NEW LOW CHARGE HEAT PUMP USING PROPANE: DESIGN AND EXPERIMENTATION

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Abstract: The design and the experimental performance of an innovative heat pump using propane and having a heating capacity of about 100 kW are presented in the paper. This unit has been designed for laboratory tests and has been realized in the framework of the European project SHERHPA in collaboration between University of Padova and Hiref SpA. Since propane is a flammable fluid, charge minimization is a priority in the design of such heat pump. Prototype shell-and-tube heat exchangers using minichannels and providing low charge have also been installed in the unit, along with conventional brazed plate heat exchangers, in order to compare different working configurations.
Very similar values of coefficient of performance and heating and cooling capacities have been measured when using the minichannel condenser instead of the brazed plate heat exchanger, even if this device provides a 65 % reduction in terms of condenser internal volume and the propane charge reduction is about 0.8 kg.
The experimental performance was also measured when adopting a minichannel internal heat exchanger showing in this case a slight improvement of the coefficient of performance.

Key Words: ***heat pumps, natural refrigerants, propane.***

1 INTRODUCTION

As a consequence of the phase out of chlorine refrigerants to avoid their negative influence on the ozone layer, hydrofluorocarbons (HFCs) like R410A, R407C and R134a are nowadays being used by the air-conditioning and heat pump industry in their more recent units. These fluids display negligible ODP, but are strong greenhouse gases with GWP over 100 years varying from about 1300 (in the case of R134a) to about 1700 (in the case of R410A). Atmospheric emissions from HVAC equipment adopting halogenated fluids cannot be avoided during the whole length of their lifetime, from manufacture to dismantling, and these have some effect on the anthropogenic global warming.

Chlorofluorocarbons (CFCs) were invented in the ‘30s as completely inert and harmless fluids. Only 50 years later, in 1974, their effect on the ozone layer destruction was discovered. Since HFCs nowadays being used are quite new man-made fluids (like CFCs were when first introduced into the market) their potential negative effects cannot be completely known.

Part of the scientific community has suggested that natural fluids with no environmental impact on the ozone layer and very low impact on the greenhouse effect like hydrocarbons, ammonia and carbon dioxide should be used as substitutes for synthetic refrigerants. Furthermore, the adoption of natural fluids, which are much better known than man-made ones, should provide some kind of intrinsic safety against potential unknown negative effects of man-made substances.

A future environmental legislation concerning refrigerants in relation to the greenhouse effect is expected. Within the EU, in mobile air conditioning systems, the gradual phase out of fluids with GWP higher than 150 starting from 1st January 2011 has been regulated by the directive 2006/40/EC. Regarding the heat pump applications, the ecological criteria for the award of the Community eco-label ((EC) No 1980/2000) to heat pumps have been revised with the Decision 2007/742/EC. According to the new regulation, in case of a working refrigerant with GWP lower than 150 (that is the case of natural fluids), the minimum requirements for the coefficient of performance (COP) and primary energy ratio (PER) in heating mode and the energy efficiency ratio (EER) in cooling mode shall be reduced by 15 % in comparison to those applied to systems using a fluid with higher GWP. Furthermore regulations on HFCs are discussed within the Kyoto protocol and later amendments.

The use of HCs would not involve major changes in the equipment design, since their thermodynamic properties and material compatibility are similar to those of traditionally used synthetic fluids. Particularly, a system designed for R22 could be used with propane (R290) as the working fluid without any problem. An overview on the use of hydrocarbons in refrigeration systems with regard to energy efficiency, environmental impact and safety standards can be found in (Granryd 2001).
The main problem of hydrocarbons as refrigerants is their flammability which has prevented their use in large scale; additional safety restrictions are required and, since the possible hazards can be considered proportional to the total amount of refrigerant trapped in the system, the charge inventory minimization can be considered a major design constraint.

In a paper by (Harms *et al*. 2003), the charge inventory distribution among the components of three unitary air conditioners with capacity varying from 9 kW to 26 kW and using R22 and R407C has been estimated. Most of the heat exchangers used standard microfin tubes, with internal diameter varying from 6 mm to 9 mm. According to their results, most of the charge was expected to be trapped in the heat exchangers; in particular the computed charge in the condenser varied from 30% up to 70% of the total amount, while the charge in the evaporator was about 15%.
Minichannels technology appears to be a very good opportunity to minimize the charge without loss in energy performance.
In a paper by (Palm 2007), the main amount of charge in a single family house heat pump with brazed plate heat exchangers (BPHEs) using R290 is experimentally shown to be trapped in the heat exchangers, and mainly in the condenser, where about 40% of the total charge was found. Prototype liquid-to-refrigerant heat exchangers using minichannels and providing a lower charge inventory when compared to BPHEs have been installed and studied (Fernando *et al.* 2004, Fernando *et al.* 2007, Palm 2007).
A similar experimental analysis of the charge distribution among the components has been performed at the University of Illinois, Urbana-Champaign, where air-to-propane minichannel heat exchangers with smaller internal volumes than traditional fin-and-tube heat exchangers have been developed and installed in a 2 kW cooling capacity refrigeration system (Hrnjak and Hoehne 2004).

The heat pump described in the present paper has a heating capacity of about 100 kW and is devoted to laboratory tests. It has been designed and realized in collaboration between the University of Padova and Hiref SpA in the framework of the European project SHERHPA (Sustainable Heat and Energy Research for Heat Pump Applications) dealing with the development of heat pumps that are in compliance with the future environmental regulations. A description of the unit is reported in (Cavallini *et al.* 2007).

2 HEAT PUMP DESIGN

Because of the safety restrictions due to the propane flammability, the heat pump has been planned to work situated outside in the open air on the roof of the Fisica Tecnica department building. The main design goal has been to achieve low propane charge inventory and high energy efficiency at the same time.
It has been chosen to design a water-to-water heat pump, because an indirect system with a secondary fluid both at the cold and hot side is a good choice to reduce the propane side volume. In fact, the size of the evaporator and the condenser can be reduced as compared to air-R290 heat exchangers by exploiting the more favorable heat transfer properties of water as compared to air.
Furthermore, in this way the present unit could be easily inserted in a geothermal system, using the ground as heat source both during the winter and the summer.

The heat pump facility is shown in Figure 1. The present prototype has been designed for testing applications, and five heat exchangers have been installed in the unit: two commercial brazed plate heat exchangers (BPHEs) are installed as the evaporator and the condenser, while low charge prototypes shell-and-tube (S&T) heat exchangers using minichannels are used as the condenser, the evaporator and the internal heat exchanger (IHX).These low charge heat exchangers are segmentally baffled shell-and-tube heat exchangers using 2 mm i.d. and 4 mm o.d. copper circular minichannels instead of conventional tubes. It has been adopted a single shell pass with two tube passes by using a U-tube bundle. The prototype heat exchangers have been designed and manufactured in collaboration with Onda SpA. Views of the tube bundle with baffles of the condenser and the internal heat exchanger are shown in Figure 2 and Figure 3, respectively.

By switching the on/off valves 8 different testing configurations can be obtained. Since BPHEs are the current industrial benchmark to achieve internal volume minimization, the configuration with BPHEs both as the condenser and as the evaporator is the reference one to experimentally quantify the advantages of operating the heat pump using the S&T minichannel heat exchangers.

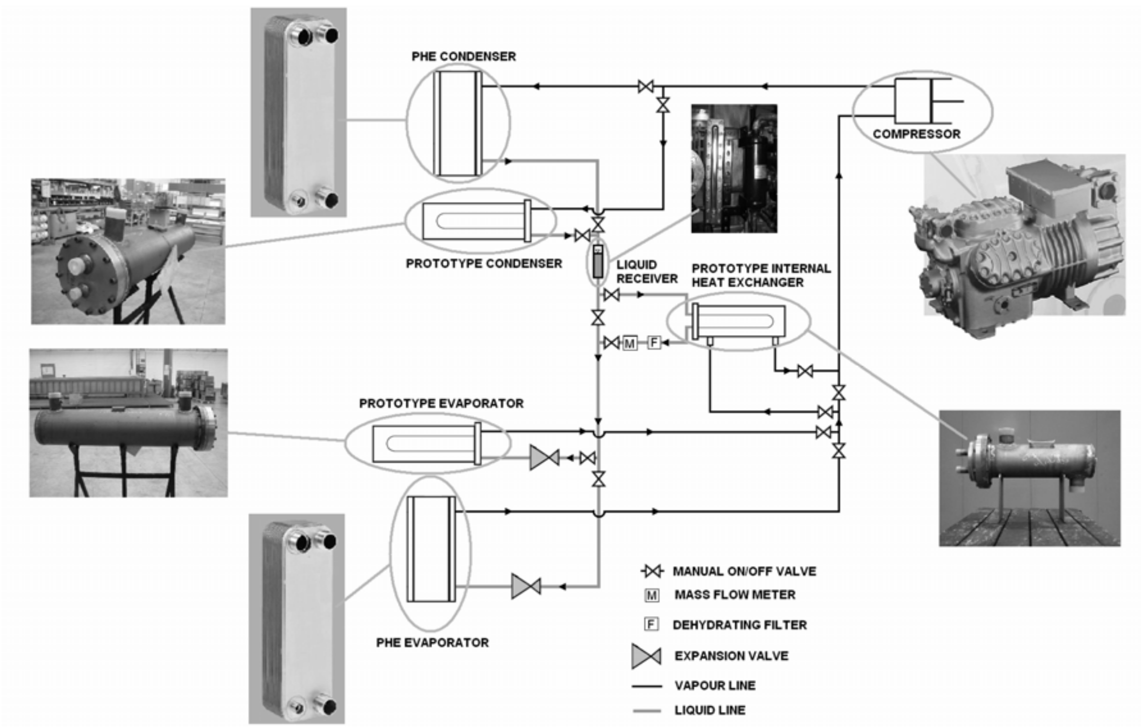

Figure 1: Heat pump facility

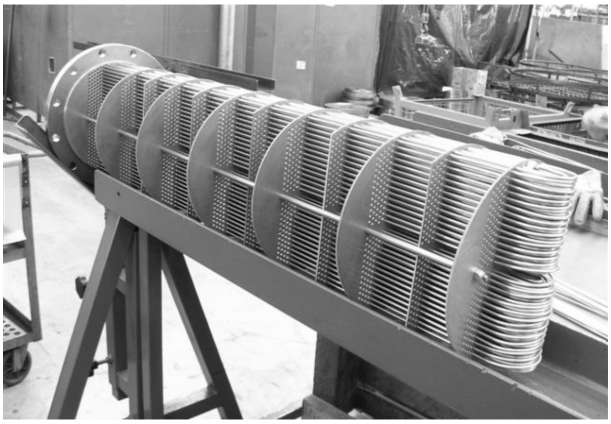

Figure 2: Tube bundle with baffles of the minichannel condenser

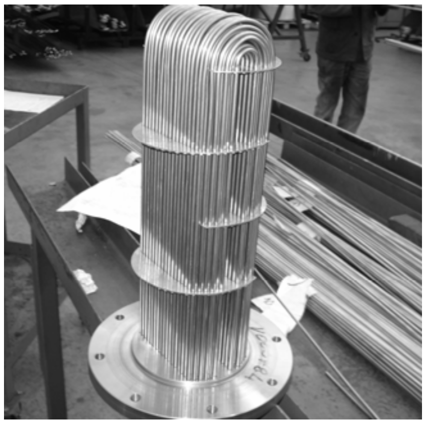

Figure 3: Tube bundle with baffles of the internal heat exchanger

The propane side internal volume of both the BPHE condenser and the BPHE evaporator is 8.4 L, while it is 2.9 L and 5.8 L in the case of the minichannel condenser and the minichannel evaporator, respectively. Regarding the total propane side internal volume of condenser plus evaporator, a 48% reduction can be obtained when using the prototype heat exchangers as compared to the BPHEs.
The design and the performance of the shell-and-tube evaporator and condenser are presented in (Del Col *et al.* 2008) and (Cavallini *et al.* 2008), reporting also experimental tests with R22.

From a theoretical point of view, some energy efficiency benefits can be achieved for propane by using an internal heat exchanger between the hot liquid line at the condenser outlet and the cold suction vapour. As reported in (Granryd 2001), at 0°C evaporating temperature and 40°C condensing temperature, the COP in cooling mode when using R290 as the working fluid should increase by 0.09% per degree of superheating, assuming isentropic compression. In practice, greater benefits should be achieved since superheat is expected to increase the compressor efficiency.
In the present facility, both configurations with and without the internal heat exchanger can be tested at the same operating conditions, to measure its influence on the system performance.

A semi-hermetic reciprocating compressor by Bitzer designed to operate with propane is used in the present heat pump. Propane has a very high solubility with conventional lubricants and ester oils. Although this characteristic is advantageous for a good circulation of the oil in the system, it can lead to a substantial decrease of the lubricant viscosity, which may be harmful to the compressor, specially at low oil temperature and high suction pressure. For this reason, the compressor is charged with 4.75 L of an higher basic viscosity mineral oil (cinematic viscosity 68 cSt at 40°C) and special measures are required by the compressor manufacturing company: a sufficient suction superheat (i.e. at least 20 K), preferably with an internal heat exchanger, and a minimum discharge gas temperature 20 K above the condensing one are suggested (Bitzer, 1997).

As shown in Figure 1, a 2 L liquid receiver is installed at the propane liquid line downstream the condenser outlet. This device helps to fast reach the stable operation of the heat pump. An indicator of the liquid level in the receiver has also been installed to get an immediate control of the active charge during the tests.

3 TEST RIG

The set-up of the experimental facility is shown in Figure 4: on the right side the propane heat pump with five heat exchangers is depicted, on the left side the hot and cold water loops are shown. The two hydraulic circuits allow to independently fix the mass flow rate and inlet water temperature both at the condenser and at the evaporator. The experimental tests can thus be performed at the desired operating conditions.

The water mass flow rate is measured by means of a Coriolis effect mass flow meter on the cold side and by a electromagnetic flow meter on the hot water side. Inlet and outlet temperatures to the heat exchangers are measured by means of T-type thermocouples (copper/constantan) using a reference ice-point. Reservoirs are present at both the cold and hot hydraulic circuits to provide a sufficient thermal inertia for a stable adjustment of the cold and hot water temperature. The propane pressure and temperature at inlet and outlet of all the heat exchangers, the compressor and the expansion valves are measured by means of pressure transducers and T-type thermocouples (copper/constantan). In addition, the electric power supplied to the compressor is measured by an electrical power analyzer, while the propane mass flow rate is measured by a Coriolis effect mass flow meter. The measurement uncertainty (with 68% level of confidence) of the instrumentation used is reported in Table 1.

Since the heating and cooling capacities are very high (the heating capacity is about 100 kW), a heat exchanger is installed between the two hydraulic circuits and the cooling capacity is dissipated by the hot water of the condenser; an automatic valve controlled by the cold water reservoir temperature adjusts the flow rate in the dissipator, to get a stable secondary fluid inlet temperature at the evaporator.

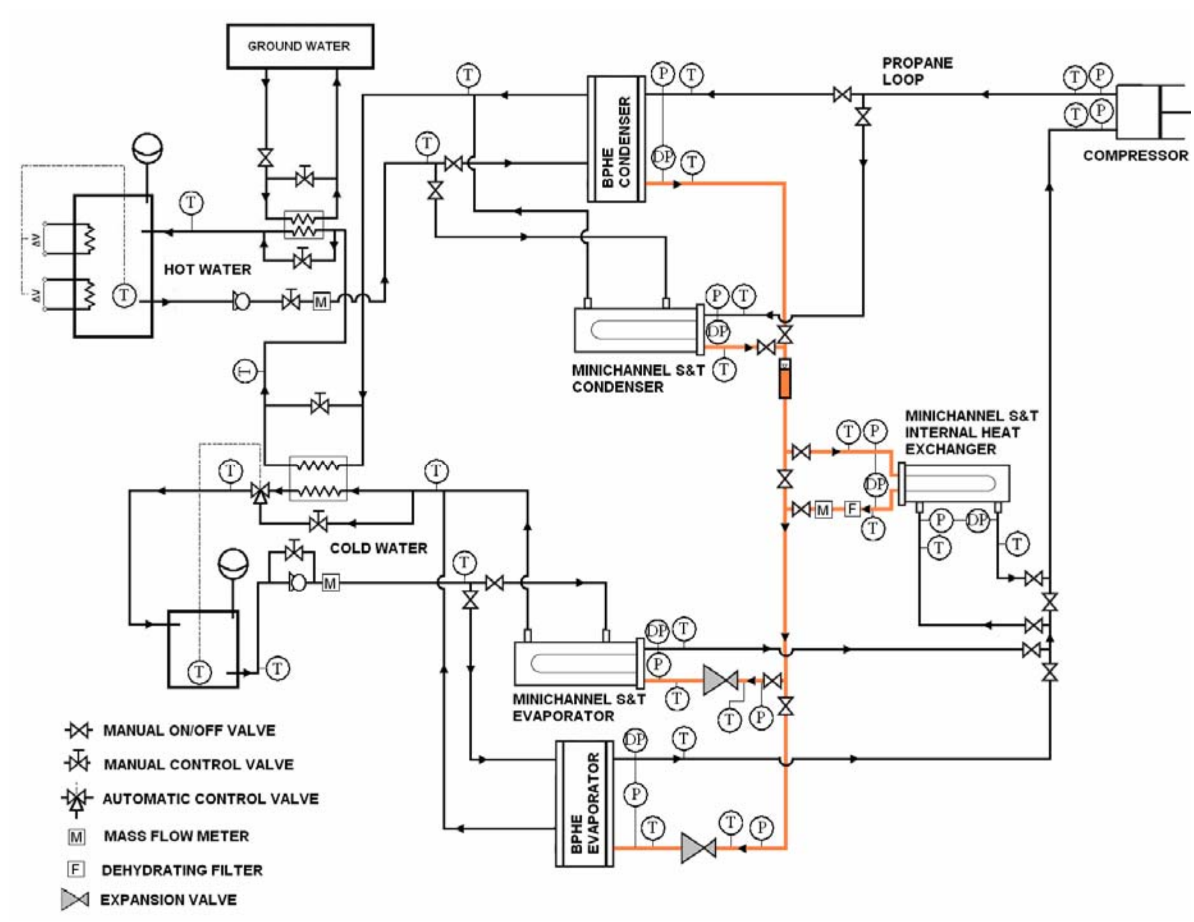

Figure 4: Test rig

Table 1: Instrumentation measurement uncertainty (68% confidence level)

Instrument	Uncertainty
R290 mass flow meter (Coriolis effect)	±0.1% of reading
Cold water mass flow meter (Coriolis effect)	±0.1% of reading
Hot water flow meter (electromagnetic)	±0.3% of reading
T-type thermocouples + ice point	±0.05 K
Relative pressure transducers	±0.15% of full scale
Relative pressure transducers (ATEX)	±0.1% of full scale
Differential pressure transducers	±0.1% of full scale
Electric power analyzer	±0.35% of reading

The heating capacity which is not dissipated by the cold water circuit can be computed by subtracting the cooling power from the heating one and is equal to the electrical power supplied to the compressor (i.e. about 30 kW at nominal operating conditions). A thermal power slightly higher than the required is dissipated to ground water; then the hot water temperature in the reservoir is adjusted by an electric heater to achieve a more stable control.

4 EXPERIMENTAL PERFORMANCE

Laboratory tests have been carried out and two different data sets have been obtained. All the tests have been carried out using the brazed plate heat exchanger as the evaporator.

The water temperature rise in the condenser, as well as the water temperature drop in the evaporator, was around 5 K for all the tested conditions reported here.

The first data set refers to the reference configuration, i.e. the configuration with BPHEs both as the condenser and the evaporator; while in the second data set the performance when using the minichannel condenser is compared to the performance when using the BPHE condenser. The heat pump has not been tested so far with the minichannel evaporator.

The power consumption of the circulating pumps has not been measured, therefore in this paper only the COP of the equipment is reported.

4.1 Reference configuration

Experimental data for the reference configuration (plate condenser plus plate evaporator) of heating capacity and equipment coefficient of performance in heating mode are plotted versus the temperature of the water supplied by the condenser in Figure 5 and Figure 6, respectively. Solid diamonds refer to experimental data obtained at around 12-7 °C evaporator water temperature, while square symbols refer to experimental data obtained at around 10-5 °C evaporator water temperature.

All these data have been obtained with the receiver downstream the condenser half filled with liquid, that means that the presence of liquid trapped at the condenser outlet is avoided and the liquid subcooling is minimized.

As one can see, by increasing the temperature of the hot water, or by decreasing the temperature of the water at the evaporator, both the heating capacity and the COP coherently decrease.

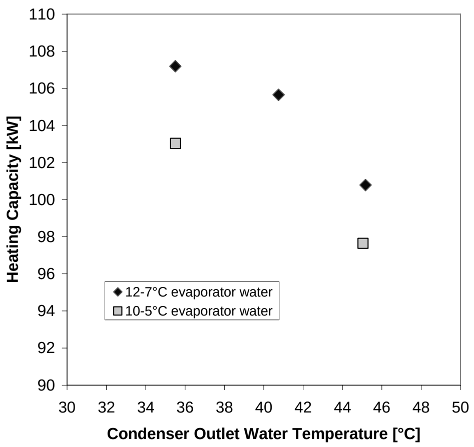

Figure 5: Heating capacity versus condenser outlet water temperature

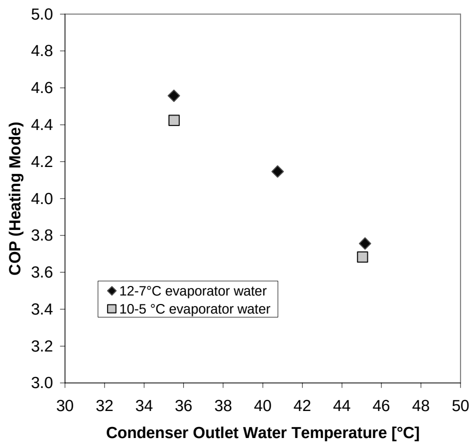

Figure 6: Coefficient of performance versus condenser outlet water temperature

According to the new ecological criteria for the award of the Community eco-label ((EC) No 1980/2000) revised by the Decision 2007/742/EC, as reported in the Official Journal L 301 of 20.11.2007, the required system COP in heating mode for a water-to-water electrically driven heat pump at 40-45 °C condenser water temperature and 10-7 °C evaporator water temperature is equal to 3.57 when the refrigerant GWP is lower than 150. For the present heat pump in the reference configuration, the measured equipment COP at 40-45 °C condenser water temperature and 10-5 °C evaporator water temperature is 3.68.

In Figure 7, a comparison between the COP in heating mode for the reference configuration without internal heat exchanger is compared to that obtained when inserting the internal heat exchanger. These data points refer to around 12-7 °C evaporator water temperature, and 40-45 °C condenser water temperature.
The vapour superheat at the evaporator outlet is around 4 K, and the additional superheat obtained when using the internal heat exchanger is around 22 K in the case reported.
About a 3% improvement on the COP in the heating mode, and about 4% improvement in the cooling mode have been obtained from experimental data. Besides, the cooling capacity increases by around 5%.

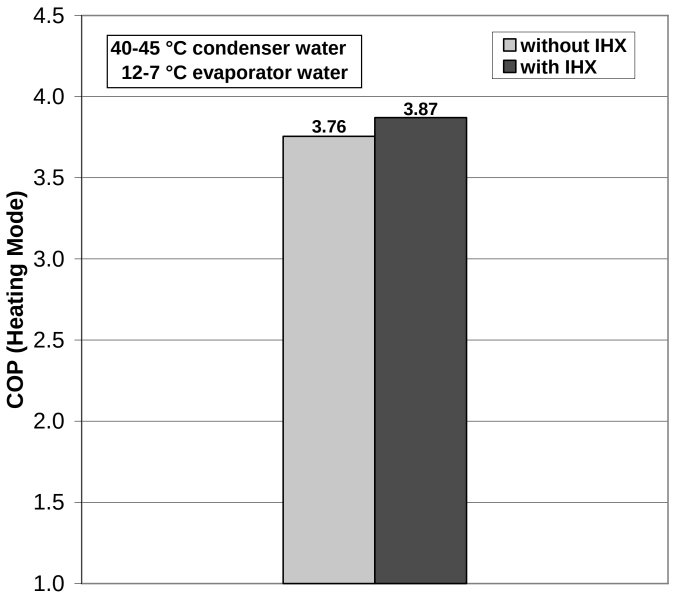

Figure 7: COP improvement when adopting the internal heat exchanger for the reference configuration

4.2 Configuration with the minichannel condenser

The performance of the unit when using the minichannel shell-and-tube condenser is here compared to that measured when adopting the BPHE, both with and without the internal heat exchanger. The present data have all been obtained with the receiver downstream the condenser half filled with liquid.
The data reported for the four configurations considered (minichannel / plate condenser, with / without internal heat exchanger) have all been obtained at the same operating conditions: i.e. around 12-7 °C evaporator water temperature, and around 40-45 °C condenser water temperature.

Experimental heating capacity, cooling capacity and equipment coefficient of performance (COP) in heating mode for the four test configurations considered are reported in Figure 8, Figure 9 and Figure 10, respectively. The uncertainties in the measurement of the heating and cooling capacities and the coefficient of performance (with 95% confidence level) are around 3.5 kW, 2.5 kW and 0.15, respectively.
As one can see, when the minichannel condenser is adopted instead of the brazed plate heat exchanger, very similar values for the equipment coefficient of performance and the heating and cooling capacities are measured.

It should be considered that the propane side internal volume is 2.9 L in the case of the minichannel condenser and 8.4 L in the case of the BPHE condenser; hence, 65 % reduction in terms of condenser internal volume has been obtained without any loss in terms of energy performance. The nominal heat transfer area of the BPHE condenser is around 7 m^2, while the propane side heat transfer area of the minichannel condenser is around 5 m^2.

The experimental water side pressure drop of the minichannel condenser is around 9 kPa at test conditions. No experimental data are available for the BPHE condenser. According to the rating software provided by manufacturer, the expected water pressure drop of the brazed plate condenser is around 40 kPa at the same flow rate.

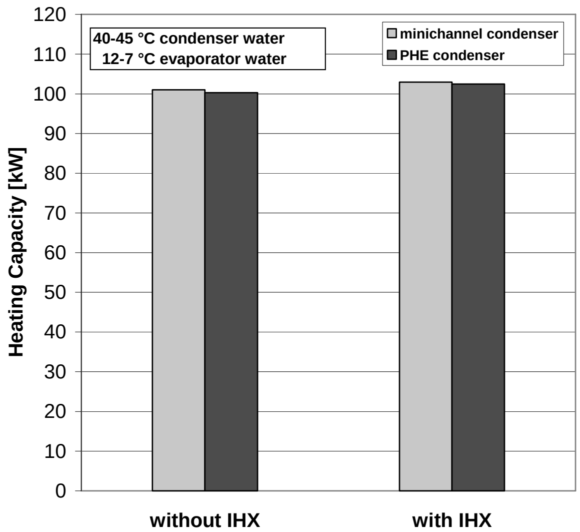

Figure 8: Experimental heating capacity for different test configurations at the same operating conditions

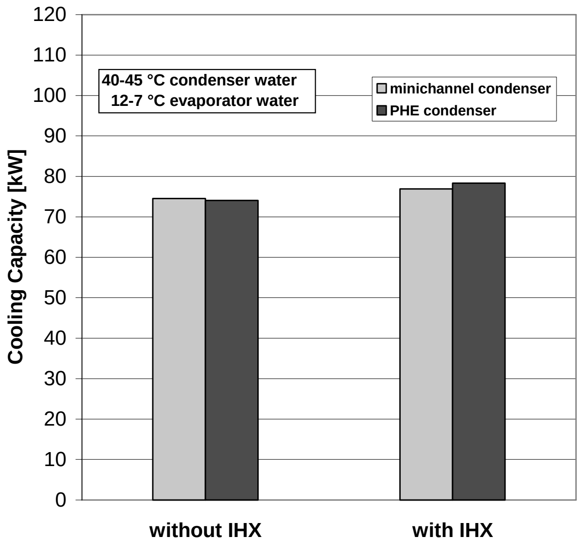

Figure 9: Experimental cooling capacity for different test configurations at the same operating conditions

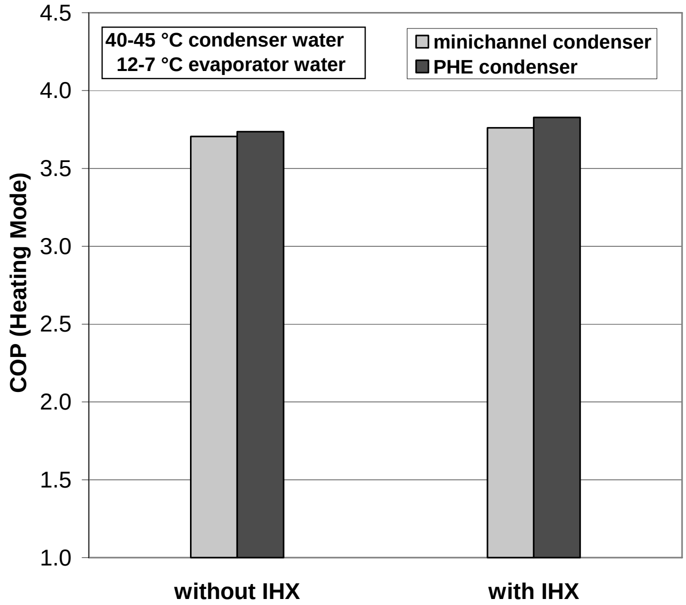

Figure 10: Experimental COP for different configurations at the same operating conditions

4.3 Propane charge

The propane charge required by the present heat pump when operating in the reference configuration (brazed plate evaporator plus brazed plate condenser) was around 4 kg when the IHX was not used and around 5.2 kg when the IHX was inserted.

Since the prototype has been devoted only to test applications and five heat exchangers have been installed in it, the piping length has not been minimized at all in the present heat pump. For this reason, about 0.9 kg of propane in the configuration without IHX and about 1.1 kg in the configuration with IHX are trapped in the piping.
The dehydrating filter and the liquid receiver contain large amount of refrigerant. Around 0.4 kg of propane is expected to be trapped in the filter. For both the configurations considered, tests were performed with the receiver half filled with liquid and thus containing around 0.5 kg of propane.
As shown in Figure 1, the dehydrating filter, as well as the Coriolis mass flow meter, have been installed in the liquid line bypass downstream the internal heat exchanger. Then, the filter is bypassed when the IHX is not used, to avoid the formation of flash vapour due to the pressure drop before entering the expansion valve, when saturated liquid propane is expected to exit from the condenser. Hence the amount of refrigerant trapped in the filter is not taken into account in the data reported for the configuration without IHX.

By reducing the piping length, therefore, a heat pump with BPHEs, a liquid receiver and a dehydrating filter, and providing around 100 kW heating capacity, could be run with less then 4 kg of propane. Besides, discarding the liquid receiver and the dehydrating filter, it could be operated with about 3 kg propane charge.

The additional charge trapped in the internal heat exchanger is around 0.3 kg. In practice, a slightly higher additional charge is required in its presence, because some propane will be trapped in the piping, mainly in the liquid line between the condenser outlet and the IHX inlet.

No experimental data about the charge required by the present heat pump when operating with the brazed plate evaporator and the minichannel condenser are available at the moment. The results reported in section 4.2 have been obtained using the same total charge inventory, but when using the minichannel condenser the liquid receiver was almost completely filled with liquid, while it was almost empty when using the BPHE condenser. By comparing the difference of the level in the liquid receiver, it could be estimated that the expected propane charge reduction using the minichannel condenser instead of the brazed plate condenser is about 0.8 kg.

5 CONCLUSIONS

The design and the experimental performance data of an innovative heat pump using propane and having a heating capacity of about 100 kW are presented. Since propane is a flammable fluid, charge minimization is important for such kind of equipment and the main design goal has been to achieve low propane charge inventory and high energy efficiency at the same time.

Five heat exchangers have been installed in the present prototype unit for testing applications: two conventional brazed plate heat exchangers are present as the evaporator and the condenser, while low charge shell and tube heat exchangers using 2 mm i.d. minichannels are also used as the condenser, the evaporator and the internal heat exchanger. Different configurations can be tested in order to compare the data.

Experimental data measured with the heat pump working with brazed plate heat exchangers are here reported. The experimental equipment COP in heating mode is higher than the system COP required by the new ecological criteria for the award of the Community eco-label ((EC) No 1980/2000), as revised by the Decision 2007/742/EC.

About 3% improvement of the COP in the heating mode, and 5% improvement in the cooling capacity are reported when using the minichannel internal heat exchanger at 40-45 °C condenser temperature and 12-7 °C evaporator temperature. The internal heat exchanger provides about 22 K additional vapour superheat. The additional charge trapped in the internal heat exchanger is around 0.3 kg.

Very similar values of coefficient of performance and heating and cooling capacity have been measured when the minichannel condenser was used instead of the brazed plate heat exchanger, even if this device provides a 65 % reduction in terms of internal volume. The expected propane charge reduction using the minichannel condenser instead of the brazed plate condenser is about 0.8 kg.

The propane charge required by the present heat pump when operating in the reference configuration (i.e. plate condenser plus plate evaporator) was around 4 kg when the IHX was not used and around 5.2 kg when the IHX was inserted. However, the piping length has not been minimized in the present heat pump. By reducing the piping length a heat pump with BPHEs, a liquid receiver and a dehydrating filter, and providing around 100 kW heating capacity, could be run with less then 4 kg of propane.

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