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# The performance of a mobile air conditioning system with a water cooled condenser

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**Abstract.** Vehicle technological evolution lived, in recent years, a strong acceleration due to the increased awareness of environmental issues related to pollutants and climate altering emissions. This resulted in a series of international regulations on automotive sector which put technical challenges that must consider the engine and the vehicle as a global system, in order to improve the overall efficiency of the system. The air conditioning system of the cabin, for instance, is the one of the most important auxiliaries in a vehicle and requires significant powers. Its performances can be significantly improved if it is integrated within the engine cooling circuit, eventually modified with more temperature levels.

In this paper, the Authors present a mathematical model of the A/C system, starting from its single components: compressors, condenser, flush valve and evaporator and a comparison between different refrigerant fluid. In particular, it is introduced the opportunity to have an A/C condenser cooled by a water circuit instead of the external air linked to the vehicle speed, as in the actual traditional configuration. The A/C condenser, in fact, could be housed on a low temperature water circuit, reducing the condensing temperature of the refrigeration cycle with a considerable efficiency increase.

## Introduction

The use of mobile A/C system for cabin cooling and drying is about 7 billion liters of fuel per year, which is the 3,2% of the global fuel consumption of vehicles [1, 2]. Similar studies evaluate the vehicles overconsumption due to the A/C system is between 1,8 and 2 l/100 km, equivalent to 20-25% of a mean sedan car fuel consumption on the homologation cycle [3].

The importance of A/C system in terms of vehicle energy balance and environmental issue invites to consider new technologies and opportunities to improve its efficiency. On the thermodynamic point of view, the most significant opportunity is to decrease the condensing temperature. However, this temperature is strictly related to the external cooling source that usually is the external air: actually, the condenser is placed in the front end of the vehicle and its heat removal rate is linked to the external air temperature and speed. Therefore, if the condenser was cooled by a liquid circuit [4], it could be managed at a lower temperature; the mean temperature difference could be reduced thanks to the higher specific heat of the liquid (i.e. water/glycol) and, also, with a suitable flow rate. An interesting opportunity is related to the possibility to integrate the condenser in the engine cooling system, particularly if it is split into two separate circuits [5-7]. In all the cited papers [4-7] the idea of integrating the A/C vehicle's system in the cooling circuit was discussed making reference to a double

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temperature circuit. In [4] the attention was focused to the heat exchanger integration inside the engine bay without considering the A/C performances as a function of the cooling power requested by the cabin. A more oriented systematic approach was discussed in [5] verifying the experimental performances achieved on a small vehicle. In [6,7] the performances of a double temperature cooling system were deeply analyzed, considering the A/C condenser as a lumped thermal load, represented by a specific power, without entering in the A/C modeling.

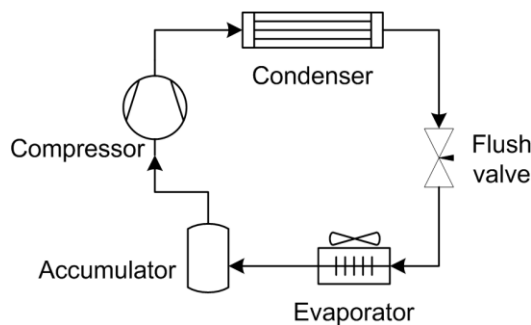
The thermodynamic equilibrium of the A/C system is not straightforward to determine and depends on several factors: ambient temperature, solar irradiation, vehicle speed or fan operation, amount of the refrigerant and, above all, components design. The frequent operation at partial load or in off-design conditions makes the A/C equilibrium even more complex.

In this paper, the Authors present and discuss a comprehensive mathematical model of the A/C system, which includes the off design conditions of the evaporator and of the condenser, the flush valve and the compressor according to conservation equations (mass, momentum and energy).

The paper discusses the behavior of the A/C system at different loads. The benefits related to a liquid cooled condenser inserted in the low temperature circuit in a double temperature level engine cooling layout are discussed in comparisons with a traditional A/C system with an air cooled condenser, considering a new interesting fluid (R1234yf [8]) instead of the traditional R134a. The two configurations have been studied in term of thermodynamic equilibrium and, in particular, the compressor energy saving have been assessed and compared to the effects of the coolant circuit modification.

## 2. Cabin air conditioning system model description

In air conditioning systems, a working fluid flows in a thermodynamic circuit composed by a compressor, a condenser, an evaporator, a flush valve and a little tank (Figure 1). The modelling of the A/C system has an inner complexity, due to the continuous matching of the condenser and evaporator temperatures with the external conditions and cabin cooling requests.



**Figure 1:** traditional A/C circuit layout: evaporator is placed in the cabin side, having the role to transfer the heat from the cabin to the refrigerant fluid (evaporating it), while the condenser is in the front end of the vehicle, having to carry out the condensation heat of the fluid towards the environmental air.

### 2.1 Compressor

The compressor usually used in mobile A/C is a volumetric one. In ideal case, the pressure is fixed by the circuit and the mass flow rate depends on the revolution speed  $\omega$  of the compressor.

$$\dot{m} = \eta_{vol} \cdot \rho_{refr} \cdot V \cdot \omega \quad (1)$$

Volumetric efficiency  $\eta_{vol}$  is slightly dependent on pressure delivered and speed, which is imposed by the electric motor. Refrigerant density  $\rho_{refr}$  refers to the inlet section of the compressor and depends on the thermodynamic conditions in the accumulator. The isentropic compressor efficiency, assumed by manufactures data, allows the calculation of outlet fluid enthalpy and mechanical power requested [9].

### 2.2 Accumulator

The accumulator is placed on the low pressure side of the A/C circuit, downstream the evaporator. It is represented by a capacity which separates the evaporator outlet and the compressor inlet, storing the extra-liquid during the refrigerant low request conditions. Moreover, it avoids that the liquid enters in

the compressor, condition that can damage the compressor itself and strongly reduces its efficiency. The accumulator acts, also, as a dryer and as a filter.

### 2.3 Evaporator and condenser

The evaporator is the heat exchanger which cools down the cabin. In this component, the refrigerant fluid usually enters as a two-phase mixture (downstream the flush valve) and exits as saturated vapor or slightly superheated. Once the exchanger geometry and the entering conditions of both fluids are known, outlet condition can be evaluated. Thermodynamic conditions of the refrigerant fluid at the inlet are known in terms of pressure (saturation pressure) and quality.

The condenser is the heat exchanger used to extract heat from the refrigerant fluid and to exchange toward the air coming from the external environment. In this component, the refrigerant usually enters as superheated vapor (conditions fixed by the compressor) and exits as saturated liquid (or as a mixture with saturated vapor or also in some conditions as sub-cooled fluid) depending from the heat exchanger behavior whose geometry is fixed. Its thermal balance gives the temperature of the fluid that crossed it (hotter than the inlet one): usually, it is represented by environmental air whose flow rate is fixed by the vehicle or by auxiliary fan.

Condenser and evaporator are represented in one dimensional way (uniform conditions in the cross section area) and divided in subvolumes: the geometry of the fluids passages is known and, therefore, the heat exchange surface. Continuity, momentum and energy equations are solved step by step following the fluid motion inside a 1D passage, considered in steady state conditions.

Forced convective heat exchange coefficients are evaluated considering proper correlations: Dittus-Boelter for single phase, Shah's and Klimenko's equations [10, 11] for two phase region:

$$\begin{cases} h_{1\text{phase}} = 0.0243 \text{Re}^{0.8} \text{Pr}^n \frac{k}{D} \\ h_{\text{cond}} = 0.023 \text{Re}_{sl}^{0.8} \text{Pr}_{sl}^{0.4} (1-x)^{0.8} \frac{3.8x^{0.76} (1-x)^{0.04}}{p/p_{cr}} \frac{k}{D} \\ h_{\text{evap}} = 0.087 \text{Re}^{0.8} \text{Pr}^{1/6} \left( \frac{\rho_{sv}}{\rho_{sl}} \right)^{0.2} \left( \frac{k_{\text{wall}}}{k_{sl}} \right)^{0.09} \frac{k_{sl}}{D} \end{cases} \quad (2)$$

$n$  is equal to 0.3 for heating fluid and 0.4 for cooling fluid,  $k$  is the fluid thermal conductivity while  $D$  the equivalent cross section diameter, the quality  $x$  is currently evaluated for each subvolumes of the heat exchanger pipe, considering homogeneous flow;  $sl$  subscript refers to saturated liquid,  $sv$  to saturated vapor,  $p_{cr}$  is the critical pressure of the fluid,  $k_{\text{wall}}$  the thermal conductivity of the exchanger walls. Previous equations consent to solve the heat exchange in operating conditions, calculating the outlet thermodynamic conditions of the fluids by knowing the mass flow rates (and, so, fluid velocities) and thermodynamic inlet conditions.

### 2.4 Flush valve

The flush valve is considered as a reduction of the fluid passage section (orifice). This produces an expansion from high pressure liquid (almost saturated liquid) to low pressure liquid (two-phase conditions) downstream the orifice. At this point, a partial pressure recovery is realized when the fluid fills again the whole section of the duct, just upstream the evaporator.

When the refrigerant fluid crosses the valve, the following processes and conditions must be represented and constraints respected:

- inlet and outlet pressure ( $p_{\text{max}}$  and  $p_{\text{min}}$ ) must match temperature levels at condenser and evaporator;
- orifice inlet conditions (in particular density  $\rho_{\text{in}}$  and enthalpy  $h_{\text{in}}$ ) are fully determined by knowing the condenser outlet conditions;
- the fixed orifice geometry imposes the mass flow rate according to (eq. 3):

$$\dot{m} = C_D A_{throat} \sqrt{2\rho_{in}(p_{max} - p_{throat})} \quad (3)$$

the discharge coefficient  $C_D$  is function of the section ratio and Reynolds. It has been calculated according to the Lichtarowicz's correlation [12];

- since pressure recovery downstream the orifice is negligible for short orifices, in a first approximation,  $p_{throat}$  can be assumed equal to the outlet pressure (i.e. evaporation pressure  $p_{min}$ ). However, in a more precise way, pressure recovery  $\Delta p_{rec}$  can be evaluated to Borda-Carnot formula;
- the momentum equation (in steady condition), continuity and energy equation are applied from the inlet to the outlet section of the valve (eq. 4), considering the flow as homogeneous:

$$\begin{cases} \Delta p_{rec} = \frac{A_{throat}}{A} \left(1 - \frac{A_{throat}}{A}\right) \rho_{in} u_{throat}^2 \\ (p_{max} - p_{min})A + \dot{m}(u_{in} - u_{out}) = 0 \\ (h_{in} - h_{out}) = \frac{1}{2}(u_{out}^2 - u_{in}^2) \end{cases} \quad (4)$$

$A$  is the full pipe cross section,  $u_{out}$  is the fluid speed downstream the orifice where the fluid fill the overall pipe section; knowing orifice geometry and considering momentum and continuity equations through it, fluid-dynamic conditions in the throat section (in particular, fluid density  $\rho_{throat}$  and fluid speed  $u_{throat}$ ) are evaluated and, therefore, the outlet enthalpy  $h_{out}$  of the fluid;

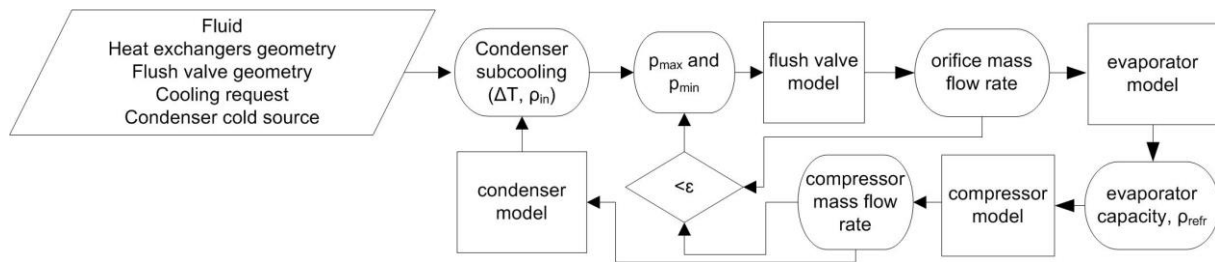
- once evaporation pressure and valve outlet enthalpy are known the outlet thermodynamic conditions are fully determined; in particular, fluid quality  $x$ .

Definitely, when the pressure drop through the valve is known from both condensing and evaporating temperatures, the valve imposed the mass flow rate (eq. 3) and the momentum equation gives the fluid speed at the valve outlet, being known the inlet conditions. Once also momentum equation is solved, enthalpy downstream the valve can be evaluate according to energy conservation (eq. 4) and, so, fluid quality. This procedure is, therefore, an iterative method: in fact, pressure levels ( $p_{max}$  and  $p_{min}$ ) have to be adjusted at each iteration, in order to match the flow rate imposed by the orifice and that one of the volumetric compressor.

### 2.5 Model integration

The sequence of the equations representing the A/C system is solved according to an iterative procedure as in Figure 2. Heat exchangers are sized and their geometry is chosen; at the same time, the valve geometry is fixed (orifice dimensions) and A/C ducts defined. Refrigerant fluid is chosen too.

Knowing the temperature of the fluids which behave as source at two heat exchangers and their flow rates, condensing and evaporating temperature can be hypothesized. Then, knowing the condenser outlet conditions, in particular fluid density  $\rho_{in}$ , equation (3) gives a flow rate value of the refrigerant fluid that must correspond to the one imposed by the compressor (eq. 1). At the same time, vapor quality is evaluated from equations (4). At this point, the heat exchangers model of the evaporator gives the thermodynamic conditions of the fluid at its outlet, i.e. compressor inlet. So, the flow rate given by eq. (1) can be verified and compared to the one given by the eq. (3): if their difference is within a given tolerance  $\varepsilon$ , the solution of all the equations is found. If the two flow rates do not correspond, the two pressure levels ( $p_{max}$  and  $p_{min}$ , condensation and evaporation) are adjusted and the procedure is re-iterated (Figure 2). The procedure is closed by the condenser behavior: once the compressor outlet conditions are known, the heat exchanger model of the condenser gives the fluid conditions at its outlet. In particular, fluid density  $\rho_{in}$  must match that of equation (3).



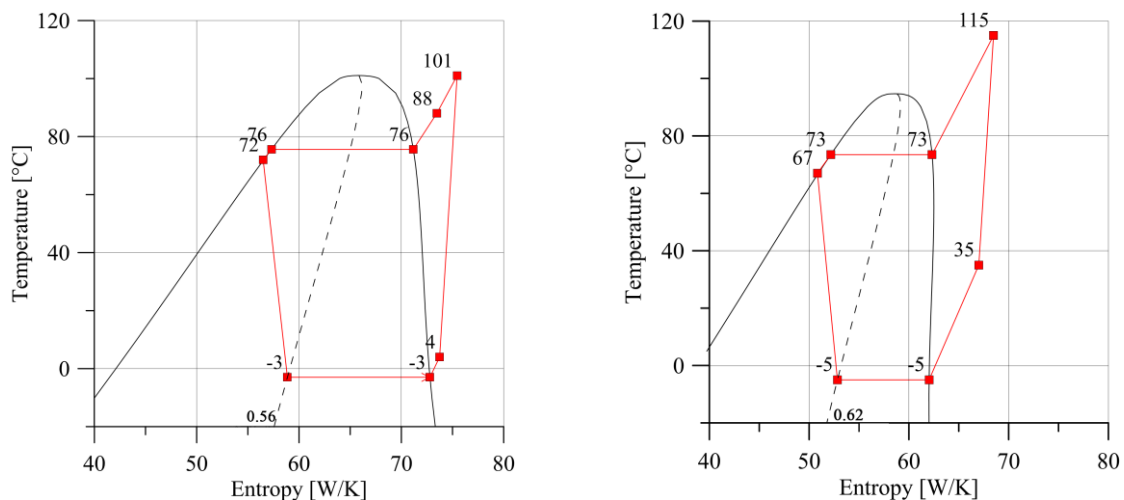
**Figure 2:** procedure for the A/C system model. Rounded shapes represent the outputs of the models.

### 3. Results

The A/C model can be considered as a virtual platform of the conditioning system and it can be used to study different layouts and arrangements. The availability of low temperature water flow is necessary to fulfill a liquid cooled condenser: an interesting solution is to split the traditional cooling system into two circuits each of them operating at different temperature [7].

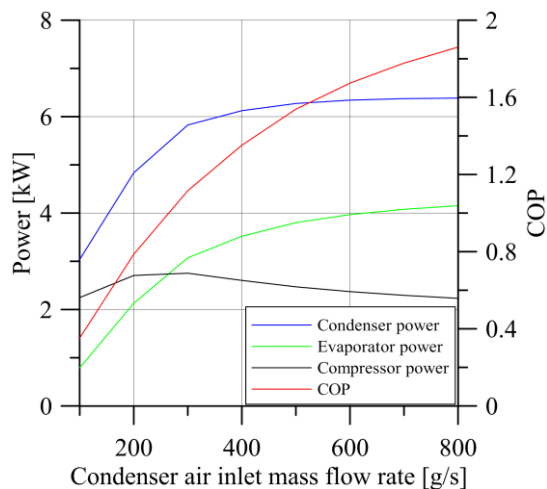
The most common operating fluid for automotive A/C is R134a, but its use is forbidden in new vehicles and will be replaced by R1234yf, which represents the actual technological standard [13, 14]. Figure 3 show the thermodynamic cycles of A/C system with R134a and R1234yf as operating fluids. Both cycles have been compared at the same cooling power (about 4 kW) required by the cabin and exchanged within the evaporator. They are evaluated in steady conditions: the heat surface of the evaporator is about 6 m<sup>2</sup> and that one of the condenser is 10 m<sup>2</sup>; air flow rate at evaporator is 190 g/s at a temperature of 45 °C, while at condenser it is 530 g/s at 35°C, corresponding to a vehicle speed of about 60 km/h. The revolution speed of the volumetric compressor is 1650 RPM. Tubes connecting the exit of the condenser have 10 mm diameter while the orifice is 1.183 mm. In this condition, the two pressure levels of R1234yf cycle are 2.6 bar and 22 bar, while R134a cycle had a maximum pressure of about 24 bar and similar minimum pressure (Figure 3).

The mathematical model demonstrates that the new fluid has an average pressure level slightly lower than the R134a, but with a maximum temperature about 10°C higher. The condensing temperatures are similar, but the condensing heat of the R1234yf is lower and, so, it is necessary to overheat more the fluid in order to keep the same power exchanged in the evaporator. Moreover, the R1234yf has a higher specific heat at constant pressure which leads to a lower mass flow rate inside the A/C circuit [15]. The cycles have been reported making reference the absolute entropy with respect to specific mass flow rates of the fluids (42 g/s for R134a and 39 g/s for R1234yf).

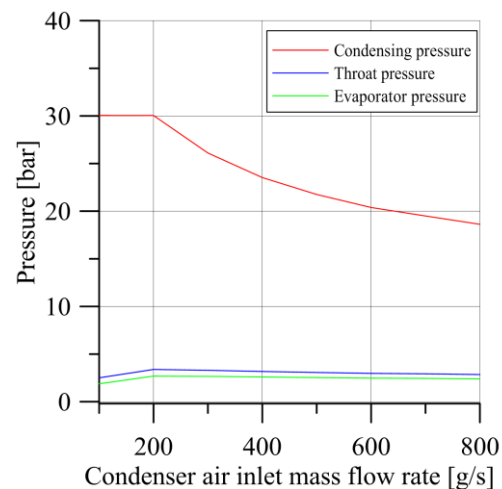


**Figure 3:** comparison between thermodynamic cycles of A/C system with R134a (on the left) and R1234yf (on the right).

Figure 4 shows the performance of the A/C system with R1234yf as working fluid and air cooled condenser: the air mass flow rate which crosses the condenser is strictly related to the vehicle speed (it is placed in the front end of the vehicle) and, so, modifies the heat rejection capacity of the condenser itself. When the air mass flow rate grows, its power rejected increases till to a saturation value due to the exchanger's fixed geometry; the evaporator power rises as well, increasing the cooling power at cabin side. At low vehicle speed, instead, this cooling capacity is limited and COP value is very low: compressor power increases as well as the refrigerant mass flow rate. Figure 5 shows the pressures reached in the circuit with different condenser air inlet flow rate: orifice throat pressure is slightly higher than evaporating pressure, indicating the low pressure recovery downstream the orifice. Condensing pressure decreases when external air flow rate increase, confirming the COP increase.

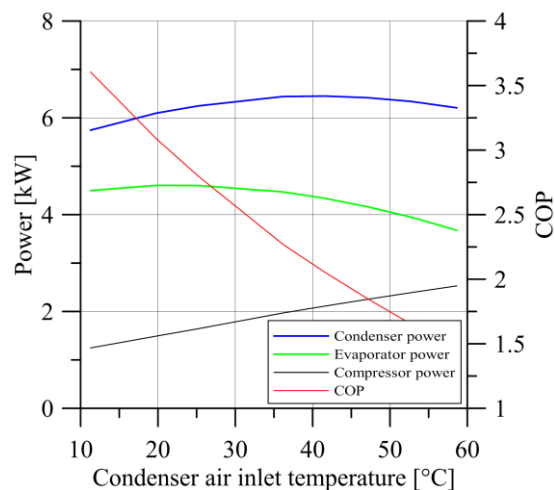


**Figure 4:** A/C performances operating with R1234yf and different air mass flow rate at condenser side

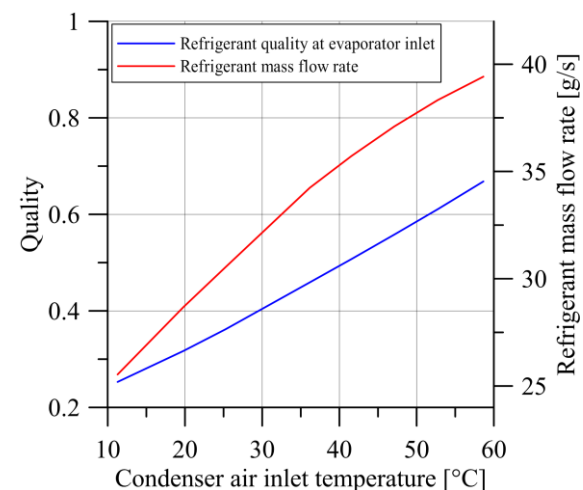


**Figure 5:** pressure levels in the circuit with reference to orifice

A significantly changing parameter is the air inlet temperature at the condenser, which actually is the external air temperature. When the air inlet temperature increases (for instance, in summer), COP decreases because of the higher mean temperature of the thermodynamic cycle. However, the system is able to keep the cooling capacity of the evaporator increasing the compressor work (Figure 6).



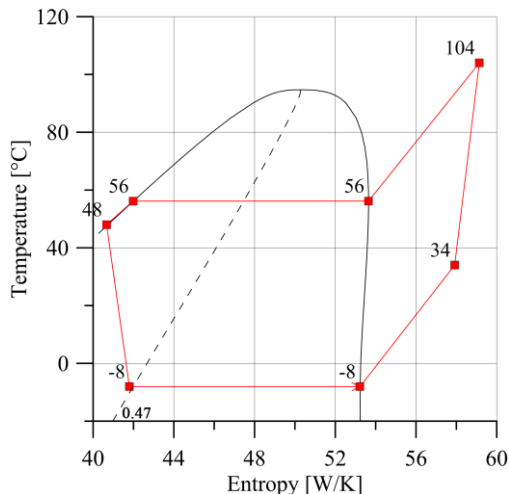
**Figure 6:** A/C performances operating with R1234yf and different air inlet temperature at condenser side



**Figure 7:** refrigerant mass flow rate and evaporator inlet quality of the A/C system for different air temperature at condenser

The performances variation on Figure 4 and Figure 6 is due to a re-arrangement of the A/C parameters, in particular of refrigerant mass flow rate and its quality at the evaporator inlet. When the external air temperature increases, the quality of the refrigerant at the very inlet of the evaporator increases as well and the cooling capacity would decrease; therefore, the A/C system reacts increasing the mass flow rate of the working fluid, in order to satisfy the cabin cooling request (Figure 7).

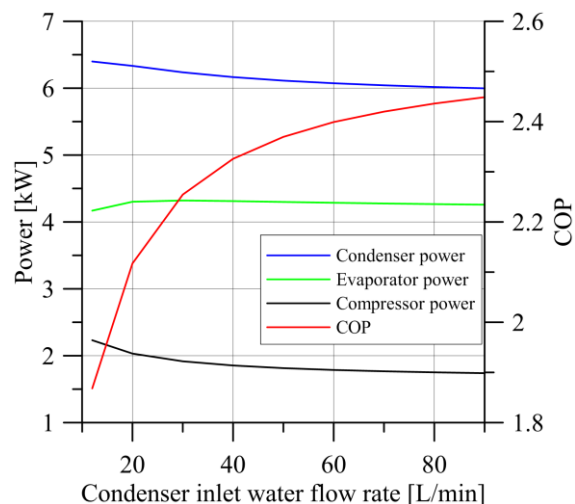
The opportunity to cool-down the condenser by water can be very interesting: the heat transfer coefficient is higher and the overall equilibrium of the A/C system could be modified (Figure 8).



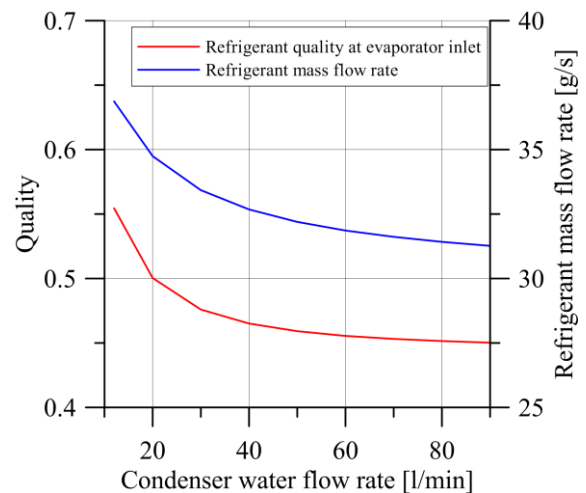
**Figure 8:** thermodynamic cycle of A/C system with R1234yf as working fluid and water cooled condenser (WCC). The cabin cooling request is fixed at 4 kW (evaporator side) and the water flow rate is 35 l/min, entering in the condenser at 42°C

The higher heat transfer coefficient of the water produces a better heat exchange within the condenser and a lower condensing temperature with respect to the air cooled one: the maximum pressure decrease (it is about 15 bar), leading to a lower compression ratio, with benefit in term of specific work required by the compressor.

The effects of the variation of the water flow rate at the condenser are showed in Figure 9. Such flow rate can be easily changed using an electrical pump on a secondary engine cooling circuit. A higher water flow rate leads to a lower condensing temperature and, so, a lower condensing pressure. The performances are, however, more stable than in the case of air cooled condenser.



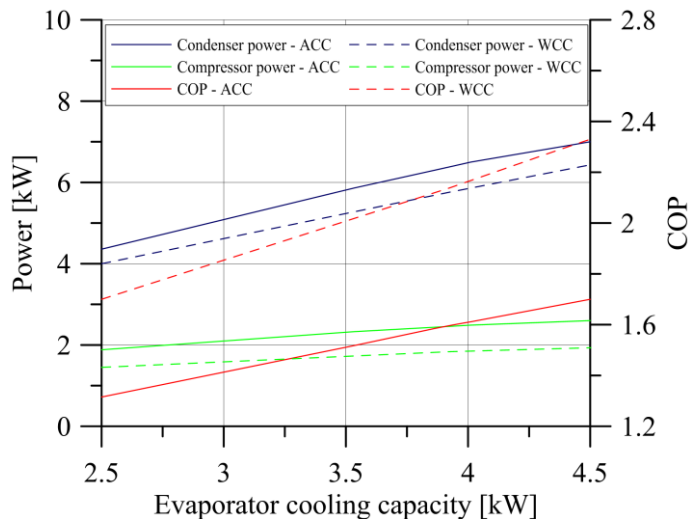
**Figure 9:** A/C performances with liquid cooled condenser at different water flow rate



**Figure 10:** R1234yf mass flow rate and quality at evaporator inlet for A/C equipped with liquid cooled condenser at different flow rate

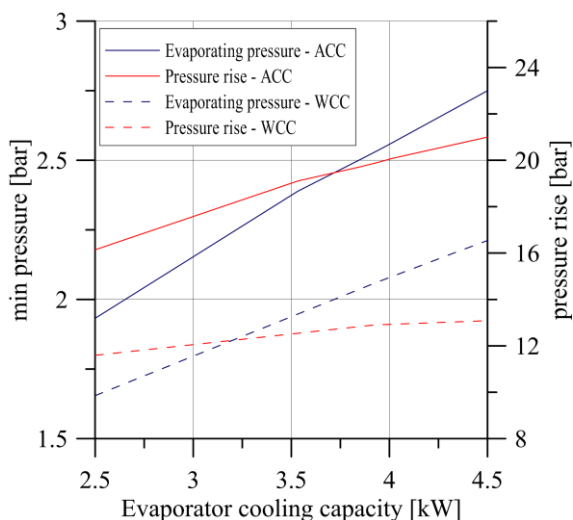
The saturation value of the power is noticeable also in this case and it is consequence of the fixed geometry of the heat exchanger. COP increases when the water flow rate grows, indicating that the two pressure levels are approaching each other. The higher COP produces a decrease of the working fluid flow rate, which is compensated by a reduction of the quality at the evaporator inlet in order to preserve the cooling request by the cabin (Figure 10).

The benefits of a liquid cooled condenser are clearly in Figure 11: the power absorbed by the compressor decreases and COP increases with a water cooled condenser.

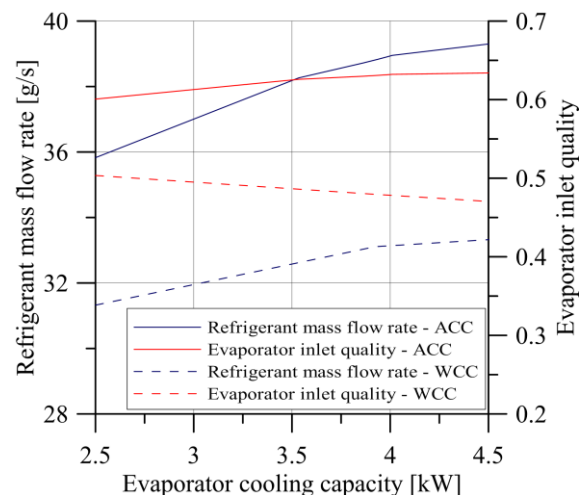


**Figure 11:** comparison between performances of the A/C system varying the cabin cooling request. Straight lines are referred to ACC, while dashed lines to WCC

In Figure 12 and Figure 13 are represented the A/C parameters that rule the performances highlighted in Figure 11. A water cooled condenser slightly reduces the minimum pressure, (at the evaporator) with respect to the air cooled one, because of the higher heat exchange; so, a higher degree of subcooling is reached in the condenser outlet. The maximum pressure level (in the condenser) decreases as well, producing a higher thermodynamic efficiency.

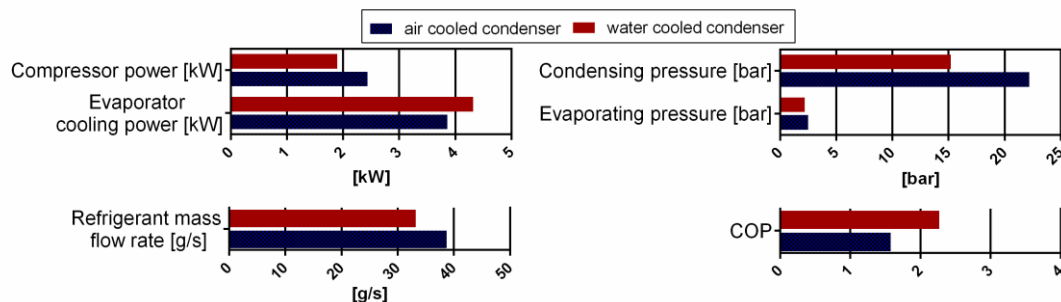


**Figure 12:** comparison between air cooled condenser (ACC) and water cooled condenser (WCC) A/C system: evaporating pressure and pressure rise



**Figure 13:** comparison between air cooled condenser (ACC) and water cooled condenser (WCC) A/C system: refrigerant mass flow rate and evaporator inlet quality

Figure 14 shows a quantitative comparison between a mobile A/C system equipped with air cooled condenser and a water cooled one, keeping the same condensing power (6 kW): with a water cooled condenser, the cooling capacity at the cabin side increases as demonstrated by the 12 % higher power exchanged in the evaporator; the most important results is, however, the decrease of the mechanical power absorbed by the compressor of about 22%. These results produce a really higher COP (40%). The refrigerant mass flow rate is lower, indicating a further reduction of the compression work (Figure 14). When the cabin cooling request increases, the quality of the working fluid at the evaporator inlet is almost constant, but it is really lower when the condenser is water cooled: this demonstrates a higher potential in terms of cooling capacity.



**Figure 14:** comparison between A/C systems with air cooled and water cooled condenser

Further benefits are represented by the reduction of the pressure levels of the system, with lower leakages probability and saving on the refrigerant mass flow rate (that can lead to a lower overall amount of the refrigerant on-board); moreover, with a water cooled condenser, the A/C components can be re-allocated within the engine bay, reducing the ducts length; a WCC, in fact, can be placed closer to the evaporator and it has not to be placed in the front end of the vehicle, as in air cooled configuration.

All this benefits lead to a more reliability of the whole A/C system in terms of fluid leakages and environmental aspects. Finally, a water cooled condenser, having higher heat transfer coefficient, can be more compact of an air cooled one, but it is necessary to introduce a secondary liquid circuit in the engine bay. This, however, can fulfill also other thermal issues, like the charge air cooling, integrating different engine and vehicle thermal needs.

## Conclusions

Mobile air conditioning systems have some critical operating issues due to the very often off-design and transient working conditions. Heat transfer coefficients at the two heat exchangers, in fact, strongly depend on the vehicle speed and on the temperatures of the two sources with which heat is exchanged (in particular outside temperature).

The paper presents a comprehensive mathematical model of the A/C system, conceived to represent specific operating conditions of the mobile units. The model catches the typical off-design conditions of the single components (compressor, heat exchangers, flush valve) when external parameters (cabin cooling request, external air temperature and vehicle speed) are significantly changed. This required an iteration procedure in which condensing and evaporating temperature are unknown and they are the result of the model.

Thanks to the model, which behaves as a virtual platform of the unit, some technological improvements for the sector have been studied regarding to:

- the performances of a new refrigerating fluid (R1234yf), which will replace by laws the conventional R134a, shows slightly lower operating pressures and lower mass flow rates;
- a water cooled condenser fed by a secondary engine cooling circuit, on which low temperature vehicle's needs could be fulfilled (intercooler, cabin heater, A/C condenser as well), realizes a lower and steadier condensing temperature, because it is not related to the changing of external air temperature and flow rate. Considering R1234yf as working fluid, a reduction of about 0.5 kW

- (22%) of the compressing work is demonstrated, with respect to the ACC case. Very interesting is, also, the stronger increase of the COP (about 40%), which leads to a more efficient system. This power reduction is, in any case, higher than the one needed by the secondary cooling circuit [7], which is in the order of 0.1 kW and can be easily compensated by the other benefit related to the double temperature cooling system [6].
- c) a reduction of the mean pressure level and the refrigerant mass flow rate can be achieved, implying a lower leakages probability and lower amount of refrigerant on board, with savings on management costs and environmental safety.

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