Air Conditioning without Refrigerant

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The requirement of air conditioning in modern passenger cars need to fullfill enviromental standards with respect to refrigerant concerning global warming potential and ozone depletion with flour chlor carbon hydrogen mixtures. This leads to compromises with respect to energy consumption and cost of components. As an alternative to standard refrigeration cycles, an inexpensive direct air cooling process is presented and discussed with respect to performance. The process is specially suited for electric vehicles or vehicles equipped with an electric boosting device (E-charger).

Modern passenger cars are equiped with air conditioning units allowing a comfortable transport and fullfilling safety issues like windscreen defogging. Todays vehicles rely on the use of a standard refrigeration cycle making use of the phase change properties of the refrigerant. Refrigerants such as R22 containing flour chlor carbon hydrogen components are prohibited since the 1990's due to their ozon depletion potential. Refrigerants as R134a are prohibited in vehicles due to global warming potential when leaking from the system. Therefore modern refrigeration cylcles rely on new refrigerants as R1234yf having the drawback of beeing flameable or R744 (CO2) requiring high compression ratios. This leads to the idea of proposing a refrigerant free cooling process compatible especially with modern electric vehicles with respect to weight. The proposed cooling process uses the air to be supplied to the inner of the vehicle directly without any secondary medium.

1 The basic cooling process

The basic cooling cycle consists of three components. The first component is a compressor that rises the pressure of the air. Due to compression, the air temperature rises. In the second component,a following heat exchanger, the compressed air is cooled down as close as possible to ambient temperature. The third component consists of a turbine. In the expansion process in the turbine, work is done by the air on the turbine. This causes a temperature drop of the air. This air can than directly be applied to the vehicle compartment. A sketch of the process is given in fig. 1. In the thermodynamic descrip-

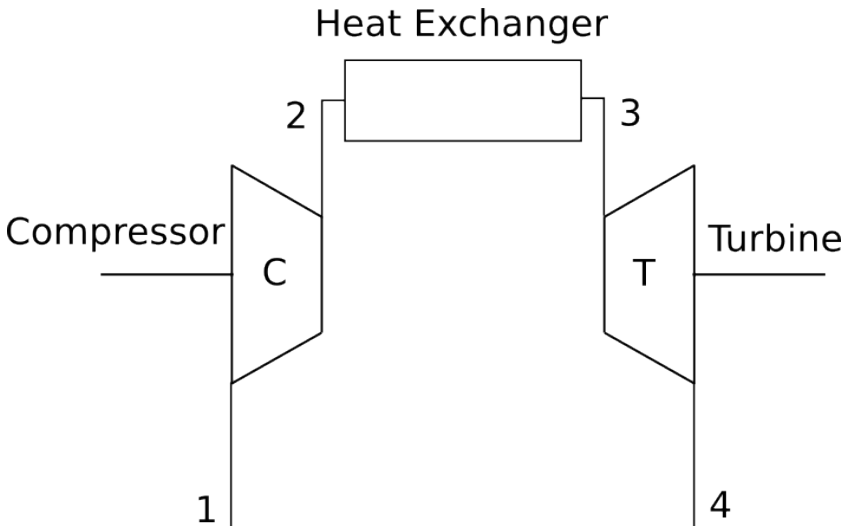

Figure 1: Sketch of the simple cooling process

tion of the process, computations are simplified by assuming air to be an ideal, caloric perfect gas ($p = \rho RT$,$h = c_p T$, $R = 287\,J/kgK$, $\kappa = c_p/c_v = 1.4$, $c_p = 1004\,J/kgK$).

1.1 Process description

The process can be summarized in the following steps.

1. Compression in a compressor with the necessary specific work w_{t12}. The compression ratio is $\Pi_c = p_2/p_1$.

$$w_{t12} = \frac{1}{\eta_C} RT_1 \left(\frac{\kappa}{\kappa - 1}\right) \left(\Pi_c^{(\kappa-1)/\kappa} - 1\right) \tag{1}$$

   As the work is supplied to the air, w_{t12} is positive in the used convention.

2. Cooling the air in heat exchanger with the specific heat q_{23}.

$$q_{23} = c_p \left(T_3 - T_2\right) \tag{2}$$

   As heat is removed from the air, q_{23} is negative.

3. Expansion in turbine with the specific work w_{t34}. The expansion ratio is $\Pi_t = p_3/p_4$.

$$w_{t34} = \eta_t RT_3 \left(\frac{\kappa}{\kappa - 1}\right) \left(\Pi_t^{(1-\kappa)/\kappa} - 1\right) \tag{3}$$

   As work is done by the air on the turbine, w_{t34} is negative.

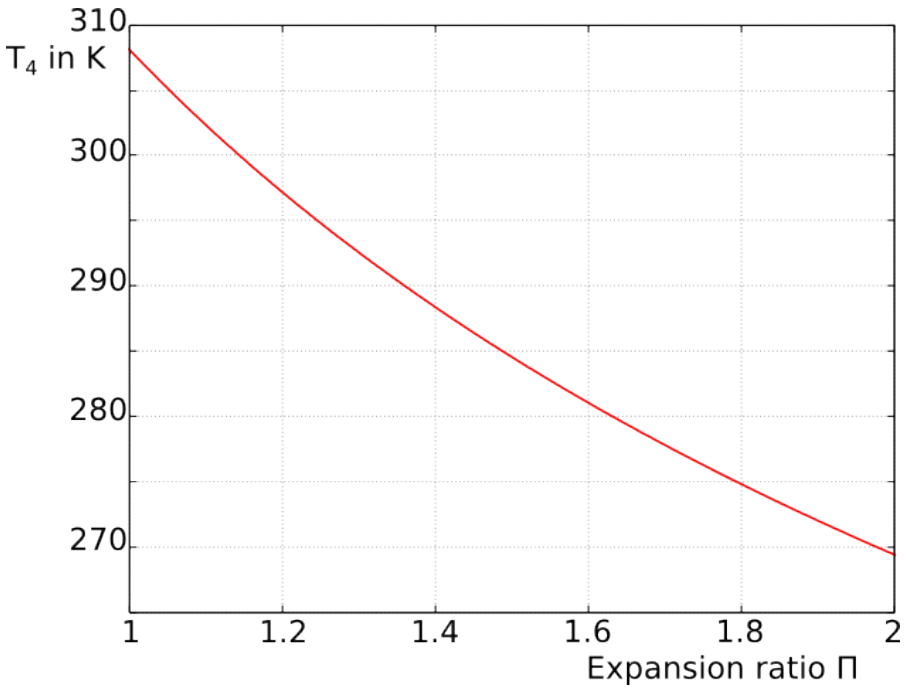

Figure 2: Cooled air temperature in Kelvin as a function of expansion ratio with $\eta_t = 0.7$,$T_3 = 35°C = 308.15K$

This process is not a closed cycle and the real process includes some pressure losses (Δp_{Cooler}) in the heat exchanger. Neglecting the pressure losses in a first analysis, pressure ratio of the compressor $\Pi_c = p_2/p_1$ and expansion ration of the turbine $\Pi_t = p_3/p_4$ are identical. The temperature after the heat exchanger (T_3) depends on the cooling performance of the heat exchanger.Typically this is slightly larger than the ambient temperture and the difference can be limited to roughly (5-10 K).

1.2 Cooling Temperature

With the isentropic efficiency definition for the turbine the air temperature after expansion only depends on the turbine inlet temperature T_3, turbine efficiency η_t and the turbine expansion ratio Π_t.

$$T_4 = T_3 \left(1 - \eta_t \left(1 - \Pi_t^{(1-\kappa)/\kappa}\right)\right) \tag{4}$$

The temperature after expansion depends on temperature after the cooler T_3, turbine efficiency η_t and the expansion ratio Π_t. To reach the best cooling, T_3 should be as low as possible and η_t as large as possible. Then cooling can be achieved with the lowest expansion ratio Π_t. Assuming the efficiency of the turbine to $\eta_t = 0.7$ and the temperature of the turbine inlet to be $T_3 = 35°C = 308.15K$, the temperature at turbine exit is given in fig. 1.2 as a function of turbine expansion ratio. With the assumed

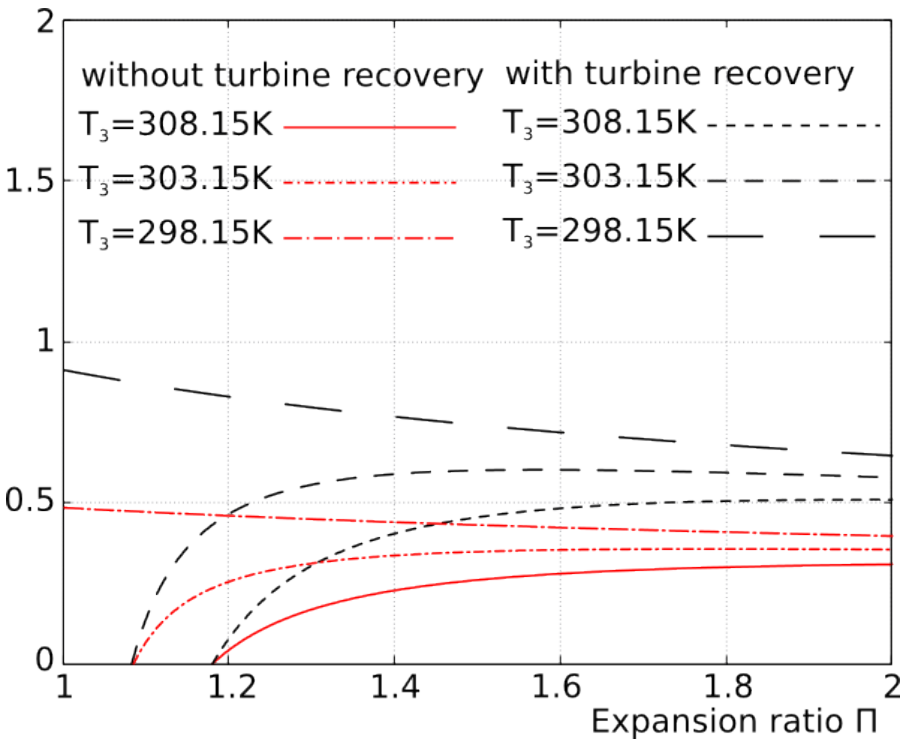

Figure 3: coefficient of performance as a function of expansion ratio with $\eta_c = 0.7$,$\eta_t = 0.7$,$T_1 = 298.15\,K$

parameters, an expansion ratio of 1.4 is sufficient to cool the air down to temperatures required for the cabin in most conditions.

1.3 Coefficient of performance

The performance of the process is given by the ratio of the specific heat removed from the air, compared to the specific energy necessary to drive the compressor.

$$\begin{aligned} \varepsilon_1 &= \frac{|c_p(T_4 - T_1)|}{w_{t12}} \\ &= \eta_c \frac{1 - T_3/T_1(1 - \eta_t(1 - \Pi^{(1-\kappa)/\kappa}))}{\Pi^{(\kappa-1)/\kappa} - 1} \end{aligned} \tag{5}$$

The performance of the process can be significantly improved, if the work done by the turbine is used to drive the compressor by coupling. The coupling can be mechanical or electrical. We do not specify the coupling and simply propose a coupling efficiency of η_m.

$$\varepsilon_2 = \frac{|c_p(T_4 - T_1)|}{w_{t12} + \eta_m w_{t34}} \tag{6}$$

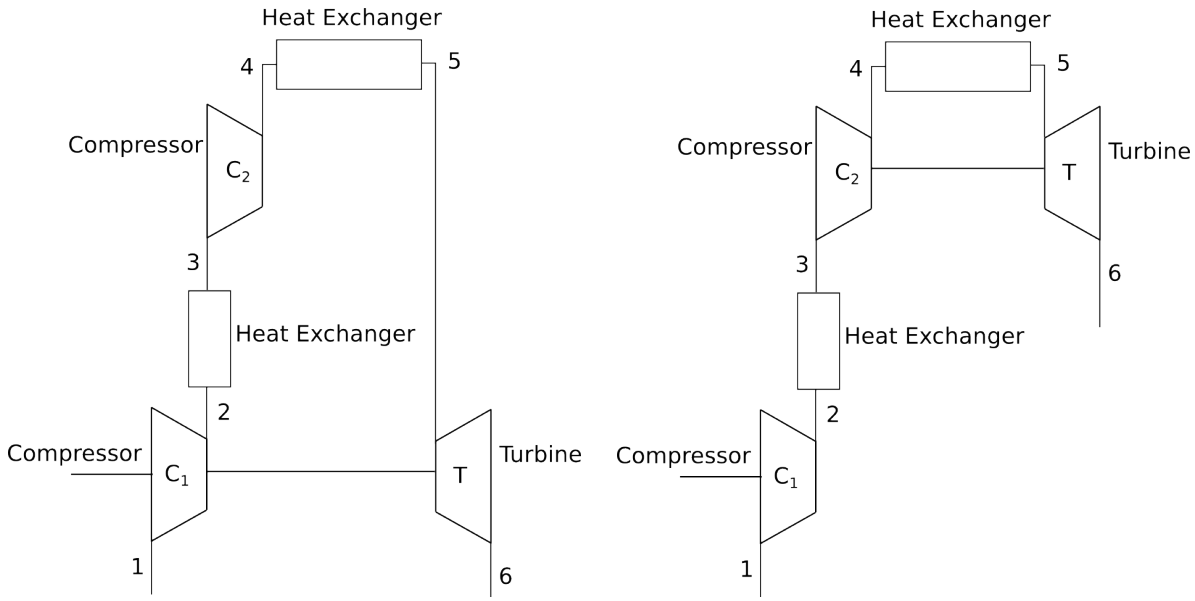

Figure 4: Two stage systems with primary compressor driven by the turbine (system A, left) and two stage system with secondary compressor driven by the turbine (system B, right)

$$= \frac{1 - T_3/T_1(1 - \eta_t(1 - \Pi^{(1-\kappa)/\kappa}))}{\eta_c^{-1}(\Pi^{(\kappa-1)/\kappa} - 1) - \eta_m \eta_t T_3/T_1(1 - \Pi^{(1-\kappa)/\kappa})}$$

The coefficient of performance of the system is illustrated in fig. 1.3. It shows that the coeficient of performance is well below the achieved coefficients of common refrigeration cylcles.

2 Advanced two stage systems

Since very small radial turbines turn at very high revolutions to achieve the necessary circumferential speed, driving the compressor with such speed and power is a challange itself. One possibility is to use the work recovered by the turbine in a seperate compressor. This leads to a two stage compression system with two compressors and a single turbine. There are two possibilites to couple the turbine to the compressor in a two stage system. In the first case (System A) the turbine is coupled to the first compressor and in a second case (System B) the turbine is coupled to the second compressor.

2.1 Description of two stage process (System A)

System A as illustrated in fig. 2 can be summarized by the following steps.

1. Compression in a first compressor with the necessary specific work w_{t12}. The

compression ratio is $\Pi_{c1} = p_2/p_1$.

$$w_{t12} = \frac{1}{\eta_{c1}} RT_1 \left(\frac{\kappa}{\kappa - 1}\right) \left(\Pi_{c1}^{(\kappa-1)/\kappa} - 1\right) \tag{7}$$

As the work is supplied to the air, w_{t12} is positive in the used convention.

2. Cooling the air in heat exchanger with the specific heat q_{23}.

$$q_{23} = c_p \left(T_3 - T_2\right) \tag{8}$$

As heat is removed from the air, q_{23} is negative. Intercooling the air can improve the overall efficiency of the system. For a simplification of the system this step can be omitted.

3. Compression in a second compressor with the necessary specific work w_{t34}. The compression ratio is $\Pi_{c2} = p_4/p_3$.

$$w_{t34} = \frac{1}{\eta_{c2}} RT_3 \left(\frac{\kappa}{\kappa - 1}\right) \left(\Pi_{c2}^{(\kappa-1)/\kappa} - 1\right) \tag{9}$$

As the work is supplied to the air, w_{t34} is positive.

4. Cooling the air in heat exchanger with the specific heat q_{45}.

$$q_{45} = c_p \left(T_5 - T_4\right) \tag{10}$$

As heat is removed from the air, q_{45} is negative. This cooling step is essential for a well working system and can not be omitted.

5. Expansion in turbine with the specific work w_{t56}. The expansion ratio is $\Pi_t = p_5/p_6$.

$$w_{t56} = \eta_t RT_5 \left(\frac{\kappa}{\kappa - 1}\right) \left(\Pi_t^{(1-\kappa)/\kappa} - 1\right) \tag{11}$$

As work is done by the air on the turbine, w_{t56} is negative.

As in the cases of the simple process, this process is not a closed cycle and the real process includes some pressure losses (Δp_{Cooler}) in the two heat exchangers. Neglecting the pressure losses in a first analysis, pressure ratio of the compressors $\Pi_{c1} = p_2/p_1$, $\Pi_{c2} = p_4/p_3$ and expansion ration of the turbine $\Pi_t = p_5/p_6$ are linked by $\Pi_t = \Pi_{c1}\Pi_{c2}$. The temperature after the heat exchangers (T_3,T_5) depend on the cooling performance of the heat exchangers.

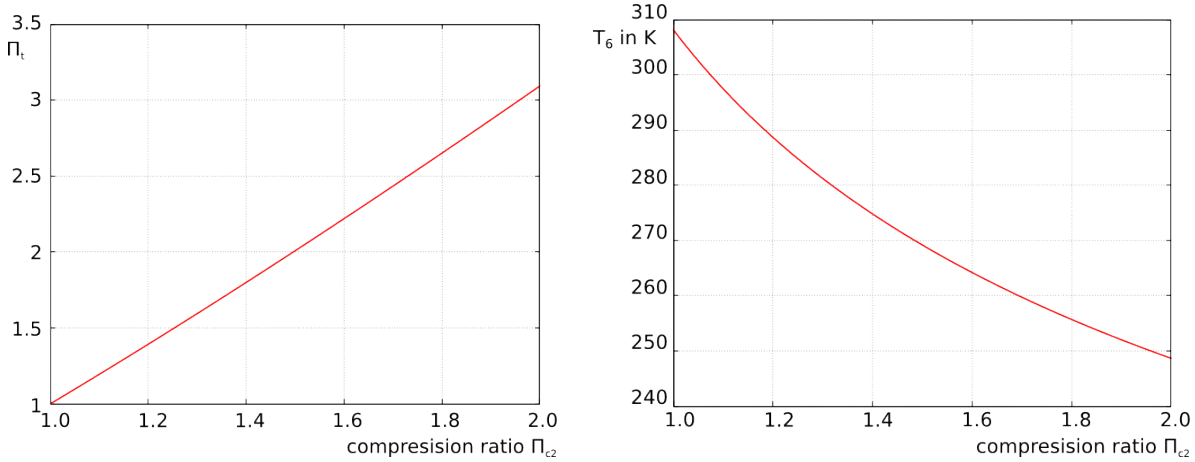

Figure 5: Turbine expansion ratio as a function of secondary compressor compression ratio (left) and cooled air temperature in Kelvin as a function of compression ratio of secondary compressor Π_{c2} (right) with $\eta_t = 0.7$,$T_3 = 35°C = 308.15K$

2.2 Cooling Temperature

For the cooled air temperature the identical relation as in the case of the simple system holds true.

$$T_6 = T_5 \left(1 - \eta_t \left(1 - \Pi_t^{(1-\kappa)/\kappa}\right)\right) \tag{12}$$

The expansion ratio depends on the compression ratio of the second compressor Π_{c2} and is obtained by combining the power relation $w_{t12} = -\eta_m w_{t56}$ and the pressure ratio relation $\Pi_t = \Pi_{c1}\Pi_{c2}$ neglecting the pressure losses in the system.

$$\Pi_t = \left[\frac{1}{2}\Pi_{c2}^{(\kappa-1)/\kappa}(1 + \eta_c\eta_m\eta_t T_5/T_1) + \sqrt{\frac{1}{4}\Pi_{c2}^{2(\kappa-1)/\kappa}(1 + \eta_c\eta_m\eta_t T_5/T_1)^2 - \eta_c\eta_m\eta_t T_5/T_1 \Pi_{c2}^{(\kappa-1)/\kappa}}\right]^{\kappa/(\kappa-1)} \tag{13}$$

The cooling temperature can then be expressed as a function of the compression ratio Π_{c2} of the externally driven second compressor. The turbine expansion ratio is shown on the left in fig. 2.2 as a function of the compression ratio of the secondary compressor compression ratio. A secondary compressor compression ratio of 1.2 already leads to a total expansion ratio of roughly 1.4 including the compressor efficiencies. This leads to a significant temperature drop in the turbine as shown on the right side of fig. 2.2.

2.3 Coefficient of performance

The performance of the process is given by the ratio of the specific heat removed from the air, compared to the specific energy necessary to drive the secondary compressor (C2).

$$\varepsilon_A = \frac{|c_p(T_6 - T_1)|}{w_{t34}} \tag{14}$$

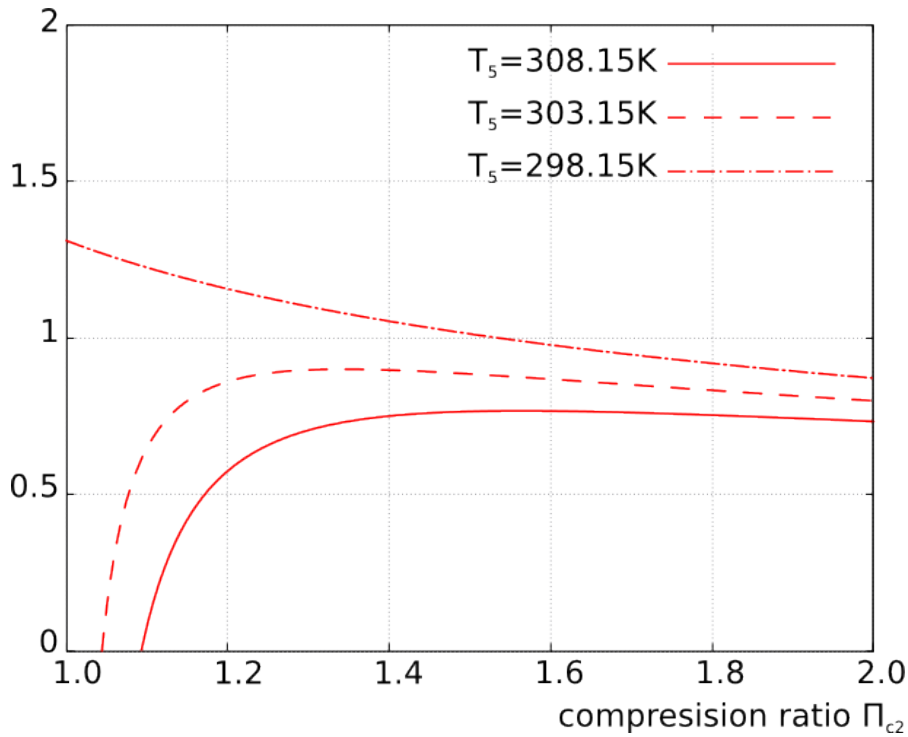

Figure 6: coefficient of performance as a function of secondary compressor compression ratio with $\eta_m = 0.95$,$\eta_c = 0.7$,$\eta_t = 0.7$,$T_1 = 298.15\,K$

The coefficient of performance of the two-stage system is shown in fig. 2.3. This is slightly better then the coefficient of performance of the simple system and due to the intercooling stage of the two stage system. However, the additional pressure loss effects are not taken into account in the figure.

2.4 Extension to System B

The extension to system B is straightforward. The only major change is the coupling condition of the second compressor to the turbine ($w_{t34} = -\eta_m w_{t56}$). This leads to similar results concerning the temperature drop and coefficient of performance as a function of the first compressor pressure ratio Π_{12}. Sytem B is especially suited for vehilces already equipped with a electric boosting charger. Exisiting electric boosters installed to increase driving performance can be used to drive the air conditioning system. This only requires the installation of additional valves.

3 Choice of materials and components

The arising temperatures are small compared to other turbomachinery working with gases. Therefore temperature does not limit the use of polymer substances for the hous-

ing components. The necessary yield strength for the compressor and turbine materials can be estimated by the square of the circumferential velocity of the compressor u_c and turbine u_t.

$$u_c = \left[c_p T_{in} \frac{1}{\lambda \eta_c} \left(\Pi_c^{(\kappa-1)/\kappa} - 1 \right) \right]^{1/2} \tag{15}$$

For typical applications the with low pressure ratios, the circumferential speed is limited to values well below 250 m/s. As the tensile forces increase with the square of the circumferential speed, the values are small compared to turbocharger compressor tensile strength and even polymer materials as polyamides may be considered as materials with an intelligent compressor wheel design.This opens the possibilities for a cost effective manufacturing of the components.

4 Summary

An alternative system for air conditioning units in mobile applications was presented. The preliminary estimations result in small pressure ratios showing the feasibility of the systems. The coefficient of performance shows to be lower compared to standard refrigeration cycles. However the components of the presented systems are simple and can be manufactured from inexpensive materials. Small packaging and low weight of the components can make such type of systems an inexpensive alternative to standard air conditioning systems. Further investigations and setup of prototypes are necessary to proove the concept and its performance in electric vehicles.