Refrigerant charge optimisation for propane heat pump water heaters

Redouane Ghoubali, Paul Byrne, Frédéric Bazantay

REFRIGERANT CHARGE OPTIMISATION FOR PROPANE HEAT PUMP WATER HEATERS

Redouane Ghoubali[(a)], **Paul Byrne**[(a)], **Frédéric Bazantay**[(b)]

[(a)]) Université Européenne de Bretagne, Equipe MTRhéo, Laboratoire LGCGM,
INSA de Rennes et Université de Rennes 1
IUT Génie Civil, 3 rue du Clos Courtel, BP 90422, 35704 Rennes Cedex 7, France
[(b)] Pôle Cristal - Centre Technique Froid et Climatisation
49, avenue René Cassin 22100 Dinan, France

paul.byrne@univ-rennes1.fr
tel: +33223234297 / fax: +33223234051

Highlights

- Hydrocarbons are interesting refrigerants for heat pump water heaters
- Flammability of hydrocarbons imposes a refrigerant charge optimisation study
- Tank-wrapped D-tube, roll-bond and microchannel condensers are tested
- Roll-bond shows, with lower charge, best COP (3.17) and heating time (under 7 h)

ABSTRACT

Hydrocarbons can be interesting refrigerants in domestic facilities such as heat pumps for domestic hot water production called heat pump water heaters (HPWH). However, due to flammability and submission to severe regulations, the use of hydrocarbons requires improvements in design and optimisation of components to reduce the refrigerant charge. This paper first reviews the regulations applied to hydrocarbon technologies then discusses possibilities of charge reduction for a R290 HPWH. Mainly by selecting condensers with high thermal efficiency and low internal volume, the charge may be reduced to extremely low levels. Using the same base of refrigerating system, three types of condensers were tested: tank-wrapped D-tube, roll-bond and microchannel heat exchangers. The D-tube has a low compactness. Microchannels are penalized by the diameter of their distributors. The best performance is obtained by the roll-bond HPWH with a charge of 224 g: COP of 3.17 and heating time under 7 h.

Keywords: heat pump; water heater; propane; refrigerant charge

NOMENCLATURE

A: surface area (m^2)

Cp: specific heat ($J\ kg^{-1}\ K^{-1}$)

COP: coefficient of performance (-)

Δt: time step (s)

E: energy (J)

h_0: system installation height (m)

LFL: lower flammable limit ($kg\ m^{-3}$)

m: mass (kg)

ρ: density ($kg\ m^{-3}$)

V_t: tank volume (m^3)

$\dot{W}$: Electrical power (W)

1. INTRODUCTION

EU Regulation No 517/2014 on fluorinated greenhouse gases effect (commonly known as F-Gas) entered into force on the 1st of January 2015 throughout the European Union (European Commission, 2014). This text repeals the former Regulation 842/2006 and introduces a program to reduce emissions of greenhouse gases by 2030 and imposes a drastic phase-down of HFCs starting from 2015. In this context, the return to natural refrigerants is a promising alternative. These fluids offer interesting thermodynamic performance but their large scale development still remains limited. Several types of hydrocarbons (HC) were used as refrigerants (Byrne et al., 2014). Propane (R290) and isobutene (R600a) were among those used before the 1930s and the development of CFCs (chlorofluorocarbons). The vast majority of facilities in which the use of hydrocarbons is potentially interesting are domestic (refrigerators, air conditioners) or commercial facilities. Isobutene (R600a) is the most frequently used in hydrocarbon refrigerators. Propane (R290) begins to be implemented by the heat pump manufacturers. It is also used in air conditioners and commercial refrigeration systems. Palm (Palm, 2008) provides a non-exhaustive list of European heat pump manufacturers using propane. National regulations can also limit the use of hydrocarbons as refrigerants in higher capacity refrigerating devices.

This article first reviews regulations and standards applied to hydrocarbons in heat pumps and scientific articles dealing with this topic. The following experimental study then focusses on how to reduce the refrigerant charge in an air-source heat pump water heater. Using the same refrigerating system architecture, different types of condensers are tested: tank-wrapped D-tube, roll-bond and microchannel. A refrigerant charge optimisation is finally carried out with a maximum COP objective.

2. SAFETY CLASSIFICATION AND REGULATIONS

Two parameters have to be taken into account in terms of safety during fluid handling: flammability and toxicity. Beside the chemical formula, the global warming potential and the ODP, **Error! Reference source not found.** shows the lower and upper explosive or flammable limits, respectively LEL and UEL or LFL and UFL. These limits are equivalent and characterize the interval of concentration of the substance in air in which the mixture can ignite. If the gas concentration in air is lower than the LFL or higher than the UFL, the risk of explosion is very low. The values of LFL and UFL and of the auto-ignition temperature are quite similar for the three hydrocarbons. Propylene also appears as a valuable refrigerant. However, propane presents the lower flammability interval.
EN378-1-2008 classifies buildings into three categories depending on the occupation conditions (European Commission, 2008). The refrigerant charge admitted in a room is limited by the occupation type, the safety group and the type of technology. **Error! Reference source not found.** reports the authorized refrigerant charge in premises of type A, B and C buildings for direct systems installed where there is some occupation by people (other than in an engine room or a boiler room). The refrigerant charge is limited to 1.5 kg for the general occupation category and 2.5 kg for B-type premises like office buildings. EN 378-1-2008 reports in annex C that air conditioning systems or heat pumps with less than 150 g of A3 refrigerants can be located in an occupied space without restrictions. If the refrigerant charge is higher, the minimum room surface is calculated depending on the refrigerant charge, the lower flammability limit and the installation height. French regulation for public access buildings (Directive ERP, Etablissements Recevant du Public) explicitly bans all hydrocarbons without any charge limit, notably for HVAC applications (Ministère de l'Intérieur, 1980).

Some safety regulations were written for the use of hydrocarbons, including leakage simulation tests and specifications for several electrical components that can enter in contact with refrigerant leaks. The standards were prepared by the International Electrotechnical Commission (IEC). The following norms deal with design and testing rules about devices working with flammable fluids for household and similar electrical appliances:

- IEC 60335-2-24: Particular requirements for refrigerating appliances, ice-cream appliances and ice-makers (IEC, 2002),
- IEC 60335-2-34: Particular requirements for motor-compressors (IEC, 2012),
- IEC 60335-2-40: Particular requirements for electrical heat pumps, air-conditioners and dehumidifiers (IEC, 2016).
- IEC 60335-2-89: Particular requirements for commercial refrigerating appliances with an incorporated or remote refrigerant unit or compressor (IEC, 2015),

Norms IEC 60335-2-24 (IEC, 2002) and IEC 60335-2-89 (IEC, 2015) allow the use of hydrocarbons if the refrigerant charge is lower than 150 g for hermetically-sealed systems (every connection is soldered or brazed) (Granryd, 2001). This rule enables to use hydrocarbons in very low capacity refrigerating devices or heat pumps. However, for middle-capacity systems, there is no consensus. Indeed, some European companies sell household equipment with refrigerant charges up to 2.5 kg (Palm, 2008). Nevertheless, the ASERCOM (Association of European Component Manufacturers) still limits its guarantee to machines having a reduced charge, again lower than 150 g in confined spaces. Mechanical and electronic components used in explosive atmospheres must be ATEX certified. High hydrocarbon charges could be handled using risk management methods that exist for industrial facilities (Claret, 1995; Guilpart, 1999). Another issue, rarely addressed in the literature, regards the multiplication of the number of appliances in the same building. A domino effect could arise if these appliances start to explode one after the other. Even though, the study will be limited to a machine having a small amount of refrigerant, with a target under 150 g.

3. REVIEW OF HEAT PUMPS USING R290 AND STRATEGIES FOR CHARGE REDUCTION

The use of hydrocarbons in air conditioning and heat pump applications requires improvements in design and optimisation of components. The manufacturer's greater responsibility for issues related to the use of a flammable refrigerant must also be considered. The inherent risks due to the flammability of propane can be easily avoided by optimising the refrigerant charge in order to reach a specific charge ratio as low as possible. This section first presents the research studies on performance improvement achieved with propane heat pumps, on charge optimisation and on the influence of the oil type. The main condenser technologies are then presented. Lastly, a charge distribution calculation is carried out to compare three types of condensers available on the market.

3.1 Performance improvement with hydrocarbons

Chang et al. (Chang et al., 2000) studied the performance of a heat pump with different hydrocarbons (propane, isobutane, butane and propylene). The heat exchangers were divided into equal portions. The heat transfers were analysed by measuring the average heat transfer coefficients on each portion. They demonstrated that the heating and cooling capacities of a heat pump using R290 are slightly lower than the ones obtained with R22 but that the COP is higher. They also showed that the capacities delivered by the circuit with R1270 are higher than those offered by R22 and that the COP is improved significantly. Fernando et al. (Fernando et al., 2004) realized a heat pump prototype with a heating capacity of 5 kW with a low charge of propane. The authors demonstrated its ability to operate with a charge of 200 g with the outdoor temperatures of a Swedish climate, without reducing the COP.

Later, Park et al. (Park et al., 2008) made an experimental evaluation of a water-to-water heat pump using R433A (a mixture of 30 % propylene and 70 % propane) as an alternative to R22. They used a compressor originally designed for R22. The results showed that the COP with R433A is from 4.9 % to 7.6 % higher than the one with R22. The performance of a water-to-water heat pump using a R170 / R290 (ethane / propane) mixture was studied by Park and Jung (Park and Jung, 2007). The R170 / R290 mixture had a compressor discharge temperature between 16.6 and 28.2 K lower than the one with R22 under the same conditions. A decrease in the discharge temperature involves less thermal stress at the compressor and increases its fatigue life. Finally, Yu et al. (Yu et al., 2010) conducted the thermodynamic analysis of a transcritical cycle of a high temperature heat pump with a R32 / R290 mixture. They managed to produce domestic hot water at 90 °C. According to this first part of the literature review, the hydrocarbons facilities are performing and interest exists for refrigerant mixtures. It reveals that two paths can be studied, either using a mixture of hydrocarbons with refrigerants having a lower flammability or reducing the refrigerant charge if hydrocarbons are used solely.

3.2 Charge optimisation

Several studies have been carried out to optimise the charge in refrigerating systems using R290. Corberán et al. (Corberán et al., 2008) found that the optimum charge of a water-to-water heat pump using R290 at constant operating conditions (W30/W7 and W10/W45) is 550 g. At this optimum charge, a cooling capacity of 14 kW is obtained. Authors also studied the charge distribution within the circuit. The influence of source and sink temperatures on the optimal charge was analysed by Corberán et al. (Corberán et al., 2011) using a numerical model, the IMST-ART software. They found that the optimum charge depends strongly on the source temperature and that the variation of the sink temperature has no effect on the optimum charge. The simulation results showed that the optimal COP is obtained when the subcooling degree is around 5 to 7 K. The liquid nitrogen method (LNM) was used by Li et al. (Li et al., 2015) to determine the refrigerant charge distribution inside the components of a R290 split-type air conditioner. They found that the refrigerant charge in the heat exchangers was about 60% of the whole charge, either in cooling or heating mode. Using a liquid-to-suction heat exchanger may also reduce refrigerant charge in vapour compression refrigeration cycles (Hermes, 2015). To go further on this topic, Poggi et al. (Poggi et al., 2008) present a complete review on charge reduction in refrigerating systems. The main conclusion of this review is that the analysis of the charge distribution and of the architecture of the refrigeration system enables to identify the components that need to be optimised to reach a low refrigerant charge. They also inventory the different possibilities to minimize charge in refrigeration circuits. Xu et al. (Xu et al., 2016) used a microchannel condenser for a propane air conditioner of 3.3 kW of cooling capacity. They obtained by simulation a total refrigerant charge under 150 g for a fully optimised machine. Kheiri et al. (Kheiri et al., 2011) worked on a new design optimisation criterion for compact heat exchangers: the ecological cost. They varied the tube diameter. The charge reduction impacting the TEWI (Total Equivalent Warming Impact) of the heat pump, they showed that the minimum TEWI was correlated with the minimum ecological cost and that there was a compromise between charge reduction and pressure drop. Palm (Palm, 2007) studied refrigerant charge reduction techniques for hydrocarbons and also HFCs because of their environmental impact. He concludes that using minichannel heat exchangers reduces substantially the refrigerant charge in the circuit. The industry has been working for a long time on the subject of refrigerant charge reduction for safety and environmental reasons and because of the prices of refrigerants. For example, patents were delivered on microchannel heat exchangers (Xu et al., 2011), on D-type (De Forest et al., 1990; Silva, 1955), kidney-type (Knabben et al., 1985), flattened (Gerstmann and Swenson, 1998) or rectangular wrapped tubes (Pierce, 1984). Roll-bond condensers (Palandre, 2013; Stiebel Eltron GmbH & Co KG, 1998) are already commercialized for heat pump water heaters (AUER, 2016; Stiebel Eltron, 2016). However, system optimisation in terms of COP, heating time and refrigerant charge limitation still seems to be possible. The aim of the work is the analysis of the characteristics of existing heat pumps that must be overcame so that the optimal charge matches a charge amount below the target value of 150 g.

3.3 Oil type

Propane is highly miscible with naphthenic mineral oils (ASHRAE, 2006). Corberán et al. (Corberán et al., 2008) used a compressor with mineral oil and they found that the quantity of R290 refrigerant dissolved in the lubricant oil represent approximately 30 % of the total charge. Mineral oils are less costly and are considered as a good choice in hydrocarbons applications with high capacities and located outside buildings (categories B - Supervised occupancy and C - Authorized occupancy) where the authorized charge is higher. In a previous study (Ghoubali et al., 2014), a heat pump for simultaneous heating and cooling with a heating capacity of 20 kW was developed with a charge of 4 kg of propane. This heat pump is at the moment in an industrialization process and the choice of the mineral oil has not been a problem for the industrial partner. Nevertheless, using mineral oil can result in a significant decrease in viscosity, which can be harmful for the compressor.

Alkyl benzene Oils (AB) are less soluble but have lower viscosity than mineral oils. They have been used in small capacity installations such as refrigerators, air conditioners. Oils with lower viscosity require additives to avoid wear (Beattie and Karnaz, 2015).

Propane is also soluble in ester oils (POE). However, POE oils have less solubility in propane than mineral oils. Martinez-Galvan et al. (Martínez-Galván et al., 2011) lead to the same conclusion of a better unit performance with POE oil. Some alkylene oils (PAG) are immiscible with propane (Palm, 2008) but the global cost and high moisture absorption may limit their use in some types of refrigeration systems. To reduce the refrigerant inside the circuit, it would be preferable to implement an oil that is not soluble with the refrigerant.

3.4 Condenser technology

Plate heat exchangers are interesting solutions for compactness. The use of a double-wall plate heat exchanger allows to comply with the food safety requirements of domestic hot water heat pumps (Règles de l'Art Grenelle Environnement 2012, 2014). The requirements are dictated by regulations on sanitary water such as European standard BS EN 1717 (European Commission, 2001) and French national standard (règlement sanitaire départemental) (Préfecture d'Ille-et-Vilaine, 1997). In addition, plate heat exchangers maintain good performance with direct condensation and finally significantly reduce the refrigerant charge. However, this kind of heat exchangers is not very competitive compared to other condenser technologies. A serious issue is the fouling of the heat exchanger after accumulation of unwanted materials such as scale. Another solution is an immersed condenser for the HPWH, a choice that could not be retained because of more possible water contamination and corrosion.

In this study, a focus is made on condenser technologies with an indirect condensation located on the external surface of the tank. The wrapped-tank D-tube heat exchanger is presently the most used in the hot water tank technology. **Error! Reference source not found.**a) shows the scheme of a 10-coil tank, **Error! Reference source not found.**b), the section of a tube and **Error! Reference source not found.**c), the image of a 36-coil tank manufactured for R134a. This comparison shows the extreme configurations between a HFC HPWH and a propane machine. The number of turns is sensibly reduced with propane because of better thermodynamic properties and charge reduction constraint. Here, the expected performance is slightly the same but the expected heating time is shorter. The advantage of shifting to a propane technology is not only environmental but also financial in terms of cost reduction of fluid and matter. The D-shape was formed from a 3/8 inches round tube with a special pair of pliers then folded while wrapped around the water tank. This solution seems the simplest to ensure a satisfactory performance at the lowest manufacturing cost.

Roll-bond heat exchangers are widely used in household refrigeration applications as evaporators (Hermes et al., 2008) and as solar collectors (evaporators) (Del Col et al., 2013; Sun et al., 2014). **Error! Reference source not found.**a) shows the roll-bond placed on the tested HPWH, **Error! Reference source not found.**b) presents a section of the refrigeration circuit and **Error! Reference source not found.**c) details the shape of the refrigerant channels. They have the advantage of offering a greater heat exchange surface and a low internal volume. However, the shape of the standard roll-bond evaporator with inflation on both sides would penalize the heat transfer between the heat exchanger and the tank surface. One-side-flat roll-bonds are used in freezer sections of refrigerators. They help reduce the risk of deterioration of the tubes in contact with food and during manual defrosting. In the case of the HPWH, a roll-bond used as a condenser with a flat surface on one side offers a better thermal contact with the tank.

Microchannels are used in different applications, particularly the cooling of electronic components and automobile air conditioning. **Error! Reference source not found.**a) presents the scheme of the condenser, **Error! Reference source not found.**b) the section layout and **Error! Reference source not found.**c) shows an image of a tank equipped with a microchannel heat exchanger. Microchannel heat exchangers usually offer a considerable advantage in terms of minimization of the refrigerant charge and maximization of the heat exchange surface. This type of heat exchanger has the advantage of reducing the internal volume and intensifying heat exchange. One can achieve a significant reduction of the amount of refrigerant contained in a finned tube heat exchanger (over 80 %), and therefore a significant charge reduction over the entire refrigeration system (30 to 60 %). The technology of microchannel heat exchangers has been discussed by various authors (Cavallini et al., 2013; Fernando et al., 2008; Maqbool et al., 2012).

3.5 Charge distribution

The charge distribution in a refrigeration circuit can be calculated using the parameters of Rigot's method (Rigot, 1995). The operating conditions assumed in the following calculation are a condensation temperature of 45 °C, an evaporation temperature of 0 °C and a subcooling of 16 K (observed in a normal operation of our industrial partner's HPWH). The solubility of propane in oil is assumed equal to 10 % of total charge mass (Fernando et al., 2003). The charge calculated in the circuit is 335 g for D-tube, 145 g for roll-bond and 280 g for microchannel. With these assumptions, the objective of a heat pump water heater with less than 0.15 kg of propane is achievable only by the roll-bond.

Error! Reference source not found. shows the distribution of the refrigerant charge in the HPWH equipped with the 3 types of condensers studied in the experimental part of this article. The component holding the largest amount of refrigerant is the condenser (more than 50 % in mass). The proportion of charge in the condenser reaches 70 % in the case of the D-tube. Between 13 % and 23 % of the charge is concentrated in the liquid line because the manufactured condensers studied here are designed for low pressure drops and not low refrigerant charge. In the case of split systems, the refrigerant charge in the liquid line can reach 32 % (Poggi et al., 2008). Therefore, the second major action for charge reduction is the diminution of the internal pipe diameter of the liquid line. However, the reduction of the pipe diameter increases pressure drops and thus the risk of partial vaporization of the fluid by the flash gas phenomenon at the inlet of the throttling valve and decrease the efficiency of the refrigeration system.

The following experimental study will evaluate if these refrigerant charge calculations are representative of the optimised values for maximum COP. It will focus on the following paths emphasised by the literature review:

- Global architecture,
- Heat exchangers technology,
- Lubricating oil,
- Reducing the size of pipes and accessories.

4. EXPERIMENTAL APPARATUS

Error! Reference source not found. presents the scheme of a mono-block heat pump water heater. The thermodynamic head of the refrigeration circuit, shown in the doted square, is placed on the top of water tank. Only the condenser is wrapped around the tank. The water tank is insulated by foam injected between the tank and the outer casing. The characteristics of the components of the HPWH are listed in Tables 3 to 6.

Table 6 shows the three tested condensers. The compactness is a geometric criterion defined as the ratio of the exchange surface over the internal volume. The roll-bond heat exchanger has the higher compactness. The internal volume integrates a large part of the liquid line, which appears very voluminous for the microchannel heat exchanger headers (distributors). This fact comes from manufacturing process limitations due to mechanical constraints appearing when wrapping the microchannel heat exchanger around the tank. Actually, the headers have to host the ends of the microchannels so that a minimum length of the tubes is inside the header. The inner diameter of the header is thus necessarily quite important (9.7 mm).

The sizing of the three condensers is based on characteristics of existing HPWH using HFCs or R290. Simulations were carried out to define the minimum of number of D-tube turns needed for a propane HPWH. Some of our simulation results have shown that a minimum of 10 turns is necessary for heating up the tank within 9 hours. The roll-bond was a manufactured product for propane. For the microchannel condenser, the same exchange surface as an existing R134a D-tube HPWH was chosen. The thermal contact was realized by mechanical pressure and thermal paste (D-tube and microchannel) or thermal glue (roll-bond).

An experimental test bench was designed for the characterisation of heat pump water heaters (**Error! Reference source not found.**). The heat pump unit is composed of a rotary compressor of 8.9 cm^3 (A), initially a D-tube condenser wrapped on a 200 l tank (B), a thermostatic expansion valve (C), a finned tube evaporator coil (outdoor heat exchanger) (D) and a four-way valve (E) to switch between heating and defrosting mode. This heat pump unit has a heating capacity around 2 kW. Mineral oil (NM100PM) was implemented in the compressor by the manufacturer for cost and practicality reasons. The tests are continued with the same thermodynamic head (compressor, evaporator, air vent, thermostatic expansion valve with the same adjustment, 4-way valve and refrigerant circuit) but with a roll-bond then microchannels wrapped around the same tank type of 200 l, insulated the same way.

Calibrated K-type thermocouples (T) were used to measure temperatures along the refrigerating circuit and to evaluate the superheat and subcooling degrees. Hygrometers (RH) are located at the inlet and the outlet of the external heat exchanger to measure the moisture content in the air duct. The refrigerating circuit includes pressure transducers (P), located at the suction and the discharge lines of the compressor, at the inlet and the outlet of the condenser and finally at the outlet of the evaporator. The water temperature at the inlet and the outlet of the tank and in the middle of the tank (at the control sensor position) are measured with calibrated PT100 thermo-resistances. A wattmeter is used to measure the electrical consumption of the heat pump (compressor and fan) in order to calculate the COP.

The HPWH is located in a climatic chamber, in which the air temperature and humidity can be well controlled, ensuring the stability of the test conditions. A test was considered valid if the test conditions are maintained within the tolerances equivalent to those of standard EN14511 (European Commission, 2013) during the temperature rise. The permissible variation on the arithmetic mean of the values compared to reference values is ± 0.5 K. Allowable deviation of measured individual values against target values is ± 2 K. Table 5 shows the measurement uncertainties of the instrumentation.

An initial temperature of the fully mixed tank of 10 °C is maintained before testing by a thermal regulation loop during 30 min. The tests are carried out until the temperature of the entire tank rises up to 53.3 °C. Each manufacturer selects a reference hot water temperature (with a minimum requirement of 52.5 °C) that should result in a short heating time. The higher the reference hot water temperature is with a low heating time and the better is the COP_{pivot}. The value of the reference hot water temperature (θ'_{WH}) is used in the IdCET tool for the calculation of COP_{pivot} required in the French thermal regulation 2012 (CSTB, 2012). Reference hot water temperature and heating time of several heat pump water heaters are listed in the website of LCIE Bureau Veritas (LCIE, 2016). This specific reference hot water temperature of 53.3 °C was chosen to approach the performance of a commercial

HPWH available in this public list that uses propane and a roll-bond condenser (AUER, 2016). The conditions of temperature and relative humidity in the climatic chamber are 7 °C and 87 %. A significant downtime between each test (at least 2 hours) is respected to permit the cooling of the compressor bell.

During the test, the condensation pressure increases as the average water temperature increases. The heating energy of the unit during the temperature rise of the tank is given by Eq. 1, between water temperatures from 28 °C to 38 °C at the centre of the tank. Between 28 °C and 38 °C, the compressor operates under optimal conditions and the COP reflects the behaviour of the HPWH without the transient phase during starting-up and the performance loss at the end of the heating-up. This criterion is employed by manufacturers to compare new systems with other existing HPWH.

$$E_{heating} = \rho \cdot V_t \cdot Cp \cdot (38 - 28) \tag{1}$$

The electric energy consumed during the test is evaluated as the product of the time step and the sum of the electric power consumed, including auxiliaries, during the water temperature elevation from 28 °C to 38 °C (Eq. 2). The time step Δt between two measurement acquisitions is equal to 10 s.

$$E_{electric} = \Delta t \cdot \sum_{28}^{38} \dot{W} \tag{2}$$

Finally, the coefficient of performance (COP) is calculated using Eq. 3.

$$COP = \frac{E_{heating}}{E_{electric}} \tag{3}$$

5. RESULTS AND ANALYSIS

5.1 Performance and heating time

Fig. 7 shows the evolution of the COP when the charge is varied. Among all tests, the maximum COP is obtained with the microchannel with a charge of 300 g. For the roll-bond heat exchanger, the COP increases with increasing refrigerant charge with a stabilization for 200 g. As observed, the COP presents a clear maximum with a charge of about 250 g for the roll-bond. The minimum charge that ensures a correct operation to the heat pump is 200 g for D-tube and microchannels and 120 g for the roll-bond. An operation is considered as correct when the subcooling degree is sufficient to avoid flash gas. The minimum quantity for D-tube and microchannels exceeds by far the maximum refrigerant charge (150 g) authorized in refrigeration systems for household use. The higher roll-bond performance with lower charge is due to first, a relatively low liquid line volume and second, a good contact between the metals of the roll-bond and the water tank. In the case of D-tube and microchannel condensers, a thermal paste was added because the contact was not perfect between the heat exchanger and the tank. This induced locally an additional thermal resistance to the heat transfer.

Fig. 8 shows the heating time or the time needed to heat up the middle of the tank at 53.3 °C (at the position of the control sensor). It is noticeable that the heating time decreases with the increase of the refrigerant charge for the 3 types of condensers. The energies exchanged at the desuperheating and condensation remain almost constant when increasing the charge. However, the subcooling portion increases significantly by increasing the refrigerant charge as shown in Fig. 9. This explains the significant reduction of the heating time and the improvement of the COP. This is observed for the 3 types of condensers. The tank takes a long time to warm up with the D-tube condenser.

The layout of the end of the heat exchange surface of the condenser is decisive for a good subcooling management that leads to better performance. On the tested roll-bond condenser, the area of subcooling is on the top of the exchanger, in thermal contact with nearly the hottest part of the water tank after a short heating time due to the thermosiphon effect in the tank water. The highest layers of water are warmer so less dense. The refrigerant in the subcooling region of the roll-bond condenser may even be heated by the tank water. The refrigerant would undergo a higher subcooling if the outlet of the condenser were located at the bottom of the tank.

Error! Reference source not found. shows the superheat evolution when the charge is varied for the three types of condensers. Superheat remains stable throughout the tests for the four charges tested with the tank-wrapped D-tube and microchannels. Superheat slightly decreases with the increase of the charge in the installation. However, in the case of the roll-bond, superheat decreases with the charge increase up to 170 g and becomes stable after. This behaviour can be explained by the fact that the thermostatic expansion valve is driven by the temperature of the refrigerant at the outlet of the evaporator. The superheat degree increases at low refrigerant charge because the thermostatic expansion valve does not work in its nominal operating range.

For an application to hydrocarbons, 3/8-inch D-tube cannot meet the charge criteria. The exchange surface appears too low for a tank volume of 200 l. Further tests would eventually be needed to evaluate the performance and the heating time with smaller tanks of 150 or 100 l.

5.2 COP optimisation

Error! Reference source not found. shows the results of the charge optimisation for maximum COP according to the polynomial interpolations of Fig. 7. Even if the number of experimental points is limited for D-tube and microchannel condensers, a tendency can be observed. The microchannel technology would provide the best COP but the higher liquid line volume leads to a higher optimal charge. Following the interpolation, the best performance is achieved with the roll-bond with a charge of 224 g with a COP of 3.17 and a heating time under 7 h because of the condensation part nearer to the position of the control sensor than with the other two condensers. However, with higher charge levels, the HPWH equipped with microchannels would have given lower heating times. The power-type interpolations presented in **Error! Reference source not found.** estimates the heating time at 5 h 26 min for 423 g.

The COP and the heating time were calculated for a refrigerant charge of 150 g using the interpolation equations from **Error! Reference source not found.** and **Error! Reference source not found.** in the case of the roll-bond. This condenser offers an acceptable COP of 2.76 and a heating time of 9 h 7 min.

The minimum room floor surface area according to annex C of EN 378-1-2008 (European Commission, 2008) is calculated using Eq. 4, in which *m* is the refrigerant charge, *LFL* is the lower flammable limit of propane equal to 0.038 kg m^{-3} and h_0 is the system height equal to 0.6 m when installed on the ground. As a comparison, the minimum surface area for a system implemented with a charge of 150 g is 36 m^2.

$$A_{min} = \left(\frac{m}{2.5 \cdot LFL^{5/4} \cdot h_0}\right)^2 \quad (4)$$

5.3 Evolution of pressures

Fig. 11 and Fig. 12 show the evolutions of maximum condensing and evaporating relative pressures when the charge is varied. The evaporating pressure remains stable throughout the tests for the three condensers tested. The average value of the low pressure is 3.8 bar relative. This pressure corresponds to an evaporating temperature of 0 °C.

The high pressure (Fig. 11) increases when the charge increases. A maximum of nearly 29 bar relative is reached at the end of the heating cycle with a charge of 300 g with the roll-bond. With this charge in the installation, the pressure reached the cutting pressure of the compressor of 29 bar relative.

As shown in Fig. 12, a higher mean evaporating pressure observed for 300 g with the roll-bond condenser can be highlighted. It can be explained by the fact that every extra-charge is mainly directed to the condenser, enlarges the subcooling part in the condenser, decreases the surface allocated to condensation heat transfer and increases the temperature difference between the refrigerant and the tank water in the vicinity of the condenser. As the roll-bond heat exchanger has the lowest internal volume, the phenomenon is particularly marked. This effect is also linked to the amount of energy absorbed in the evaporator, which would be higher for larger charges under equivalent operating conditions. To compensate the charge increase, the thermostatic valve closes slightly and the amount of refrigerant flowing through the system is reduced. This causes an increase of the high pressure.

5.4 Influence of lubricating oil

Some compressor manufacturers recommend a difference of 5 K between an evaluation of the oil temperature and the condensation temperature. The oil temperature is approximated by a thermocouple placed and insulated at the bottom of the compressor crankcase. Fig. 13 shows this differential in the case of the microchannel condenser. The temperature differential is greater than 5 K during the heating-up of the tank and this was observed for the three condensers tested. With a lower refrigerant charge (e.g. 202 g), less fluid should be dissolved in the oil because the temperature difference is higher. Rotary compressors offer generally less desuperheating energy because the degassing of the fluid contained in the oil refreshes the discharge gases for a considerable time period during the starting-up (approximately 1 hour). Due to their internal geometry, reciprocating compressors are not affected by this desuperheating energy loss but their efficiencies are generally lower (according to previous tests conducted by their manufacturer).

6. CONCLUSION

After reviewing standards and scientific research on the topic, this paper presents a COP optimisation study depending on the refrigerant charge for a R290 heat pump water heater with three different types of condensers. All condensers should be able to lead to very similar performance in terms of COP and heating times, the difference relies in the total heat exchange area and the refrigerant charge inside the condenser. The main conclusions that can be drawn from this study are the following:

- The 3/8-inch D-tube condenser does not allow to reach a high compactness ratio for a use with hydrocarbons. The heat exchange coefficients on the refrigerant side are too small relatively to the necessary amount of fluid. Despite a very low surface area of 0.24 m^2, the internal volume is 730 cm^3.
- The microchannel heat exchanger is known to provide the best local heat transfer coefficients but the refrigerant charge needed is still very high for domestic use. Other shapes of microchannels can be designed to maintain the same surface area and a low internal volume. Some other improvement paths were pointed out such as decreasing lengths and diameters of the condenser outlet tubes.
- The roll-bond heat exchanger offers a good compromise between high local heat transfer coefficients, good thermal contact with the water tank and low liquid line volume. With a charge of 150 g, the HPWH has a COP of 2.76 and a heating time around 9 hours.

Nevertheless, the objective of 150 g with satisfactory COP and heating time is not completely achieved in the present study. Indeed 145 g is the estimated charge for a HPWH with a roll-bond whereas the experimental optimisation result is 224 g. This difference is partly due to the particular layout of the tubes in the roll-bond used in the experiments. The area of subcooling is on the top of the heat exchanger, in thermal contact with the hottest part of the water tank after a short heating time.

However, the main effect seems to come from the position of the condensation part in the heat exchanger, higher and closer to the control temperature sensor if there is more liquid in the lower part. To reach a satisfactory performance with a refrigerant charge of 150 g in a HPWH heating a tank of 200 l, more paths should be explored such as reducing the diameter of distributors and liquid lines, testing compressors working with POE oil and modifying the layout of the condenser heat exchange surface along the tank. These measures should continue to improve the COP and decrease the heating time using less refrigerant charge.

ACKNOWLEDGEMENTS

The authors would like to acknowledge the support of GFC industries and Pôle Cristal, technical centre specialized in technologies of refrigeration and HVAC.

REFERENCES

ASHRAE, 2006. Lubricants in refrigeration systems, in: ASHRAE Handbook of Refrigeration, SI Edition.

AUER, 2016. EDEL : Chauffe-eau thermodynamique 200 & 270 L [WWW Document]. URL http://www.auer.fr/fr/produits/chauffe-eau-thermodynamique/edel/ (accessed 12.12.16).

Beattie, R.J., Karnaz, J.A., 2015. L'alliance des fluides à faible GWP et des lubrifiants. Rev. Général du Froid 1154, 27–38.

Byrne, P., Ghoubali, R., Miriel, J., 2014. Scroll compressor modelling for heat pumps using hydrocarbons as refrigerants. Int. J. Refrig. 41, 1–13. doi:10.1016/j.ijrefrig.2013.06.003

Cavallini, A., Del Col, D., Rossetto, L., 2013. Heat transfer and pressure drop of natural refrigerants in minichannels (low charge equipment). Int. J. Refrig. 36, 287–300. doi:10.1016/j.ijrefrig.2012.11.005

Chang, Y.S., Kim, M.S., Ro, S.T., 2000. Performance and heat transfer characteristics of hydrocarbon refrigerants in a heat pump system. Int. J. Refrig. 23, 232–242. doi:10.1016/S0140-7007(99)00042-0

Claret, J.L., 1995. Retour d'expérience. L'ammoniac et la réfrigération.

Corberán, J.-M., Martínez-Galván, I., Martínez-Ballester, S., Gonzálvez-Maciá, J., Royo-Pastor, R., 2011. Influence of the source and sink temperatures on the optimal refrigerant charge of a water-to-water heat pump. Int. J. Refrig. 34, 881–892. doi:10.1016/j.ijrefrig.2011.01.009

Corberán, J.M., Martínez, I.O., Gonzálvez, J., 2008. Charge optimisation study of a reversible water-to-water propane heat pump. Int. J. Refrig. 31, 716–726. doi:10.1016/j.ijrefrig.2007.12.011

CSTB, 2012. Règlementation Thermique des Bâtiments Neufs. Fiche d'application : Saisie des chauffe-eau thermodynamiques à compression électrique.

De Forest, L., Roberts, I.D., Dixon, C.W.S., 1990. Heat exchanger. US4918938.

Del Col, D., Padovan, A., Bortolato, M., Dai Prè, M., Zambolin, E., 2013. Thermal performance of flat plate solar collectors with sheet-and-tube and roll-bond absorbers. Energy 58, 258–269. doi:10.1016/j.energy.2013.05.058

European Commission, 2014. Regulation (EU) No 517/2014 : On fluorinated greenhouse gases and repealing Regulation (EC) No 842/2006. Off. J. Eur. Union.

European Commission, 2013. EN 14511: Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling.

European Commission, 2008. EN 378-1-2008: Refrigerating systems and heat pumps - Safety and environmental requirements - Parts 1: Basic requirements, definitions, classification and selection criteria.

European Commission, 2001. BS EN 1717: Protection against pollution of potable water in water installations and general requirements of devices to prevent pollution by backflow.

Fernando, P., Palm, B., Ameel, T., Lundqvist, P., Granryd, E., 2008. A minichannel aluminium tube heat exchanger - Part II: Evaporator performance with propane. Int. J. Refrig. 31, 681–695. doi:10.1016/j.ijrefrig.2008.02.012

Fernando, P., Palm, B., Lundqvist, P., Granryd, E., 2004. Propane heat pump with low refrigerant charge: design and laboratory tests. Int. J. Refrig. 27, 761–773. doi:10.1016/j.ijrefrig.2004.06.012

Fernando, W.P.D., Han, H., Palm, B., Granryd, E., Lundqvist, P., 2003. The Solubility of Propane (R290) with Commonly Used Compressor Lubrication Oils, in: International Conference on Compressors and Their Systems, IMECHE Conference Transactions. Professional Engineering Publishing.

Gerstmann, J., Swenson, P.F., 1998. Two-pipe heat pump system with isolated tank coil for domestic hot water. US5772113.

Ghoubali, R., Byrne, P., Miriel, J., Bazantay, F., 2014. Simulation study of a heat pump for simultaneous heating and cooling coupled to buildings. Energy Build. 72, 141–149. doi:10.1016/j.enbuild.2013.12.047

Granryd, E., 2001. Hydrocarbons as refrigerants - an overview. Int. J. Refrig. 24, 15–24. doi:10.1016/S0140-7007(00)00065-7

Guilpart, J., 1999. Evaluation des risques liés aux fluides frigorigènes naturels. Tech. l'Ingénieur

BE9750.

Hermes, C.J.L., 2015. Refrigerant charge reduction in vapor compression refrigeration cycles via liquid-to-suction heat exchange. Int. J. Refrig. 52, 93–99. doi:10.1016/j.ijrefrig.2014.12.014

Hermes, C.J.L., Melo, C., Negrão, C.O.R., 2008. A numerical simulation model for plate-type, roll-bond evaporators. Int. J. Refrig. 31, 335–347. doi:10.1016/j.ijrefrig.2007.05.005

IEC, 2016. IEC 60335-2-40: Household and similar electrical appliances - Safety - Part 2-40: Particular requirements for electrical heat pumps, air-conditioners and dehumidifiers.

IEC, 2015. IEC 60335-2-89: Household and similar electrical appliances - Safety - Part 2-89: Particular requirements for commercial refrigerating appliances with an incorporated or remote refrigerant unit or compressor.

IEC, 2012. IEC 60335-2-34: Household and similar electrical appliances - Safety - Part 2-34: Particular requirements for motor-compressors.

IEC, 2002. IEC 60335-2-24: Household and similar electrical appliances – Safety – Part 2-24: Particular requirements for refrigerating appliances, ice-cream appliances and ice-makers.

Kheiri, A., Feidt, M., Boussehain, R., Costea, M., Popescu, T., 2011. Refrigerant charge reduction: On a new design optimization criterion for compact heat exchangers. Int. J. Refrig. 34, 1462–1470. doi:10.1016/j.ijrefrig.2011.04.010

Knabben, H., Kunz, J., Kordon, R., 1985. Récipient d'échangeur de chaleur avec au moins une portion de paroi garnie d'un tube. FR2549214.

LCIE, 2016. LCIE Bureau Veritas [WWW Document]. URL http://www.lcie.fr/datacert/pdf/Chauffe-eau_thermodynamique.htm (accessed 12.12.16).

Li, T., Lu, J., Chen, L., He, D., Qiu, X., Li, H., Liu, Z., 2015. Measurement of refrigerant mass distribution within a R290 split air conditioner. Int. J. Refrig. 57, 163–172. doi:http://dx.doi.org/10.1016/j.ijrefrig.2015.05.012

Maqbool, M.H., Palm, B., Khodabandeh, R., 2012. Experimental investigation of dryout of propane in uniformly heated single vertical mini-channels. Exp. Therm. Fluid Sci. 37, 121–129. doi:10.1016/j.expthermflusci.2011.10.012

Martínez-Galván, I., Gonzálvez-Maciá, J., Corberán, J.-M., Royo-Pastor, R., 2011. Oil type influence on the optimal charge and performance of a propane chiller. Int. J. Refrig. 34, 1000–1007. doi:10.1016/j.ijrefrig.2011.01.012

Ministère de l'Intérieur, 1980. Arrêté du 25 juin 1980 portant approbation des dispositions générales du règlement de sécurité contre les risques d'incendie et de panique dans les établissements recevant du public (ERP).

Palandre, L., 2013. Condenseur pour chauffe-eau thermodynamique. FR2963416.

Palm, B., 2008. Hydrocarbons as refrigerants in small heat pump and refrigeration systems - A review. Int. J. Refrig. 31, 552–563. doi:10.1016/j.ijrefrig.2007.11.016

Palm, B., 2007. Refrigeration systems with minimum charge of refrigerant. Appl. Therm. Eng. 27, 1693–1701. doi:10.1016/j.applthermaleng.2006.07.017

Park, K.-J., Shim, Y.-B., Jung, D., 2008. Performance of R433A for replacing HCFC22 used in residential air-conditioners and heat pumps. Appl. Energy 85, 896–900. doi:10.1016/j.apenergy.2007.11.003

Park, K.J., Jung, D., 2007. Thermodynamic performance of R502 alternative refrigerant mixtures for low temperature and transport applications. Energy Convers. Manag. 48, 3084–3089. doi:10.1016/j.enconman.2007.05.003

Pierce, J.E., 1984. Energy efficient water heating device and system. US4452050.

Poggi, F., Macchi-Tejeda, H., Leducq, D., Bontemps, A., 2008. Refrigerant charge in refrigerating systems and strategies of charge reduction. Int. J. Refrig. 31, 353–370. doi:10.1016/j.ijrefrig.2007.05.014

Préfecture d'Ille-et-Vilaine, 1997. Réglement sanitaire départemental.

Règles de l'Art Grenelle Environnement 2012, 2014. Installations d'eau chaude sanitaire. Confort, prévention des risques et maîtrise des consommations.

Rigot, G., 1995. Prédiction de la charge en fluide des échangeurs. Rev. Générale du Froid 47–50.

Silva, W.C., 1955. Water heating systems of the heat pump type. US2716866.

Stiebel Eltron, 2016. Accelera® E heat pump water heaters [WWW Document]. URL http://www.stiebel-eltron-usa.com/sites/default/files/pdf/brochure-accelera-e_0.pdf (accessed 12.12.16).

Stiebel Eltron GmbH & Co KG, 1998. Warmwasserspeicher. DE29811953U1.

Sun, X., Wu, J., Dai, Y., Wang, R., 2014. Experimental study on roll-bond collector/evaporator with optimized-channel used in direct expansion solar assisted heat pump water heating system. Appl. Therm. Eng. 66, 571–579. doi:10.1016/j.applthermaleng.2014.02.060

Xu, B., Wang, Y., Chen, J., Li, F., Li, D., Pan, X., 2016. Investigation of domestic air conditioner with a novel low charge microchannel condenser suitable for hydrocarbon refrigerant. Meas. J. Int. Meas. Confed. 90, 338–348. doi:10.1016/j.measurement.2016.04.034

Xu, X., Cheng, S., Fan, Q., Lai, J., 2011. Micro-channel heat exchanger for heat pump water heater. CN201706810.

Yu, J., Xu, Z., Tian, G., 2010. A thermodynamic analysis of a transcritical cycle with refrigerant mixture R32/R290 for a small heat pump water heater. Energy Build. 42, 2431–2436. doi:10.1016/j.enbuild.2010.08.016

Fig. 1: 200 l tank wrapped around with a D-tube

Fig. 2: Roll-bond heat exchanger

Fig. 3: Microchannel heat exchanger

Fig. 4: Charge distribution in HPWHs equipped with different condensers

Fig. 5: Scheme of the HPWH

Fig. 6: Scheme of the HPWH experimental test bench

Fig. 7: Evolution of the COP depending on the refrigerant charge

Fig. 8: Evolution of the heating time depending on the refrigerant charge

Fig. 9: Evolution of the subcooling degree depending on the refrigerant charge

Fig. 10: Evolution of the superheat degree depending on the refrigerant charge

Fig. 11: Maximum condensing pressure depending on the refrigerant charge

Fig. 12: Mean evaporating pressure depending on the refrigerant charge

Fig. 13: Temperature difference between the bottom of compressor and the condensing fluid with a microchannel condenser for 4 refrigerant charges

Table 1: Some characteristics of three hydrocarbons

Fluid	Chemical formula	$GWP_{100years}$ ($kg_{eq}CO_2\ kg^{-1}$)	ODP	LFL % volume	UFL % volume	Auto-ignition temperature (°C)
R290 Propane	$CH_3CH_2CH_3$	3	0	2,2	10	470
R600a Isobutane	C_4H_{10}	3	0	1,8	9,8	460
R1270 Propylene	C_3H_6	2	0	2	11,2	485

Table 2: Quantity of fluid permitted in buildings of type A, B and C (direct systems)

Fluids		Hydrocarbons
Security group (EN 378 1-2012/ASHRAE 34 -2010)		A3
European Directive for under-pressure equipment		Class 1
Occupational category	A - General occupancy (residential, hotels, schools...)	< 1.5 kg
	B - Supervised occupancy (offices, laboratories, manufacturing space...)	1 kg below ground level and 2.5 kg above ground level
	C - Authorized occupancy (refineries, cold storage...)	1 kg below ground level and 10 kg above ground level

Table 3: Specifications of the compressor

Type	Rotary compressor
Displacement ($cm^3\ rev^{-1}$)	8.9
Oil quantity (cm^3)	250
Oil type	Mineral (NM100PM)
Solubility (%)	10 %

Table 4: Specifications of the evaporator

Type	Finned tube
Volume (cm^3)	119
Tube material	copper
Tube length (mm)	350
Tube outer diameter (mm)	5
Tube inner diameter (mm)	3
Tube pitch (mm)	19.05
Depth row pitch (mm)	12.7
Number tube rows	3
Number of tubes	48

Table 5: Specifications of the piping

Type	Liquid line			Compressor-to-condenser pipe	Evaporator-to-compressor pipe
	D-tube	Roll-bond	Microchannel		
Length (m)	2	1.4	1.6	0.5	0.25
Inner Diameter (mm)	7.53	7.53	9.7	4.35	6.53
Volume (cm^3)	92	62	118	22	11

Table 6: Specifications of condensers

Condenser	Number of turns	Inner section of the channel (cm^2)	Heat exchange surface (m^2)	Internal volume (cm^3)	Liquid line volume (cm^3)	Compactness (m^2 m^{-3})
D-tube	10	0.465	0.24	730	92	1096
Roll-bond	-	0.08	0.66	223	62	2960
Microchannel	-	0.0036	0.784	441	118	1778

Table 7: Specifications of the instrumentation

Instrument	Type	Precision
Thermocouple	Type K (contact probe)	± 1 K
Pressure sensor	Electronic manometer	± 0.5 %
Thermo-resistance	PT 100	± 0.1 K
Power meter	High precision wattmeter	± 2 %
Hygrometer	Capacitive hygrometer	± 1 %

Table 8: Results of charge optimisation for maximum COP

Condenser type	**COP max**	**Charge (g)**	**Estimated heating time (hh:mm:ss)**	**Minimum room size according to EN378-1-2008 Annex C (m^2)**
D-Tube	2.59	357	07:52:44	201
Roll-bond	3.17	224	06:45:53	79
Microchannels	3.71	423	05:26:05	283

Figure 1.

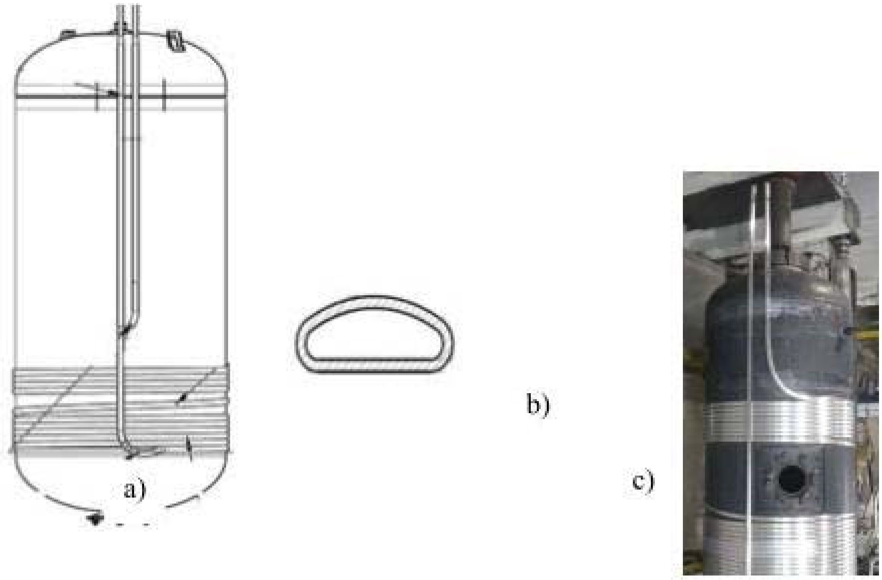


Figure 2.

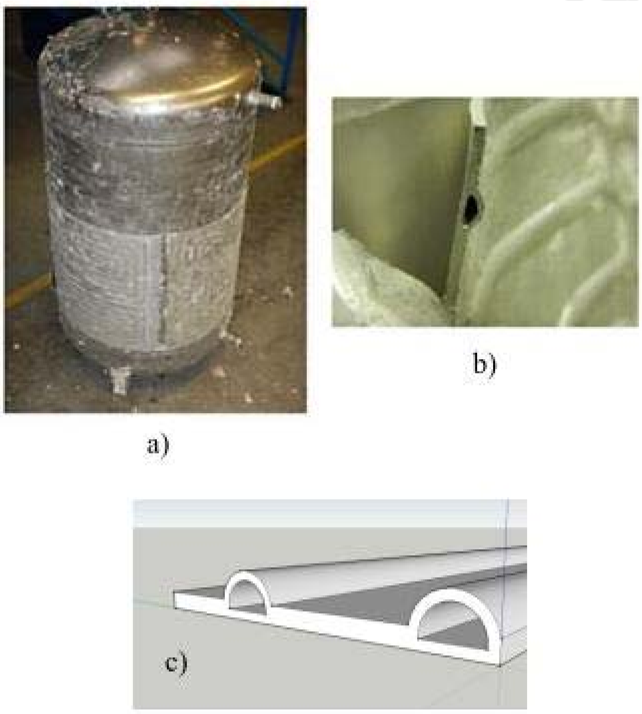


Figure 3.

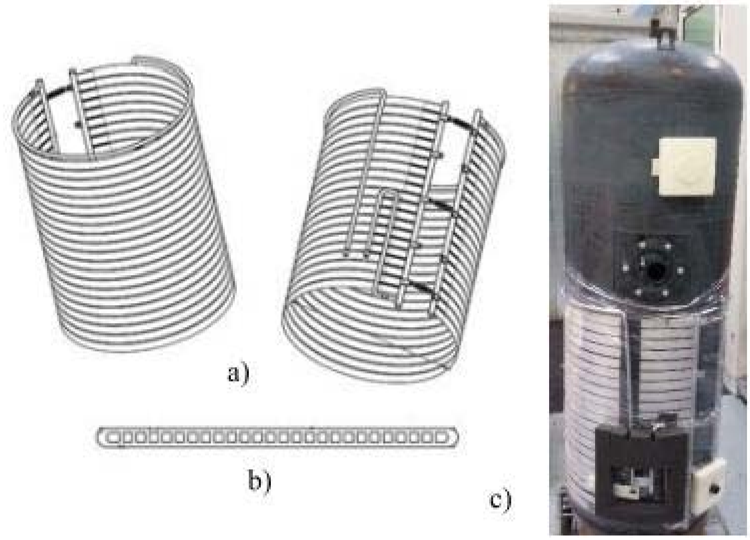
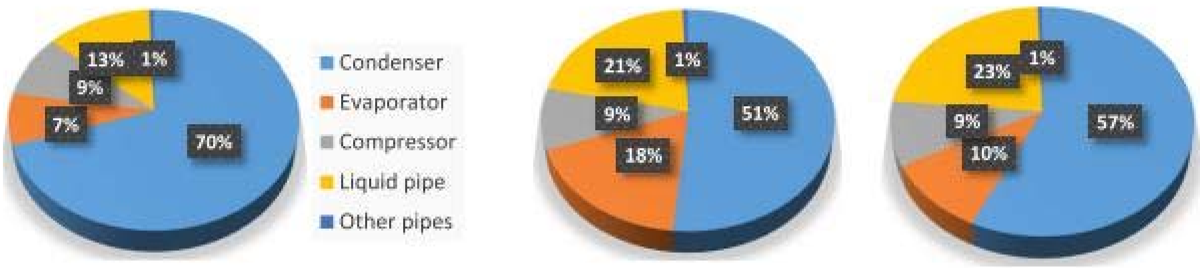


Figure 4.



a) D-tube　　b) Roll-bond　　c) Microchannel

Figure 5.

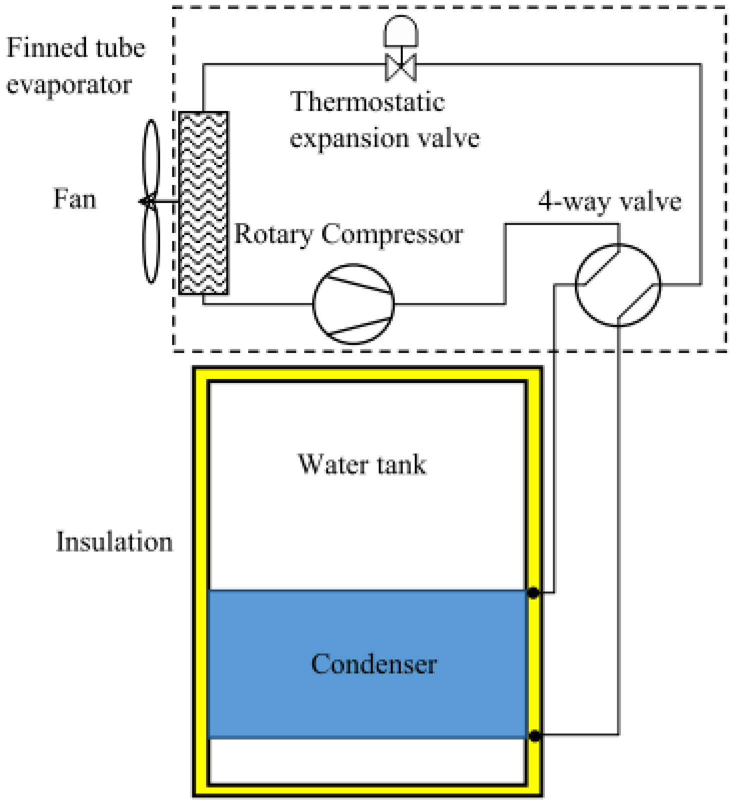


Figure 6.

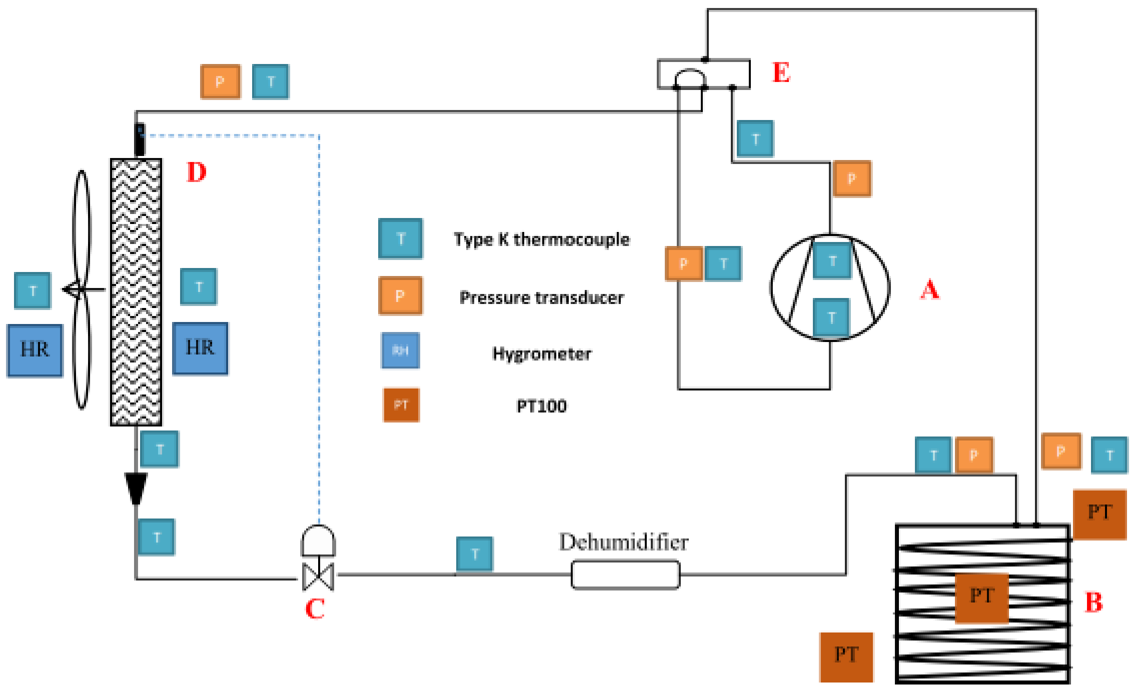


Figure 7.

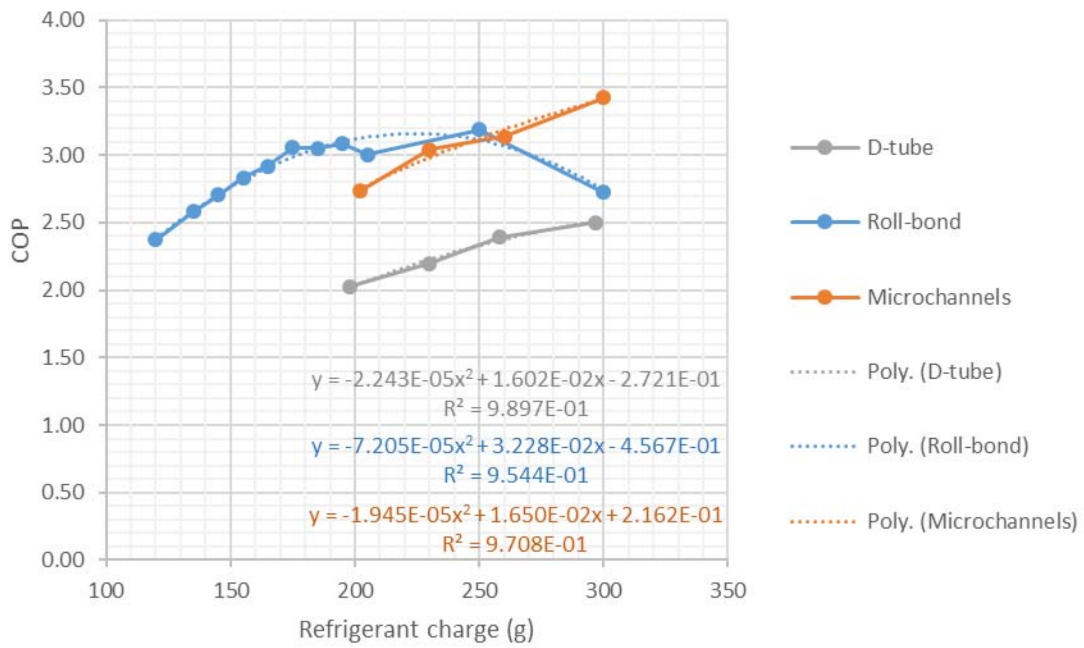


Figure 8.

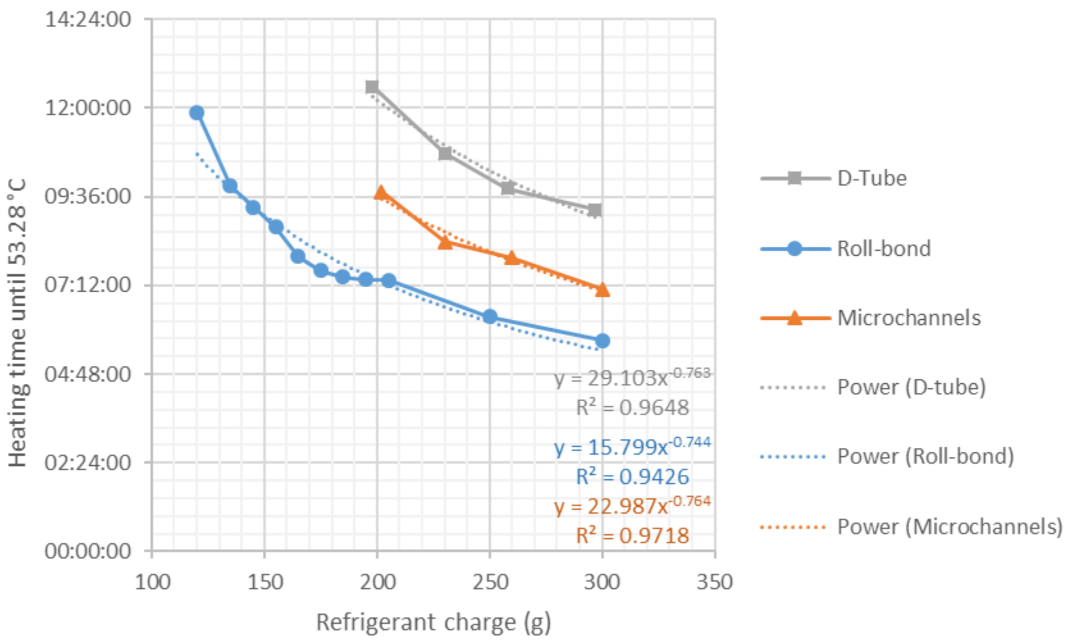


Figure 9.

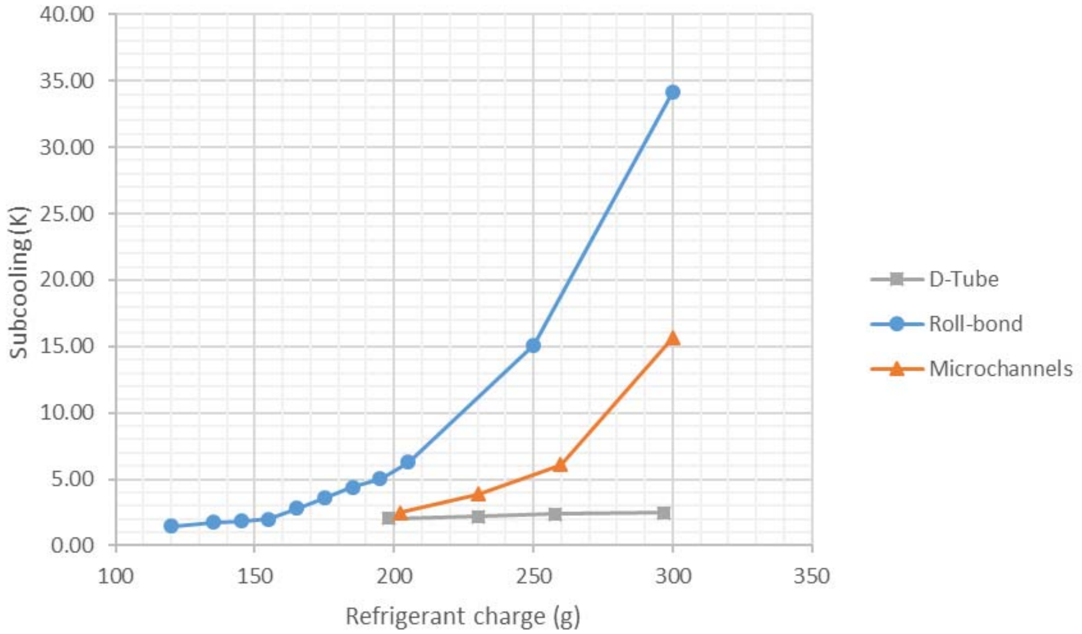


Figure 10.

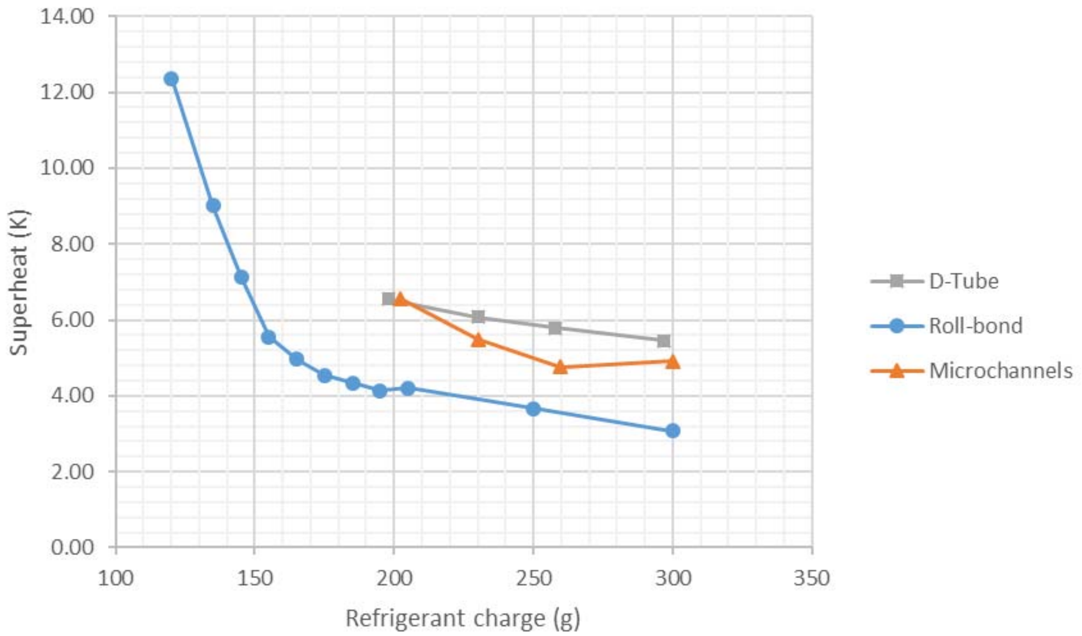


Figure 11.

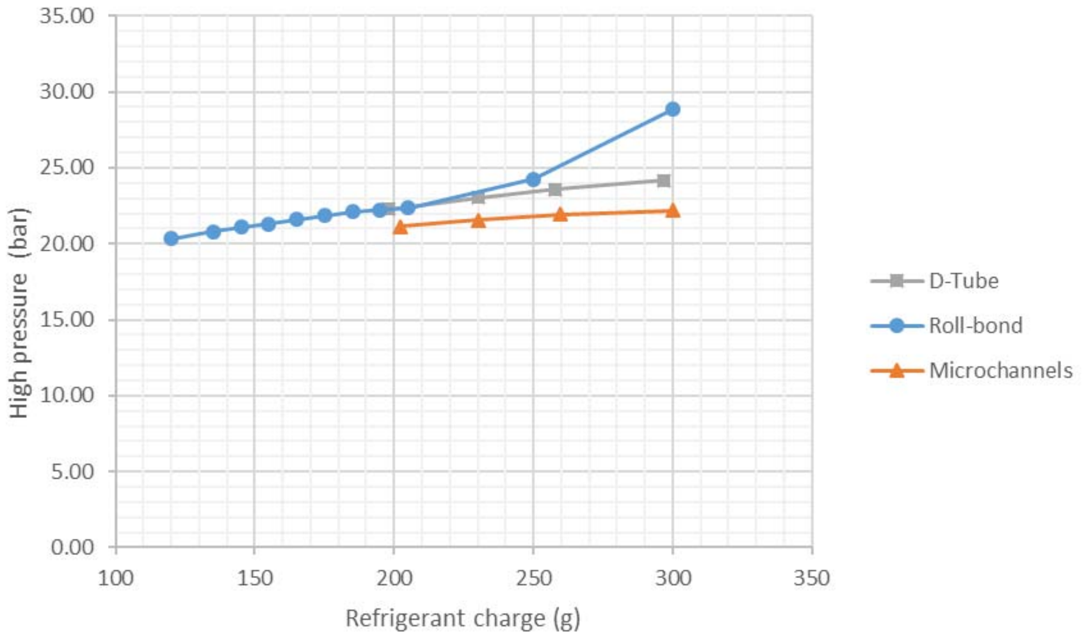


Figure 12.

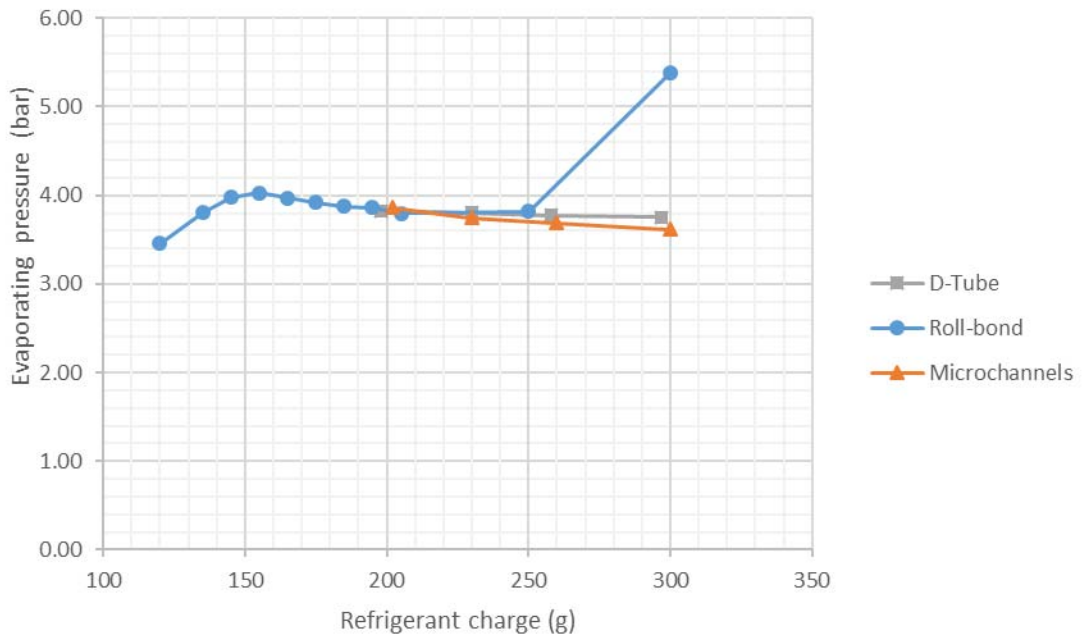
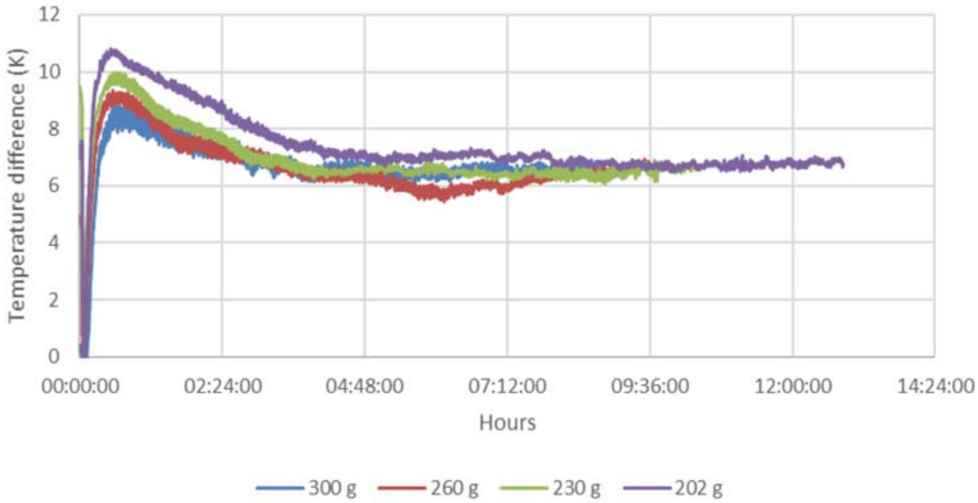


Figure 13.


12
10
8
6
4
2
0
Temperature difference (K)
00:00:00
02:24:00
04:48:00
07:12:00
09:36:00
12:00:00
14:24:00
Hours
300 g
260 g
230 g
202 g