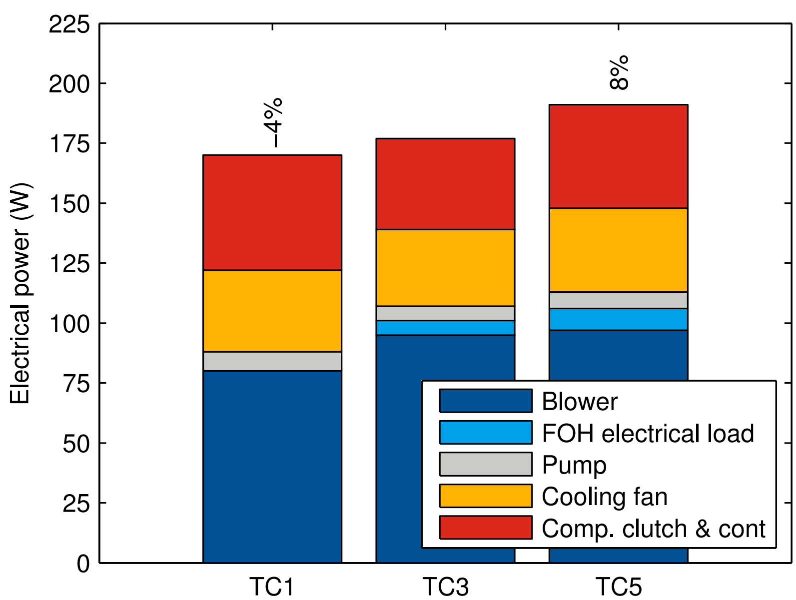

Figure 6.12 Average electrical loads, test cycle comparison

In Figure 6.13 the different mechanical loads can be seen, the same results as for the electrical loads was also valid for the compressor load. That is, that the number of included tests had a small effect.

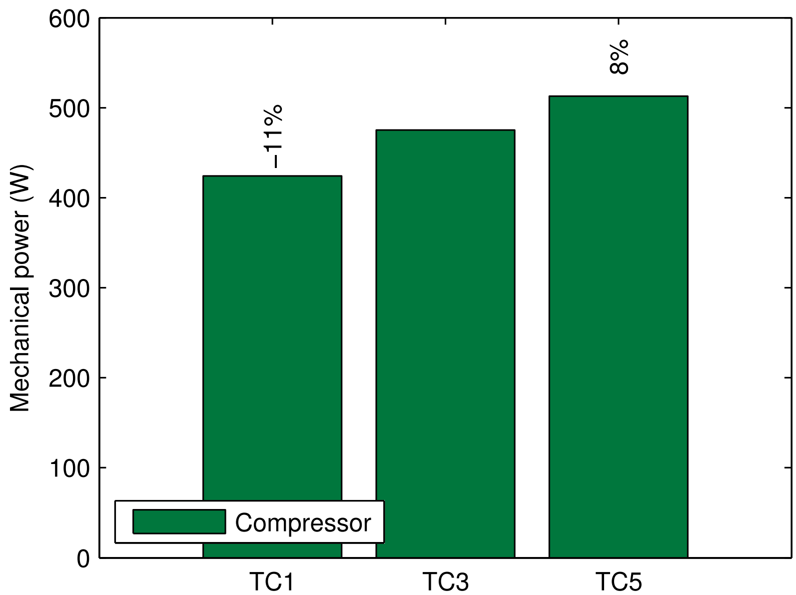

Figure 6.13 Average mechanical loads, test cycle comparison

However, for the FOH heat flow the difference between the cycles were significant, see Figure 6.14. TC1 only included a test in 15°C, in this temperature the FOH was not engaged and this introduced a major difference. Furthermore, for TC5 the weight of tests including an activated FOH increased with 50% compared to the TC3, from 0.2 to 0.3. The heat flow also increased approximately 50%.

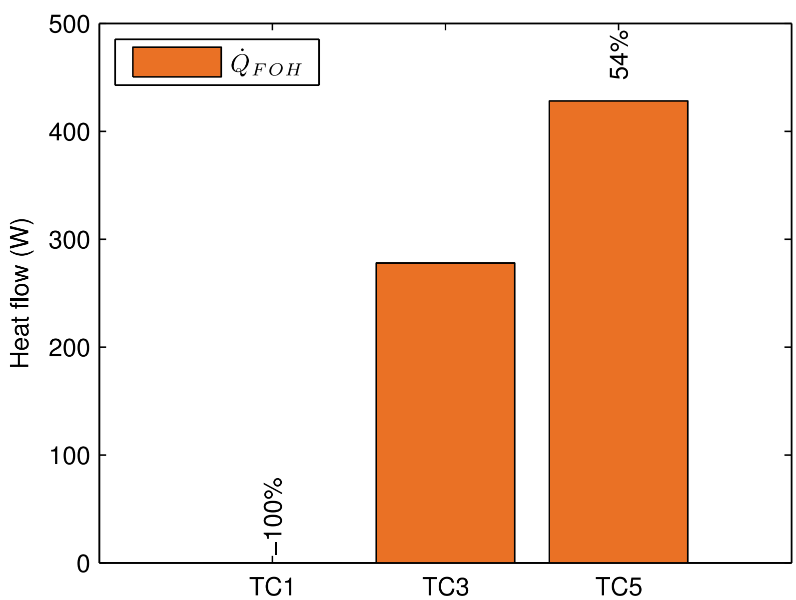

Figure 6.14 Average heat flow from the FOH, test cycle comparison

The average heat flow from the sources are presented in Figure 6.15. A decrease for TC1 and an increase for TC5 can be seen. Furthermore, TC1 lacks three sources, FOH, recirculation and latent cooling.

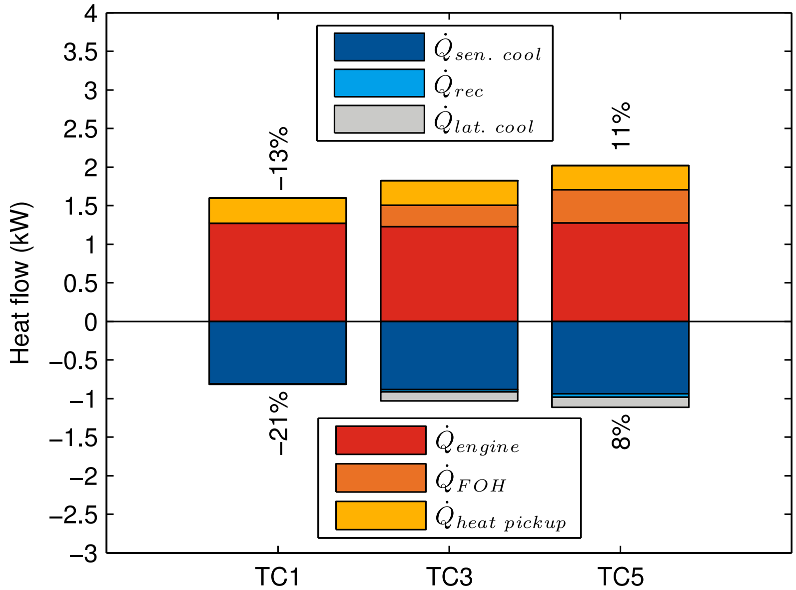

Figure 6.15 Average heat flows from sources, test cycle comparison

For the heat flow into the passenger compartment TC1 clearly shows differences with the other cycles, see Figure 6.16. No cooling was present and the heat flow was significantly smaller compared to the other two cycles. The difference between TC3 and TC5 were small, i.e. the TC3 captured the required energy for the passenger compartment.

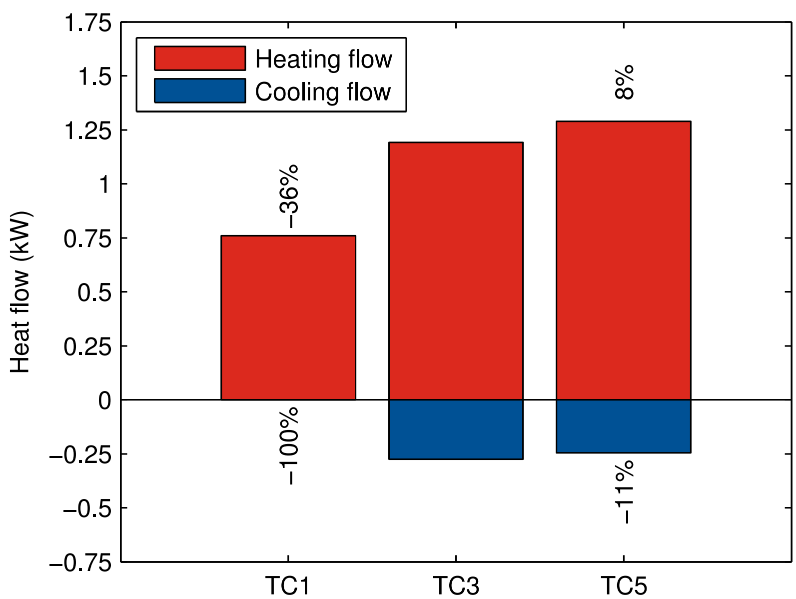

Figure 6.16 Heating and cooling flows into the passenger compartment, test cycle comparison

The same tendency as for the heating and cooling flow into the passenger compartment can also be seen in Figure 6.17. That is, TC1 does not capture all the effects on the sinks. TC3 and TC5 were very similar, only the mass had a noticeable difference.

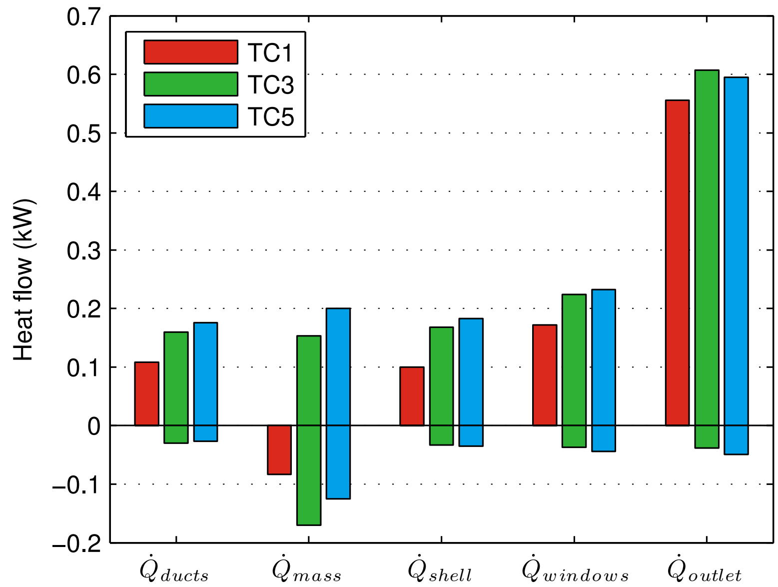

Figure 6.17 Average passenger compartment sinks, test cycle comparison

In summary, the chosen amount of tests in the test cycle was good enough. TC1 did not capture all the effects, for instance no cooling of the passenger compartment and no use of the FOH. Clearly, it cannot be used for evaluating different energy saving measures, however, for the current vehicle the electrical and mechanical load were very similar for all cycles. This is a significant result by itself, it shows that the intermediate condition is the most important condition for the electrical and mechanical loads and that the influence of other conditions were small in comparison. The only significant difference between TC3 and the TC5 was the heat flow from the FOH. It seems that the developed test cycle underestimate the use of the FOH, however, it does not by itself justify the increased complexity by using 5 tests.

7 ENERGY SAVING MEASURES

One of the main question of the thesis is "*How should future vehicles be designed with the aim of using much less energy for the interior climate compared to vehicles designed today*?" In this chapter all the results from the evaluation of the different energy saving measures are reviewed. The main focus is on the two papers that reported results regarding energy saving measures, though the chapter also includes some additional results regarding glass and maximum performance.

7.1 Single Measures

The objective of paper IV, "Potential Energy Consumption Reduction of Automotive Climate Control Systems", was to investigate the theoretical potential of several energy saving measures. By using the developed test cycle the three different operation modes of the climate system would be included in the evaluation. In total 21, different energy saving measures were tested, divided into two categories; source and sink. The two categories were source, how the heating and cooling is generated, and sink, the required heat flow into the passenger compartment. For the different energy saving measures see Table 7.1.

Table 7.1 Investigated energy saving measures

Case no:	Short description	Case no:	Short description
1	Base case		
2	No recirculation		
	Source category		**Sink category**
3	Full recirculation	13	Decreased comfort
4	Perfect HVAC blower	14	No sun soak
5	Perfect condenser cooling fan	15	Perfect preconditioning
6	No heat pickup	16	No duct losses
7	Heat exchanger valve	17	No interior masses
8	AC off 15°C and below	18	Perfect shell insulation
9	Increased evap. temperature	19	Perfect glass insulation
10	Increased AC-system heat trans.	20	Opaque glass, absorption
11	Perfect AC compressor	21	Opaque glass, reflection
12	No AC pressure drop	22	No windows
		23	100% reflecting exterior color

Many of the energy saving measures were unrealistic, for example having no interior mass in the passenger compartment or having a climate system blower with an efficiency of 100%. However, the approach demonstrated the potential energy saving of each measure.

The results were divided into five different groups; electrical power, mechanical power, FOH use, heating and cooling sources and heat flow into the passenger compartment. The most notable reduction of average electrical power was the climate system blower with an efficiency of 100% which reduced the total electrical power with 46%, otherwise the potential savings were small. For the average mechanical power, the best saving measures was, to increase the compressor efficiency to 100%, but other measures such as reducing the AC-system use in intermediate conditions also showed large potential. The FOH was only used in the cold test, thus the only potential for reducing the FOH energy use was measures that affected the heat up, for example full recirculation and no interior masses. In the results the effect of reheat and heat pickup could clearly be seen, the heat flow into the passenger compartment were much smaller than the heat flow from the sources. A more reflecting glass naturally decreased the need for cooling, however, in intermediate climate a more reflecting glass reduced the sun load when heat was needed. This increased the required heat flow provided by the climate system.

In summary paper IV demonstrated that there were few energy saving measures that singlehanded could reduce the electrical load significantly, however, for the mechanical load there were more opportunities. Measures on the sink side, i.e. passenger compartment, could decrease the passenger compartment heat flow significantly, nonetheless the total decrease of energy was much smaller, mainly due to reheat in intermediate conditions. How the system operates in intermediate conditions determines to a large degree the overall energy use.

7.2 Combined Measures

Paper III, "Reduction of Energy Used for Vehicle Interior Climate", presented an evaluation of a combination of realistic energy saving measures divided into the two categories, source and sink. The purpose of the evaluation was to see the potential energy saving and better understand how different sink and source measures affected the system.

The energy saving measures were analysed with the simulation model and developed test cycle. Five energy saving measures on the sink, that is, the passenger compartment, and five on the source were investigated, see Table 7.2. The results were divided into four different cases, base case without any modifications, sink case with measures only on the sink, source case with measures only on the sources and the combined case where the sink and source measures were combined. The average passenger compartment temperature was comparable between cases both during the transient and steady state parts of the tests.

Table 7.2 Summary of energy saving measures

Sinks	
Ducts	Decreased convective heat transfer coefficient and mass by 30%
Interior mass	Decreased mass by 20%
Shell	Increased thermal resistance by 20%
Windows	Increased thermal resistance by 20%, decreased transmission up to 12% and increased reflectivity up to 26%
Color	Increased reflectivity from 25% to 70%
Sources	
Recirculation	Below 5°C: 35% recirculation in transient and 20% in steady state
FOH	Request dependent on heat demand
Blower	Blower and blower motor efficiency increased with 10% each, changed control unit
AC-system	Deactivated AC-system below 15°C, increased maximum evaporator set point from 8° to 12°C
Heat pickup	Decreased convective heat transfer coefficient by 25%

The measures could decrease the average electrical load substantially, mainly by measures in the source category. For the source case the average could be decreased with 43% and the combined case with 50%. Similar decreases were also possible for the average mechanical load, that is a decrease of 36% for the source measures and 44% for the combined case. Sink measures were much smaller, 11% for the electrical load and 8% for the mechanical load. Similarly, the largest effect on the FOH use was by changing the control, i.e. not the required energy for the sink. With modified control and recirculation it was possible to decrease the FOH use with 27% in the source case and 28% in the combined case. The heat flow from the different sources could be reduced significantly, to a large extent by reducing reheat. However, the actual heat flow into the passenger compartment was difficult to reduce. All sink measures could only decreased the heating flow with 14% and cooling flow with 18%.

In summary the paper showed how a combination of energy saving measures would affect the total energy required for the interior climate. The most important result was that the operation of the system have a much larger effect on the energy consumption compared to the required energy into the passenger compartment. This means that source measures must be implemented first before sink measures can have a substantial effect.

7.3 Effect on Maximum Performance

The developed test cycle included mild conditions, that is no extreme cool down or heat up. Although these extreme conditions are uncommon they can be interesting to investigate, especially from a performance stand point. In this section the same energy saving cases as in paper III, i.e. combined energy saving measures, are investigated in hot and cold conditions. For the conditions see Table 7.3. In the hot case the vehicle was sun soaked to an average passenger compartment air temperature of roughly 60°C before the test and in the cold test the interior mass had ambient temperature at the start. The velocity profile was WLTP and the climate control setting was auto 22/22. Note that in these cases the passenger compartment temperature was not controlled to be comparable. That is, both the energy use and performance are compared.

Table 7.4 Maximum performance test conditions

	Weight	Temperature (°C)	Dewpoint (°C)	Sun load (W/m²)
Cold	-	-18	-20	0
Hot	-	43	26.5	1000

7.3.1 Heat up simulation in -18°C

The average electrical power was decreased with approximately 36% in the heat up test for the combined measures. The main energy saving measure was the changed blower control unit. See Figure 7.1 for a comparison of the average electrical load for the different cases in -18°C.

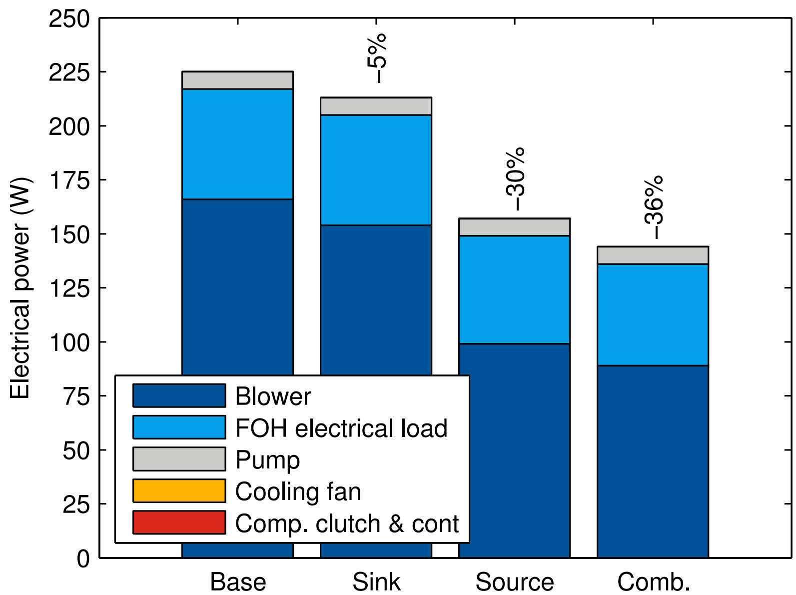

Figure 7.1 Average electrical loads, heat up in -18°C

The heat flow from the FOH is presented in Figure 7.2. A reduction of heat flow was only seen for the combined case with measures on the FOH control and a reduced heat demand.

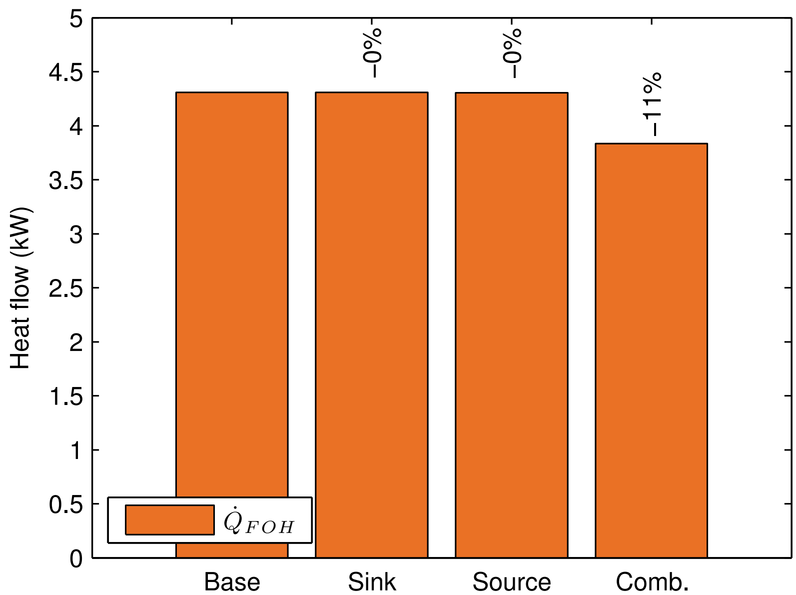

Figure 7.2 Heat flow from the FOH, heat up in -18°C

A new sources of heat for the source and combined case was recirculation. However, the effect was relatively small and mainly decreases the heat flow from the engine, see Figure 7.3.

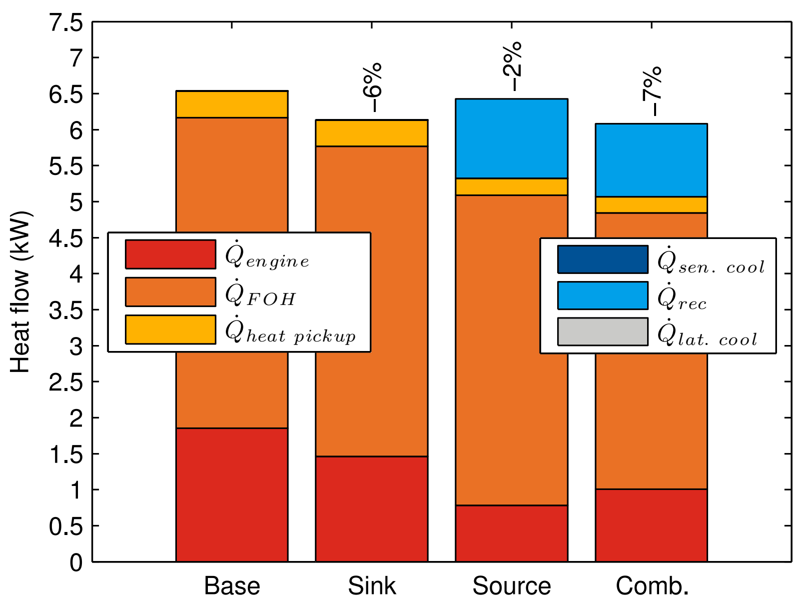

Figure 7.3 Heat flows, heat up in -18°C

Due to the focus on performance the heat flow into the passenger compartment did not differ much between the different cases, see Figure 7.4.

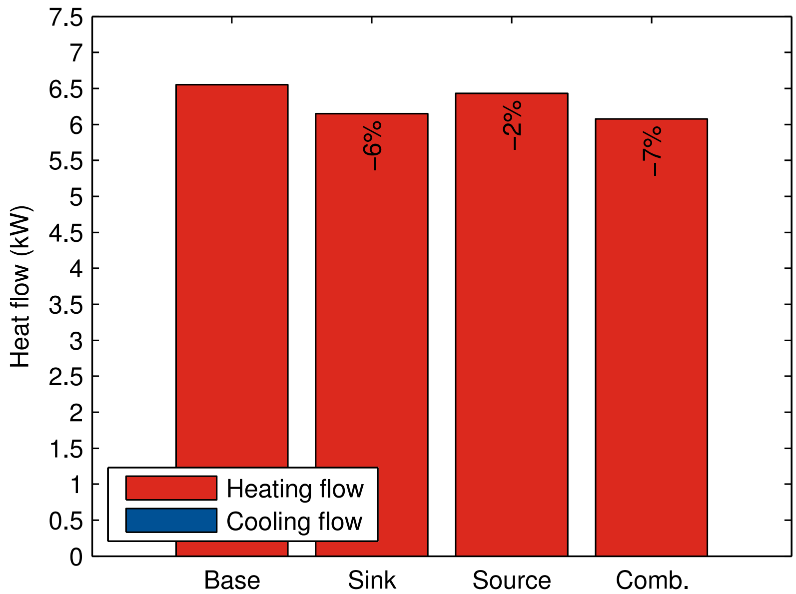

Figure 7.4 Heat flow into the passenger compartment

In Figure 7.5 the different sinks for the passenger compartment is presented. For all cases the airflow out from the passenger compartments was the largest sink. It also increased for the energy saving cases compared to the base case.

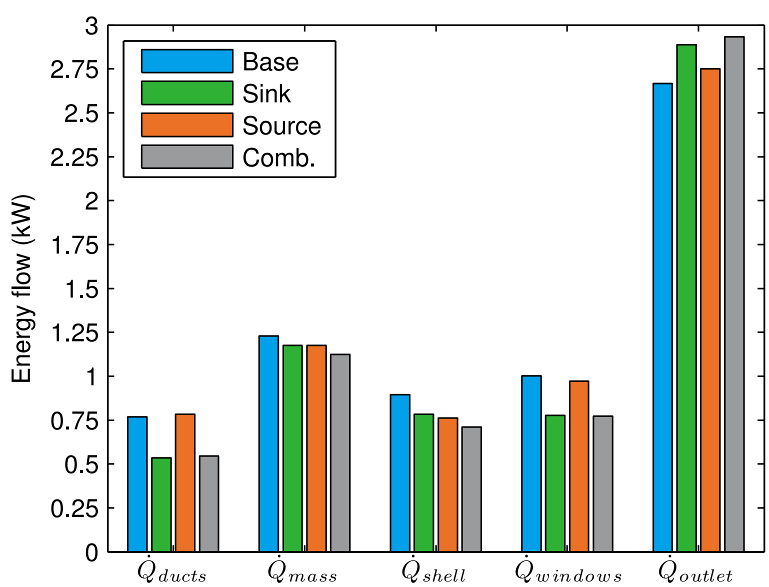

Figure 7.5 Passenger compartment sinks, -18°C

The average passenger compartment air temperature is presented in Figure 7.6. The heat up was faster for all energy saving cases. For the source case the major increase was thanks to the recirculation. Measures in the sink category increased the passenger compartment temperature in steady state, this because it was not compensated in these cases. It affected both the sink and combination case.

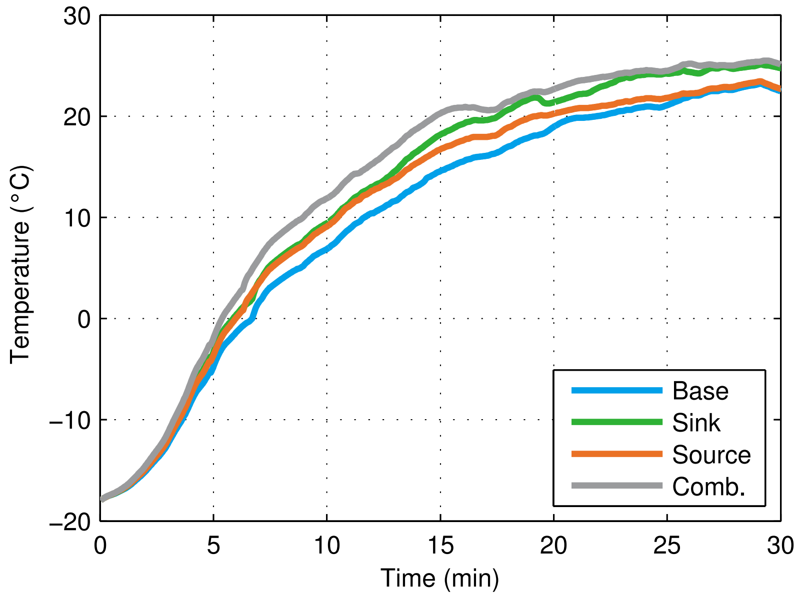

Figure 7.6 Transient average passenger compartment temperature, -18°C

A rough measure of the relevant time to comfort could be defined as the time when the temperature reaches three degrees within the end temperature of the base case. In this test the temperature was ~19.5°C and the decrease of time to comfort can be seen in Figure 7.7.

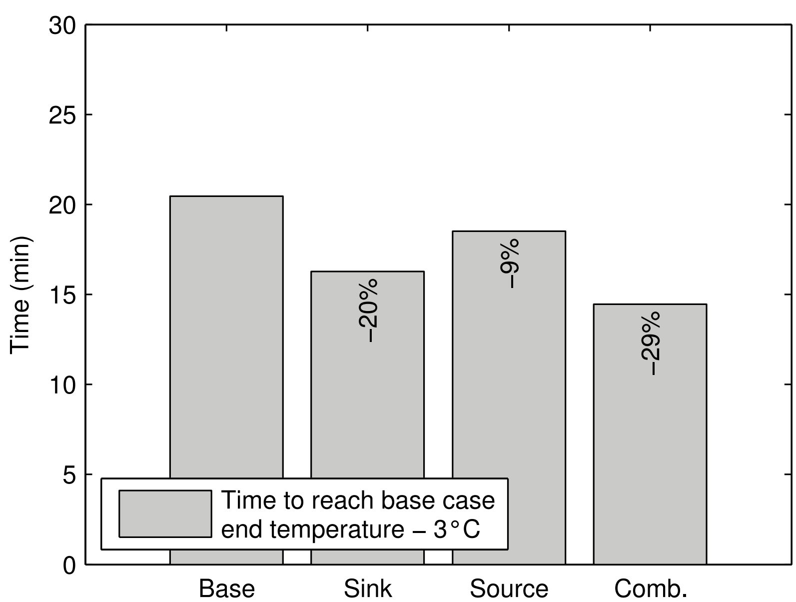

Figure 7.7 Time to reach base case end temperature minus three degrees, -18°C

7.3.2 Cool down in 43°C, 1000 W/m², sun soaked vehicle

The average electrical power can be found in Figure 7.8. The decrease is much smaller compared to the cold case. The blower control unit does not dissipate heat for high blower voltages, i.e. not much energy can be saved then by changing the control unit.

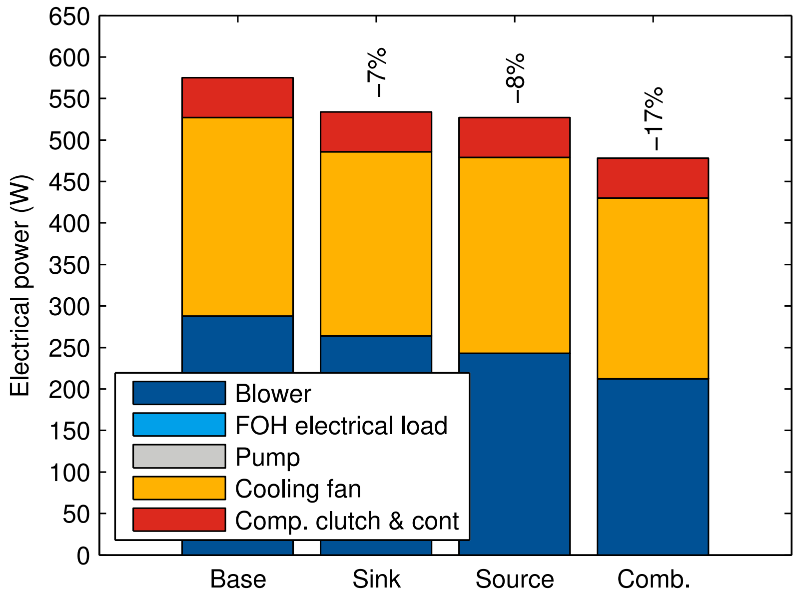

Figure 7.8 Average electrical loads, cool down in 43°C, 1000 W/m² sun load

The average mechanical power was decreased by 10% in the combination case, mainly due to the effects of the sink measures. See Figure 9.

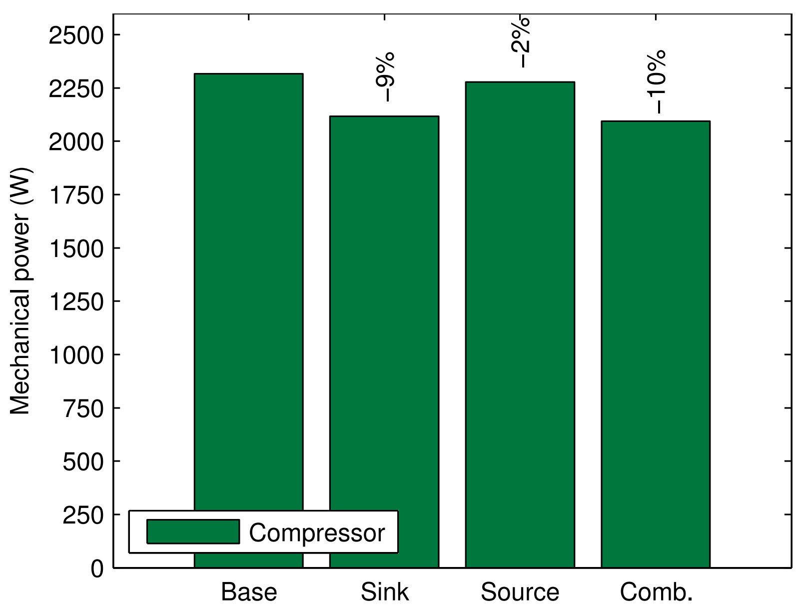

Figure 7.9 Average mechanical loads, cool down in 43°C, 1000 W/m² sun load

Even though the compressor mechanical power was reduced by 10% roughly the same amount of cooling was provided in the combination case, see Figure 7.10. The main reason for this was the decreased load on the passenger compartment, consequently more cooling available through recirculation. Furthermore, the heat pickup was also reduced which also decreases the compressor load.

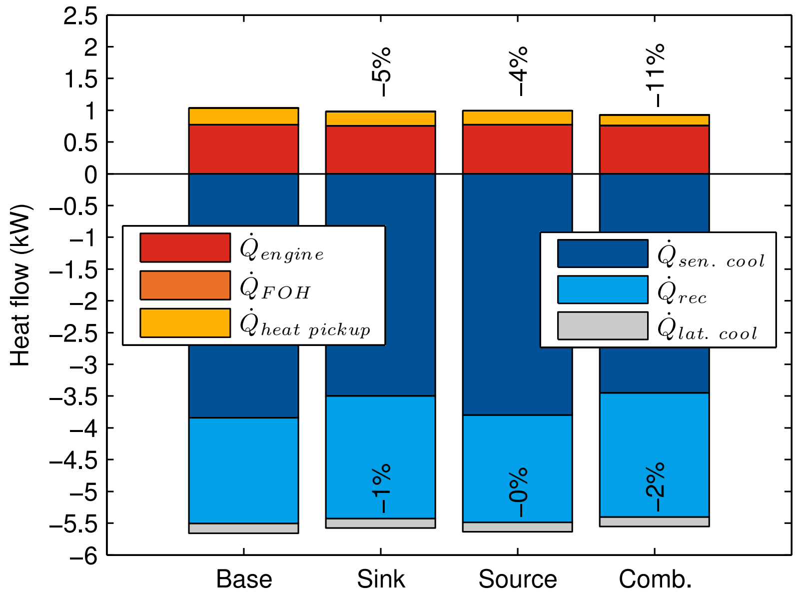

Figure 7.10 Heat flows, cool down in 43°C, 1000 W/m² sun load

Figure 7.11 clearly shows that the same amount of cooling was provided in all four cases.

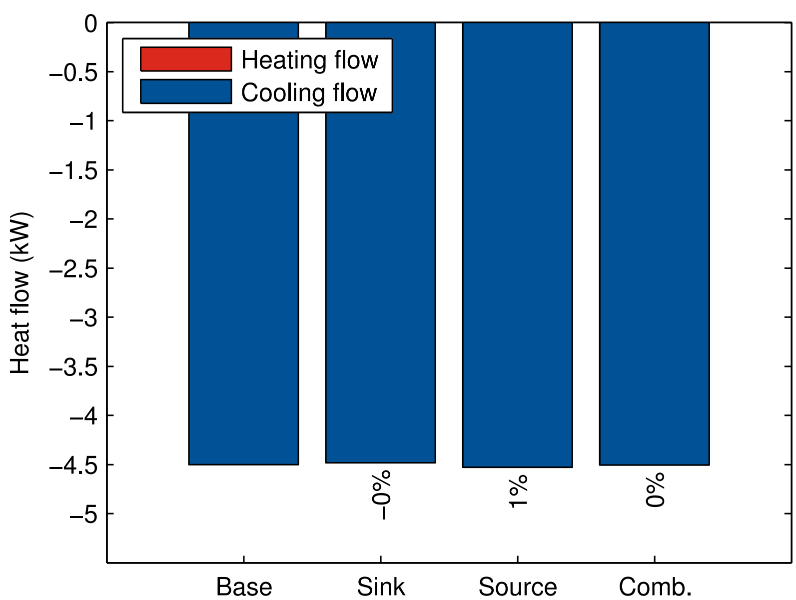

Figure 7.11 Heat flow into the passenger compartment, cool down in 43°C, 1000 W/m² sun load

As in the cold case the exiting airflow was the largest sink, however, the mass was almost as large, see Figure 7.12. The reason for this was the sun load which acted through the mass. Moreover, the airflow was recirculated in this case and therefore no cooling was actually lost with the airflow out.

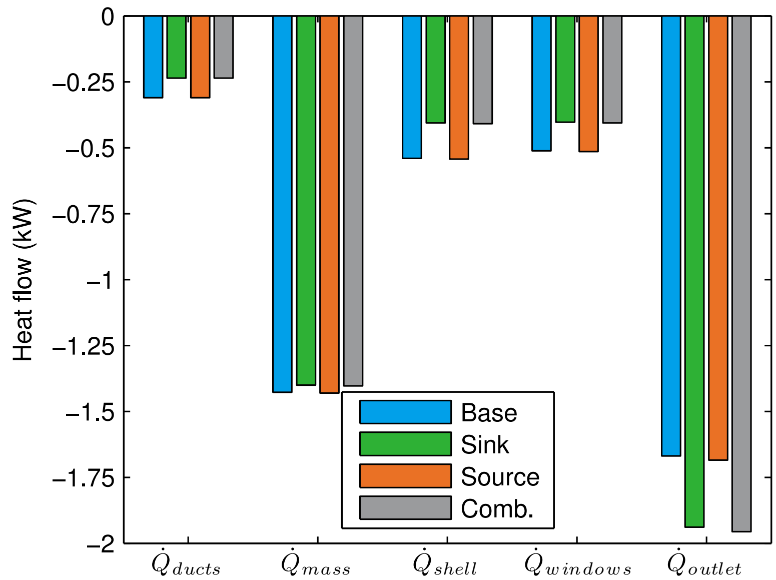

Figure 7.12 Passenger compartment sinks, cool down in 43°C, 1000 W/m² sun load

In Figure 7.13 the average passenger compartment air temperatures are presented. The base case and source case were very similar, the only difference which affected this test was the blower efficiency. Because the energy savings in the source category does not affect the average temperature the sink case and combination case also were very similar. Note that there was a difference in start temperature, mainly due to the difference in windows and exterior colour, the sink and combination case absorbed less energy in the windows and shell. This effect can also be seen in Figure 7.12.

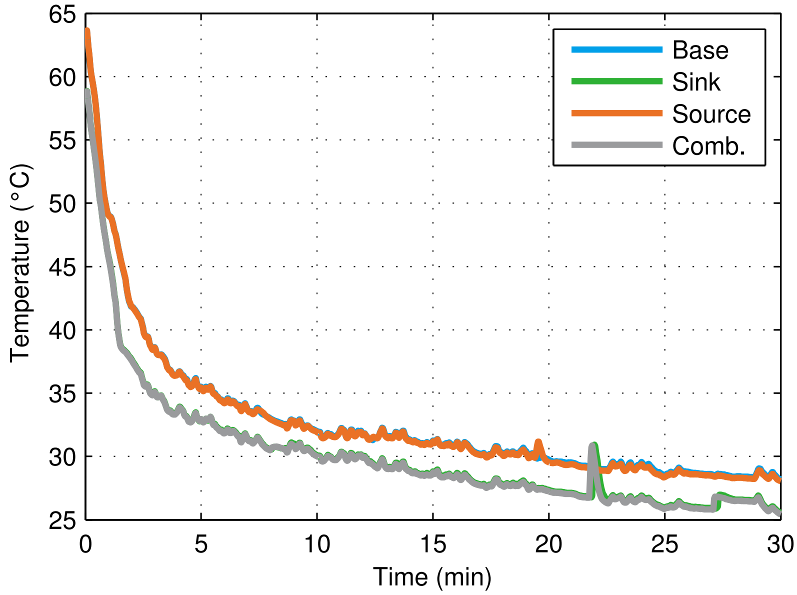

Figure 7.13 Transient average passenger compartment temperature, cool down in 43°C, 1000 W/m² sun load

The difference in time to comfort is presented in Figure 7.14. Time to comfort was defined similar as in cold conditions; time to reach base case end temperature plus three degrees. The effect of the energy saving measures in the sink category was very large. Mainly the reflecting windows and the reflecting exterior colour contributed to the large decrease in time to comfort.

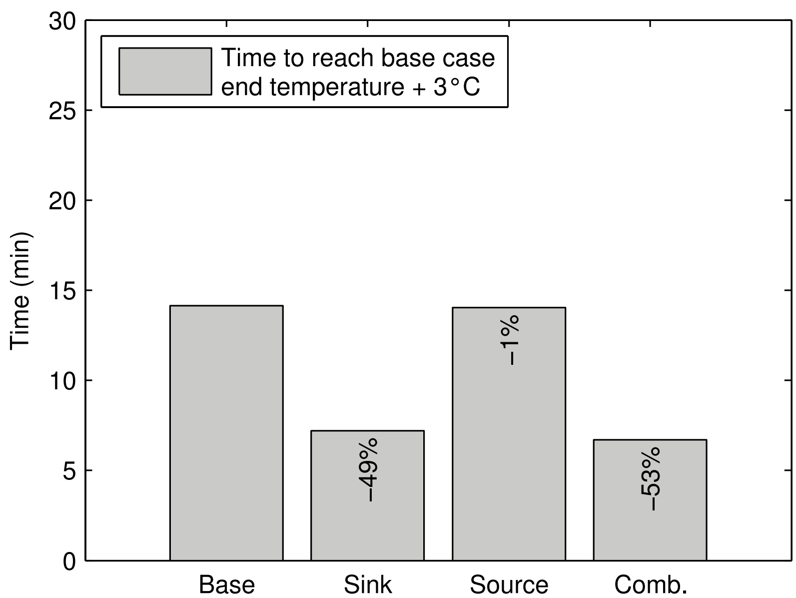

Figure 7.14 Time to reach base case end temperature plus three degrees, cool down in 43°C, 1000 W/m²

7.3.3 Summary of maximum performance tests

Compared to the evaluation of average energy use the energy reduction was much smaller for the performance tests. Particularly the source measures did not decrease the energy use as much as for the less extreme conditions. The sink and combined measures decreased the transient time considerably, 20-30% for the heat up and 50% for the cool down. These results can be an indication on why the source measures generally have been neglected, the effect is relatively small in performance tests.

7.4 Different Types of Glass

The influence of the different types of glass on the energy used for vehicle interior climate have been investigated by many, see for example [81–83]. However, in most research the effect is investigated under severe conditions, that is, a high sun load and warm temperatures which are beneficial for the measures. The potential of different glass was investigated in much more representative real-world conditions by using the developed test cycle described in chapter 6.

Four different cases were included in the investigation. First, the base case which have been used for all comparisons, see chapter 6.2 for more details. Second, IR-reflecting windows. Third, a combination of IR reflecting and tinted windows. The fourth case was a 100% reflecting window, the purpose of this case was to see the theoretical potential. See Table 7.5-7.8 for glass properties for the different cases.

Table 7.5 Glass properties, base case

	Transmittance (%)	Reflectance (%)	Absorption (%)
Windshield	58	7	35
Front side windows	48	6	46
Rear side windows	44	5	51
Rear window	43	5	52

Table 7.6 Glass properties, IR case

	Transmittance (%)	Reflectance (%)	Absorption (%)
Windshield	46	31	23
All side windows	46	22	32
Rear window	46	31	23

Table 7.7 Glass properties, IR and tinted case

	Transmittance (%)	Reflectance (%)	Absorption (%)
Windshield	46	31	23
Front side windows	46	22	32
Rear side windows	15	5	80
Rear window	15	4	81

Table 7.8 Glass properties, 100% reflecting case

	Transmittance (%)	Reflectance (%)	Absorption (%)
All windows	0	100	0

In Figure 7.15 the electrical loads for the different cases are presented. Different types of glass clearly have small influence on the electrical load. The glass reduces the airflow required during the cool down, however, this was such a small part of the blower average electrical power that the decrease was almost insignificant.

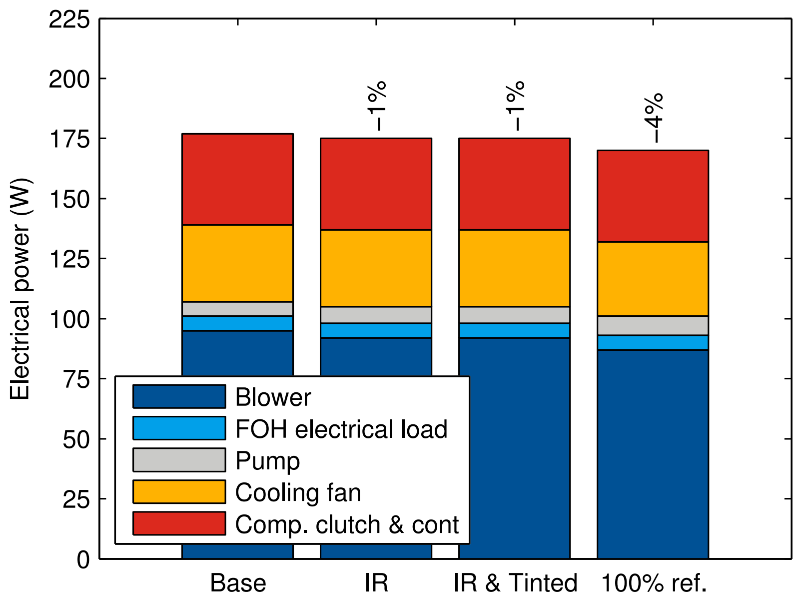

Figure 7.15 Electrical loads, glass cases

The decrease of the compressor load is presented in Figure 7.16. As for the electrical loads the decreases was small, one major factor for this were the maximum allowed evaporator temperature. A decrease of the required cooling into the passenger compartment could not be translated to lower compressor load because the system was not allowed to increase the evaporator temperature. The decrease of cooling was then achieved by increased heating.

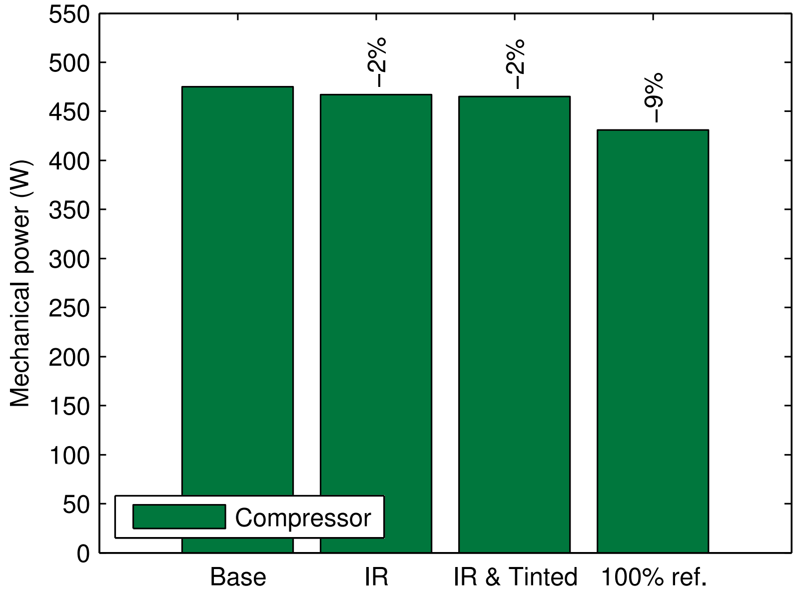

Figure 7.16 Mechanical loads, glass cases

The heat flow from the sources can be found in Figure 7.17 and the heat flow into the passenger compartment in Figure 7.18. Even though the cooling into the compartment was decreased by 7-8% for IR and IR and tinted cases the actual decrease of the source was much smaller, roughly 2%. This because the intermediate case was weighted much heavier compared to the warm case. Another noticeable result was the increased need for heating. The decreased effect of the sun load required that more heat was supplied by the climate system to the passenger compartment in intermediate conditions.

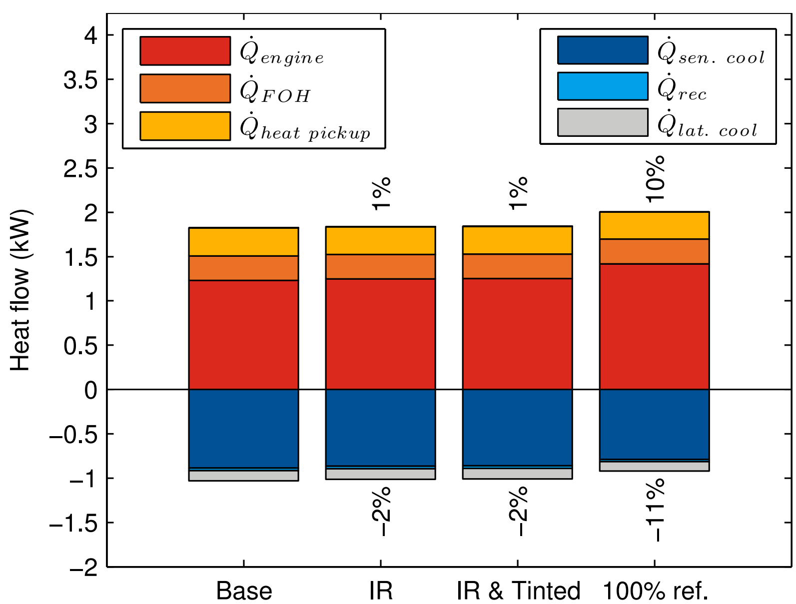

Figure 7.17 Heat flows from sources, glass cases

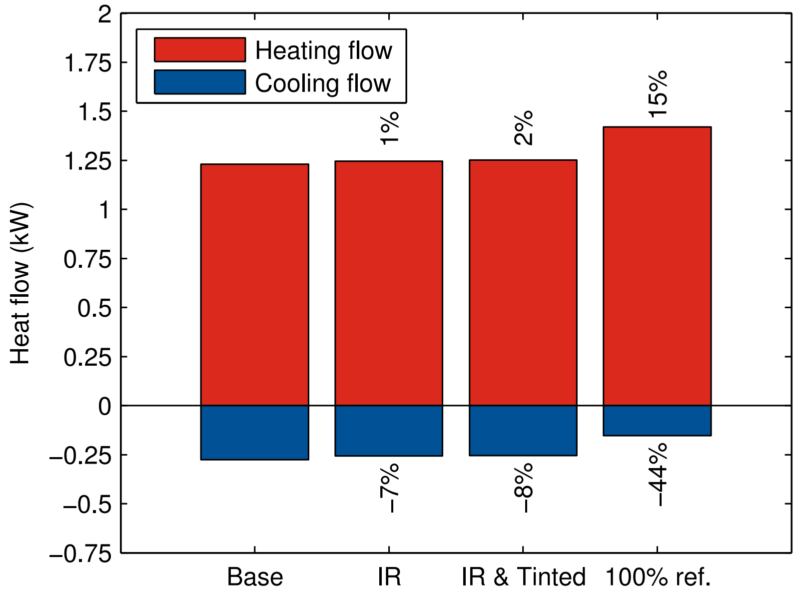

Figure 7.18 Heat flow into the passenger compartment, glass cases

These results indicated that the potential for energy savings with increased reflectance or absorbance of the windows was quite small, at least for the entire market. However, for sunny and warm markets, there could be substantial gains, especially if the performance is considered. In Figure 7.19 the transient average passenger compartment temperature is presented for a cool down test in 43°C with a sun load of 1000 W/m². The vehicle was soaked in these conditions for approximately one hour before the start of the test.

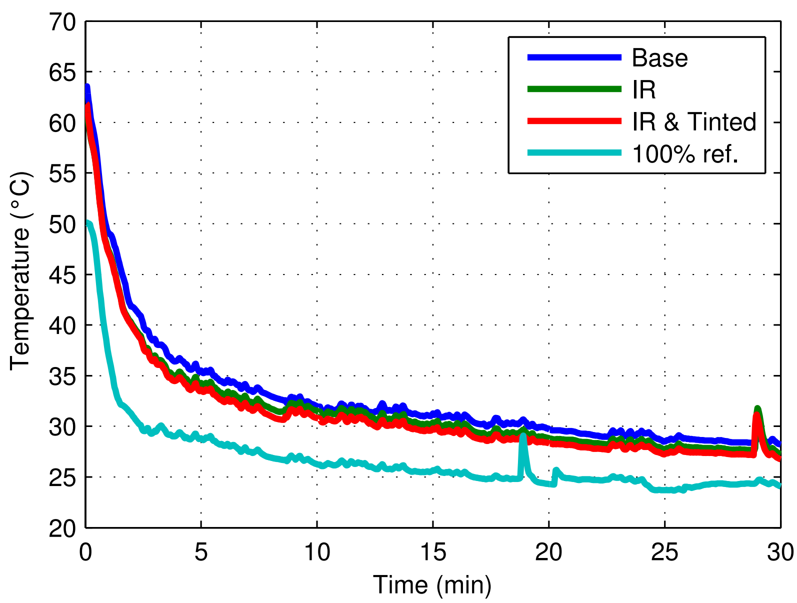

Figure 7.19 Transient average passenger compartment temperatures, glass cases

The case with 100% reflecting windows was significantly better than the other cases. The IR and IR and tinted case also performed much better than the base case, however, this can be slightly difficult to deduce from Figure 7.19. Therefore the time to reach the base case end temperature plus three degrees was calculated, see Figure 7.20.

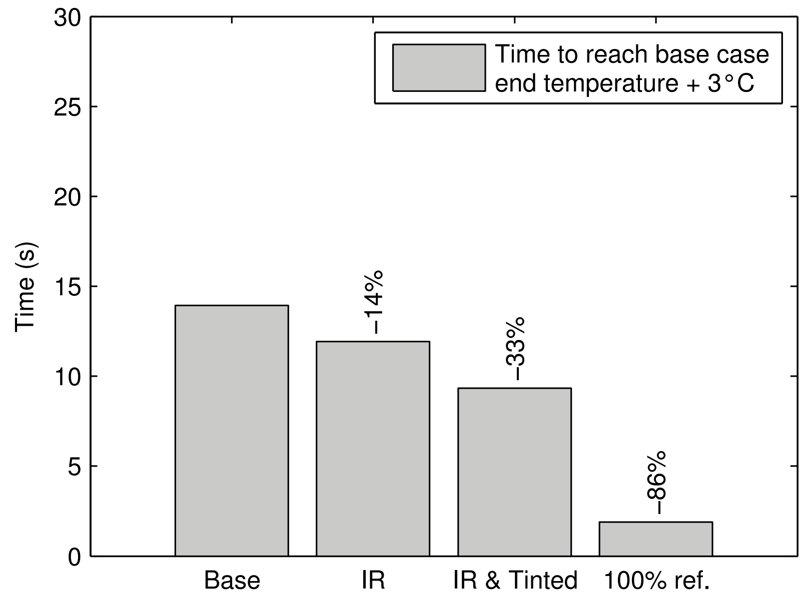

Figure 7.20 Time to reach base case end temperature plus three degrees, glass cases

In summary, the potential for a general decrease of the energy used for interior vehicle climate by implementing reflecting windows was small. However, for some markets and especially considering performance these types of windows can be very beneficial.

8 DISCUSSION AND CONCLUSION

In this chapter the strengths and limitations of the work are discussed and the main conclusions are summarized. Some additional remarks regarding the electrical and mechanical loads are also included.

8.1 Some Aspects on the Electrical and Mechanical Load

In general, the electrical and mechanical loads have not been compared to any other type of load in this thesis. However, to better get an understanding of the relative size of the energy use of the climate system some rough approximations are included in this section.

The electrical power is provided through an alternator which converts mechanical energy to electrical energy. Assume 70% alternator efficiency, $\eta_{alternator}$ and an average electrical power, $\dot{W}_{electrical}$, of 178 W, see Figure 6.2. Further, assume average compressor power, $\dot{W}_{comp}$ of 475 W, see Figure 6.3. The average mechanical load, $\dot{W}_{mech}$, of the climate system is

$$\dot{W}_{mech} = \frac{\dot{W}_{electrical}}{\eta_{alternator}} + \dot{W}_{comp} = \frac{178}{0.7} + 475 \approx 725\ W. \tag{8.1}$$

The average power for propulsion for WLTP is approximately 6 kW for a vehicle of the same size as the investigated S60. The climate system share of the total energy used is

$$\frac{725}{6000+725} \approx 11\% \tag{8.2}$$

if all other loads are neglected. The climate system clearly has an effect on the total energy use of the vehicle, i.e. fuel consumption.

In more demanding ambient conditions the impact can increase significantly. Using the loads for the cool down presented in chapter 7.3.2 the average mechanical load of the climate system was approximately 3 kW. This corresponds to 50% of the power used for propulsion and 33% of the total energy used if all other loads are neglected.

The mechanical pump in the coolant circuit was not included among the mechanical loads since its main purpose is to cool the engine and not to provide waste heat to the climate system. However, some of the used energy could be attributed to the climate system. According to Wang et al. [84] the mechanical pump uses approximately 185 W. The climate system's share of this load is unknown, however, less than 25% should be a safe assumption.

Another neglected load that is included in the climate system is seat heating. Lorenz et al. [85] uses approximately 60 W for one seat, unfortunately, the test was not comparable to the developed test cycle used in this thesis. However, the magnitude of the load can still be used for an estimation of the load. Assume that seat heating is used only in the cold test, 0°C, in the developed test cycle. Further assume that the load is 100 W throughout the test for one activated seat, i.e. only one person in the passenger compartment as in the test cycle. A load of 100 W for 30 minutes is, however, exaggerated, the seat would be very warm. The weight of the cold test is 0.2, consequently the total electrical load should be increased with at most 20 W if seat heating is included.

8.2 Limitations

There are four main limitations or shortcomings with this work: First, only one specific vehicle was used for measurements. Second, only one general type of climate system was investigated. Third, throughout the project there were problems with the compressor torque measurement and the correlation between the measurement and simulation regarding compressor power. Fourth, the passenger compartment model limited what types of energy saving measures that could be investigated. These four limitations are discussed in detail in this chapter.

A large share of the work was based on one specific vehicle. For instance, all measurements were made with the same vehicle and if this particular vehicle was different from the standard vehicle or did not operate as intended this could possibly influence all measurement results. Furthermore, the simulation results could also be affected because some models were based on complete vehicle testing. A similar weakness also applies to the measurement system, i.e. the same measurement system was used throughout all measurements. Using the same system could make the measurements sensitive to systematic errors even though the sensor accuracy was well defined.

However, there are a couple of factors which improves the conditions for achieving representative results. The vehicle was of a well-known type which had been in production for a couple of years. Many qualities, such as cool down performance, were known beforehand and were checked and compared to earlier results. Understandably, this approach did not apply to properties that had not been investigated earlier, but in general the vehicle was considered representative. The measurement equipment was installed by experienced mechanics, most according to internal measurement standards. Furthermore, the tests were performed by experienced test engineers and test facility staff in a familiar climatic wind tunnel. These conditions improves the possibility of representative results, however inaccuracies can still be present in the results, see for example the discussion on compressor power.

The second shortcoming was that only one type of climate system in one specific type of vehicle was investigated. In this case the vehicle was equipped with a 2.4 l diesel engine, a fuel operated heater and a sedan shaped passenger compartment. Furthermore, the climate system had a specific configuration of the components, it also had 14 outlets into the passenger compartment and an automatic climate control which engaged the AC-system extensively. All these features are not general for vehicles or even a specific brand of vehicles and can vary significantly. For example, different engines, different shapes of the passenger compartment, different outlet configurations and manual control of the climate system are all plausible variations. These features affect many of the energy related properties.

However, in this work the general energy use have been divided into many different categories. That is, not all results are applicable on a different type of vehicle or climate system, but some results most likely are. For instance, vehicles of the same size and configuration of the passenger compartment will probably have similar requirements on heating and cooling and this is independent of the engine type. Vehicles without a fuel operated heater supply the heat by other means, for example an electrical heater or a less efficient engine. In other words it is relatively easy to see which loads that are applicable for a different setup of the climate system. Furthermore, the used configuration of the climate system in this work is quite common in the automotive industry.

The third shortcoming was the compressor power results. In paper I the compressor power varied more than the sensor accuracy indicated for two cases with similar load. Caution with the results were advised. However, when comparing the measured compressor power with the simulated an even larger difference was revealed. In general the measured compressor power was much larger than the simulated. Reasons for the difference was discussed both in paper II and here in chapter 5.2. The discussions pointed to problems with the measurement, however uncertainties still remains. Unfortunately no new measurements were performed, first because it was deemed to be unnecessary, later due to lack of time. In hindsight, these measurements should have been prioritized in order to finally close the issue.

The main body of work have assumed that the simulated compressor power is correct, but what are the consequences if the simulation is incorrect, i.e. the simulated compressor power should be larger? The average energy use of the climate system presented in chapter 6.2.1 would be too low, i.e. the mechanical load on an average vehicle would be larger. However, comparisons of energy saving measures focused on relative decrease of compressor power, that is, an increase of the base compressor would only affect the results if it was nonlinear for the different cases. Furthermore, the compressor power was almost never compared to any other load, i.e. the absolute level was less important. To sum up the discussion; the compressor power results were acceptable although no good.

The fourth limitations was related to the passenger compartment model and its discretization. The accuracy of temperature and airflow in the individual volumes in the passenger compartment was not as accurate as the average temperature, furthermore the volumes were few and the interior radiation balance was not included. For these reasons it was not possible to evaluate thermal comfort for the passengers, besides, this was never included in the objectives of the project. Because of this limitation some energy saving measures were not possible to evaluate despite large potentials. In other words, a broad type of energy saving measures were ignored in this project. Examples of this type of measures were localized cooling or heating, heated panels and partial heating of passenger compartment as a function of occupancy. However, if the effect of one of these types of measures is known, for example, assume that with heated seats the average air temperatures could be reduced by 3°C, the effect on the total energy used for interior climate can easily be estimated.

8.3 Strengths

There are some special approaches that distinguish this project from other research in the area of energy and climate systems. The first difference was that the model included all the important effects on the energy use for vehicle interior climate. Important effects and systems such as heat pickup and air-handling unit were included. The second difference was the integration of the automatic climate control into the model, i.e. realistic operation of the system. The development and use of a test cycle representative for real-world conditions for evaluating energy saving measures was the third special feature.

There are not many models that combine the complete vehicle approach with detailed models of the climate system. The system models are frequently simpler, thus neglecting important effects. The many combinations of more detailed AC-system and passenger compartment model ignores other important effects, such as condenser cooling fan and additional electrical loads. Heat pickup and the modelled air-handling unit are two parts that stand out compared to other models.

For a realistic operation of the system the integration of the automatic climate control system was invaluable. Any condition was possible to simulate with correct airflow, air distribution, recirculation degree, evaporator set point, condenser cooling flow and temperature control. All the special functions that are included in the automatic climate control of a real vehicle was also included in the simulation.

The creation of the real-world representative test cycle with the three distinct modes of operation for the climate system enabled a realistic estimation of the average energy use. Furthermore, it allowed an unbiased evaluation of different energy saving measures, finally totally different measures could be compared.

The combination of these three special approaches generated a unique method for a realistic evaluation of the energy use for vehicle interior climate.

8.4 Main Conclusions

The main objective of this work was to understand the energy used for vehicle interior climate. This was achieved by analysing and presenting results both for specific conditions and overall average energy use. The results were divided into electrical and mechanical work and heat flows. Both measurements and simulations contributed to the understanding. The main findings can be divided into three different parts: First, how the current system works regarding electrical and mechanical work, heat flows and how to analyse the results. Second, how a simulation model can be developed. Third, what energy saving measures makes sense and why. The second part, development of the simulation model roughly answers the question of which method to use for evaluation. Furthermore, the third part, which energy saving measures are most interesting directly answers the objective "*How should future vehicles be designed with the aim of using much less energy for the interior climate compared to vehicles designed today*?"

The presented division of all the different work and heat flows have, to the author's knowledge, never been done before for an automotive climate system. Earlier works are dominated by the AC-systems influence on fuel consumption. The AC-system is the largest load on a vehicle equipped with an internal combustion engine, in other words a logical priority. However, the interaction between the different systems was not captured with this approach. The division into source and sink in this work clearly demonstrates that the interaction of the different system, especially the automatic climate control system, AC-system and the requirement of de-humidification have a large influence on energy use, mainly in intermediate conditions. One very important finding was that the required cooling and heating of the passenger compartment does not by itself decide the required energy. The operation of the system, i.e. how the required heating and cooling is supplied is very important for the average energy used for vehicle interior climate.

One interesting observation was that for the maximum performance tests the relative importance of the sink increases, i.e. in these cases the required heating and cooling flow for the passenger compartment have a larger influence on the total energy used. Perhaps the general focus on performance have also coloured the view on energy use. Other characteristics such as heat pickup and recirculation have not been thoroughly investigated earlier. These types of areas, which are closely coupled to how the system actually operates in a real-world situation, have received their proper place as important factors for the energy use in this work. In summary, all the factors influencing the energy use for vehicle interior climate, have been included in the same comparison. This makes it possible to holistically understand the complete system, maybe for the first time.

There were not any realistic alternatives for better understanding the energy use and evaluate new energy saving measures than a simulation model. However, the level of detail for all included systems was generally a difficult balance between accuracy and possible modifications on one hand and simulation speed, robustness and complexity on the other hand. The chosen level was good enough although significant improvements are probably possible, especially regarding computation time. That is, with a small or no reduction in accuracy some model can be significantly faster if more time would be devoted to their design. In this project it was more important with an accurate model rather than a fully predictive model, therefore some models or part of models were based on complete vehicle testing. For future work, especially in the industry, a model without the need for complete vehicle tests for calibration would be a very powerful tool. In this project simulations were an invaluable tool and the development of this method was successful, even when the compressor power issue is included in the evaluation.

What are the conclusions regarding which energy saving measures that should be implement? This is very dependent on how the system is structured and how it's controlled. For the system evaluated in this work there are three specific measures that should have priority. As the first measure deactivate the AC-system below 15°C, or in a more general expression: Only use the AC-system for cooling and de-humidification when actually needed. This requires an improved control of the humidity levels to avoid condensation on the windows. However, it can probably be done without any changes in hardware and the average compressor load can be decreased by 27% from this single measure, a significant improvement. Note that many systems, in particular manual systems, probably already are used in this way. The second measure is to change the blower control unit, this single measure would decrease the average electrical load by approximately 25%, furthermore it will also reduce the cooling load thanks to reduced heat pickup. The third step would be to decrease the amount of reheat even further by increasing the maximum allowed evaporator temperature. However, this can be very challenging due to issues with noise, smell and controllability. An alternative solution could be to implement a controllable bypass of the evaporator. Without increasing the maximum evaporator temperature many measures that decreases the need for cooling will have quite small effect, thus this is a prerequisite to achieve significant reduction in warm climate.

The specific requirements influences the next choice of energy saving measures. If there is a lack of available heat, part-recirculation would probably be an interesting measure. However, implementing this measure also requires the need to closely watch the humidity levels to avoid condensation on windows. If the focus is on mechanical loads then measures such as reducing the heat pickup can be beneficial but also measures in the sink category are viable because the main limitations of the system have been corrected. However, to really take advantage of the energy saving measures in the sink category the airflow must be reduced or a much more efficient recovery of energy in the air is needed. This is already achieved in current vehicles in warm conditions by recirculation. There are many challenges with reducing the airflow, for instance the requirements on removal of humidity and pollutants from the passenger compartment must still be satisfied.

In summary there is a large potential to reduce the energy used for vehicle interior climate, initial measures must be directed on the source category otherwise the effect of the measures will be below their potential.

9 FUTURE WORK

In this chapter different suggestions for future work are briefly discussed.

- With the current passenger compartment model energy saving measures focused on comfort were not possible to investigate, for example localized heating and cooling. However, the potential of this type of measures could be large and the measures should be compared with the previously investigated measures. The model developed in this thesis could be used for analysing energy saving if a decrease, or increase, of the passenger compartment temperature is provided. The temperature increase or decrease have to come from another model, such as 3D CFD with a thermal comfort model, or complete vehicle tests.
- Many energy saving measures did not have a large effect because the influence of really cold or warm conditions was small. However, there are markets which are much warmer or much colder than the average of the developed test cycle. In these markets substantial energy savings are possible. The energy saving measures should therefore be investigated with a more appropriate test cycle or with different weights on the tests in the developed test cycle for these markets.
- There are many interactions between the engine and climate system. With the developed model these interactions are possible to investigate. For example, increased load on the engine due to the AC-system can heat up the engine faster, consequently increase the efficiency. The increased efficiency can offset some of the increased fuel consumption.
- Investigate how the climate system operates in current and future proposed certifications cycles such as SC03, AC17 and the European Union's "MAC Test Procedure".
- Convert the developed model from an ICE vehicle to an EV and investigate current energy use and future energy saving measures for that type of drivetrain.
- The current model required complete vehicle testing for some of the investigated systems and attributes, primarily the passenger compartment, the total pressure drop of the air-handling unit and the heat pickup of the incoming air. For early development, models that are available before the physical vehicle are very useful. These predictive models needs to be developed.
- Investigate and verify that the mechanical load can be estimated with simulation, that is, model and measure exactly the same AC-system in a complete vehicle.

- The operation of the climate system in intermediate conditions has the largest effect on the current energy use and obscures other, future types of energy saving measures. Consequently, the energy savings should be re-evaluated with the most obvious ones implemented, such as deactivated AC-system below 15°C, increased maximum allowed evaporator temperature and replaced blower control.
- The airflow has a large influence on the energy used for vehicle interior climate, as a result reduced airflow should be investigated. Focus of the investigation have to be on the consequences for humidity, de-misting, de-icing, thermal comfort and air quality.
- The condenser airflow was simplified in the model, that is, no temperature increase was included when air was recirculated from the engine compartment during idling. Furthermore, no heat from other components affected the condenser air temperature, it was always supplied with ambient tempered air. A more advanced model could include these factors.
- A simple method of measuring the airflow during vehicle use.
- The mechanical efficiency of the compressor was relatively low, especially for the small loads that were fairly common in the developed test cycle. This should be investigated further.

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