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Replacing batteries with water by an innovative evaporative cooling process for vehicle air conditioning

Replacing batteries with water by an innovative evaporative cooling process for vehicle air conditioning

Nicolò Morselli[a], Michele Cossu[a], Alberto Muscio[a] *

[a] Dip. di Ingegneria "Enzo Ferrari" – Univ. di Modena e Reggio Emilia, Italy
* Corresponding author e-mail: alberto muscio@unimore.it

Abstract. In plug-in hybrid and battery electric vehicles, the air conditioning system can absorb a significant fraction of the energy stored onboard in the battery pack, thus causing a decrease of the range, or the need of additional battery capacity with its associated mass and cost. An alternative to standard air conditioning systems is provided by evaporative cooling. This was used at the dawn of the automotive industry in the form of direct evaporative cooling, however it did not take hold due to the high relative humidity induced in the vehicle cabin by the supply of almost saturated air, with the consequent risk of condensation and health issues. In recent times, Maisotsenko developed an innovative indirect evaporative cooling system, in which the air introduced in the conditioned compartment is not humidified and it can also be cooled down to dew point rather than to the wet bulb temperature thanks to the system architecture. Water consumption is relatively low, so the same cooling energy provided by a vapor compression system powered by a given mass fraction of the battery pack can be obtained by evaporating a comparable mass of water. The approach is investigated here by theoretical means, in order to explore its potential and identify possible critical issues.

1. Introduction

The progressive electrification of road transport is one of the main goal of the European Commission's "A clean planet for all" strategy, which lays the foundations for reducing the use of fossil fuels by encouraging the use of energy mix with a lower environmental impact. In this context, electric mobility is spreading more and more rapidly in the European scenario, differentiating itself mainly according to three different technologies: hybrid electric vehicles (HEVs), plug-in hybrid-electric vehicles (PHEVs) and battery electric vehicles (BEVs) [1]. In HEVs, the internal combustion engine (ICE) is coupled with an electric motor which is powered by a battery pack with capacity of few kWh's. The electric motor actively contributes to the traction of the vehicle by supporting ICE in operating conditions where it is less efficient (*e.g.* frequent start and stop) or when maximum power is required. PHEV vehicles have an architecture similar to HEVs, with the substantial difference that the electric motor is often able to autonomously move the vehicle, that is without contribution of the ICE, for a longer range. To this end, the battery that powers the electric motor has a larger capacity and generally allows a few tens of kilometers to be driven in electric mode, easily covering home-work commuting. Another difference compared to simpler HEVs consists in the possibility of recharging the battery by connecting to the charging points and thus allowing a full-electric use of the vehicle for most of the time. The third architecture is represented by BEVs, which are vehicles without ICE but driven only by one or more electric motors powered by a battery pack with capacity of tens of kWh. This type of vehicle can be recharged from public or private charging stations at high power (over 100 kW), considerably reducing recharging times. HEVs and PHEVs are undoubtedly a means to accompany infrastructures towards a progressive decarbonization of the transport sector, while it is clear that BEVs are the real target in the medium-long term (2035-2050). This transition period is mainly determined by the relative immaturity of battery packs [2,3], which heavily affects the costs of BEVs [4], and their low energy density (100-300 Wh/kg [5]) when compared to fossils fuels [6]. Enormous efforts, public [7] and private [8], are underway to constantly search for new materials and technologies that can reduce the specific cost of batteries, while increasing their energy density and therefore the mileage of a PHEV or a BEV. However, these efforts are often hindered by energy consumption that goes beyond the vehicle's traction alone such as that related to summer air conditioning of the passenger compartment (HVAC), which is difficult to estimate and mainly depend on the ambient conditions and the solar load [9].

The most common HVAC system used in recent vehicles is a vapor-compression refrigeration (VCR) cycle with an electric compressor, whose need of electric energy can significantly reduce the car mileage. Several studies are focusing on the development of energy storage systems based on the accumulation of thermal energy in the form of sensible and latent heat during recharging periods, in order to reduce the use of electricity stored in the vehicle battery [10]. Promising systems are those based on phase change materials (PCMs), however they still suffer from poor thermal conductivity and low energy density. An alternative that has not yet been studied in the automotive sector is represented by indirect evaporative cooling cycles (IECs), in which the latent heat of water evaporation is used to cool a (wet) air flow that is humidified and saturated, but unlike direct evaporative cycles (DEC) this air flow is not introduced in the air conditioned space, in this case the vehicle cabin. Instead, it is introduced in a heat exchanger, where it exchanges heat with, and cools down, another (dry) airflow, whose absolute humidity remains unchanged. Standard IEC systems does not allow to cool the wet and, consequently, the dry air flows below the wet bulb temperature at ambient conditions. In recent times, a promising variation of the IEC scheme, the Maisotsenko cycle (M-Cycle), has been implemented. This theoretically allows cooling the dry air flow down to the dew point temperature at ambient conditions, with coefficients of performance (*COP*) far superior to vapor-compression refrigeration (VCR) cycles [11,12].

In this work, an analytical model developed to calculate the performance of a IEC system based on the M-Cycle was applied to the summer air conditioning of a C-segment vehicle cabin.

2. Materials and methods

2.1. Description of the M-cycle system considered for the scope

The M-Cycle uses the cooling energy available with the latent heat of water which is evaporated into the air. It combines thermodynamic processes of heat transfer and evaporative cooling to facilitate cooling of an air flow. More specifically, it is a modified indirect evaporative cooling system, through which the air can be cooled, potentially, down to the dew point temperature instead of the wet bulb temperature as for the classic direct (DEC) and indirect evaporative systems (IEC). Furthermore, apart from the cooled air, the M-Cycle produces, as waste, a saturated and relatively cool air flow that can be exploited for many uses. The M-cycle in counter-current configuration can be realized in different ways, basically consisting of two types of primary channels: dry channels, in which the ambient air always flows without increasing its moisture content, and wet (humid) channels, in which water in the liquid state is injected in the air flow and evaporates, thus cooling the air flow thanks to the latent heat. The moisture content is also increased, ideally up to saturation conditions. This work investigates a specific configuration of the M-Cycle cycle (Fig. 1), which consists of a heat exchanger with two primary channels, wet channel and dry channel, where the same fluid flows. Ambient air is introduced in the dry channel (1) and then cooled. At the end of the dry channel, a fraction of the cooled air (2) is diverted and conveyed into the wet channel, where it will be humidified. The fraction of diverted air enters the wet channel in a pre-cooled condition with respect to ambient air, as shown in Fig. 2, thereafter it remains relatively cool while it absorbs heat from the air flow in the dry channel, until it is released (3). The sensitive cooling of the dry channel, together with the counter-current configuration, can lead, in an ideal heat exchanger with infinite length, to the coincidence between the temperature of the product air at the outlet of the dry channel and the dew point temperature of the air at ambient conditions. For this reason, the performance of the M-cycle is estimated through the so-called dew point efficiency, defined as the ratio between the difference in dry bulb temperatures at the inlet and outlet conditions in the dry channel compared to the difference in dry bulb temperatures at inlet and the corresponding dew point temperature. Instead, for conventional evaporative cooling systems, the performance is estimated through the wet bulb efficiency, defined as the ratio between the difference in dry bulb temperatures at the inlet and outlet conditions of the in the dry channel compared to the difference in dry bulb temperatures at the inlet and the corresponding wet bulb temperature. An M-Cycle can have a wet bulb efficiency in excess of 100%.

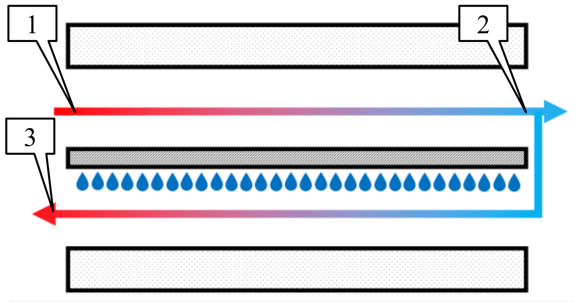

Figure 1. Schematic representation of the M-cycle.

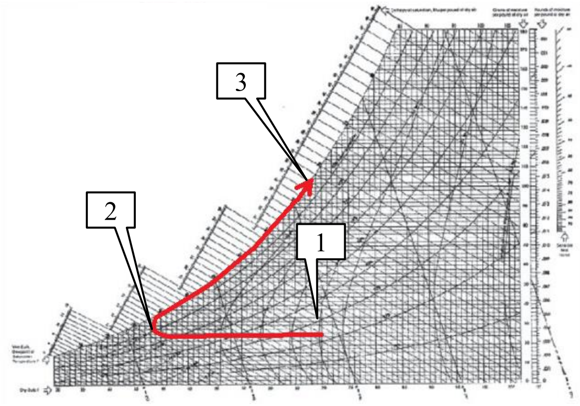

Figure 2. Psychrometric chart representation.

2.2. Analytical model description

Ambient air is sucked into the dry channel of the heat exchanger at temperature $T_{dry,in} \equiv T_{amb}$ (°C) and relative humidity $RH_{dry,in} \equiv RH_{amb}$ (%) (point (1) in Figs. 1-2), then it is cooled down and exits the dry channel at temperature $T_{dry,out}$ (°C) (point (2)). The absolute humidity, or humidity ratio, defined as the mass of water vapor in the unit of mass of dry air, does not change along the dry channel, so that its value at the outlet $x_{dry,out}$ (kg_v/kg_a) is equal to the inlet value $x_{dry,in}$ (kg_v/kg_a). The latter can be calculated from the inlet (*i.e.* ambient conditions).

In the M-cycle, a fraction r (with $0 < r < 1$) of the cooled air at temperature $T_{dry,out}$ is diverted into the wet channel, and the remaining fraction $(1 - r)$ is the product air supplied to the cooled space. The ideal dew point efficiency is generally estimated by numerical modelling of the heat exchanger. However, an alternative, analytical approach has been made available through this research. The approach is based on the reasonable assumption that the fraction of cooled dry air that is diverted in the wet channel, not saturated the dew point efficiency %, undergoes an almost adiabatic saturation process in the very first part of the wet channel, thereafter it is warmed up along the wet channel by heat transfer from the dry channel. Liquid water is continuously evaporated along the wet channel and water is kept always close to saturation, so most of the heat is absorbed as latent heat and the process in the wet channel follows the saturation curve. This process can be implemented by passing the diverted flow of cooled dry air through an evaporative pad, *e.g.* a cellulose honeycomb pad soaked with water; thereafter, while the air flows along the wet channel, water is continuously sprayed onto the surface of the wall separating it from the dry channel, as well as on the other channel surfaces and/or in the airflow.

In the initial adiabatic saturation process, the wetted air introduced in the wet channel can theoretically be cooled down to the wet-bulb temperature $T_{wet,in}$ (°C) corresponding to the dry-bulb temperature and relative humidity of the cooled dry air, $T_{dry,out}$ and $RH_{dry,out}$. On the other hand, $T_{wet,in}$ (°C) controls $T_{dry,out}$ as it is the temperature of the air introduced in the wet channel and used to cool down, in the counter-flow heat exchanger, the air flowing in the dry channel. The advantage of the M-cycle is that $T_{wet,in}$ (°C) and therefore $T_{dry,out}$ are lower than the wet-bulb temperature $T_{wb,in}$ (°C) resulting

IOP Conf. Series: Earth and Environmental Science 1106 (2022) 012020 doi:10.1088/1755-1315/1106/1/012020

from the dry-bulb temperature and relative humidity of the ambient air prior to be cooled, that is the minimum temperature allowed in a simple evaporative cooling system (be it direct or indirect).

The specific enthalpy of the air-vapor mixture after the initial adiabatic saturation process $J_{wet,in}$ (J/kg$_a$), referred to the unit mass of dry air, must equal the specific enthalpy of the air-vapor mixture $J_{dry,out}$ (J/kg$_a$), before saturation, that is at the outlet of the dry channel, slightly increased by the sensible heat of the evaporated liquid water:

$$J_{wet,in} \cong J_{dry,out} = c_{p,a} \cdot T_{dry,out} + x_{dry,out} \cdot \left(c_{p,v} \cdot T_{dry,out} + q_{lv} \right) \tag{1}$$

where

$x_{wet,in}$ absolute humidity after the initial adiabatic saturation (kg$_v$/kg$_a$)
$c_{p,a}$ specific heat at constant pressure of the dry air (J/(kg·K))
$c_{p,v}$ specific heat at constant pressure of the water vapor (J/(kg·K))
q_{lv} latent heat of vaporization/condensation of water (J/kg)

In the previous equation, the enthalpy of the liquid water has been neglected with acceptable approximation.

The specific enthalpy J_{sat} of the air-vapor mixture along the saturation curve (RH = 100%) is related to the temperature and, through the saturation pressure of water, to the absolute humidity at saturation. The correlation between specific enthalpy and temperature on the saturation curve is one-to-one and can be expressed by a polynomial, from which an equivalent specific heat at constant pressure $c_{p,eq}$ (J/(kg$_a$·K)), including both sensible and latent heat, can be associated to the temperature increase of the air-vapor mixture along the saturation curve.

$$c_{p,eq}\left(T\right) = \frac{dJ_{sat}\left(T\right)}{dT} \tag{2}$$

Due to latent heat effects, $c_{p,eq}$ is much larger than the specific heat at constant pressure of dry air $c_{p,a}$, moreover it is highly temperature dependent.

As already mentioned, heat is exchanged between dry flow and wet flow in a counter flow heat exchanger. This can be analyzed by means of the ε-NTU method. In this regard, the instantaneous heat capacity C_{dry} (W/K) of the dry air flow is:

$$C_{dry} = \dot{m}_a \cdot \left(c_{p,a} + x \cdot c_{p,v} \right) \tag{3}$$

The mass flow rate of cooled dry air can be calculated from the volume flow rate of the ambient air and the dry air density at the inlet of the dry channel. An instantaneous heat capacity C_{wet} (W/K) can also be defined for the wet air flow:

$$C_{wet} = r \cdot \dot{m}_a \cdot c_{p,eq,mean} \tag{4}$$

Since the equivalent specific heat $c_{p,eq}$ changes significantly with temperature, a mean value $c_{p,eq,mean}$ (J/(kg$_a$·K)) has been introduced in the relationship above. This is in principle unknown as the outlet conditions of the wet channel are also unknown. An approximate yet effective initial estimate can be calculated considering the value of $c_{p,eq}$ at the wet bulb temperature of the ambient air $T_{wb,in}$ (°C) as this is mostly intermediate between the temperature $T_{wet,in}$ after dry cooling and initial saturation and the temperature $T_{wet,out}$ of the wet air flow before it is released from the heat exchanger. The estimate can then be refined from the calculated outlet conditions of the wet channel, possibly through an iterative approach.

In the ε-NTU method, the effectiveness ε (%) of the heat exchange process is defined as the ratio of achieved heat transfer rate and maximum theoretical heat transfer rate, for which the fluid with the lowest instantaneous heat capacity exits at the inlet temperature of the other fluid. With reference to the M-cycle, one has:

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}} = \frac{\dot{Q}}{C_{min} \cdot \left(T_{dry,in} - T_{wet,in} \right)} \tag{5}$$

The ε-NTU method requires that a minimum instantaneous heat capacity C_{min} is identified, for which that of the dry flow seems a convenient choice as it does not change during the dry cooling process. In fact, a relatively low minimum value of r exists, for which C_{dry} does not exceeds C_{wet}. If this condition is satisfied, C_{min}=C_{dry} and the effectiveness ε of the counter-flow heat exchanger depends on the ratio of the instantaneous heat capacities and the number of transfer units NTU as follows:

$$\varepsilon\left(\frac{C_{dry}}{C_{wet}}, NTU\right) \tag{6}$$

The number of transfer units NTU is defined as

$$NTU = \frac{U \cdot A}{C_{min}} \equiv \frac{U \cdot A}{C_{dry}} \tag{7}$$

where U (W/(m^{2}K)) is the overall heat transfer coefficient between the two channels, through the separating wall, and A (m^2) is the effective heat transfer area of such separating wall.

In the M-cycle, the maximum heat transfer rate that can be exchanged is that at which the dry air enters the dry channel at inlet/ambient temperature $T_{dry,in}$ and exits at the wet bulb temperature after adiabatic saturation of the cooled dry air, $T_{wet,in}$. The heat rate that is actually exchanged is that at which the dry air flow enters at temperature $T_{dry,in}$ and exits at temperature $T_{dry,out}$, at which a fraction r of the mass flow rate is then diverted in the wet channel, and the remaining is supplied to the cooled environment. Therefore, one has:

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}} = \frac{C_{dry} \cdot \left(T_{dry,in} - T_{dry,out}\right)}{C_{dry} \cdot \left(T_{dry,in} - T_{wet,in}\right)} \equiv \frac{T_{dry,in} - T_{dry,out}}{T_{dry,in} - T_{wet,in}} \tag{8}$$

With ambient conditions assigned in terms of $T_{dry,in} \equiv T_{amb}$, $RH_{dry,in} \equiv RH_{amb}$, and ambient pressure, a one-to-one correlation exists between $T_{dry,out}$ and $T_{wet,in}$. As a results, a unique couple of $T_{dry,out}$ and $T_{wet,in}$ values exists for a given value of ε.

The enthalpy $J_{wet,in}$ (J/kg$_a$) of the air-vapor mixture at the wet channel inlet can be calculated from $T_{wet,in}$. The enthalpy $J_{wet,out}$ (J/kg$_a$) of the air-vapor mixture at the wet channel outlet can also be calculated in order to improve the estimate of $c_{p,eq,mean}$:

$$J_{wet,out} = J_{wet,in} + \frac{\dot{Q}}{r \cdot \dot{m}_a} \tag{9}$$

The consumption of liquid water $\dot{m}_{lv}$ (kg$_l$/s) is:

$$\dot{m}_{lv} = r \cdot \dot{m}_a \cdot \left(x_{wet,out} - x_{dry,in}\right) \tag{10}$$

where a one-to-one correlation again exists at the wet channel outlet between $J_{wet,out}$ and the absolute humidity $x_{wet,out}$ (kg$_v$/kg$_a$).

Finally, an efficiency of use of the latent heat of the consumed liquid water can be calculated:

$$\eta_{lv} = \frac{\dot{Q}_{dry}}{\dot{m}_{lv} \cdot q_{lv}} = \frac{1-r}{r} \cdot \frac{\left(c_{p,a} + x_{dry,in} \cdot c_{p,v}\right) \cdot \left(T_{dry,in} - T_{dry,out}\right)}{\left(x_{wet,out} - x_{dry,int}\right) \cdot q_{lv}} \tag{11}$$

2.3. Test case scenario

The analytical model was used to evaluate the indirect evaporative conditioning of a C-segment PHEV or BEV via an M-cycle system. A reference vehicle is a Kia Soul full-electric, in which the performance of the air conditioning system has been obtained from previous studies that report an average coefficient of performance (COP) equal to 1.82, resulting from a cooling capacity of 1750 W compared to an electrical power of 960 W drawn by the compressor [13]. In particular, the cooling capacity value was used to calculate the water consumption of the M-cycle applied in partial substitution of the VCR cycle. To this end, a survey was performed, applied to a counter-current and recirculating heat exchanger operating in M-cycle mode, the geometry of which is shown in Fig. 3. The heat exchanger was taken 1 m long and 0.15 m wide. The heat transfer area is 0.940 m^2 thanks to the creation of the corrugated surface as shown in Fig. 4.

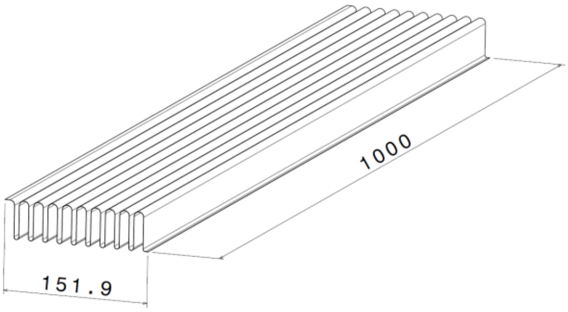

Figure 3. CAD representation of the heat exchanger geometry implemented in the analytical model. Dimensions in mm.

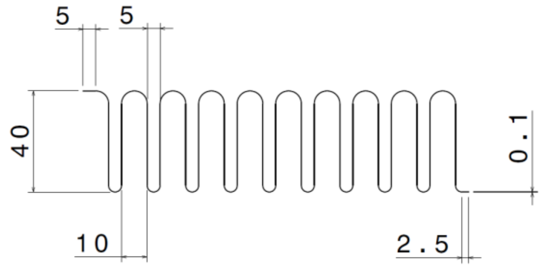

Figure 4. Detail of the geometry of the heat transfer surface. Dimensions in mm.

The heat exchanger behavior was evaluated through the analytical model by varying the ambient air input parameters: dry bulb temperature ($T_{dry,in}$≡$T_{amb,}$) from 25°C to 42.5°C and relative humidity ($RH_{dry,in}$≡RH_{amb}) from 20% to 90%. For each combination of these two parameters, the following parameters were calculated: temperature ($T_{dry,out}$) and relative humidity ($RH_{dry,out}$) of the product air at the dry channel outlet; cooling heat rate, that is the heat transfer rate subtracted from the product air; specific cooling capacity (calculated in Wh of cooling energy subtracted from the product air per kg of evaporated water). All of these parameters were calculated at a fixed recirculation rate *r* of 40% and at an inlet volumetric flow rate of 81.1 m^3/h at sea level pressure. From the results, the heat transfer surface area and the water consumption needed to meet the reference cooling heat rate of 1750 W were eventually calculated.

3. Results

The results of the application of the analytical model to the heat exchanger in question are shown in Figs. 5-8, in which a maximum cooling heat rate of 194 W was estimated at the ambient conditions of 42.5°C and 20% relative humidity, obtaining a dry channel outlet temperature of 29.8°C. However, it is rare for a PHEV or a BEV to operate in these conditions of high temperature and low ambient relative humidity. It was thus convenient to perform a more accurate analysis of the applicability of the M-cycle by identifying a wider RH_{amb} range, assumed to vary between 40 and 80% (dashed lines in Figs. 5-10). In that range, the cooling heat rate varies of the modeled system between 27 W and 128 W, obtaining temperatures of the product air leaving the dry channel between 19°C and 40°C, with RH_{amb} ranging between 57% and 91%.

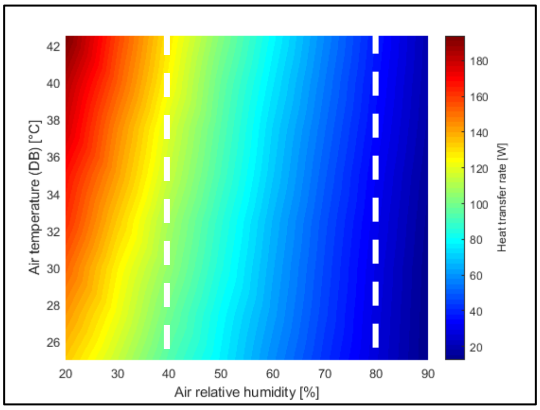

Figure 5. Cooling heat rate.

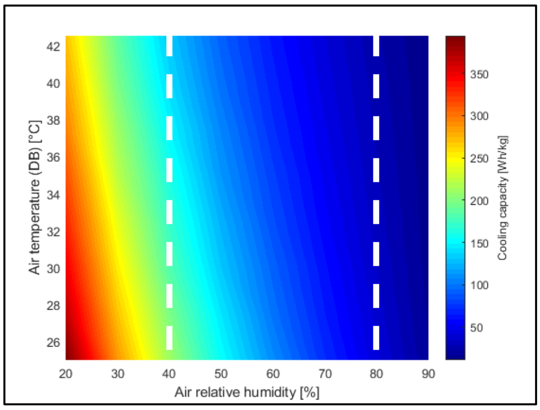

Figure 6. Specific cooling capacity.

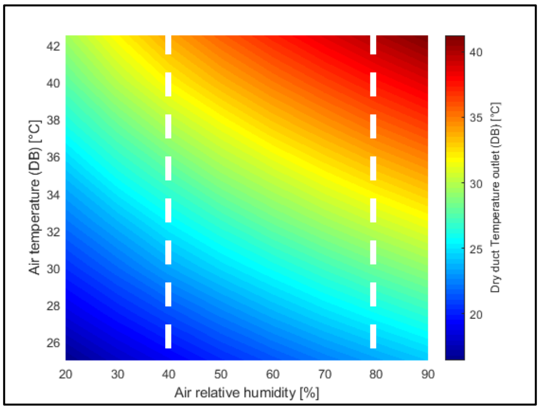

Figure 7. Dry channel outlet temperature.

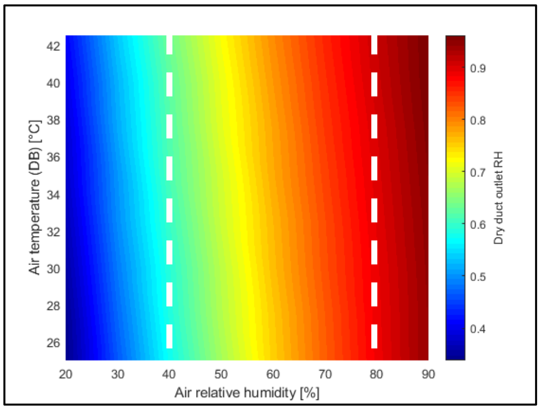

Figure 8. Dry channel outlet relative humidity.

For RH_{amb} values around 40%, the specific cooling capacity ranges between 370-410 Wh/kg$_{water}$, resulting not far from the specific cooling capacity of 400-600 Wh/kg$_{battery}$ when a VCR system with $COP \approx 2$ and battery pack with specific capacity of 200-300 Wh$_{el}$/kg$_{battery}$ are considered.

In order to meet the target of 1750 W of cooling heat rate for the reference vehicle considered in this work, the amount of water that need to be vaporized ranges between 4.27-4.73 kg$_{water}$/h. Considering an average vehicle speed of 50 km/h in the home-work commuting, the water consumption results between 8.54 and 9.46 L$_{water}$/100km which can lead to the use of sufficiently small water tanks that can be refilled every 2-3 days. The heat transfer surface area and the specific cooling capacity were also scaled according to the vehicle cooling performances and the obtained results are reported in Figs. 9-10.

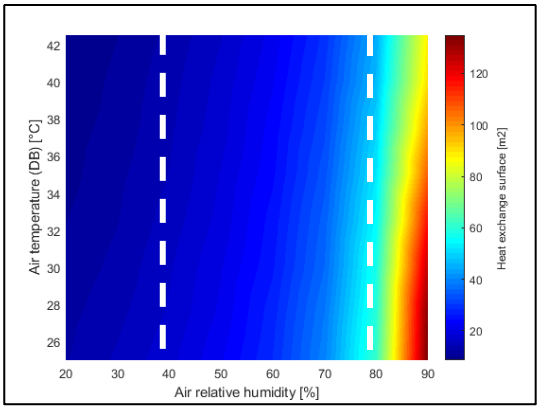

Figure 9. Heat exchange surface.

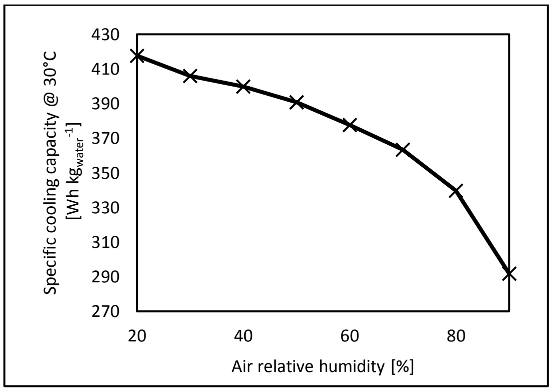

Figure 10. Specific cooling capacity

In this case, a heat exchange surface area of 15.9 m^2 is needed which makes mandatory to test more performing materials for the heat transfer surface (e.g. fibrous membrane [14] and nylon-based sheet covered by micro sized lint [15]) in order to reduce the dimensions of the exchanger and make it compatible with the volumes available in a vehicle.

Finally, a preliminary calculation was carried out on heat transfer effectiveness (Figure 11) and water use efficiency (η_{lv} - Figure 12) according to the variation of the rate of air recirculated within the M-cycle. It is noted that, at the same NTU value, there are no great advantages, in terms of effectiveness, in increasing recirculation rate above 40% (value used for this study) and it is observed that, in order to increase η_{lv}, it is crucial to minimize the recirculation rate r and increase the NTU value at least above 1.

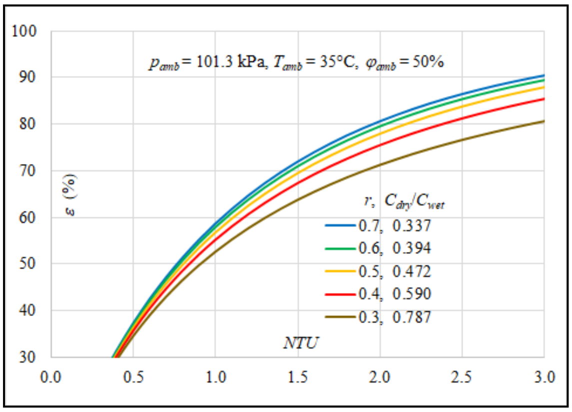

Figure 11. Variation of the heat exchanger effectiveness as a function of the recirculation rate r and the NTU value.

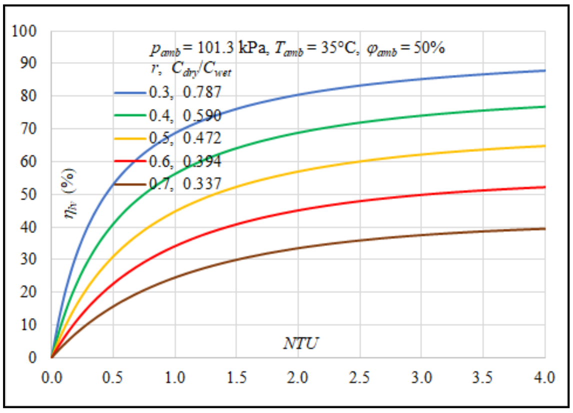

Figure 12. Variation of the heat exchanger water use efficiency as a function of the recirculation rate r and the NTU value.

4. Conclusions

In this work, an analytical model that describes the operation of a counter-current and recirculating heat exchanger operating according to the Maisotsenko cycle was applied to investigate the partial or total replacement of vapor compression refrigeration (VCR) systems as currently used for air conditioning of plug-in hybrid (PHEVs) and battery electric vehicles (BEVs), taking as a case study that of a fully electric production car.

From the calculated temperature and RH conditions at the exit of the heat exchanger dry channel, it can be seen that not all the obtained values are compatible with thermal comfort of car occupants. Therefore, it is necessary to foresee the operation of the M-cycle in hybrid mode, in which the air leaving the dry channel can be mixed with drier air coming from the VCR, which, in this way, must process a much lower flow rate and therefore can operate at reduced absorbed power. Moreover, the saturated but relatively cool air leaving the wet channel of the M-cycle system can be used to enhance cooling of the VCR condenser and, in this case, slightly improving the overall efficiency of the air conditioning system. Above all, the specific cooling capacity per unit mass of evaporated water is comparable with that of the unit mass of batteries providing electric energy to a VCR system with electric compressor. At low ambient humidity values (20%), the mass of water to be evaporated may even equal that of batteries supplying the energy to an electric VCR. Generally speaking, the result shows once again how the M-cycle is successful if coupled to a second VCR system that allows to reduce the incoming humidity and, at the same time, to make smaller the heat transfer surface area and, as a result, compatible with use in passengers cars.

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