

## Thermal analysis of car air conditioning

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**Abstract** Thermodynamic analysis of car air cooler is presented in this paper. Typical refrigerator cycles are studied. The first: with uncontrolled orifice and non controlled compressor and the second one with the thermostatic controlled expansion valve and externally controlled compressor. The influence of the refrigerant decrease and the change of the air temperature which gets to exchangers on the refrigeration efficiency of the system; was analysed. Also, its effectiveness and the power required to drive the compressor were investigated. The impact of improper refrigerant charge on the performance of air conditioning systems was also checked.

**Keywords:** Thermal analysis; HVAC; Air conditioning

### Nomenclature

$AC$	–	air conditioning
$COP$	–	coefficient of performance
$h_3$	–	refrigerant enthalpy at the evaporator inlet, kJ/kg
$h_4$	–	refrigerant enthalpy at the evaporator outlet, kJ/kg
$\dot{M}$	–	refrigerant factor mass flow rate, kg/s
$\dot{Q}_{evap}$	–	rate of heat in evaporator, W
$\dot{Q}_{comp}$	–	rate of heat in compressor, W

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## 1 Thermal comfort in car cabin

The aspect of proper thermal condition inside the compartments is seriously treated nowadays. The techniques of achieving quite good effects of the thermal condition are well organized in the buildings, but in mobile spaces, such as cars, trains and buses they are still being developed. Temperature in a vehicle cabin is closely related to the occurrence of traffic accidents [1]. Zlatoper [2] has created the ranking list of the factors which affect the traffic accidents in United States and placed the cabin temperature on the third position. On hot summer days the internal temperature often exceeds  $+40\text{ }^{\circ}\text{C}$ . So it is obvious that the thermal conditions in the cabin of vehicles directly influence the driver's and passengers' safety. Both too high and too low ambient temperature influences human physical and mental state. Research into driver's efficiency indicate that it can be even 35% higher at  $+20\text{ }^{\circ}\text{C}$  than at  $+35\text{ }^{\circ}\text{C}$ . Decrease of efficiency at  $+5\text{ }^{\circ}\text{C}$  can be the same as that at  $+35\text{ }^{\circ}\text{C}$  [2,3].

There are also additional parameters, which influence the thermal comfort: air flow speed, air humidity, outer wall temperature and, what is important in the vehicles cabins: sun radiation. In many cases thermal parameters are controlled by regulation of the air temperature and the mass flow rate, only. According to this air flow speed can locally exceed its reasonable value.

It is hard to strictly define the thermal comfort. Usually thermal comfort means that temperature is between  $20\text{ }^{\circ}\text{C}$  and  $22\text{ }^{\circ}\text{C}$ , humidity is about 50% and air velocity is under  $0.5\text{ m/s}$ . These can be called the independent factors.

There is also a second group of factors affecting thermal comfort – individual human feelings, which are much harder to define, because each person has his or her own preferences for thermal comfort. One may say there is a thermal comfort in a certain situation while at the same time the other states “I fell bad here” [4]. The symptoms of thermal discomfort are an intensive sweat production, increase of heart beat frequency, and as a result, the decrease of driver's concentration and efficiency.

The investigations presented in this paper are a part of a larger project, comprising also the CFD (Computational Fluid Dynamics) model of car interior. The first part should answer the question how the air mass flow rate distribution at the inflow to the cabin influences the temperature and air flow inside the cabin, especially in the passengers surroundings. The full numerical model of cabin and the numerical simulations will be helpful in

optimization of air inlets, air mass flow rate, humidity conditions etc. with the respect to the thermal comfort. In the paper the very early results of the simplified model are presented. The second part is a thermodynamic modelling of air cooling unit in order to estimate the influence of basic cabin parameters on the air conditioning (A/C) unit *COP* (Coefficient of Performance), power consumption of the unit and fuel consumption of the vehicle. Different types of working fluid will be tested. In the paper the initial results of this part are shown. The A/C unit model and CFD model are assumed to be coupled. In the third part of the project the experimental verification of both models will be performed.

## 2 Thermal analysis of refrigerator cycle

The rate of heat which should be transferred out of the car cabin is about 2 kW [6,7]. Heat balance for the car cabin is shown in Fig. 1.

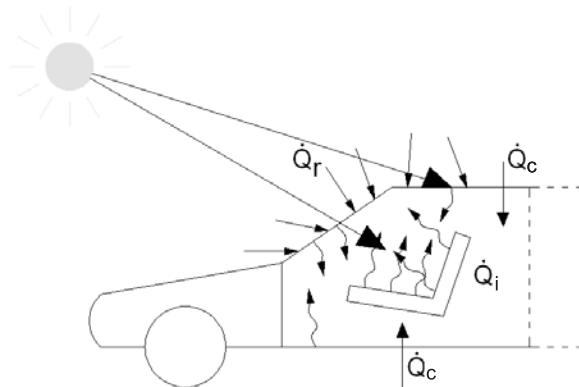


Figure 1. Heat balance for the car cabin:  $\dot{Q}_r$  – radiation rate of heat,  $\dot{Q}_c$  – convection rate of heat,  $\dot{Q}_i$  – internal rate of heat.

An optimum A/C unit should assure thermal comfort under the time varying thermal loads with minimal energy consumption. Compressor in the unit is driven by the vehicle engine and therefore considerably increases the fuel consumption. In the paper two types of unit are considered. The first with an uncontrolled orifice and uncontrolled compressor (a fixed piston displacement), the second with thermostatic controlled expansion valve assuring 1 K superheating of refrigerant at compressor inlet and an externally controlled compressor (a piston displacement from 60 to 120 cm<sup>3</sup>). Both

cases work with the compressor speed 1000 and 3000 rpm. Nominally refrigerant charge (medium – R134a) is 0.44 kg. The charges 0.22 kg (50% nominal) and 0.055 kg (12% nominal) were also considered. Air temperature changes from 20 °C to 45 °C. The refrigerator scheme is shown in Fig. 2. The commercial software Kuli was used [9] for the simulations.

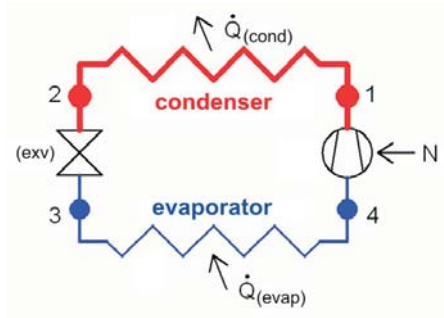


Figure 2. Scheme of refrigerator.

As it was mentioned, the aim of the investigations was to influence the air A/C unit parameters by the refrigerant charge and inlet air temperature on the air A/C unit parameters:  $COP$ , heat transferred in evaporator ( $Q_{evap}$ ), refrigerant temperature inside evaporator and compressor driving power ( $Q_{comp}$ ). The parameters are defined in the following way:

$$COP = \frac{\dot{Q}_{evap}}{\dot{Q}_{comp}}, \quad (1)$$

$$\dot{Q}_{evap} = \dot{M}(h_4 - h_3), \quad (2)$$

$$\dot{Q}_{comp} = \dot{M}(h_1 - h_4), \quad (3)$$

where  $COP$  denotes coefficient of performance,  $\dot{Q}_{evap}$  evaporator rate of heat,  $\dot{Q}_{comp}$  equivalent of the rate of heat required to drive the compressor,  $\dot{M}$  refrigerant factor mass flow rate,  $h_3$  and  $h_4$  refrigerant enthalpy at the evaporator inlet and outlet respectively.

### 3 Uncontrolled cycle

In the uncontrolled case the system of parameters depends on the orifice throttle area diameter. If the area is too big the compressor works properly

(without fluid droplets) only in some ranges. If the effective throttle area is too low the refrigerant at evaporator outlet will always be superheated but the temperature at the compressor outlet may be too high. Such high temperature involves other problems. The first one is the higher compressor material durability, the second is a problem with liquid phase at the condenser outlet, especially at higher ambient air temperature. When the refrigerant is not liquid, or more precisely, the subcooled liquid, efficiency of the entire system decreases significantly. In this case, at ambient temperature of 20 °C without regard to the refrigerant charge, COP is about 5 for 3000 rpm and 3 for 1000 rpm. If decreases from the value of 1 at 45 °C ambient air temperature. For 0.055 kg charge the drop is about 3 for 1000 rpm and 2 for 3000 rpm. It is shown in Fig. 3.

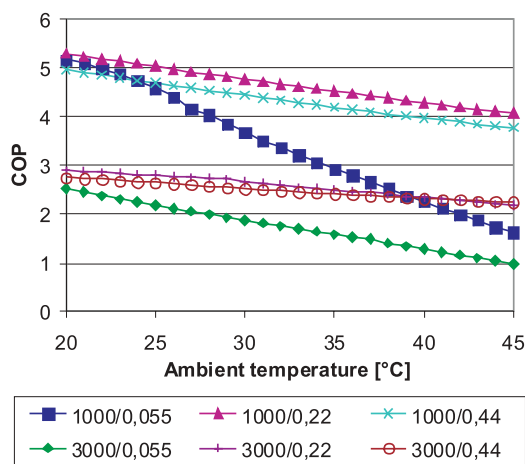


Figure 3. *COP* as a function of ambient temperature (R134a).

Similar trend can be observed with heat transferred in evaporator. For the charge 0.44 kg and 0.22 kg, the rate of heat increases with air temperature. It is shown in Fig. 6. For 45 °C the rate of heat is two times higher than for 20 °C. In the case of the charge 0.055 kg the situation is opposite – heat flux decreases with air temperature.

There is one advantage of 0.055 kg charge case: compressor power is low, at 3000 rpm is below 1 kW while for 0.44 kg it is twice higher. In such case it is almost impossible to achieve the required temperature inside the car cabin in this case. Heat transferred in the evaporator is simply too low (Fig. 4).

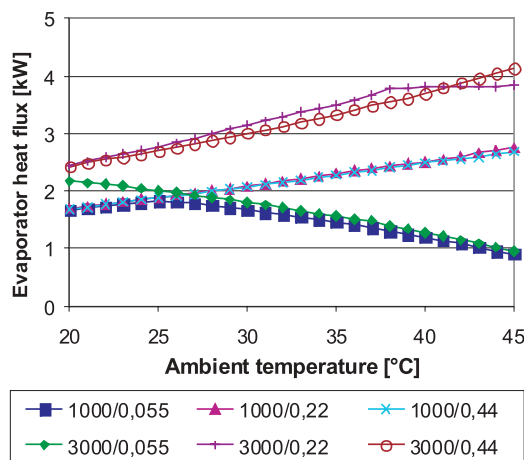


Figure 4. Evaporator heat flux as a function of ambient temperature (R134a).

## 4 Controlled cycle

In the controlled case there is no problem with too high temperature, because the expansion valve always prevents 1 K from the superheating. So, there is no problem with high temperature material durability because temperature at the compressor outlet is lower too. If the temperature of the compressor outlet is lower it is also easier to obtain the liquid phase at the condenser outlet.

One can say the controlled cycle is more “flexible”. In this case the rate of heat in the evaporator can be about 1.5 kW higher than in the uncontrolled one, which means that we can reject 1.5 kW of heat flux more from the car compartment. For 0.055 kg charge, compressor inlet (evaporator outlet) temperature is about 40 °C at 45 °C air temperature. It means that temperature at compressor outlet can be about 100 °C.

For other charges inlet temperatures are similar and always below 15 °C. *COP* tendency is similar to the uncontrolled cycle but the values are a little bit lower (Fig. 5). The compressor power is always higher than in the case of uncontrolled cycle. There is also a higher rate of heat in evaporator for both 1000 and 3000 rpm cases (Fig. 6). For the charge 0.055 kg *COP* is equal to 0.25 at 45 °C ambient air temperature.

Comparing the cases one can say that uncontrolled case is better than the controlled one. *COP* values are higher for uncontrolled, compressor

power is lower (lower fuel consumption), but on the other hand the rate of heat in evaporator is lower and refrigerant temperature is higher. Additionally it should be emphasised, that for the charge of 0.055 kg the compressor works only with vapour phase. For the charge of 0.11 kg it works with the vapour phase above 35 °C air temperature. For the charge 0.44 kg the liquid always appears in compressor.

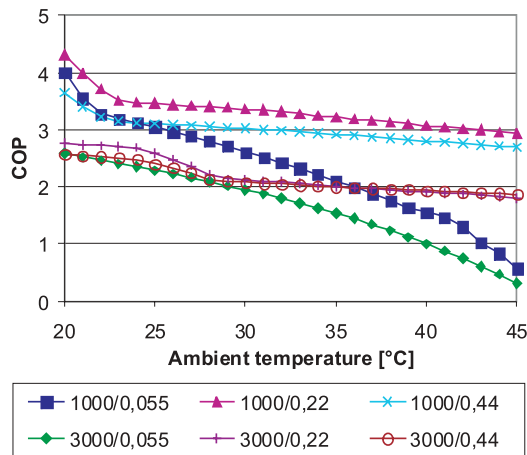


Figure 5. COP as a function of ambient temperature (R134a).

All the simulations show how important is the proper refrigerant charge in A/C systems. It is impossible to estimate the amount of refrigerant in the cycle without special measuring instruments. The only case in which we can say that the refrigerant charge is too low is when the outlet air temperature from A/C is too high.

Air-conditioning cycle with CO<sub>2</sub> (R744) as refrigerant was also considered. In this case the refrigerant is in supercritical phase between the compressor outlet and the expansion valve inlet. Pressure levels are also different. At the compressor outlet it is up to 200 bar (about 15 bar for R134a cycle) and at the evaporator outlet is about 80 bar (about 1.5 bar for R134a). This cycle can also be called the “controlled cycle” because expansion valve always achieves 1 K superheating at the evaporator outlet.

It can be noticed that evaporator capacity is much lower for 0.075 kg charge both, at 1000 and at 3000 rpm. The difference increases with the air temperature. Capacity is over twice lower for 0.075 kg than for other charges. It is almost impossible to achieve the required temperature inside

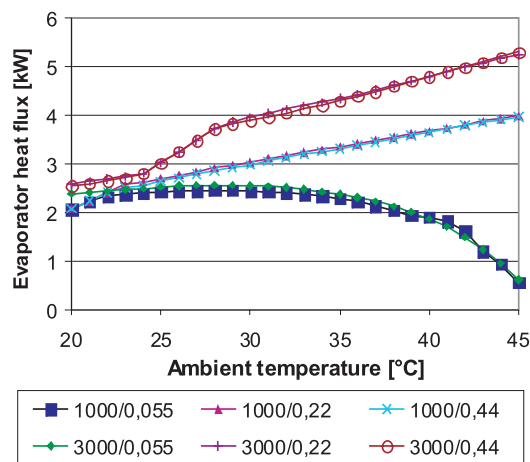


Figure 6. Evaporator capacity as a function of ambient temperature (R134a).

car cabin in this case. The heat transfer in the evaporator is just too low. It is shown in Fig. 7. *COP* value is similar for all charges in the whole air temperature range. Except for 0.075 kg, it is between 1.5 and 2. This value is lower than R134a cycle but still over 1, even for 45 °C air temperature. For 0.075 kg charge *COP* value is near 1 at 1000 rpm and decreases to 0.5 at 3000 rpm (Fig. 8).

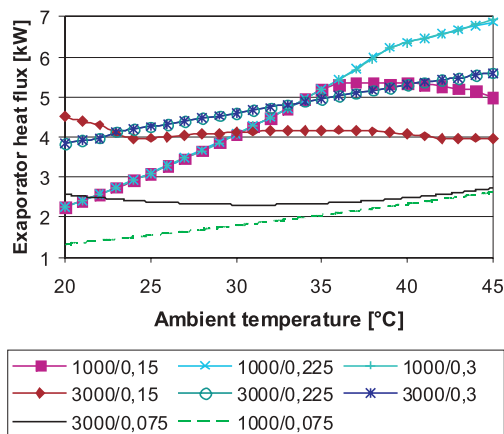


Figure 7. Evaporator capacity as a function of ambient temperature (R744).



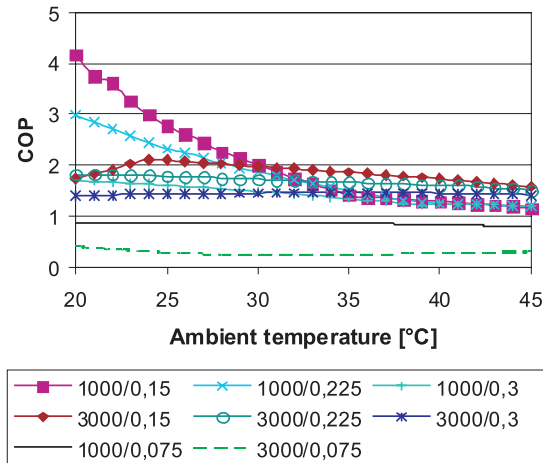


Figure 8.  $COP$  as a function of ambient temperature (R744).

## 5 Conclusions

In the paper the thermal analysis of A/C unit cycle is presented. It can be noticed that the decrease of the refrigerant charge decreases  $COP$  for both considered refrigerants i.e. R134a and  $CO_2$ . For R134a case, the decrease is higher in controlled cycle because the temperature in evaporator is higher which causes the lower rate of heat in evaporator and higher compressor power. Beneficial is the fact that the compressor works always with the gas phase. For non controlled R134a cycle efficiency is higher but the hazard of compressor work with liquid appears. For  $CO_2$  case  $COP$  values are at the same level for all temperatures, except for 0.15 kg charge when it decreases significantly with the air temperature. It can be noticed that  $COP$  values are lower than for R134a cycle. In all cases  $COP$  is lower when the refrigerant charge is below its nominal value.

There is one advantage of the usage  $CO_2$  as a refrigerant over R134a. Global Warming Potential (GWP), which is a measure representing potential of a substance to contribute to the global warming, is 1300 for R134a and only 1 for  $CO_2$ . It means that 1 kg R134a emission is equal to 1300 kg of  $CO_2$ . The Ozone Depletion Potential is zero in both cases.

Car units are not so hermetic as home air conditioning devices. If the fact how many cars are equipped with A/C systems will be taken into con-

sideration, the emission from mobile systems is going to be a significant part in the global green gas emission.

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