

Study of the Performances of a Residential Air Conditioner Converted respectively to *Propane* and *Isobutane* in Tropical Conditions

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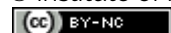
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ABSTRACT

Refrigeration and air conditioning are often incriminated for their contributions to the greenhouse effect. An alternative solution to curb this problem is the use of refrigerant with low global warming potential (GWP). To meet this goal, most hydrocarbons are potential candidates. Thus, the objective of this work is to compare the energy performance of propane (R290) and isobutane (R600a), which are low GWP fluids used as refrigerants in a residential split air conditioner under tropical climatic conditions (typical case of Benin). The investigation aimed to analyse the thermodynamic performance of these two fluids from their refrigeration cycles. The studies were conducted on a residential split air conditioner in Benin converted respectively to R290 and R600a refrigerants. The air conditioner was charged in increments of 50g of refrigerant until reaching the maximum load of 400g. Series of measurements (of temperature, hygrometry, air velocity, and mass of fluid, pressure, current intensity and voltage) were performed to calculate the cooling capacity and the coefficient of performance for each load of refrigerant. The so-named Engineering Equation Solver and MATLAB software were also used to simulate the performance of the studied fluids. The results of the study showed that the best operation conditions are guaranteed for a charge of refrigerant comprises within 150g and 300g for both fluids. The equipment showed better performances with R600a.

Nomenclature

\dot{Q}_p :	Heat lost by the compressor by warming up	(kW)
\dot{W}_{eff} :	Power actually consumed by the compressor	(kW)
\dot{m}_f :	Mass flow of the refrigerant through the compressor	(kg/s)
h_1 :	Enthalpy of the refrigerant at the compressor inlet	(kJ/kg)
h_2 :	Enthalpy of the refrigerant at the outlet of the compressor	(kJ/kg)
h_2^{is} :	Enthalpy of the refrigerant at the outlet of the compressor for an isentropic compression	(kJ/kg)
n_{eff} :	Effective efficiency of the compressor	-
q_v :	Air volume flow rate	(m ³ /h)
v_{moy} :	Mean air speed in the open face	(m/s)
A:	Area of the open space for the air flow	(m ²)
$\Delta h_{air} = h_s - h_e$:	Variation of enthalpy of the air at the evaporator	(kJ/kgas)
ε :	Efficiency of the evaporator	-
ξ :	The proportion of energy lost	-
ρ :	Density of air	(kg/m ³)

1. Introduction

Emissions of greenhouse gases contribute to disturbing climate change. Over the course of this century, it is to be feared an overall increase in temperatures between 1.3°C and 4.3°C [1]. This range corresponds to a stabilization of the current level of CO₂ emissions in the best case and to a late implementation of measures to reduce pollutant emissions. The consequences of global warming (desertification, floods, cyclones, mass migrations, rising sea levels, etc.) are alarming and could become catastrophic. International authorities have now made firm and serious commitments at the highest levels. Thus, on October 14, 2016 was signed "Kigali agreement" to stabilize, then eliminate the use of Hydrofluoro-carbons (HFCs), powerful greenhouse gases. The Kigali Amendment would be implemented on 1 January 2019, provided it is ratified by at least 20 Parties to the Montreal Protocol (or 90 days after ratification by the 20th member country, whichever is later) [3]. This agreement, considered "historic", is an amendment to the Montreal Protocol of 1987 to the "Vienna Convention for the Protection of the Ozone Layer" (1985) [3]. It is estimated that this agreement will avoid the equivalent of 8800 megatons of CO₂ equivalent per year from 2050, a reduction of 0.5 degree Celsius of warming (compared to the reduction of 2 to 1.5 degrees provided for in the Paris Agreement of December 2015) [3]. The Kigali agreement provides for the industrialised countries to begin a process to reduce HFCs by 2019 and developing countries to freeze as of 2024 [3]. By 2040, the level of use of HFCs should not exceed 15% of current levels. It is therefore incumbent upon the industry to develop substitutes (some of which already exist, which probably explains compliance with the Kigali agreement) and favour the production of products such as air conditioners and refrigerators that consume less energy [3] and use eco-friendly refrigerants. It is urgent to intervene to create

a new ecological balance respectful of the rights of future generations. Changing current trends and reducing energy consumption are essential.

2. Literature review

This brief literature review focuses on works related to hydrocarbons because of GWP which is about 3. Chang et al. (2000) studied the performance of a heat pump with different hydrocarbons (HCs) as propane, isobutane, butane and propylene. The heat transfers were analysed by measuring the average heat transfer coefficients. They showed that the heating and cooling capacities of a heat pump using R290 were slightly lower than those obtained with the refrigerant R22, but that the coefficient of performance (COP) was higher. The capacities delivered by the circuit with R1270 are superior to those offered by R22 and the COP is significantly improved [4]. Fernando *et al.* (2004) constructed a prototype heat pump with a heating capacity of 5 kW with a low propane load. The authors demonstrated its ability to operate with only 200 g load within Swedish climate without reducing the COP [5]. Park and Jung (2007) studied the performance of a water/water heat pump using an ethane/propane (R170/R290) mixture. The R170/R290 mixture had a compressor discharge temperature lower (of between 16.6 and 22.2°C) than that of R22 under the same operating conditions. A decrease in the discharge temperature implies less thermal stress on the compressor and increases its service life [6].

Park et al. (2008) conducted an experimental evaluation of water to water heat pump using R433A (a mixture of 30% propylene and 70% propane, GWP = 3) as an alternative to R22. They used a compressor originally designed for the R22. The results showed that the COP with R433A was 4.9% to 7.6% higher than that with R22. Yu et al. (2010) performed the thermodynamic analysis of a transcritical cycle of a high-temperature heat pump with an R32/R290 mixture. They succeeded to produce hot water at 90°C [7].

Corberán and Martínez [8] used a compressor with mineral oil and found that the amount of R290 refrigerant dissolved in the lubricating oil represented about 30% of the total charge. Mineral oils are less expensive and are considered a good choice for high capacity hydrocarbon applications located outside buildings (categories B - Occupied occupancy and C- authorized occupancy) where the permissible load is higher [8]. Ghouali and Byrne [9] studied a heat pump for simultaneous heating and cooling with a heat output of 20 kW was developed with a load of 4 kg of propane. The choice of mineral oil did not pose a problem for the industrial customer. Nevertheless, the use of mineral oil can lead to a significant decrease in viscosity, which can be harmful to the compressor. Choudharia and Sapalib [10] made a comparative analysis of the performance of R290 and R22 over a standard vapour compression cycle for different evaporative temperatures at constant condensing temperature. It has been observed that the R290 gives a lower discharge temperature, which is an important factor in improving the life of the compressor. The required refrigerant mass flow rate with R290 was 50% lower than that of R22. The coefficient of performance of R290 closely matches that of R22. However, a higher COP can be expected by a specially designed system related to R290 properties. In general, R290 is a better substitute for R22 in real-world applications because of its excellent environmental, thermo-physical and energy-efficient properties.

Gang and Chengfeng [11] have shown that a Modified Vaporcompression Refrigeration Cycle (MVRC) system operating with the R290/R600a zeotropic mixture is recommended for domestic refrigerator-freezers (Figure 1). Unlike the Traditional Vapor-compression Refrigeration Cycle (TVRC), a phase separator is added and the previous condenser in TVRC is subdivided into two condensers and a sub-cooler (Figure 2). In order to evaluate the thermodynamic performance of the MVRC, a mathematical model based on the first and second laws of thermodynamics is first presented. The results of the simulation show better COP, volumetric refrigeration capacity, total exergy destruction and exergetic efficiency than the TVRC operating with the R290/R600a mixture under all operating conditions due to the temperature glide induced by the zeotropic mixture. Thus it appears that the MVRC offers better performance improvements for domestic refrigeration applications. However, there remains design, operational control and cost issues that need to be addressed appropriately. The theoretical studies highlighted the potential of the new cycle. Additionally further theoretical and experimental works are still needed [11].

In addition, Xu et al. [12] investigated a microchannel condenser for a 3.3 kW refrigerant capacity propane air conditioner. They predicted by simulation a total refrigerant charge of less than 150 g for a fully optimized machine. Finally, it should be noted that none of the previously mentioned studies was conducted in tropical areas. Thus, the present study attempted to test and analyse the use of HCs as refrigerants under tropical climatic conditions.

3. Materials and methods

3.1. Materials

The refrigerant charge is done at the outdoor unit of the air conditioning system (see Figure 3). The circuit charge is made by means of a manifold connected to the suction of the compressor and to the charging cylinder. The refrigerant vessel is placed on an electronic weighing scale to track the charge. The manifold used is the model 5 ways, type IV51. It consists of two isolation valves, a balancing valve, two additional purge valves and a pressure channel vent valve allowing the operator to purge one or the other or both channels of the system.

The mean outside climatic conditions are: temperature 28°C, relative hygrometry 80%

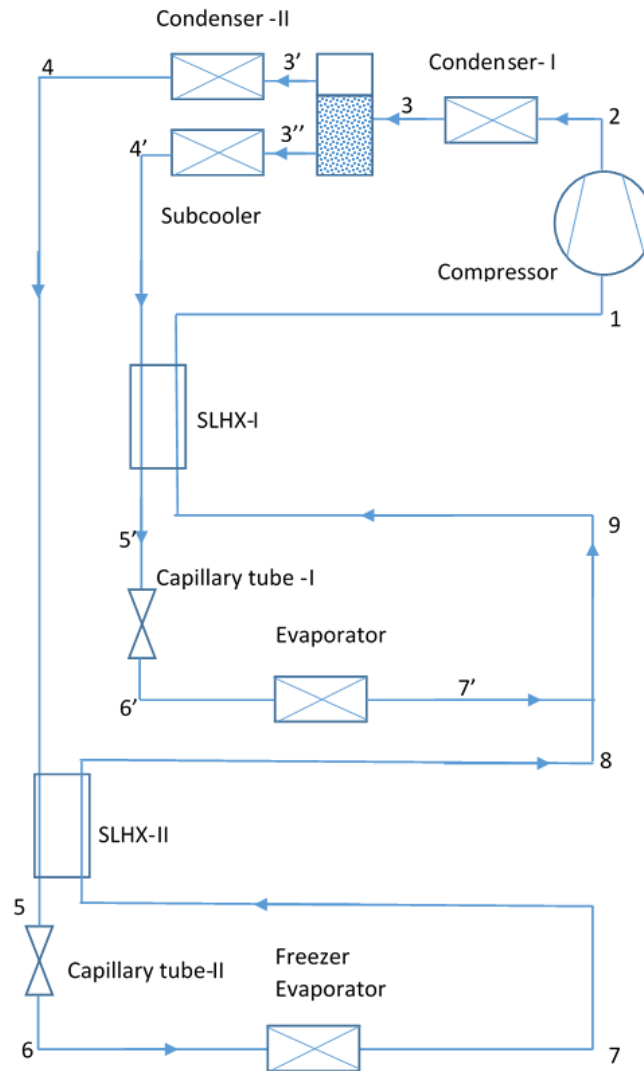


Figure 1: The schematic diagram of MVRC

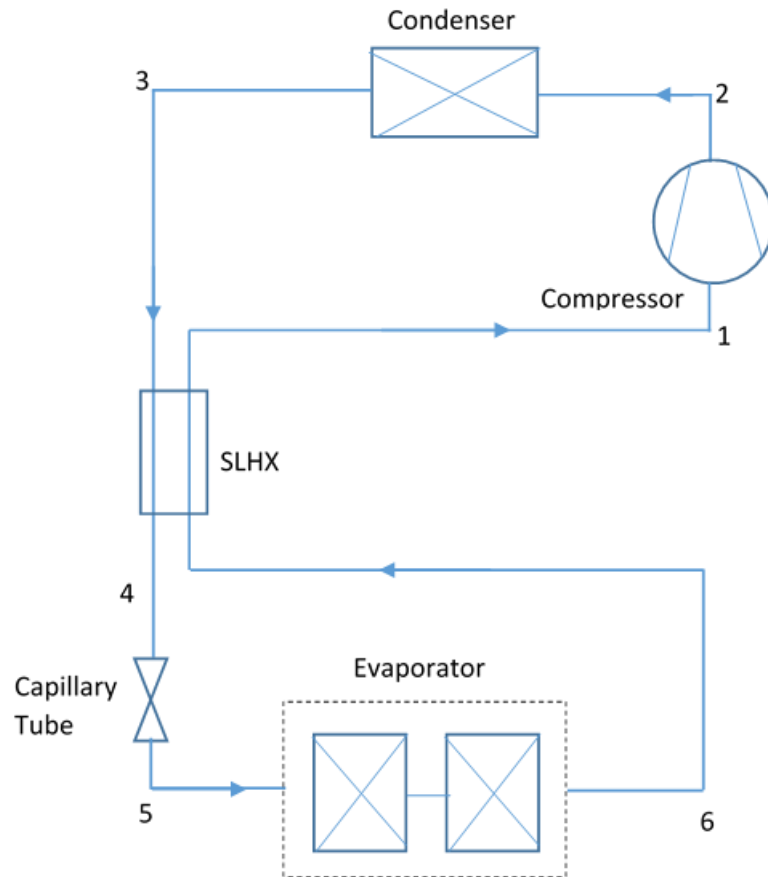


Figure 2: The schematic diagram of TVRC



Figure 3: Fluid charge on the outdoor unit.

3.2. Methods

The approach is a comparative study of the energy performance of a split air conditioner converted to refrigerants R600a and R290 under Benin climatic conditions. The calculations involved in the thermodynamic performance of the vapour compression cycle are given hereafter. The air conditioner was charged in increments of 50g of refrigerant until reaching the maximum load of 400g. With every single charge of refrigerant, a series of measurements are operated (enthalpy variation on the air passing through the evaporator, temperatures, hygrometry, fluid charge on the outdoor unit, and the current intensity consumed by the compressor). These series of measurements were post-processed on Matlab. Then for each fluid, the electrical power consumed and the coefficient of performance was discussed. Another program written on the Engineering Equation Solver (EES) software made it possible to extract the refrigeration cycle in the enthalpy diagram of each refrigerant studied.

3.2.1 Mechanical work of the compressor

Neglecting the loss of energy during energy conversion leads to:

$$\dot{W}_c = \dot{m}_f (h_2 - h_1) \quad (\text{kW}) \quad (1)$$

3.2.2 Energy balance on the compressor (Figure 4)

The energy balance on the compressor gives:

$$\dot{W}_{eff} - \dot{Q}_p = \dot{m}_f (h_2 - h_1) (\text{kW}) \quad (2)$$

Assuming that $\dot{Q}_p = \xi \dot{W}_{eff}$

$$(1 - \xi) \dot{W}_{eff} = \dot{m}_f (h_2 - h_1) \quad (3)$$

$$\dot{W}_{eff} = \frac{\dot{m}_f (h_2^{is} - h_1)}{\eta_{eff}} = UI \cos \varphi (\text{kW}) \quad (4)$$

3.2.3 The cooling capacity

The cooling capacity of an air conditioner depends on the volume flow rate and the enthalpy change of the cooled air flowing through the indoor unit:

$$Q_f = \frac{q_v}{\rho} \cdot \frac{\Delta h_{air}}{\varepsilon} \quad (5)$$

3.2.4 The volume flow

The air volume flow is important to determine to calculate the cooling capacity

$$q_v = A \cdot v_{moy} \quad (\text{m}^3/\text{s}) \quad (6)$$

The speed can be calculated according to several methods. The one we used is to average the velocities measured locally at the open space for cooled airflow:

$$v_{moy} = \frac{\sum_{i=1}^n v_i}{n} \quad \text{m/s} \quad [2] \quad (7)$$

4. Results and Discussions

4.1. Thermodynamic performance analysis

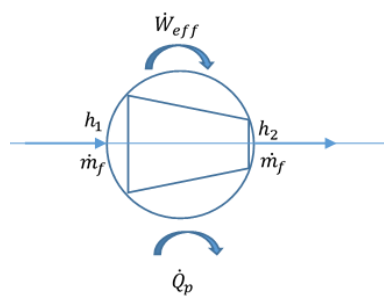


Figure 4: Moto compressor.

The analysis of the thermodynamic performance of R290 and R600a is performed on the basis of the ideal vapour compression refrigeration cycle. In this thermodynamic analysis, the pressure drops in the condenser, the evaporator and the suction and discharge valve are assumed to be zero. The subcooling and superheating degrees are taken equal to 5°C. A code of the EES software is developed to calculate the thermodynamic analysis of the vapour compression refrigeration system. The vapour compression refrigeration cycles on the enthalpy-pressure diagram for both fluids are shown in Figures 5 and 6.

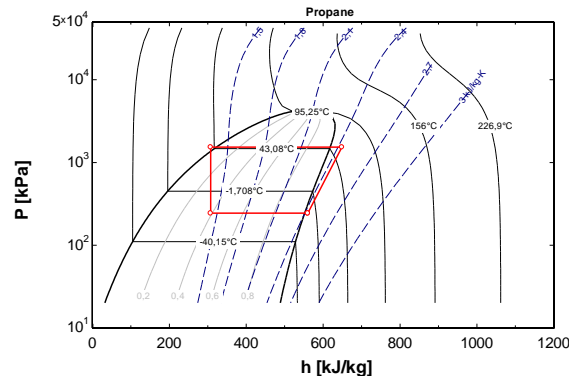


Figure 5: Ideal vapour compression refrigeration cycle of propane on the p-h diagram.

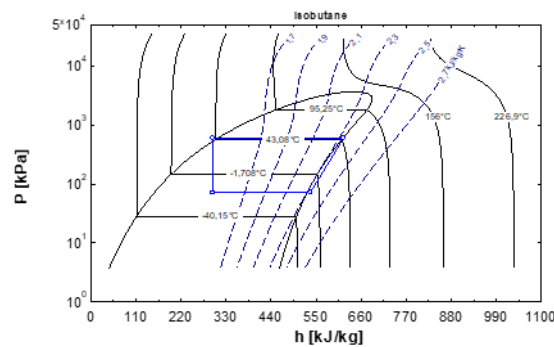


Figure 6: Ideal vapour compression refrigeration cycle of isobutane on the p-h diagram.

It can be noted that the desuperheats are respectively of 13.12 °C and 0.95 °C for propane and isobutane, which means that for the same working condition, the condenser must be oversized for a propane cycle (compared to that of isobutane).

4.2. Analysis of the energy performance

The experiments are performed on a split air conditioner of 1,47kW. Figure 7 shows the variation of the refrigerating capacity with the respective charges of refrigerants R290 and R600a. It is noted that an optimum cooling capacity prevails: 200 g with R290 and 250 g with R600a within the experimental conditions. In general, the refrigeration output of R290 exceeds that of R600a from 150 g of refrigerant charge.

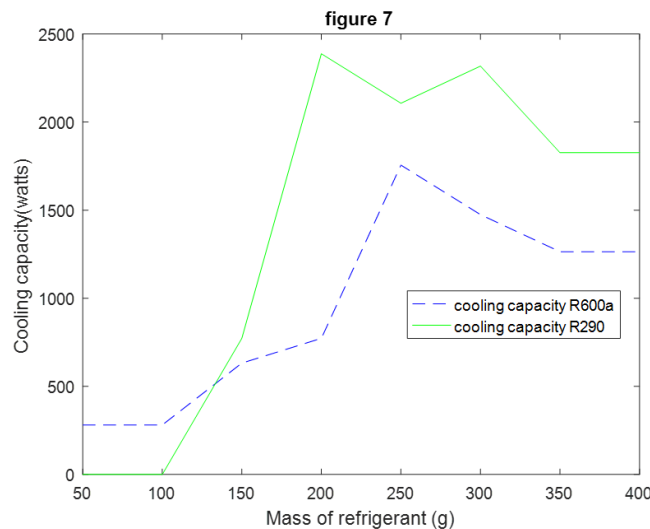


Figure 7: Evolution of the refrigerating capacity as a function of the quantity of fluid charged.

The variations of the electrical power consumed with the respective charges of the refrigerants R290 and R600a are displayed in Figure 8. The electrical power consumed increases with the charge of refrigerant for the two fluids. In general, the power consumption is higher with the R290 than the R600a.

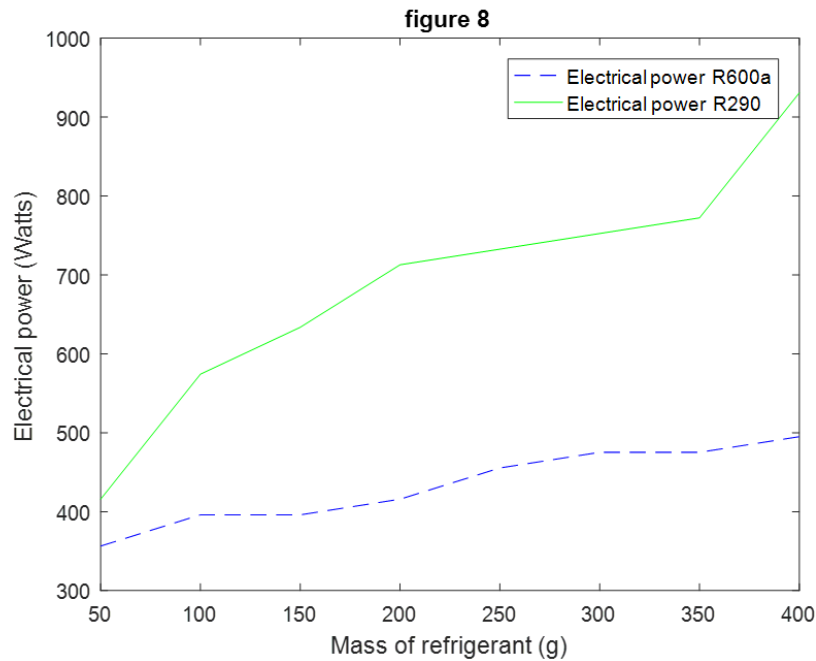


Figure 8: Evolution of electrical power as a function of the mass of refrigerant

Figure 9 shows the variation of the refrigeration efficiency (COP) with the respective charges of refrigerants R290 and R600a. It is found that an optimum efficiency prevails: 200 g with R290 and 250 g with R600a within the experimental conditions and system. The cooling efficiency is rather higher for isobutane. On the basis of the results obtained from the operation of the air conditioning system, three working zones can be defined:

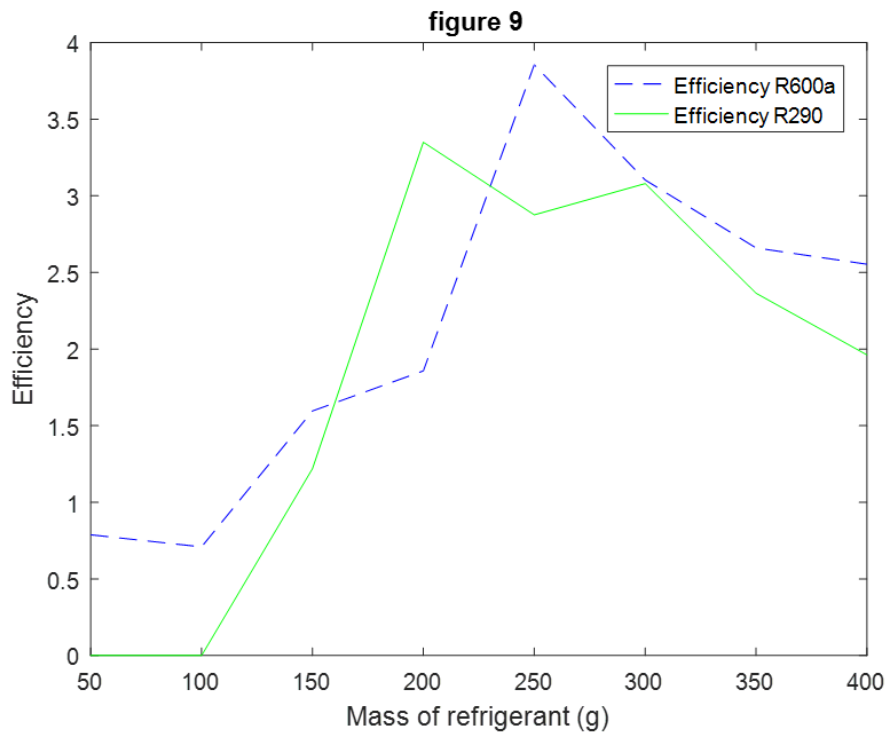


Figure 9: Evolution of efficiency as a function of the mass of refrigerant

Underload area: when the mass of refrigerant is between 50 and 150g.

The amount of refrigerant charged is lower than the normal amount required. As a consequence, the low-pressure parts of the device are covered with frost. In this zone the cooling efficiency is very low. This zone is not indicated for the economic and technical profitability of the air conditioner.

Normal charge area: the mass of refrigerant is between 150 and 300g.

The amount of refrigerant charged is approximately close to the normal amount required (250g). In this zone the cooling efficiency is high. This zone is indicated for the economic and technical profitability of the air conditioner. The R600a gives the highest refrigeration efficiency $EF = 4$. This confirms the results of the theoretical study.

Overload area: the refrigerant mass between 300 and 400g

In such a case, the amount of refrigerant charged is greater than the required one, and the unit has heated up. In this zone the cooling efficiency is low. These operation conditions are not indicated for the economic and technical profitability of the air conditioner.

5. Conclusion

This study showed the evolution of the cooling capacity supplied, the electrical power consumed and the cooling efficiency for each case of the fluid load. Three operation areas were identified for the air conditioning system: underload, normal load and overload zones. It has been found that in the indicated operating zone, i.e., the normal charge area, the best performing fluid is R600a relative to R290. The mixture of these two fluids could be studied under the conditions selected

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