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Abstract

Since fully electric vehicles (FEV) do not have enough waste heat for cabin heating as internal combustion engine (ICE) vehicles and hybrid vehicles do, it is beneficial to employ a reversible heat pump in FEVs. The reversible heat pump can be used for both the cabin cooling and heating and especially under winter conditions it is able to substantially improve the FEV driving range compared to heating with PTC heater.

This paper is focused on energy efficient control for the reversible heat pumps using either R1234yf or R744 refrigerant. The main goal of these control algorithms is to achieve a satisfactory passenger thermal comfort with as low as possible energy consumption.

Keywords: optimal control; heat pump; fully electric vehicle; thermal management system

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Nomenclature

AC	Air conditioning
COP	Coefficient of performance
EXV	Electric expansion valve
FEV	Fully electric vehicle
HVAC	Heating, ventilation and air conditioning
MIMO	Multiple-input, multiple-output
SH	Superheat
SISO	Single-input, single-output
$\dot{m}$	Mass flow rate
T	Temperature
t	Time

1. Introduction

An automotive heat pump (used as a common name for cooling and heating devices as it pumps heat between two mediums) usually consists of compressor, evaporator, condenser and expansion device. In internal combustion engine (ICE) vehicles compressor performance as well as heat pump performance is dependent on ICE speed. ICE vehicles usually utilize heat pump only for cooling as there is enough heat from ICE for cabin heating. On the other hand, in FEV the heat pump performance is dependent neither on main electric motor nor on vehicle speed, because a compressor with built-in electric motor is been used and it can be controlled independently on other parts of FEV.

Variable speed capacity control is the best possible way to adjust a heat pump performance to current cooling or heating demand (Qureshi et al. (1996)). Nowadays most of the problems blocking the widespread of variable speed control reported by Qureshi et al. (1996) are solved and it can be advantageously used for heat pump capacity control. Variable speed capacity control was proposed for energy savings in buildings in combination with predictive control methods (Salsbury et al. (2013)). The comparison of on/off control and variable speed control for ground source heat pump was done by Madani et al. (2011). Considerable attention has been paid to variable speed control of heat pumps in both the domestic and commercial buildings, industrial refrigeration plants and supermarket refrigeration (Gillies (2014), Henry et al. (2014), Lee et al. (2016) and Mastrullo et al. (2015)). It needs to be mentioned that variable speed of auxiliary devices (like evaporator fan, condenser fan, pumps etc.) is also needed, otherwise power consumption of these devices can negatively affect the COP and overall performance (Niu et al. (2016)).

Regarding the expansion device, the TXVs are widely utilized, but it was shown for example in Aprea et al. (2002) that EXVs have better characteristics (possible lower superheat, better performance during transients, not prone to "valve hunting" etc.). In combination with possibility of usage of CO_2 (R744) as a refrigerant, EXVs are better choice instead of other expansion devices.

Also, condenser and evaporator mass flow rate control is possible in FEV heat pump (using variable speed electric motor), thus some control algorithms for condenser and evaporator fans are needed. This is another difference from ICE vehicles, where only discontinuous (on/off) or stage control of fans is usually possible.

2. Automotive heat pump and HVAC models

A heat pump model was created in Dymola using ThermalSystems Mobile AC library. The resulting model is very complex and quite accurate, so it is possible to use it for control algorithms evaluation. The model was exported into Functional Mock-Up Unit (FMU) and then simulated in Matlab Simulink using FMUtoolbox (which was previously developed by us).

Also heating, ventilation and air conditioning (HVAC) models were created to allow control algorithms design and their evaluation. We created two versions of those models, as different models' complexity is needed for different tasks.

2.1. HVAC simple model

As an example, a model of vehicle cabin is shown in this section. Other parts of HVAC are modelled using the same approach.

This model is based on basic thermal equations and it presumes some simplifications. The air humidity is assumed to be constant and all parts of model are considered as lumped (ideally mixed).

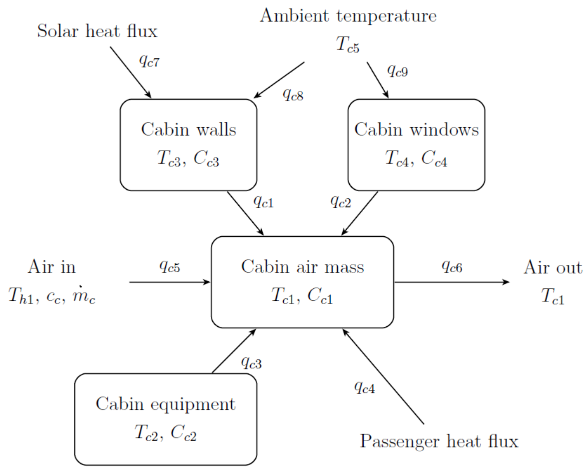

Fig. 1 Vehicle cabin simple model

The HVAC model is shown in Fig. 1. From this schematic, we can write the model equations in general

$$C_{c1}\frac{dT_{c1}}{dt} = q_{c1} + q_{c2} + q_{c5} + q_{c4} + q_{c3} - q_{c6},$$
$$C_{c2}\frac{dT_{c2}}{dt} = -q_{c3},$$
$$C_{c3}\frac{dT_{c3}}{dt} = -q_{c1} + q_{c7} + q_{c8},$$
$$C_{c4}\frac{dT_{c4}}{dt} = -q_{c2} + q_{c9},$$

and after quantification of heat flows the equations become

$$\frac{dT_{c1}}{dt} = \frac{1}{C_{c1}}\left[-\left(G_{c1} + G_{c2} + G_{c3} + \dot{m}_c c_c\right)T_{c1} + G_{c3}T_{c2} + G_{c1}T_{c3} + G_{c2}T_{c4} + \dot{m}_c c_c T_{h1} + q_{c4}\right],$$
$$\frac{dT_{c2}}{dt} = \frac{1}{C_{c2}}\left(G_{c3}T_{c1} - G_{c3}T_{c2}\right),$$
$$\frac{dT_{c3}}{dt} = \frac{1}{C_{c3}}\left[G_{c1}T_{c1} - \left(G_{c1} + G_{c8}\right)T_{c3} + G_{c8}T_{c5} + q_{c7}\right],$$
$$\frac{dT_{c4}}{dt} = \frac{1}{C_{c4}}\left[G_{c2}T_{c1} - \left(G_{c2} + G_{c9}\right)T_{c4} + G_{c9}T_{c5}\right],$$

where T denotes temperature, G stands for thermal conductance, $\dot{m}$ is mass flow rate, c is specific heat capacity and C denotes heat capacity. Index c is used for cabin variables and number indexes 1 to 4 indicate cabin air mass, cabin internal equipment, cabin walls and cabin windows respectively.

2.2. HVAC complex model

A complex model was created in the same way as the heat pump model and they were combined to produce one complex model of heat pump, HVAC and vehicle cabin (Fig. 2). The simulations of resulting model are used as a substitution of real experiments.

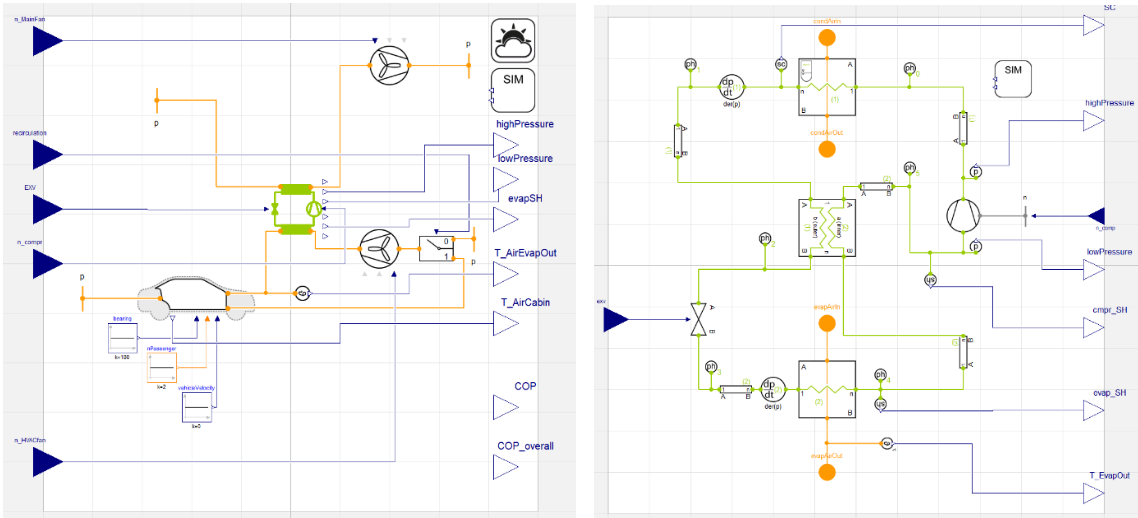

Fig. 2 (a) HVAC model in Dymola, (b) heat pump circuit model in Dymola

3. Heat pump and HVAC control

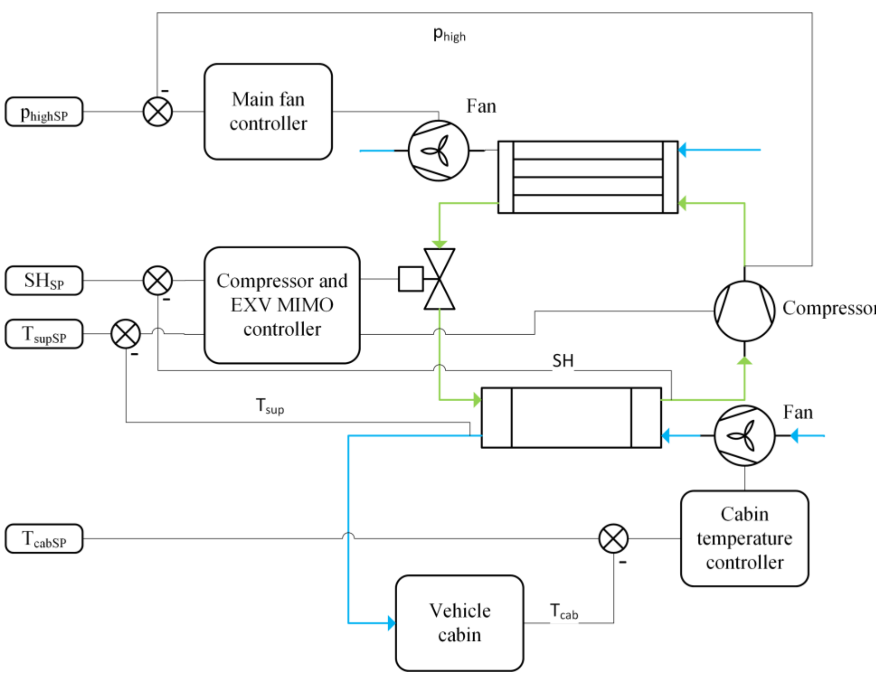

Fig. 3 Control loops overview

For heat pump and HVAC control we proposed a system of control loops. The overview of that system is in Fig. 3. Cabin cooling power is determined by speed of HVAC fan, which is supplying air with constant temperature into the cabin. The supply air temperature is controlled by compressor speed. Control algorithms of condenser fan and expansion valve ensure efficient operation under different ambient conditions.

3.1. Compressor control

In FEV, an electric compressor needs to be used. Then it is quite easy to control refrigeration capacity of the circuit using variable speed of electric machine. We propose to control compressor speed based on the air temperature behind the evaporator, which has a constant setpoint (for example 6°C). There are multiple advantages of this solution. Firstly, under partial load, low side pressure can be little bit higher and so the compressor work will be lower. Also, this approach eliminates the need of limitations for cabin supply air temperature.

3.2. Expansion valve control

There are two possible objectives of expansion valve control – desired evaporator superheat or condenser subcooling. The choice depends on refrigeration circuit design and layout. In circuits with high pressure receiver, superheat control is usually applied and in circuits with low pressure accumulator, subcooling control can be used.

As there exist quite strong coupling between compressor speed and evaporator superheat, we designed the superheat control as a Multiple Input Multiple Output (MIMO) system control. This approach allows compressor control loop to be much faster compared to individual (SISO) control of EXV and compressor speed, because MIMO decoupling helps to suppress superheat fall after compressor speed change.

Dymola heat pump model was used to obtain the transfer function of MIMO system. We got 2x2 MIMO system with inputs u and outputs y

$$u = \begin{pmatrix} u_1 \\ u_2 \end{pmatrix} = \begin{pmatrix} exv \\ c \end{pmatrix},$$

$$y = \begin{pmatrix} y_1 \\ y_2 \end{pmatrix} = \begin{pmatrix} SH \\ T_{\sup ply} \end{pmatrix},$$

where exv means opening of EXV, c denotes compressor speed, SH stands for evaporator superheat and T_{supply} is the temperature of air supplied into vehicle cabin. And then we have a matrix of transfer functions

$$G = \begin{pmatrix} G_{11} & G_{12} \\ G_{21} & G_{22} \end{pmatrix} = \begin{pmatrix} \dfrac{-918.2}{4s+1} & \dfrac{93.46}{2.5s+1} \\ \dfrac{-65(-9s+1)}{(0.64s+1)(4.86s+1)} & \dfrac{-4.721(16.6s+1)}{(0.43s+1)(3.75s+1)} \end{pmatrix},$$

where G_{ij} denotes the transfer function from input j to output i. This simple model was identified in selected operating point.

Using MIMO decoupling procedure we can compute compensators, which ensure that open loop transfer function matrix is diagonal and thus closed loop transfer functions matrix is also. In that case the system is fully compensated and control action on any input will influence only output with the same index (eg. compressor speed change will not change the superheat). As the model is very simple and measured in one operating point, the couplings will not be fully compensated, but we expect significant improvements. This anticipation is proved in Section 4.1 using heat pump simulation.

3.3. Evaporator fan control

Using evaporator fan, it is possible to control the heat flow into and from the vehicle cabin. If the cooling (or heating) demand is low, also the speed and the power consumption of the fan will be low. The control algorithm is based on the model introduced in Section 2.1. The first version consists of PI controller with limited output and anti-windup.

3.4. Condenser fan control

Condenser fan control is very important from overall energy consumption point of view. Insufficient air mass flow rate through the condenser can lead to excessive work done by compressor due to needed higher refrigerant pressure.

We denote $\dot{Q}$ as heat needed to be rejected in condenser from refrigerant to air and in very simple case we can write

$$\dot{Q} = G(T_{c1} - T_{c3}) \ ,$$

where G is condenser thermal conductance, T_{c1} is refrigerant temperature (assumed constant through condenser) and Tc3 is air temperature behind the condenser. We can describe behavior of Tc3 using equation

$$\frac{dT_{c3}}{dt} = \dot{m}cT_{c2} - \dot{m}cT_{c3} + \dot{Q},$$

where $\dot{m}$ is air mass flow rate, T_{c2} is ambient temperature and c is air specific heat capacity.

In steady state, the heat rejected in condenser needs to be constant, also the temperature split between refrigerant and the air needs to be constant. If air mass flow rate decreases (lower vehicle speed or stopped), air temperature behind the condenser starts to increase and thus $\dot{Q}$ will decrease and lower amount of heat will be rejected in condenser and cooling effect will be lower. To withstand that, a condensing temperature will be increased (by higher amount of heat inside condenser and also by using EXV and compressor control) to keep the condenser temperature split constant and as the refrigerant pressure and temperature are dependent, also pressure will increase. This brings higher pressure ratio and thus higher compressor power needed for the same cooling effect. Resulting COP will be lower and the cooling will have higher impact on battery SOC.

In this case, higher condenser air flow can be achieved by condenser fan, which can be operated at different speeds. We propose to keep the high-side pressure at such level, that the temperature split is kept constant at all vehicle speeds. This will bring additional power consumption, but it will prevent the increase of compressor power need.

We designed a PI controller for condenser fan speed control. Its performance is quite good (see the simulations), but probably it could be improved with feedforward control based on current vehicle speed and its changes. It will bring faster responses, because there will be no need to wait for the control error (as it is needed in feedback systems).

4. Simulations and measurements

4.1. *Compressor and EXV MIMO control*

In Fig. 4 there are results of simulation of heat pump connected to the vehicle cabin. The model is based on refrigerant circuit with high pressure receiver and EXV is used to control superheat to desired value. Compressor is driven to follow temperature setpoint in vehicle cabin. At the time of 70 seconds there is step of superheat setpoint from 7 K to 5 K and controllers can keep the temperature and superheat correctly. At the time of 100 s, there is temperature setpoint step from 25,5°C to 26,5°C and in Fig. 4 (a) the superheat falls to 0 K and that means liquid refrigerant flowing to compressor. In Fig. 4 (b), there is the same simulation, but MIMO control was applied and evaporator superheat is kept above 0 K, so compressor is protected against liquid flow and against possible damage. Also, the transient is faster, so the COP will be kept higher in case of MIMO control.

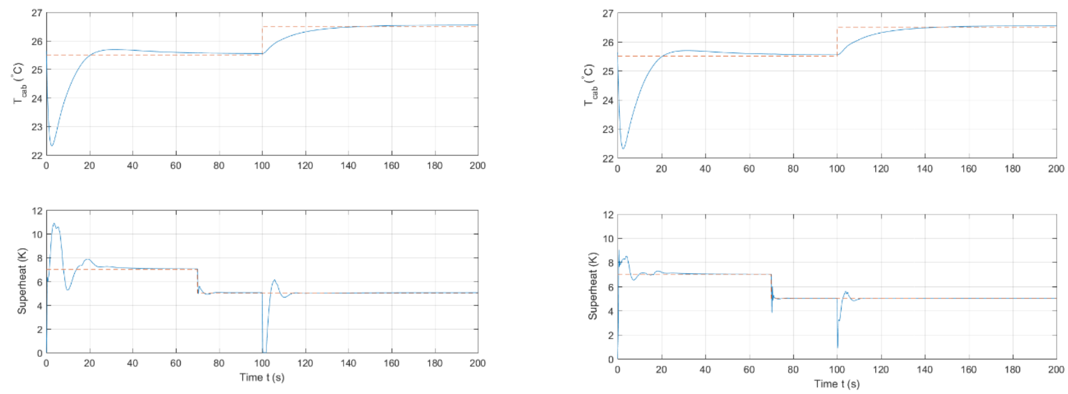

Fig. 4 Heat pump MIMO control simulation results (a) without decoupling, (b) with decoupling

4.2. Condenser fan control

In Fig. 5 there is verification of condenser fan controller, which is keeping the pressure at the defined value using condenser fan speed. At the time of 5000 seconds, the setpoint of pressure was increased by 1 bar and then it was decreased back at the time of 5200 s. The controller can keep the pressure at the setpoint with no steady-state error. The PI controller was designed quite slow to avoid oscillations of pressure.

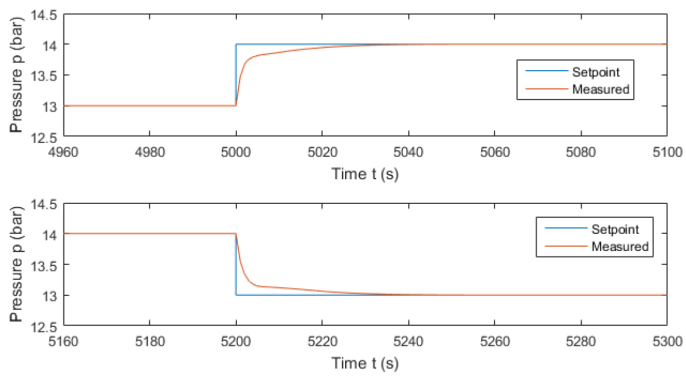

Fig. 5 Simulation of control of heat pump high side pressure using condenser fan

In Fig. 6 there is shown a dependency of COP and overall COP (includes power consumption of both the fans) on high side pressure. The data were acquired using simulations under constant conditions (ambient temperature of

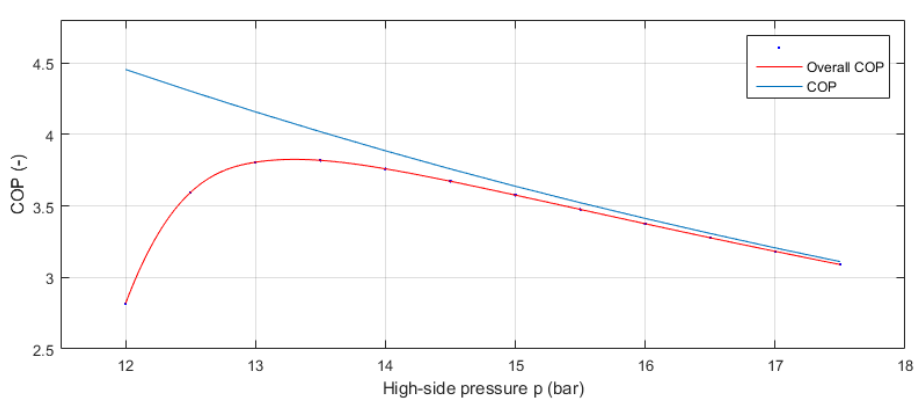

Fig. 6 Simulated COP and overall COP dependency on high side pressure

40 °C, vehicle cabin temperature 30 °C), only the mass flow rate of the air through condenser was variated (using speed control of condenser fan). A significant difference between COP and overall COP can be seen especially for lower pressure (high speed of fan) and decrease of both the COP and overall COP is evident for higher pressure (low condenser fan speed).
The goal of condenser fan efficient control is to keep the overall COP as higher as possible. The simulations

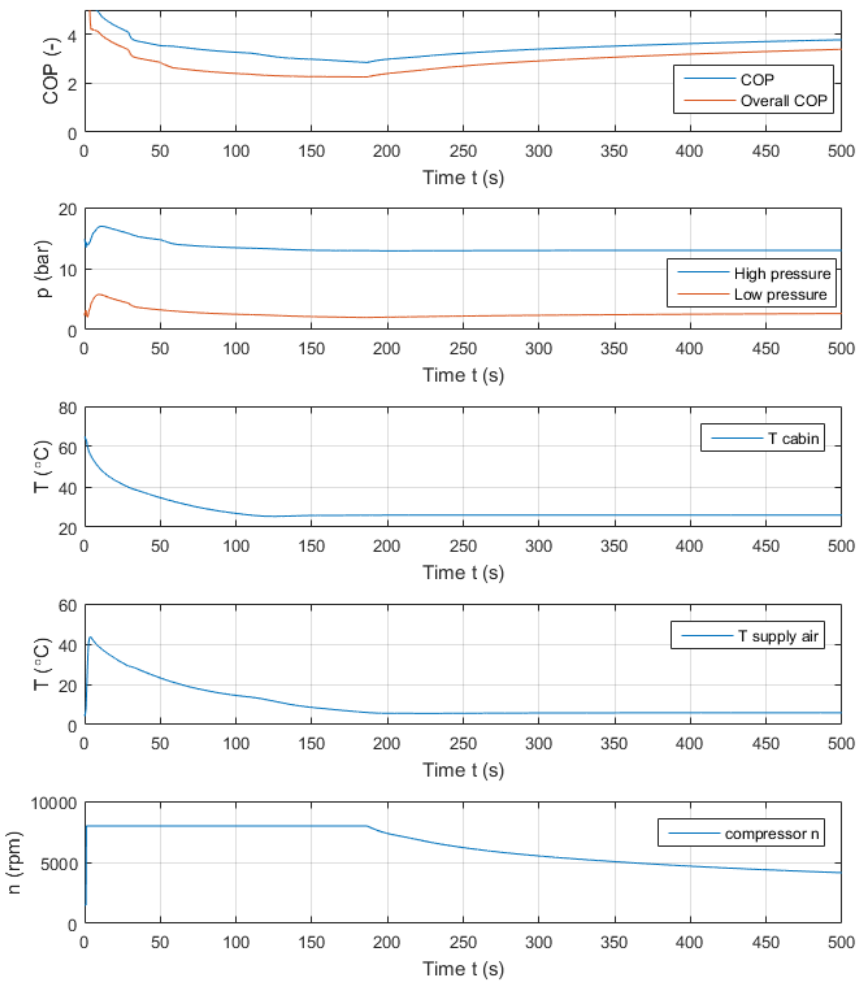

Fig. 7 Overall heat pump control simulation

confirm that our approach with keeping constant condenser split of 10 K can be a good candidate for energy efficient control. An associated saturated pressure for temperature 50 °C (ambinet temperature plus 10 K) is 13,179 bar, what fully fits to overall COP extremum in Fig. 6.

4.3. Overall heat pump control

In Fig. 7 there is result of overall heat pump control simulation. The cabin with initial temperature of 65 °C is cooled down to 26 °C. Ambient temperature was 40 °C and there were medium heat gains into the cabin (two passengers and approx. 1400 W solar flux).

The control algorithms can react on setpoint changes as can be seen in Fig. 8. In the first graph, a cabin temperature setpoint and measured temperature are shown, in the second graph the temperature of air supplied into cabin is depicted. The cabin temperature setpoint is quite well tracked and the temperature of air supplied into cabin is almost kept constant.

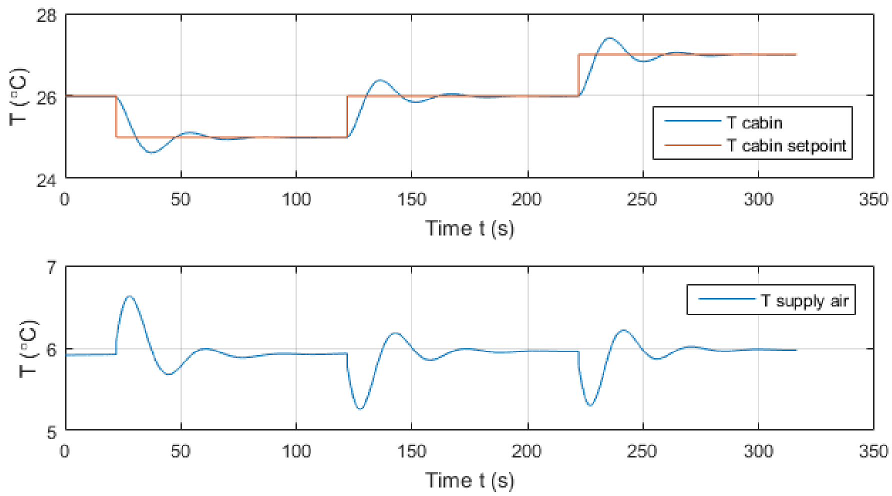

Fig. 8 Demonstration of cabin temperature control capabilities

4.4. Measurements

In Fig. 9 there is a first result of measurement on recently constructed heat pump test bench. We are working on algorithms implementation into AURIX microcontroller to allow their evaluation and verification in real operation.

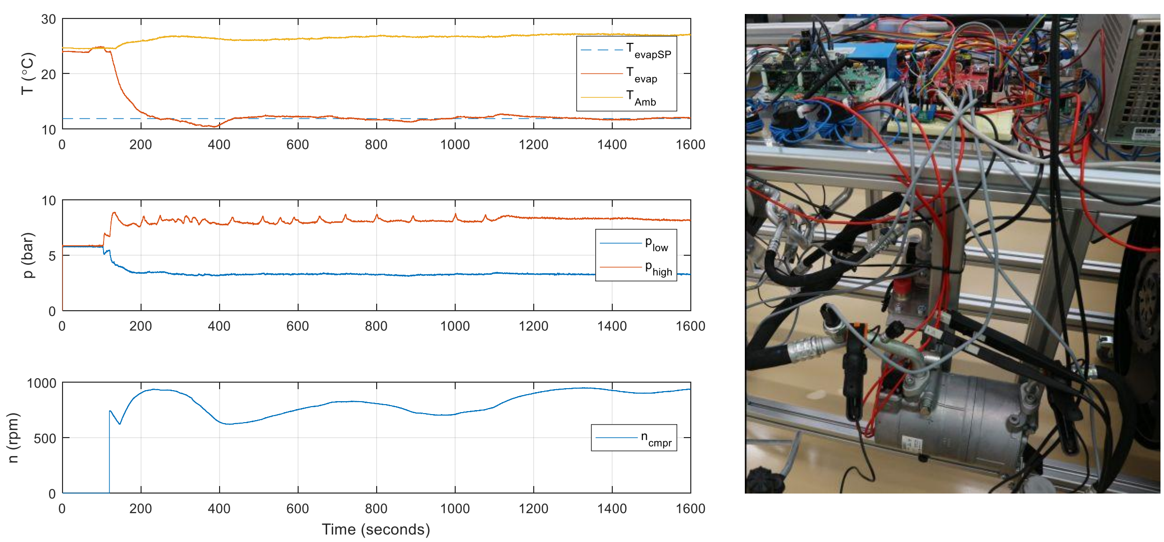

Fig. 9 Heat pump test bench (a) first measurements (b) photograph

5. Conclusion

This paper provides a comprehensive overview of possible control approaches for heat pumps in fully electric vehicles. Different control loops were examined and the best of them were reported in Section 3 and evaluated in simulations and using test bench (Section 4.4). Effective heat pump control can bring significant savings of energy and thus can provide longer vehicle range (especially under worse ambient conditions).

We demonstrated the results of simulations and measurements on heat pump in cooling mode (as we were able to operate the heat pump test bench only in cooling mode at the time of the paper preparation), however the behavior will be very similar and the control approach will be analogous in heating mode, only the overall heat flow direction would be changed (and also heat exchangers will be different, air to water heat pump is being prepared).

We developed a set of control algorithms, which can drive both R1234yf and R744 heat pumps and HVAC equipment in such a way that energy consumption of overall system is as minimal as possible. It can be also said that the highest possible COP needs to be achieved. These algorithms are able to deal with variable FEV velocity as well as with the changes of ambient temperature and user requested temperature in FEV cabin. Also, cooperation of heat pump control algorithms with overall high-level management of energy flows was formulated to achieve a maximal prolongation of FEV driving range.

This paper did not aim to cover all the aspects of energy optimal control of heat pump and HVAC system for FEV as this is very extensive topic (air recirculation, waste heat recovery, heat pump reversation etc.). Only some parts were selected and presented within this paper. Further work will be aimed to evaluation and verification of control algorithms in real experiments and detailed testing and tuning of control algorithms in both the cooling and heating modes.

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