Modelling and Control of the AC-system in Heavy Duty Vehicles

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Modelling and Control of the AC-system in Heavy Duty Vehicles

Master thesis performed at Vehicular systems
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by

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Sammanfattning
Abstract

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A model of the AC-system was developed in order for an efficient controller to be designed. The model was designed to comprise of all the basic behaviours that the AC-system has, rather than to be an exact model of the system. As the AC-system showed to be very complex, a number of limitations in the model had to be made.
The AC-system has two temperature sensors and is actuated by turning the AC-compressor on or off. Two different control strategies were tested for the control of the AC-compressor. The first was to use a controller to directly control the compressor clutch and the second one utilised a pulswidth modulated control structure were the controller stated the pulswidth to be used.
Both control structures were implemented in the computer model, the AC-rig and in a truck in a climate chamber. Both control strategies showed to fulfil the demands on the system in the somewhat idealistic circumstances during which they were tested.

Nyckelord
Keywords AC-systems, HEVAC, Temperature control, Heavy duty vehicle

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Södertälje, June 2001
Magnus Eriksson

Some notations used

In general
Q – Heat transfer rate
G(s) – Model transfer function
e(s) – The error signal
u(s) – The control signal
y(s) – The output signal
C_p *– Specific heat at constant pressure*

Subscripts referring to signals in the system
Mixair temperature – The temperature of the air leaving the cooling package
Evaporator temperature – The temperature on the coldest place on the evaporator
fh – airflow across the evaporator
rh – Relative humidity in the air

Abbreviations

AC – Air Condition
CFD – Computational Fluid Dynamics
ECU – Electronic Control Unit
EXV – Electronic eXpansion Valve
FEM – Finite Element Method
TXV – Thermostatic eXpansion Valve
HEVAC – HEater Ventilation Air Condition
PWM – Pulse Width Modulator

Contents

1. Introduction

Scania is one of Europe's leading manufacturers of heavy duty vehicles. Catering for the world's most demanding customers has made Scania one of the worlds leading manufacturers of heavy trucks, buses and engines.

In this thesis, a model of the trucks AC-system is developed and two different control strategies for AC-control is evaluated. The work has been performed at the section for Cab Testing and Evaluation and the Driver interface section at Scania technical centre.

1.1 Objectives

The purpose of this thesis is to investigate how the AC-system in Scanias trucks works and investigate different control strategies. Especially the case when low cooling capacity is needed from the AC-system is to be investigated.

1.2 Background

Traditionally the AC-system has only been used to provide maximum capacity when engaged but with increasing demands for driver comfort, the wish came to be able to run the AC-system at a lower capacity. One such case can be when the outside temperature is +25 °C and you want to cool the air to 20 °C.

The ECU:s in the automotive industry today have more computer capacity and additional sensors that make it interesting to investigate if more advanced control concept can be employed.

Some prior investigation showed for the possibility to accommodate this without having to make any radical changes to the system layout. This thesis will further explore these possibilities and give an insight into the problems that can be countered.

1.3 History

The thermodynamic basis for AC began in the time of France's Napoleon when *Nicolas Leonard Sadi Carnot (1796-1832)* wrote *Reflections on the motive power of fire* in which he formulated the basic principals governing conversion of heat work. By reversing Carnot's engine which uses heat supplied as an energy source and delivers mechanical work is supplied as the energy output, mechanical work is supplied as the energy source and heat is transferred from one energy level to another- the basis of refrigeration and air conditioning.

Almost all vehicles manufactured in the world today have air conditioning as standard or as an option. This includes off highway vehicles, passenger cars, trucks, busses, farm equipment etc. A vehicle without AC, today, is similar to a vehicle without a heater during the thirties when heater was still an option.

Before AC, there were a number of so-called comfort devices to cool a driver in hot climates. A popular item was an evaporative cooler which was filled with water and mounted outside the driver side window. Ram air flowing through the unit, at high speeds, was cooled to the wet bulb temperature and ducted into the vehicle. A more expensive unit was equipped with a fan that was wired directly to the battery. These units were useless in areas of high humidity and even added to passenger discomfort.

AC was first introduced in the late thirties by Packard, Chrysler and Cadillac. The units where supplied to the automotive industry by the Bishop and Babcock Company.

1.4 Thesis outline

The work done during this thesis and the concepts contained in the thesis are described in the following chapters.

Chapter 2, **Theory.** An overview of the basic theory for heat transfer, control theory and modelbuilding.

Chapter 3, The AC system. Description of AC-systems in general and a detailed description of the AC-system in Scanias trucks in particular.

Chapter 4, The model. Here the developed model and the simulation results are presented.

Chapter 5, Control algorithm.

Chapter 6, Discussion and conclusions. A short discussion of the thesis and some ideas for future work is presented in chapter 6.

1.5 Method

The plan for the work was as follows:

1. Create a Simulink model of the system using measurements from the AC test rig.
2. Develop a control algorithm in Simulink.
3. Implement and test the control system in the test environment.
4. Implement the developed control algorithm in the truck.
5. Test the final system in a truck in the climate chamber.

The emphasises on making as much work as possible outside of the truck are due to a number of reasons. One of the more important reasons not to do initial measurements in truck is the difficulty to measure the interesting quantities when the AC system in mounted in the truck. Further a much more precise control of the surrounding conditions can be obtained when mounting the AC system in the test rig. The biggest drawback with this approach is that it will be very hard to get a comparable system performance in some sense. The air flow distribution in the rig for instance will never be the same as in the ductwork in the truck.

2. Theory

In this chapter some theoretical background to the area will be given. The basic principles for heat transfer, control theory, model building and AC-systems are covered.

2.1 Heat transfer

Heat is generally transferred through four different processes - conduction, convection, radiation and masstransport.

2.1.1 Conduction

Heat transfer by conduction is the way heat is transferred in solids. The energy is spread by a diffusion process where the electrons and the lattice vibrations in solids are excited by frequent collisions. For metals the electron part is dominant and for dielectric matter (insulators) the lattice vibrations is dominant as the electrons are much harder bound.

2.1.2 Convection

Convection is the process that is responsible for the heat exchange between a surface and a surrounding fluid, in this case air. When the air molecules come in contact with the surface they will absorb or emit heat depending on the temperature difference between the air and the surface.

There are two different types of convection depending on if the fluid as a whole have any velocity components and by that way are forced on the surface, or if the fluid comes in contact with the surface in a more natural way i.e. diffusion. The two types are called Forced convection and Natural convection.

Natural convection
The thin layer of air that is in contact with a solid's surface will have the same temperature as the surface. If there is a temperature difference between the surface and the air temperature a small distance away from the surface, there will by the ideal gas law also be a density difference between the air layers. This density difference results by diffusion in a net transport of surface air molecules away from the surface. Their place is then taken by other air molecules that will interact with the surface and a heat transport takes place.

Forced convection
The convection is said to be forced when the surrounding fluid flows along the surface. To get a picture of heat transfer process by forced convection we have to study the fluid flow close to the surface. The viscosity of the fluid makes the flow close to the surface always to be laminar, even when turbulence appears further out.

2.1.3 Radiation

In contrast to the mechanisms of conduction and convection, where energy transfer through a material medium is involved, heat may also be transferred through regions where perfect vacuum exists. The mechanism in this case is electromagnetic radiation. The surface electrons in the material that is being subjected to the electromagnetic radiation absorb part of the radiation and then emits radiation as they are spontaneously deexcitated.

2.2 Control Theory

The most commonly used controller is the PID controller. This controller will provide sufficient result in many simple systems and is always a good "first shoot" when designing regulators. The formula for the PID controller is displayed below.

$$u(t) = K\left(e(t) + \frac{1}{pT_i}\int e(t)d\tau + T_d\frac{de(t)}{dt}\right)$$

2.2.1 Digital implementation of the PID algorithm

To be able to implement the PID algorithm in a computer system it must be time discrete. One easy way of doing this is to use the approximations below.

$$\int_{t_0}^{t} e(\tau) \approx T_s\left(e(t) + e(t-T_S) + e(t-2T_S) + ...\right)$$

$$\frac{de(t)}{dt} \approx \frac{e(t) - e(t-T_S)}{T_S}$$

There are also other more elaborate schemes of doing the discretisation for instance *Euler backwards* or *Tustins rule*.

It proves necessary to modify the simple algorithm above in some ways. One important modification is the one that prevents reset windup. This phenomenon occurs because real systems always has a limited control signal and that fact lets the I-part of the regulator sum up to very high values, slowing down the response of the regulator. There are several methods to prevent reset windup, where three main groups can be identified.

- i) Differential implementation
- ii) Permissive integration
- iii) Adjustment of the I-part

The time discrete PID regulator above can be written as follows:

$$I_n = I_{n-1} + K\frac{T_S}{T_i}e_n$$

$$v_n = Ke_n + I_n + K\frac{T_d}{T_S}(e_n - e_{n-1})$$

$$u_n = \begin{cases} u_{\max} & \text{If } v_n > u_{\max} \\ v_n & \text{If } u_{\min} \le v_n \le u_{\max} \\ u_{\min} & \text{If } v_n < u_{\min} \end{cases}$$

Differential implementation

One way is to implement the regulator on differential form by forming Δv_n as the difference in v between two samples. The algorithm will look like:

$$\Delta v_n = K\left(e_n + e_{n-1} + \frac{T_S}{T_i} e_n + \frac{T_d}{T_S}(e_n - 2e_{n-1} + e_{n-2}) \right)$$

$$u_n = \begin{cases} u_{\max} & \text{If } u_{n-1} + \Delta v_n > u_{\max} \\ u_{n-1} + \Delta v_n & \text{If } u_{\min} \le u_{n-1} + \Delta v_n \le u_{\max} u_n \\ u_{\min} & \text{If } u_{n-1} + \Delta v_n < u_{\min} \end{cases}$$

Permissive integration

In this case the I-part of the regulator is shut of if the control error is to great and by that mean prevents the reset windup. This gives the following code:

$$\text{IF (CONDITION) THEN } I_n = I_{n-1} + K\frac{T_S}{T_i}$$

$$v_n = Ke_n + I_n + K\frac{T_d}{T_S}(e_n - e_{n-1})$$

$$u_n = \begin{cases} u_{\max} & \text{If } v_n > u_{\max} \\ v_n & \text{If } u_{\min} \le v_n \le u_{\max} \\ u_{\min} & \text{If } v_n < u_{\min} \end{cases}$$

Adjustment of the I-part

This technique is similar to permissive integration but instead of completely shutting down the I-part one tries to adjust it to the situation. This is done by updating the I-part as follows when the control signal is saturated.

$$I_n := I_n + \frac{T_S}{T_t}(u_n - v_n)$$

The complete algorithm becomes:

$$I_n = I_{n-1} + K\frac{T_S}{T_i} e_n$$

$$v_n = Ke_n + I_n + K\frac{T_d}{T_S}(e_n - e_{n-1})$$

$$u_n = \begin{cases} u_{\max} & \text{If } v_n > u_{\max} \\ v_n & \text{If } u_{\min} \le v_n \le u_{\max} \\ u_{\min} & \text{If } v_n < u_{\min} \end{cases}$$

$$I_n := I_n + \frac{T_S}{T}(u_n - v_n)$$

2.3 Model building

There are three main ways of building models of a system:

- Physical modelling (white box).
- Grey box (semi-physical modelling).
- Black box (non-parametric estimation).

2.3.1 Physical modelling

A physical model is solely based on physical connections and usually doesn’t involve any parametric estimation.

2.3.2 Grey box modelling

Grey box modelling lies between the black box and physical modelling. Here simple physical relationships are used when they are known and identification methods are used in order to estimate the other properties of the system.

2.3.3 Black box modelling

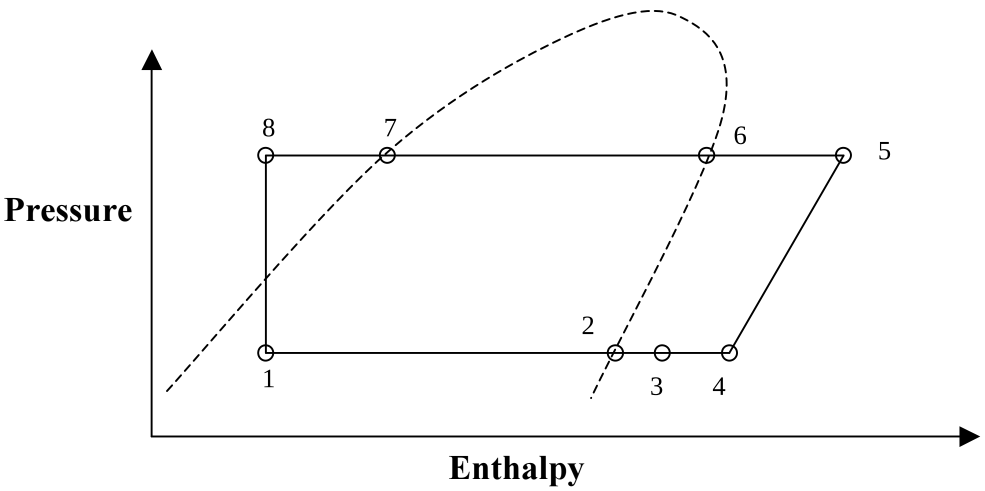
Here a model is build with no regard to the underlying physics. Typically only the orders in a linear model are determined and the parameters are then estimated using some numerical method.

3. Description of the AC system

The general layout of automotive AC-systems doesn't differ very much from system to system. Here a description of the most common components and principles are described. More elaborate descriptions and principles can be found in Kargilis [7].

3.1 The function of the AC-system

The functional description of the AC-system is described by the Carnot diagram for the thermodynamic process. In figure 1 the phase diagram for the cooling agent (dotted line) is plotted over the carnot diagram for the process, note that figure 1 only is a schematic sketch. The actual carnot diagram has less ideal appearance. In the region under the dotted line most of the cooling agent is in the liquid phase and over the line it is mostly gas phase. In a real phase diagram the actual percentage between liquid and gas phase can be determined.



1. Evaporator inlet
2. Saturation
3. Evaporator Outlet
4. Compressor Inlet
5. Compressor Outlet
6. Condenser Inlet
7. Condenser Outlet
8. Expansion Valve Inlet

Figure 1. Carnot diagram for the process plotted on the phase diagram for the cooling agent.

3.1.1 Compression, discharge

Superheated refrigerant vapor enters the compressor, isentropically at low pressure and temperature and is compressed to a high temperature and pressure. The difference in enthalpy between the high and low-pressure gas is referred to as *compressor work*.

3.1.2 Condensation

A compressor discharge line carries the high-pressure gas to the condenser where it is condensed in three phases: desuperheating, condensing and subcooling. The condensing process is done at constant pressure and temperature. The difference between the enthalpy from desuperheating to subcooling is a measure of the heat content released to the ambient air, which cools the condenser via a fan.

3.1.3 Receiver-drier

The subcooled liquid refrigerant enters a receiver-drier to remove any moisture and to separate gas bubbles from the liquid. The receiver-drier also acts as a reservoir for excess liquid refrigerant.

3.1.4 Flow control

A liquid line transfers the refrigerant to a flow control device (for example an expansion valve) where it is expanded (adiabatic expansion) to a low pressure, low temperature mixture of liquid and gas.

3.1.5 Evaporation

The low temperature and pressure liquid refrigerant enters the evaporator where it is evaporated (boiled) by ambient or recirculated air from the vehicle interior. The air is blown across the evaporator by a blower and associated ducting. The evaporation process is also at constant pressure and temperature. The difference in enthalpy between the liquid and gas refrigerant is a measure of the amount of heat (sensible and latent) that is removed from the air entering the evaporator.
The evaporator also acts as a dehumidifier by condensing the water vapour (latent heat) in the air flowing across the outside surface of the evaporator.

3.1.6 Compression, suction

The low temperature and pressure superheated refrigerant gas is carried to the suction side of the compressor through a suction line.

3.2 Components

There are a number of different types of automotive AC-systems. In general there are four major active components that is present, regardless of the type of system:

- Compressor.
- Evaporator.
- Condenser.
- Cooling agent.

These components must always be present in order for the system to function. The typical AC system in heavy duty trucks also include the following components:

- Flow control device.
- Receiver drier or accumulator drier.
- Pressure guards.
- Freeze protection.

3.2.1 Compressor

There are two main groups of compressors used for automotive AC systems, fixed and variable displacement compressors. The variable displacement compressor has the ability to match the AC demand under all conditions without cycling by adjusting the piston travel on demand. This gives the designer the ability to regulate the actual cooling effect that the system can perform. The variable displacement compressors are however not as durable as the fixed displacement type which makes them hard to use in heavy vehicle circumstances where durability demands are much higher than for passenger cars. Therefor the most used compressor in trucks is of the fixed displacement type. The fixed displacement compressors are often equipped with a magnetic clutch that enables the compressor to be engaged/disengaged. The cycling compressor allows for fuel-efficient operation and is the most widely used approach in heavy vehicles.

3.2.2 Evaporator

There are four basic types of evaporators:

- Plate FIN
- Tube and Fin
- Serpentine
- Heater Tube and Spacer

They all exhibits different properties in terms of size, airside pressure drop efficiency, water carryover etc. They are normally made of aluminium, in rare occations they are made with copper tubes. The most important evaporator factors according to [7] are:

1. Low airside pressure drop.
2. Good water drainage to prevent core icing and odour problem.
3. Uniform temperature distribution.

3.2.3 Condenser

The condenser is often part of the cooling module which consists of a radiator, condenser, intercooler, oil cooler and fan shroud. All these components are packed to fit within a specified area in the front of the vehicle. Because all these components influence each other in terms of airflow and temperature they must be designed so the operation of one unit doesn’t hinder the function of the others.

3.2.4 Cooling agent

The cooling agent is the media in which the thermal exchange takes place. Today all AC-systems contain a cooling agent without any negative effect on the ozon layer. There are no freons in the gas. The most commonly used cooling agent today is named R-134a (1,1,1,2 Tetrafluoroethane). There are some advanced concepts that experiment with other cooling agents such as carbon dioxide and ammoniac. In the cooling agent there is also a certain percentage of compressor oil that is needed to ensure sufficient lubrication of the AC-compressor. The compressor oil must be matched to the cooling agent that is used in order for it to function optimally. There are regulations regarding how to fill an AC-system with cooling agent and this operation should usually be done by a certified person.

3.2.5 Flow control device

There are a number of different flow control devices used in automotive AC-systems the two most common types are the thermostatic expansion valve (TXV) and the orifice tube.

Orifice Tube

The orifice tube is a calibrated tube that meters the liquid refrigerant flow to satisfy the evaporator demand. A typical orifice tube circuit is shown in figure 2. The orifice tube is always used in conjunction with an accumulator drier. The accumulator's function is to act as a reservoir by containing liquid that flows from the evaporator and to release only gas to the compressor. It also contains a desiccant bag that absorbs moisture that may be in the refrigerant. The orifice tube is not that common today as it was in the past, as demands for system performance got higher the orifice tube system didn't measure up.

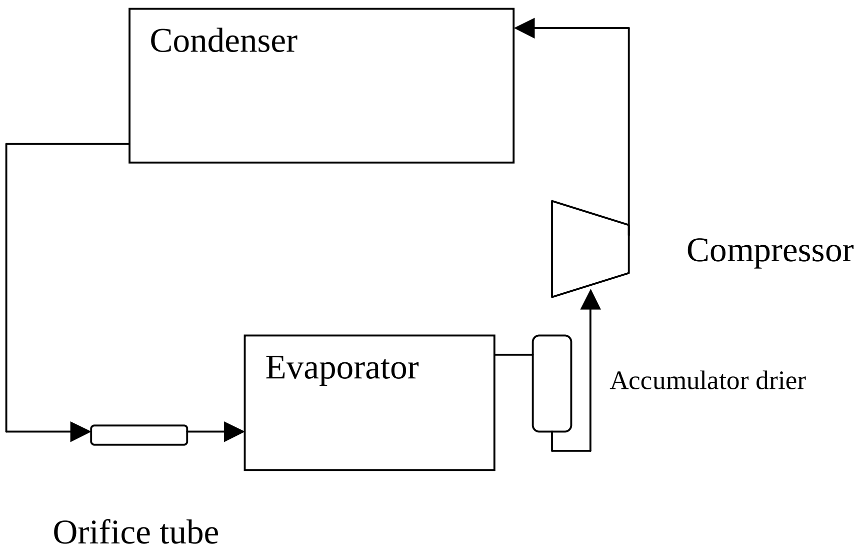


Figure 2. Typical orifice system.

Thermostatic expansion valve

The expansion valve is a modulating device that controls the flow in response to the superheat of the refrigerant gas leaving the evaporator. The expansion valve is a more complex device then the orifice tube but it does offer a wider range of control. The expansion valve controls the flow of the liquid refrigerant entering the evaporator relative to the amount of superheat in the refrigerant gas leaving the evaporator. Since the expansion valve operates in response to superheat, a part of the evaporator must be used to superheat the refrigerant gas. A cutthrough sketch of an expansion valve is shown in figure 4 and a sketch of a expansion valve system is shown in figure 3.

The expansion valve acts as a throttling device by expanding, adiabatically, the high pressure, high temperature liquid refrigerant from the condenser through a controlled opening into a low pressure, low temperature mixture of liquid and gas of about 5 to 30% quality.

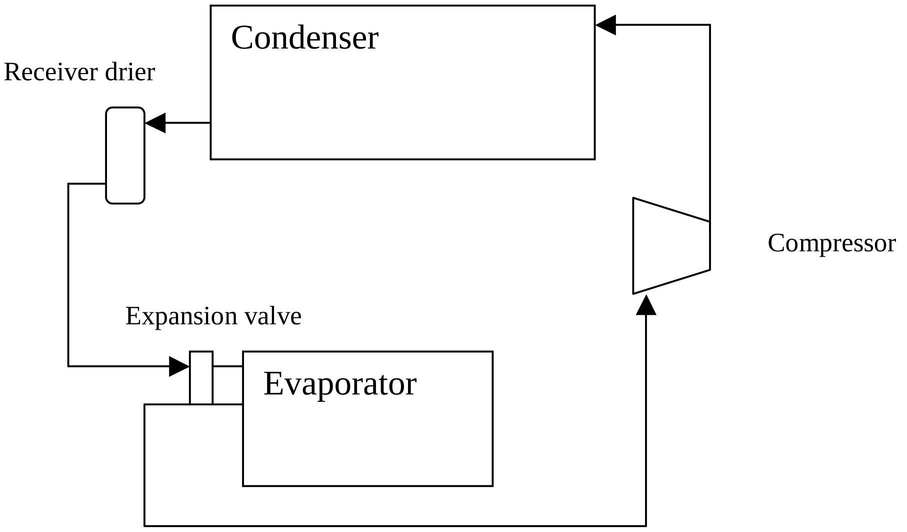


Figure 3. Typical expansion valve system.

A thermostatic bulb sense the evaporator outlet tube temperature (externally equalised) or the evaporator inlet tube temperature (internally equalised). The medium in the bulb (refrigerant) exerts a pressure on the diaphragm. An increase in temperature (heat load) causes the diaphragm to open and thereby increasing refrigerant flow. A decrease in temperature causes the diaphragm to close and thereby decreasing the flow.

The type of expansion valve used in automotive applications is generally externally equalised which has more accurate control of evaporator superheat than the internally equalised valve. This is particularly important for evaporators having a high refrigerant pressure drop.

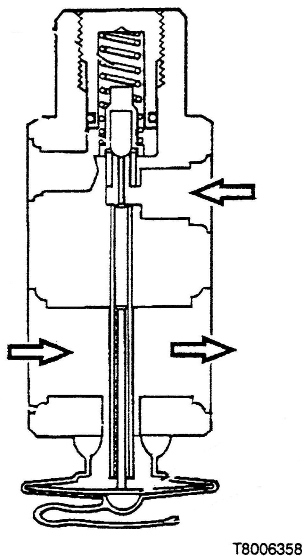


Figure 4. Thermostatic expansion valve.

3.2.5 Pressure guard

As many as four pressure-switches could be placed throughout the AC system, to protect the AC system from catastrophic failure and to protect the most expensive component, the compressor.

Figure 5. shows a typical TXV system with pressures switches.

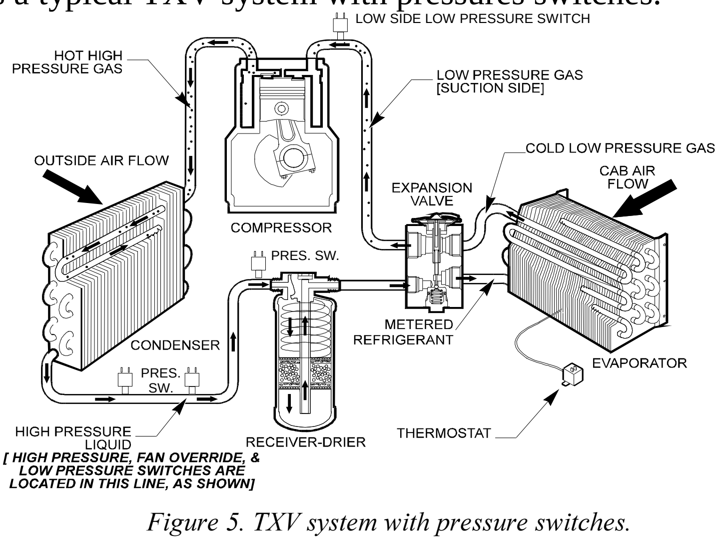


Figure 5. TXV system with pressure switches.

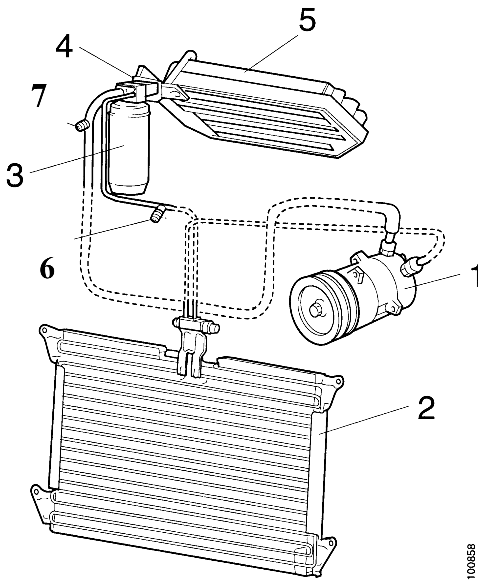
The high-pressure switch disengages the compressor clutch if the systems operating pressures exceeds its setpoint typically 2.4 MPa. The low-pressure switch keeps the compressor-clutch from engaging if the system pressure is too low due to ambient temperature or lack of refrigerant. The fan override pressure switch engages the engine fan or condenser fan to increase the airflow across the condenser to reduce the system’s operating pressure. Much less common is the fourth pressure switch, the low side, low-pressure switch. Its function is to disengage the compressor anytime the compressor suction pressures get too low. This protects the compressor against damage due to lack of lubrication from insufficient circulation of refrigerant and oil.

3.2.6 Freeze protection

When the evaporator cools humid air, moisture is condensed on the evaporator. If the evaporator temperature is below 0 °C the water on the evaporator will freeze to ice. This can result in clogging up the evaporator with ice and hereby increase the pressure drop over the evaporator. In order to prevent this there must be some kind of control of the evaporator temperature, this is usually done by a thermostat. In system where a ECU controls the compressor one tends to use a temperature sensor that is placed at the coldest place on the evaporator.

3.3 Scanias HEVAC system

The AC-system in Scanias truck is a typical TXV system with a magnetic compressor clutch. In figure 6 a sketch of the system is shown.



1. Compressor
2. Condenser
3. Reciever drier
4. Expansion valve
5. Evaporator
6. High-pressure service valve
7. Low-pressure service valve

Figure 6. Sketch of Scanias AC-system.

A typical step response curve from the system is shown in figure 7. The fluctuations that can be seen at the bottom of the curve are caused by the expansion valve which pressure limiting gives this result on the evaporator temperature.

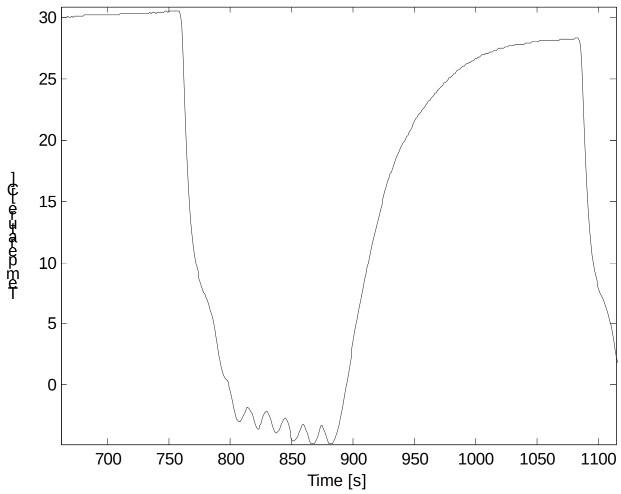


Figure 7. Typical step response curve from the AC-system.

3.3.1 Actuators

The only actuator used by the AC control is the magnetic clutch on the compressor. In this theses the compressor clutch was controlled by the different regulators, see sec. 5.

3.3.2 Sensors

There are two temperature sensors, the evaporator sensor and the mixair sensor. The evaporator sensor is used as a freeze protection for the evaporator and it is situated in the upper left corner of the evaporator. The mixair sensor is used to control the temperature of the air exiting the heating package. It is situated in the air duct according to figure 8.

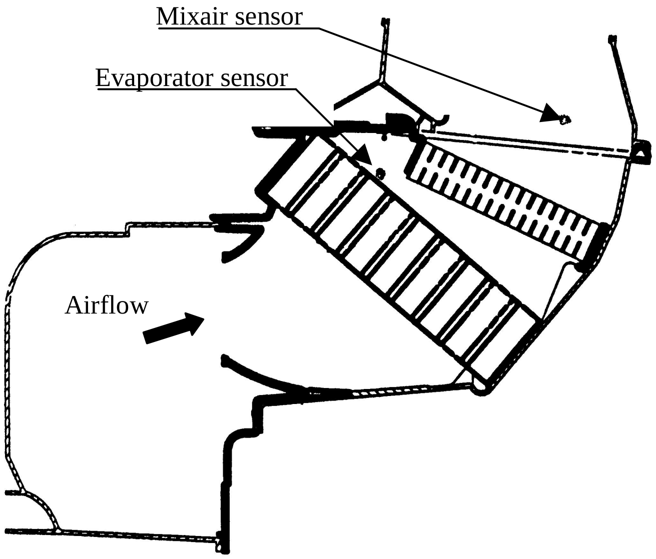


Figure 8. Sensor placement in the air channel.

This method of measuring the temperature of the air leaving the heating package has some obvious limitations. The fact that the temperature is measured only in one point, states very high demands on the placement of the sensor. Ideally one would like to measure the temperature distribution and the flow distribution along the cross-section of the air channel. With this kind of information it would be possible to determine the heating or cooling work that is added to the cab environment. To replace all this necessary information with the temperature in one point can be a very coarse approximation. This makes it necessary to verify that the sensor reading really reflects the actual situation. The temperature and flow distributions in the air channel are usually very complex and hard to measure and simulate.
Both temperature sensors are of the same type and exhibit the same performance.

4. Modelling the system

In order to effectively develop a control algorithm it was decided that a computer model of the system should be developed. This decision was made mainly for two reasons:

1) Creating a model of the system is a good way of gathering knowledge about the system. The work with the building of the model provides a structured way of gathering the information that is needed for a successful control design.
2) As this is a rather slow and complicated system it is somewhat difficult and time consuming to real tests on the system. If most of trial and error work can be performed on the computer model, much time can be saved.

As stated in section 2.3 there are three different main approaches to system modelling. After some studies and measurements the semiphysical approach seemed like the most appropriate, this because of the high complexity that the system exhibits. This approach was selected some of the system properties could be accurately decided from a theoretically stand point, for instance the over maximum cooling capacity. Other properties proved to be very complicated to model from a theoretical background and were thus approximated. A good example of this is the transient behaviour of the system was a complete model would crave a total and detailed model of all the parts in the system.

Some trials on the black box approach where tried and it proved hard to get models that responded well to big variations in some of the parameters. The black box models worked very well when applied to the same conditions as they where made from. However the systems performance changes very much with the change of many of the in signals and it would have taken a large set of black box models to cover the desired range in the signals. This approach would also require a large set of measurements from the system which could be rather time consuming to gather.

4.1 Model parameters

The first step in building the model was to identify the signals that were of significance to the system performance, some of the most important are shown in table 1.

Ambient temperature
Relative air humidity
Total airflow across evaporator
Airflow distribution across evaporator
Airflow across condenser
Compressor rpm
Expansion valve setting
Amount of refrigerant in the system
Amount of compressor oil in the refrigerant
Amount of water in the refrigerant

Table 1. Some of the variables that affects the behaviour of the AC-system.

These parameters are all important to the total system performance, there are also other parameters that exhibits minor contributions to the system. The parameters in table 2 are the one that usually is deemed to influence the system significantly, see Kargilis [7]. It is possible to measure some of these parameters in the AC-rig but many are very difficult to measure and are very likely to differ from one system to another.
One of the more tricky issues here amount of cooling agent in the system. This factor is of course very influential on the system behaviour and the amount of cooling agent that is put into an AC-system is always measured. But the thing is that no AC-system is hermetically sealed and so over time there is always some cooling agent leakage. And since there is always compressor oil present in the cooling agent a certain leakage of compressor oil is to be expected. The cooling agent and the compressor oil have rather different dispersion rates through a leaking system because they have quite different chemical compositions. It is easy to understand that these properties are very hard to from a physical point of view model, they are also very difficult to measure.

4.2 Distinctions

The AC-rig is also not comparable to the real system mounted in a truck at some points. Therefor the choice was made to train a model only from the data that could be measured in the rig. And to limit the modelling work so that it wouldn’t model any behaviour specific to the AC-rig. This led to the distinction to only model the system behaviour as far as the evaporator temperature sensor. The temperature that the evaporator sensor measures is not by far as sensitive as the mixair temperature. This is because the evaporator sensor is situated very closely to the evaporator and thus measuring the temperature that the evaporator has rather than the temperature of the air. The sensor is also situated were the cooling capacity of the evaporator is as highest, this makes it somewhat more indifferent to small different in airflow and temperature distribution.

It is rather hard to do accurate comparable measurements between the system in the AC-rig and the one that is in a truck to really verify this for all situations. Comparing measurements was made in this way that a truck was put in the climate camber and all the signals that could measured in the AC-rig was measured or approximated. The as system was then run and evaporator sensor temperature was noted. The same condition as in the climate chamber was generated in the AC-rig and the evaporator temperature was measured here. This comparison showed us that the evaporator temperature behaviour in the AC-rig was comparable to the one in the real truck. Since it was rather unlikely that these results could be employed on the mixair temperature any comparing measurements between the mixair temperatures wasn't made.

This led to decision to only try to make a model as far as the evaporator temperature. This compromise with the modelling was chosen because the main objective with the theses wasn't the modelling of the system but to investigate the control strategies. With these distinctions a reasonable compromise between the wish for a comprehensive model of the system and to fulfil the objective for the theses was reached. The fact that only the evaporator temperature was modelled didn't really affect the design of the controller since a cascade approach was chosen, see 5.

There have been attempts to make more comprehensive models of AC-systems, they often focus on one specific problem, which they try to model very accurately, see [5] and [6]. There are many FEM and CFD models that studies flow and temperature distributions in various parts of the system these models are however often limited to the steady state performance of the system.

4.3 The model

The model was split into two main parts one that modelled the cooling capacity and one that modelled the transient behaviour of the system.

4.3.1 Modelling the cooling capacity

The overall energy balance for the AC-system can be written as:

$$Q_{Condenser} = Q_{Evaporator} + W_{Compressor}$$

The compressor work is usually determined experimentally. The result is presented in a compressor performance curve, see figure 9.

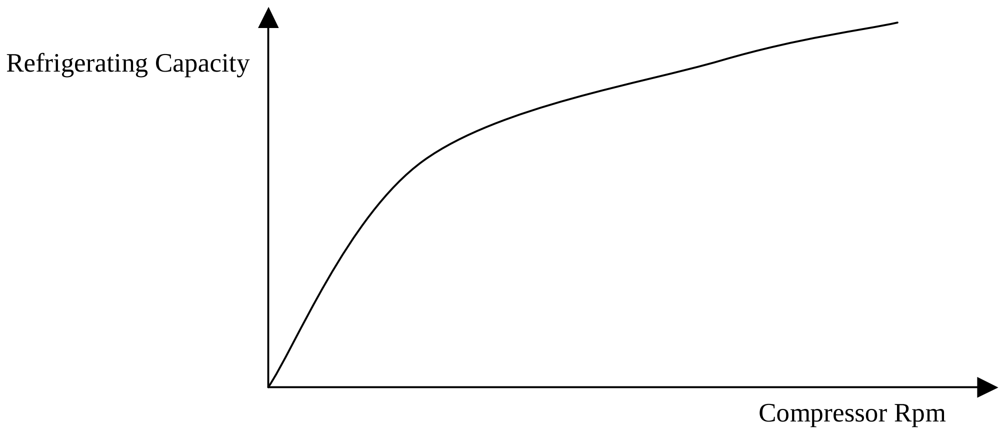


Figure 9 Compressor performance curve.

The evaporator duty consists of the dry air sensible heat and the sensible and latent heat of condensation of the moisture in the air. The sensible heat and latent heat are absorbed by the refrigerant flowing in the evaporator.

$$Q_{Evaporator} = Q_{sensible} + Q_{latent}$$

The dry air sensible heat is determined from the thermodynamics for forced convection.

$$Q_{sensible} = E \ (\Delta m \cdot Cp)_{air} (T_f - T_a)$$
$$E = 1 - e^{-NTU}$$
$$NTU = \frac{U \cdot \Delta A_i}{(\Delta m \cdot Cp)_{air}}$$

T_f, T_a: Temperature before and after evaporator
U: Overall heat transfer coefficient
ΔA_i: Effective area

Where E is the local effectiveness and Δm is the mass flow rate of the air flowing past the evaporator. The calculation of NTU is dependent on the geometry of the evaporator as well as number of other things. For a complete explanation of these calculations see Gurasan [8]. For our purposes the evaporator manufacturer had supplied us with performance curves for the evaporator. And since this was the evaporator that was independent to use there would be no gain be performing these kind of analyses further. So in the model the evaporator efficiency is determined from the curves supplied by the evaporator manufacturer.

The latent heat of condensation of the moisture in the air and the amount of condensate is traditionally determined using a psycrometric chart (mollier diagram). This chart is build from the thermodynamic properties of air and water and they gives an easy way of determine the air-cooling capacity that is lost due to the condensation of water. There are some commercial computer-programs that can calculate the properties in the psycrometric chart. In our model however the chart was implemented as lookup tables.

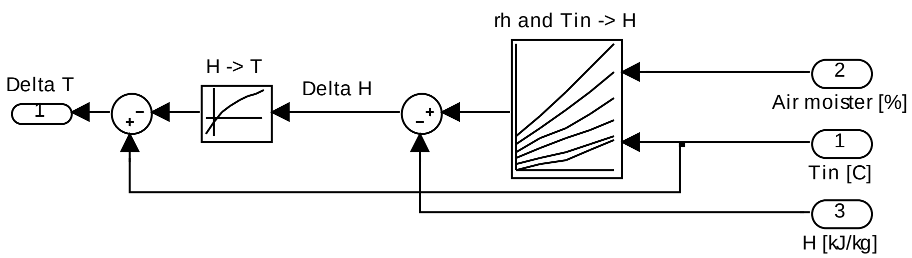


Figure 10. Simulink model of the molliere diagram.

The implemented Simulink model is showed in figure 10.
This method of determining the latent heat is commonly accepted in the industry and has proved it to give adequate results. A psychrometric chart is supplied in Appendix A.

When all this has bee determined it is possible to determine the temperature of the air after the evaporator T_a in the equation above. The resulting Simulink model for these properties for the system is shown in figure 11.

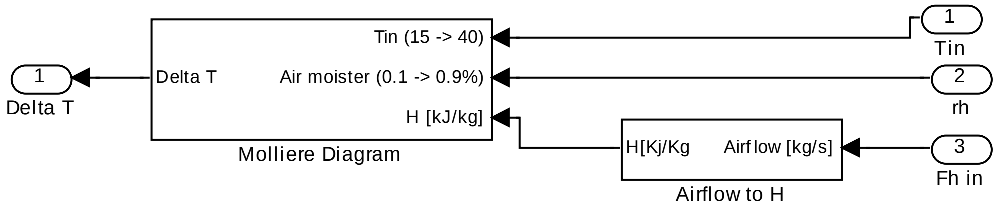


Figure 11. Model of the cooling capacity of the system

The total cooling capacity gives an approximation of the steady state behaviour of the system i.e. the cooling capacity that can be expected after running the compressor a long time (>10 min). This value is used to give the final level on the step response in the model.

4.3.2 Transient behaviour of the system

The next step in the model work was to try to model how the system reacted when the compressor was turned on and off. This was the most important part of modelling because the control algorithm would never let the system work in steady state mode when low cooling capacity is needed. The controller will switch the compressor on and off during the transients and this will post demands on accuracy to these parts of the model. It was known at Scania that the system behaves somewhat easier when using dry air. It is also preferable to run the AC-rig without the use of moist air. Therefor the first attempts in the AC-rig were done without any moist in the air a typical step response for the system is displayed in figure 12.

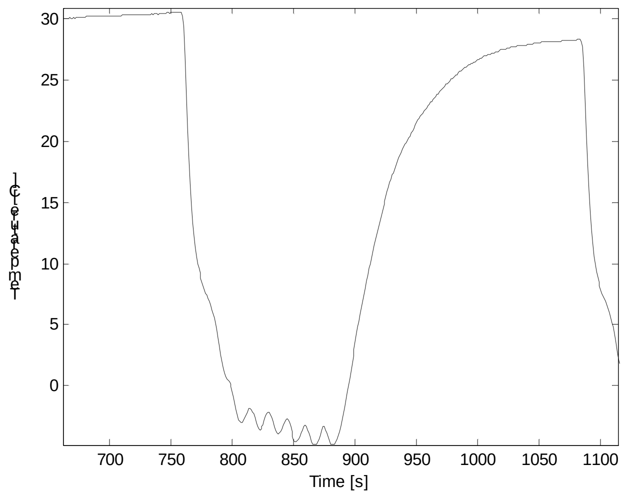


Figure 12. Step response for the evaporator temperature.

After studying step response measurements from the AC-rig it seemed natural to try and model it with some kind of simple first order system.

$$G(s) = \frac{K_p}{1 + sT} e^{-sL}$$

The static amplification K_p is the total cooling capacity and it is given as the output in figure 12. The time constant T was first appraised during constant airflow and moisture using data from the AC-rig and the NCD-block in Simulink.
As seen in figure 13 the time constant for the cooldown and the heating is quite different. This made it necessary to have one time constant for the cooldown and one for the heating if the model should work satisfactory e.i. to have different system models depending on the status of the compressor clutch. In general the cooling of the evaporator is much faster then the heating, this is natural since the compressor generates power to the system. When the correct parameters were found for the case when the ambient conditions were unchanged, trials with change in temperature, compressor rpm, airflow and moisture were performed. The condenser side of the system was never considered partly because there was no time to do a complete investigation and partly due to the fact that the cooling of the condenser in most practical cases is quite sufficient. These trials showed that the airflow and the air humidity were the two most important factors in regard of the response time. These were also the most important parameters for the cooling performance as seen in 4.3.1. Varying the airflow and humidity changed the systems timeconstants remarkably. A worst and best case for the time constants were tested in order for reasonable limits system to be established. The worst case (slowest response) was when 100% humidity and maximum flow was used. The maximum airflow that was tested was selected from the maximum airflow that could be generated by the fan in the truck. The fastest response time was achieved with no humidity and minimal airflow. When studying these measurements it was found that there was a remarkable difference between the cases. The time constant for the evaporator temperature typically differed as much as by a factor 10 between best and worst case. It also showed that the airflow was more influential than the humidity in terms of response time.

Therefor the first attempt was to include dependence between the timeconstant and the airflow. This was done by letting the timeconstant be linear dependent of the airflow across the evaporator. The constants a and b were determined experientially in the AC-rig.

$$G(s) = \frac{K_p}{1 + sT(fh)} e^{-sL}$$

$$T(fh) = a * fh \quad \text{if the AC - compressor is off}$$

$$T(fh) = b * fh \quad \text{if the AC - compressor is on}$$

The Simulink models for these properties are displayed in figure 13.

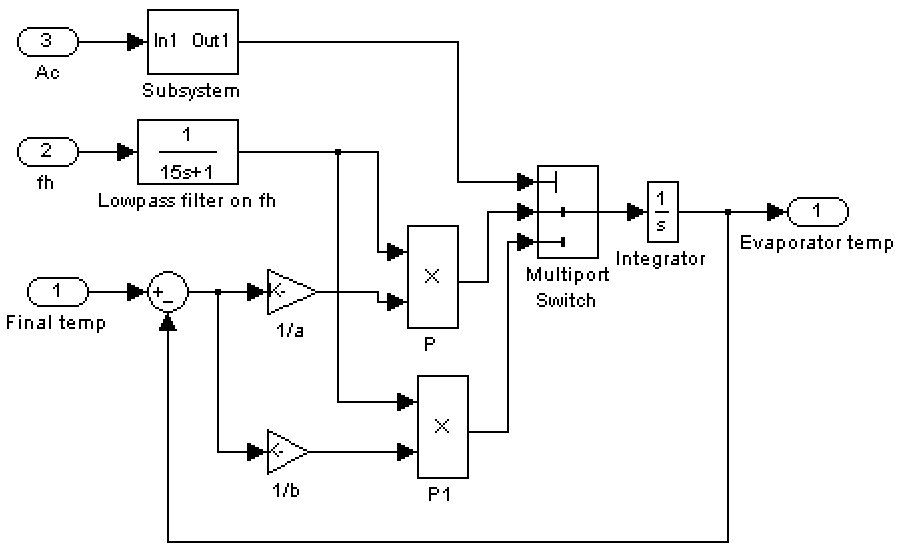


Figure 13. Model of the transient behaviour of the system.

The measured airflow signal from the AC-rig contained a considerably amount of noise. So in order for the model to be run with actual indata from the AC-rig a lowpass filter on the airflow signal was necessary.

4.3.1 Water pickup

An increase in air moisture did not only lead to a decrease in cooling capacity as stated above but it also changed the total heating curve. This can be seen in figure 14. This is due to the water or ice that is deposited on the evaporator. In the evaporators heating phase this water must be evaporated and this phase transition will crave energy. So when the evaporator temperature reach the dew point the evaporator temperature will not increase until the water has evaporated. The time it takes for all of the water to evaporate was modelled as linear depended on the time that the compressor was engaged. This is a very course approximation of a complicated process but as the accuracy demands on the model was very low in this specific sense the approximation was accepted. It was thought better to at least have the phenomenon in the model although not very accurate. When looking at the temperature distribution over the evaporator surface during the evaporation of the deposited water the complexity of the process becomes very obvious.

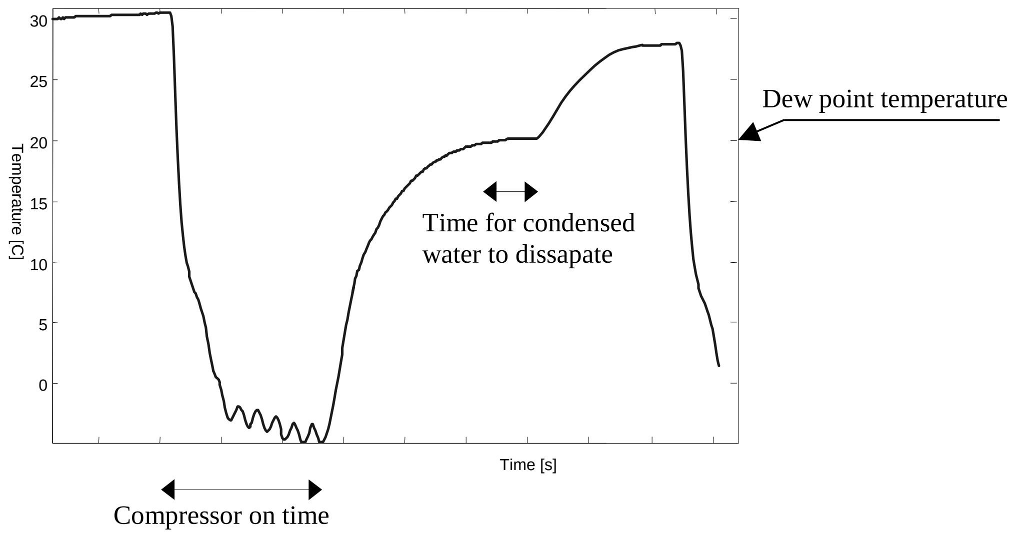


Figure 14. Typical system performance when using humid air.

The water pickup in the evaporator was modelled as dependent of the time that the compressor was active. That is the longer the time that the AC-compressor is engaged the more water was condensed on the evaporator. The amounts of water on the evaporator the determined the time that the temperature should stay at the devpoint temperature before reaching the final temperature. After some parameter adjustments a linear dependence between the compressor on time and water pickup was chosen.

This is of course a very simplificated way of approaching the problem. This way of modelling the water pickup dispersion does for instance not take into account the water that dripples of the bottom of the evaporator. It also doesn’t consider the fact that during certain conditions some icing of the evaporator is to be expected. But as a simple and fast way of implementing this system behaviour into the model it worked quite well. If the compressor was engaged for short periods of time this simple model worked quite satisfactory.

4.3.2 The complete model

The two main part of the model, the transient and the static part was put together to form a total model of the system. This model is shown in figure 15. The model can be driven with either signals from the AC-rig or with completely made up data.

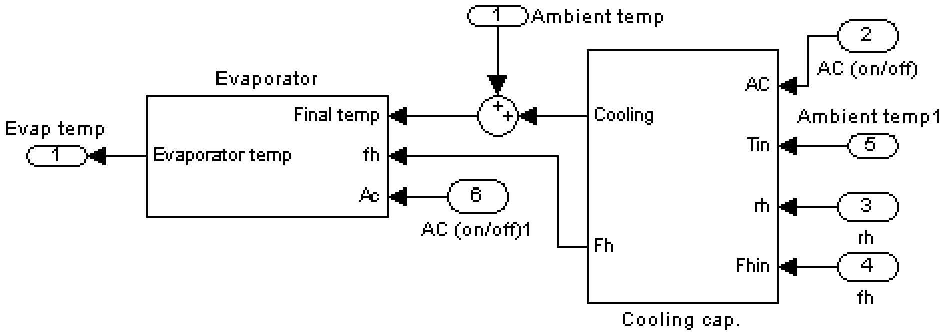


Figure 15. The complete model of the AC-system.

4.3 Model performance

The main goal of the modelling work wasn't to develop a detailed model of the actual system performance but to display the quantitative behaviour to aid the development of the control algorithm. And to that purpose the model was found to be adequately accurate. In figure 16 the modelresponse contra real measurements are shown. This good example on how well one could make this model perform. Poorer was generally conceived with more difficult conditions. The model wasn't very accurate when trying with high air moisture and much icing on the evaporator due to the very simple principle of modelling this property. However this feature wasn't important for the specified situation when low cooling was required. For a more accurate model of the water pickup/dissipation a much more detailed model that takes in account the temperature distribution over the surface of evaporator must be developed. As there wasn't any modelling of the expansion valve the oscillations that the real system has in its "stationary" phase isn't present in the model.

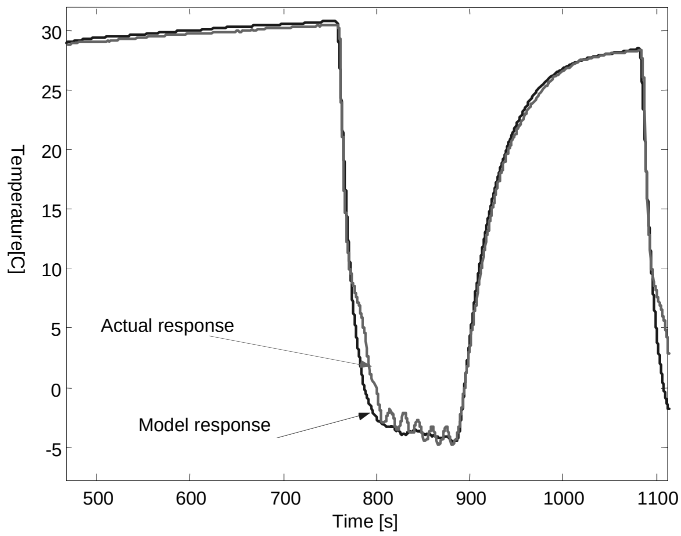


Figure 16. Simulated and measured step response.

The model was tested against measurements from the AC-rig and it proved to cover the whole spectrum of conditions that the AC-rig could produce in terms of airflow, temperature and humidity. In conclusion it could be said that the model performance was more than satisfactory for the purpose that it was developed for.

4.4 Equipment

A big part of the work was to make measurements on the system and to conduct different tests. The test equipment therefor plays a central role for the work done.

4.4.1 AC test rig

Scanias AC test rig is of an ordinary type mainly designed for performance test of evaporators and condensors. It consists of two separate air systems, one for the evaporator side and one for the condenser side. Temperature, airflow and air humidity can be set in both ducts and there are flow control devices mounted to try to ensure laminar airflow. A standard AC system is mounted in the rig and the AC compressor is driven by a speed controlled electrical motor.

Table 2. Shows a list of the quantities measured in the AC-rig.

Quantity	Unit	Comment
Air flow	Kg/s	
Low and high side refrigerant pressure	Pa	
Air moisture before and after evaporator	%	
Temperature at the moisture sensors	°C	Both pt-100 and thermocouples
Rotation speed of the compressor	Rpm	
Temperature distribution after evaporator	°C	Nine thermocouples

Table 1. Quantities measured in the AC-rig.

To collect all this data a 30-channel datalogger (Intab AAC-2) was used. The logger had a maximum sample rate of 1 Hz, so that is the sample rate used. When it was possible we tried to measure temperature both with pt-100 and thermocouples so that we had control over the inaccuracy in the thermocouples at steady state. The thermocouples are better then the pt-100 in that sense that they have a much faster response time.

The AC-rig was not optimal for doing transient experiments on the system because of the fact that the air was recirculated. This made it very difficult to maintain a constant temperature of the air going in to the evaporator. During a step response test it also

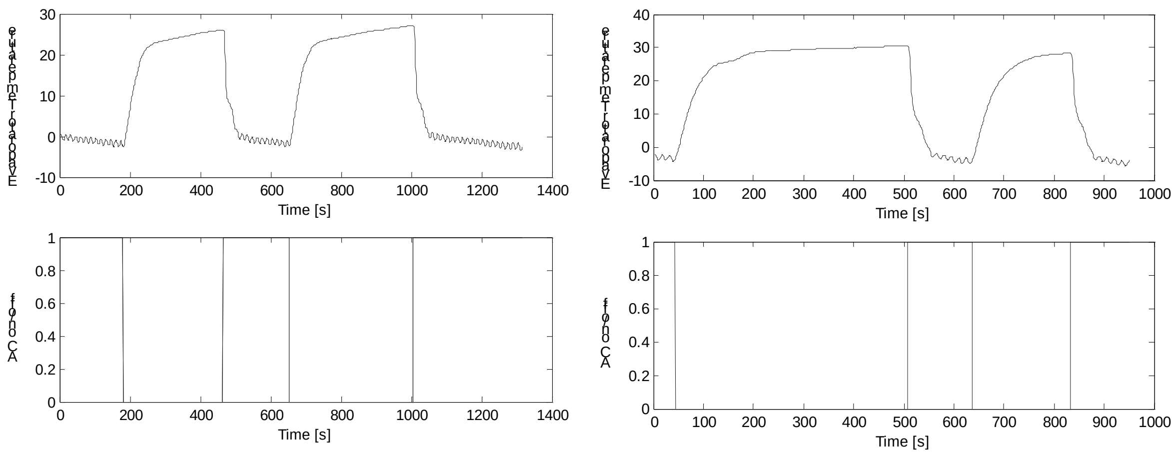


Figure 17. Two step response measurements in the AC-rig, the one to the left is before the modification and the right one after the modification.

made it virtually impossible to control the humidity of the air. To resolve this problem a slight modification of the rig had to be made. Instead of recirculating the air in the evaporator channel the air was lead out into the room. The room is a much bigger reservoir and this made it possible to maintain constant conditions for the AC-system during the experiments. Figure 17 shows two step response experiments performed in the AC-rig one before and one after the modification.

Temperature distribution

To get a picture of what the temperature distribution over the evaporator looked like, the signals of the nine thermocouples was plotted as a surface. One picture per sample was created and then put together to show an animation of the temperature distribution as the system was run. This gave an easy method of studying the temperature variations and made it easy to understand what really happened. The alternative is to use a thermal camera, this gives more detailed images to the cost of a much more complicated and expensive measurement procedure.

Ideally the flow distribution across the evaporator should also be measured but since this is very complicated and demands complicated measuring equipment, measurements of this kind wasn't performed.

4.3.2 Climate chamber

Scanias climate chamber is large enough to fit a whole truck. The temperature can be set from –40 °C to +50 °C and it is actively controlled so that the temperature isn't affected by heat generated by a running truck. There is a rolling highway fitted in the room, the dynamometer is capable of breaking the maximum power of a normal engine and there are two big fans in the front of the room, these are used to simulate the wind on the front of the truck when driving.

5. Control algorithm

Two different approaches to controlling the system were tried. One when the compressor is directly controlled by a regulator and one where a pulse width is modulated by a regulator. It should be stated that there is another way of generating low cooling capacity than cycling the compressor on/off. This is to use the heater that is situated after the evaporator to heat the air to the desired temperature after it has been cooled by the evaporator. This option is not considered in this thesis for several reasons, it is for instance not very fuel efficient to run the system this way.

The first hypothesis was to utilise a cascade control structure where the inner controller should ensure that the temperature variations in the evaporator temperature were held within the limits. The outer controller should adjust the level of the demanded evaporator temperature so that the correct mixair temperature was achieved. A sketch of the chosen control structure is shown in figure 18.

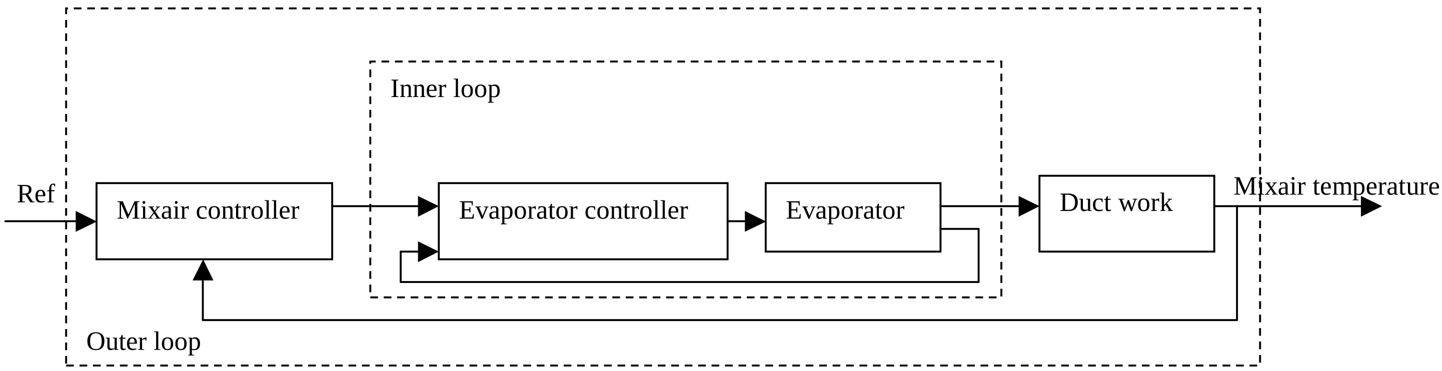


Figure 18. Controller structure.

In the choosing of this type of control structure lay the assumption that the temperature variations in the evaporator temperature would be larger than the ones in the mixair temperature. This seemed reasonable to believe since the evaporator temperature sensor was situated on the place on the evaporator where the most power was dispersed. A study of old measurements made at Scania also supported this assumption. One such measurement can be seen in figure 19. Here it clearly shows that the mixair temperature seems to have smaller amplitude on its oscillations.

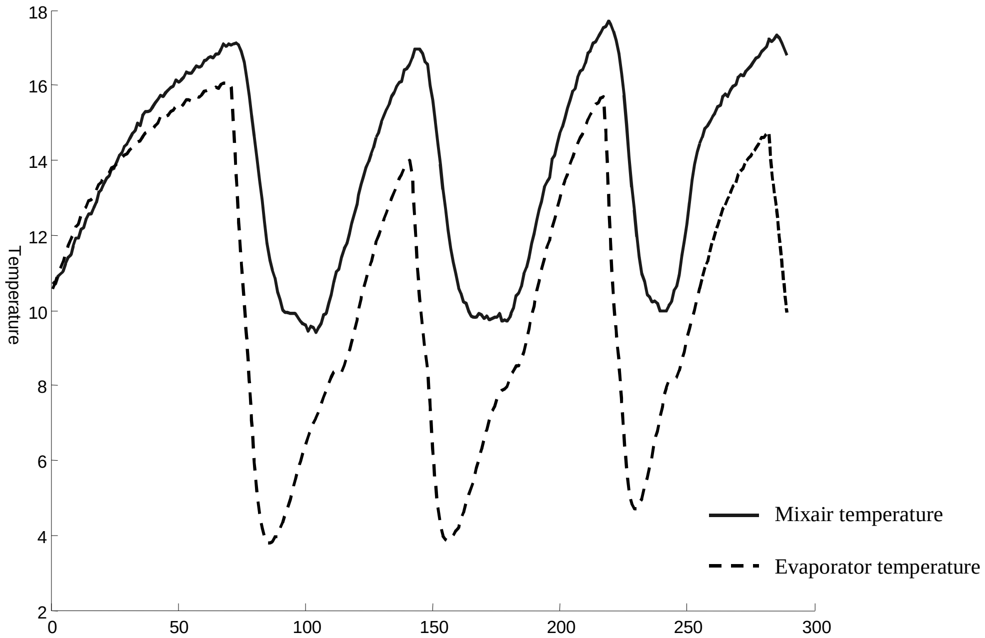


Figure 19. Previous measurement displaying mixair and evaporator temperatures.

Unfortunately this property of the system could not be tested in the AC-rig since there wasn't any accurate way to see how the mixair temperature would behave in the truck from the rig set-up.

When doing evaluation tests on a truck in the climate chamber it was found that the assumption that the mixair temperature variations were less then the one in the evaporator temperature was not entirely correct. It showed that when higher switchrates where employed on the compressor the fluctuations in mixair temperature often became larger then the ones in the evaporator temperature, as is displayed in figure 20.

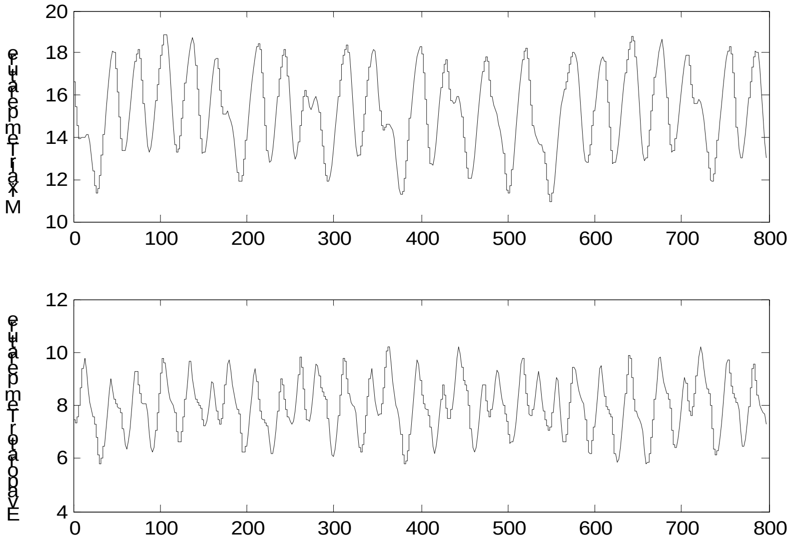


Figure 20.Comparison of the mixair temperature and the evaporator temperature.

This result was not anticipated and it was even thought that the mixair temperature would be more even with higher switchrates due to the more evenly distributed refrigerant flow through the evaporator. This thought is true for very high switchrates where the flow/pressure in the system don't have time to decrease to the same extent as with a lower switchrate. However these switchrates can never be allowed in the control algorithm due to the amount of stress it would induce on the magnetic clutch in the compressor.

This problem kind of made the cascade approach to the control structure unfeasible. Two different ways of dealing with this were tried:

1. Let the control algorithm use the mixair temperature directly as an in signal (no cascade coupling).
2. Use the cascade structure and sharpen the demands for temperature variations in the evaporator temperature so that it will allow for the mixair temperature to fluctuate more.

Both these strategies were tested on the two different control algorithms.

5.1 Demands

The most obvious demand on the regulator is that it should be able to control the mixair temperature within a allowed temperature span. This temperature span was set to ±2.5°C of the desired mixair temperature. One other obvious request is that the response time should be as fast as possible, this property is mostly limited by the capacity of the AC-system.

The AC compressor drains considerably power from the engine and it is therefore desirable to run the compressor as little as possible in order to minimise the total fuel consumption. There are also limitations on how much stress that can be induced on the compressor without endangering its lifetime. This demand is limited by the number of cycles that the magnetic clutch can perform i.e. it is not allowed to switch the compressor on an off at a high pace. No investigations regarding the durability of the system and the wear that can be allowed have been performed in this thesis. Therefor the demands on the compressor clutch have been taken from the compressor manufacturer recommendations.

5.2 Direct control of the compressor

This regulator design is somewhat more straightforward than the pulswidth modulated approach. Here the compressor is directly controlled by a regulator via a relay, according to figure 21.

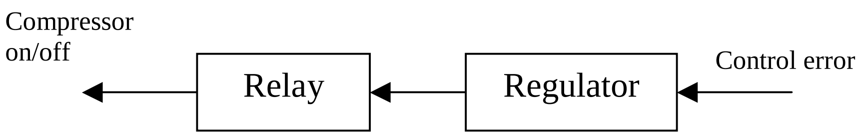


Figure 21. Control principle

This regulator structure was first tried out in the AC-rig, controlling the evaporator temperature and using thermocouples for measuring in signals. In the AC-rig a simple PID-regulator was used and its performance was really promising. The control law that was implemented looks as follows:

$$\mathrm{IF}\,(|\mathrm{e_n}| < 10)\ \mathrm{THEN}\ I_n = I_{n-1} + K\frac{T_S}{T_i}$$

$$v_n = Ke_n + I_n + K\frac{T_d}{T_S}(e_n - e_{n-1})$$

$$u_n = \begin{cases} 1 & \mathrm{IF\,v_n} > 5 \\ 0 & \mathrm{IF\,v_n} < -5 \end{cases}$$

As seen above permissive integration was used to counteract reset windup. The boundaries for the relay on the output as well as the regulator parameters where first approximated using the computer model of the system. These were then adjusted further when the system was implemented in the AC-rig.

The regulator was implemented using again the AAC-2 logger who was suitably equipped with relayed outputs. A laptop was connected to the logger via a serial interface and a simple computer program was developed in order for different controller to be evaluated in the AC-rig. The thermocouples that were already fitted in the AC-rig were used for the in signals and the relayed output on the logger controlled the magnetic clutch in the compressor.

When the regulator was transferred to the truck and the actual system, some complications arose. Firstly it showed that the temperature sensors in the actual system in the truck was considerably slower in their response time than the thermocouples. This gave the system a much slower response and a longer time delay from the input to the output of the system. Secondly the assumption of the mixair temperature behaviour proved not to be accurate. This made it necessary for some changes in the original structure.

The problem with having a slower system was countered with the use of a PID regulator with a large derivative proportion. This made it possible to regain some of the regulator performance from the AC-rig but it also made the controller very sensitive to disturbances in the temperature signal.

An Otto Schmidt deadbeat regulator structure [2] was also tried on the system in the truck in order to counter the problem with slow sensors. A simple model of the system behaviour and time delay was used.

The response of this type of regulator could be improved by adding a forward feed from the airflow across the evaporator. This property showed to have a great importance on the power that should be put out by the compressor.

5.2.1 Evaluation

The only possible way of using these regulators is to use them directly on the evaporator temperature and then having another outer loop to control the mixair temperature. This is because the response time of the system is critical to the regulator and the delay from the compressor to the mixair sensor is longer than the one to the evaporator sensor. Also no model of the mixair temperature behaviour was developed and that is a demand if a successful regulator should be developed.

The controller showed to work very well in the AC-rig. The amplitude in the evaporator oscillations corresponded well with the setting of the relay boundaries. To test the controller disturbances in the forms of step changes in the airflow and the relative humidity was introduced. The controller showed to be able to cope remarkably well with these disturbances. When changing the humidity from 10% to ~90% the temperature was held within the boundaries all the time. Changes in the airflow were somewhat harder for the controller since they change the system remarkably. The controller typically restored the evaporator temperature within 5 minutes from a step disturbance in the airflow (0.1 Kg/s to 0.2 Kg/s). Unfortunately no measurements from these experiments can be displayed because the logger was used for the regulator. The computer program that was made didn’t include any logging capability, it mearly displayed the temperature as a plot on the screen.

The standard PID approach did never work as well in the truck as it did in the AC-rig and in the model. This was mostly due to the slower sensors used in the truck. The problem with the variations in the mixair temperature also made it hard to be able to grant a certain performance. The only thing to do about this problem was to use a much higher demand on the temperature variations in the evaporator temperature and hope that it would be enough to keep the mixair temperature within its boundaries. Another problem is the high degree of derivation of the in signal that causes the regulator to be sensitive to disturbances. This must be countered with extensive filtering of the temperature signal.

The Otto Schimdt regulator proved to work well when the internal model used showed high resemblance to the actual system performance. But since the system changes very much with influence of many parameters, see section 4., it is very hard to get good performance with this kind of regulator structure. It was therefore deemed that this wasn’t going to be a successful strategy unless a very accurate model was developed and additional sensors were added to the AC-system. An interesting alternative is to use an adaptive controller that continuously trains the model in an IMC controller [2] structure. An adaptive controller should be able to counter some of the influential parameters and thus give a better controller.

Another problem with this control structure is that there is no control of how often the compressor is switched on and off. If this kind of structure is to be used a limitation on the switch rate must be implemented so that one with certainty can say that too much stress isn’t induced on the compressor clutch. This controller does, on the other hand, not drive the compressor more than it has too in order to grant the desired performance. This makes it somewhat more fuel efficient and easier for the compressor clutch.

5.3 Pulswidth modulated control

This technique is based on the assumption that there is a certain pulswidth that will grant good performance under specific conditions. The control signal is the control error and the steer signal is pulswidth. The control principle is displayed in figure 22.

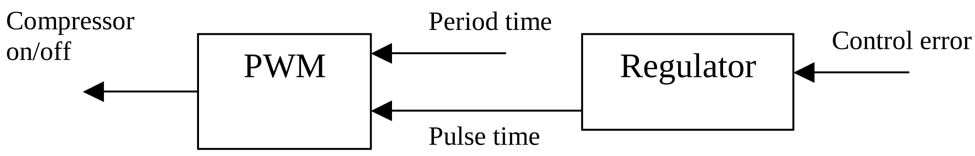
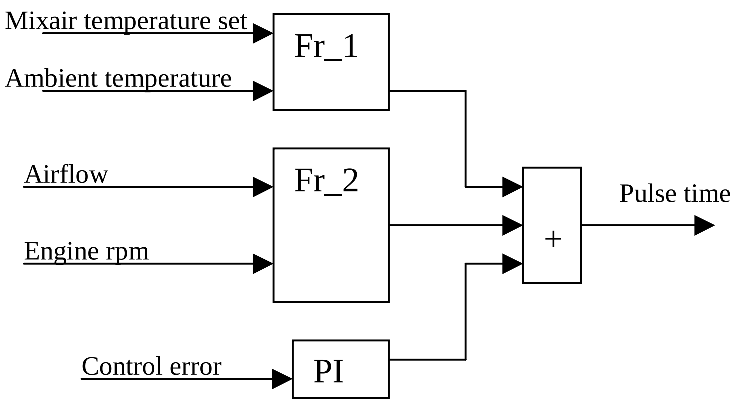


Figure 22. Control principle

The control algorithm made use of a basic PID-regulator with a rather slow response in order to get a reliable steady state performance. This gave the system good steady state performance while the regulator showed very poor response times. This was resolved by a feedforward coupling from the desired mixair temperature. The feedforward gave the regulator a much faster response time. The use of a slow regulator also made for very bad performance to disturbances in the system. For instance a change in fan speed would take several minutes to compensate. To sharpen this response behaviour of the regulator a feedforward network from the airflow and engine rpm should be employed. The structure of the final control structure is shown in figure 23.



Figur 23 Principal sketch of the final pulswidth control algorithm.

The airflow across isn't directly measured but since the fan control compensates for the vehicle speed the airflow is almost constant at the selected fan speed. This enables for the airflow to be used for forward coupling.

5.3.1 Evaluation

In order to evaluate this control structure it was implemented in an ECU and tested in truck in the climate chamber.

This control structure proved to be rather indifferent to which of the signals that were used as in signal. This is quite reasonable as the system is not directly controlled by the control signal but rather the average power output that should be put out by the compressor. Therefor the best way to use this control structure is to directly control the mixair temperature.

The temperature fluctuations are directly proportional to the cycle time used, shorter cycle time results in more precise temperature control. The response time of the system is mostly determined by the design of the feedforward loops Fr_1 and Fr_2.

Figure 24 displays the result from one experiment using the PWM controller in a truck that is placed in the climate chamber. As seen in figure 24 the steady state performance of the controller is well inside the temperature frame. However the transient behaviour is not satisfactory without some kind of forward coupling. Generally a better controller performance is achieved with lower cycle times i.e. higher switch frequencies.

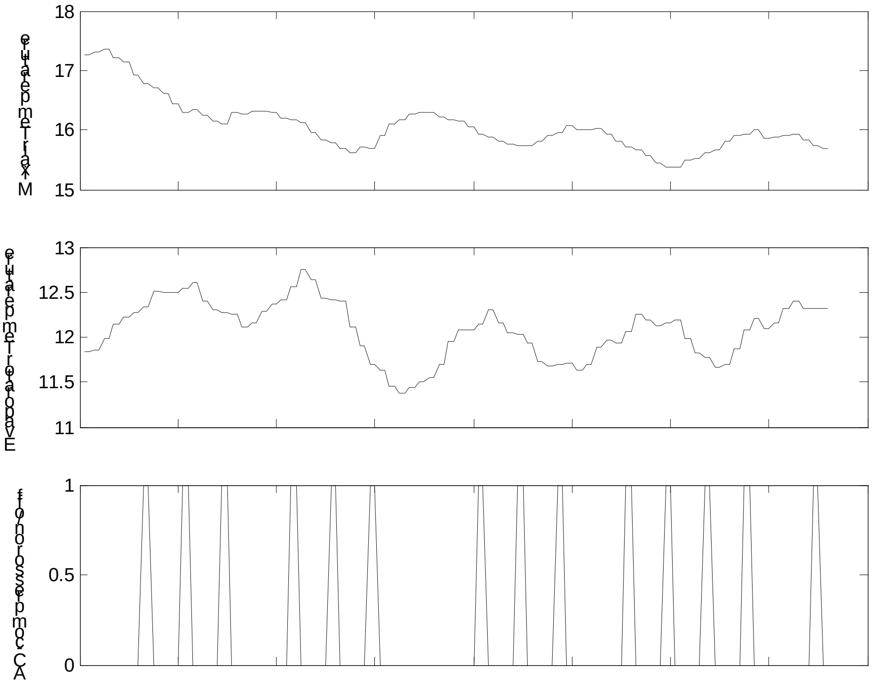


Figure 24. System output using the PWM controller.

6. Discussion and Conclusions

A control strategy for the AC-system in Scanias trucks has been developed and evaluated. The design procedure can roughly be divided into 5 steps.

1. **Gathering knowledge.** The first step in designing a control strategy is to gain as much knowledge as possible about the system that is to be controlled. The quality of the control structure is greatly dependent on the designer's experience and knowledge of the system. In order to obtain this knowledge a lot of documentation about AC-systems have been gathered and studied. Another great source of information is the knowledge bank gathered by the engineers at Scania, interviewing them often gave profound information. Some initial measurements and field trials where also conducted in order to get a feel for the specific system.
2. **Building the principle model.** In order to efficiently control the system it is preferable to have a system model that can be used for simulations. In this case a model for computer simulations also shortened the development time substantially, as the actual system was very slow. To conduct trials on the actual system it self is therefor very time consuming.

   The purpose of the model was for it to have the principal behaviour that the actual system exhibited. It was found that building a more exact model of the system would be very complicated that it wouldn't fit within the frames of this thesis. The most important aspect of the model was to be able to predict the transient behaviour of the system, this because it was in this region that the controller should work.
3. **Designing the control system.** The design of the control system starts with deciding the demands that shall be inflicted on the controlled system. When these demands are formulated a main structure of the controller must be decided. The control strategies are best developed with the help of the computer model of the system. The use of a program package like Simulink greatly aids the designer in his design work. When a suitable control structure has been chosen the design of the regulator it self can commence. This work is also preferably done in a computer environment with a system model. Depending on the model accuracy more or less parameter tuning can be done in the computer. There is however always necessary to adjust the controller in the actual system.
4. **Implementation of the controller.** When the controller performs well on the simulated system it should be tested on the real system. To be able to do this the selected controller must be implemented on the actual system. Some program packages like Simulink/RTW supports rapid prototyping and code generation to some platforms. These programs can greatly reduce the time it take to implement the controller.
5. **Evaluation of the controller performance.** The evaluation of the controller is the last step in the design chain. If the controlled system performs within the demands that are postulated then the design can be released for production. If the controller does not fulfil some of the demands the designer must restart the design process at a proper level.

6.1 Modelling the AC-system

The modelling of the AC-system was an important part of this thesis and much effort was put into learning how the system behaved. It was found that a total model of the system would be very complicated to develop. This is because of the high complexity and the many dependencies that the system exhibits. The model that was developed worked well and was as accurate as one can expect given the simplifications made. Although the designing of control algorithms for the system would benefit greatly from a more accurate and complete model, it was found that the model that was built served its purpose sufficiently. The model proved to be a great help in the design work and it helped a great deal in speeding up the process. As a comparison it can be mentioned that 1 hours work in the AC-rig can be simulated in less than 1 minute in the Simulink model. The model was used both with data measured in the AC-rig and with data that should reflect actual conditions in nature.

6.2 The controller

The two control strategies that were tested showed to exhibit different behaviours and advantages. If a proper model of the total system performance could be developed the controller with direct compressor control could be made more efficient. This controller suffered much from the slow sensors that are used in the truck and if one tries to compensate for them the problem will shift to involve the sensitivity for disturbances. Maybe if some more effort were put into this work it would result in an effective controller utilising a true bang-bang control. On the other hand this regulator structure should be able to provide a more optimal control of the system because it can be made much faster than the PWM controller. It is also this structure that will provide the lowest fuel consumption and lowest over all switching of the compressor. This is due to the fact that this controller doesn’t perform any unnecessary switching of the compressor, which could become the case with a PWM controller.

The PWM approach on the other hand provides a robust and simple controller that also has the possibility to work well in the system. The lack in performance that the PWM experiences can be helped with the use of extensive forward coupling. One further development of this controller could be to adapt the cycle time so that a more optimal control can be achieved.

6.3 Future work

It is believed that in order for an efficient controller to be developed an accurate total model of the AC-system should be developed. The model must take in to account the temperature variations over the evaporator surface and some of the “hard to measure” parameters in the system.

To only use the magnetic clutch as the actuator for the AC-system does inevitably put some restraints on the overall performance that can be achieved. It is thought that much can be gained from introducing some kind of advanced flow control device such as an electronic expansion valve. This would give the system designers much better abilities to optimise system performance in many of the critical operation cases. Such system can show many good sides, see [3], for instance do they often provide methods of detecting low levels of cooling agent.

It should also be mentioned that the AC-system isn’t designed to be run this way. All AC-system designing has the goal to maximise the cooling performance and not to grant certan performance for this type of application. This means that it is unwise to introduce a controller that switches the compressor at a high pace without first doing detailed studies on the effects that this have on the system. Things that must be investigated are such as the stress on the compressor, the amount of compressor oil needed, stress on tubing and hoses and the load induced on the auxiliary belt transmission. If all these things are investigated this way of controlling the AC-system will prove to be most successful.

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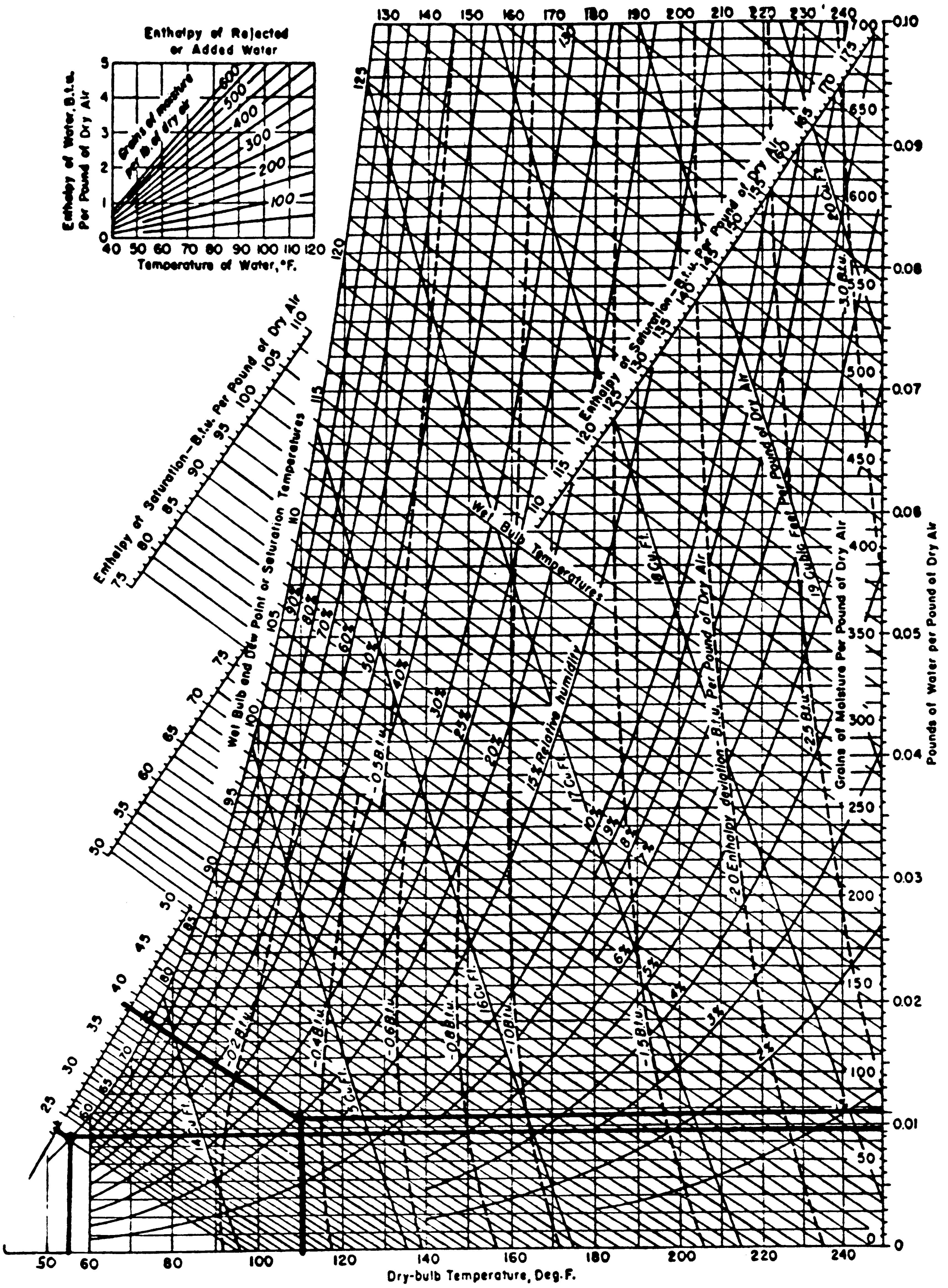
Appendix A



Enthalpy of Rejected or Added Water
Enthalpy of Water, B.t.u. Per Pound of Dry Air
Temperature of Water, °F.
Enthalpy at Saturation—B.t.u. Per Pound of Dry Air
Wet Bulb and Dew Point or Saturation Temperatures
Wet Bulb Temperatures
Grains of Moisture Per Pound of Dry Air
Pounds of Water per Pound of Dry Air
Dry-bulb Temperature, Deg. F.