EXPERIMENTAL INVESTIGATION ON A SMALL CAPACITY HEAT PUMP PROTOTYPE WITH CO2 AS WORKING FLUID

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Abstract: The recent European regulation on Fluor gases, pointing out the greenhouse effect of present refrigerants, has put in relief the need of research on new fluids as substitution. For systems based on the mechanical compression cycle, some solutions has just begun to rise, by using fluids so called natural refrigerants such as carbon dioxide. The use of CO_2 in heat pump leads to a transcritical cycle very suitable to achieve high temperature for hot tap water. On the other hand, the lower temperature level in the gascooler can be valuable for space or floor heating. A small unit prototype with a heating capacity up to 2.5 kW is tested in different conditions. Among the inlet parameters, the main performances are estimated versus the evaporation temperature from 0°C to 10°C, the high pressure from 75 bar to 110 bar and the CO_2 charge from 400 g to 600 g. The critical relevant points of the system are the gascooler, the compressor and the oil circulation. Thus, two different technologies of gascooler have been installed and tested: shell and tubes with inner diameter of 2 mm and plates heat exchanger granted to work up to 150 bar. The amount of oil in the compressor has been also investigated, leading to an improvement of the performances by reducing it.

Key Words: heat pump, carbon dioxide, prototype, heat exchanger

1 INTRODUCTION

Since the early 80's, big issues on environment have risen, mainly linked to the human activities. The change in weather involves complicated phenomena and the emission of greenhouse effect gases (GHG) is part of them. In refrigeration, air conditioning and heat pump applications, the refrigerants used are partially effective with large global warming potential (GWP) as indicated in Table 1.

Table 1: Environment effect of several refrigerants

Refrigerant	***Chemical name***	***ODP***	***GWP (100 years)***
CFC-12	CCl2F2	1	7100
HCFC-22	CHCl2F2	0.055	1500
R-407C	R32/R125/R134a (23/25/52)	0	1530
R-290	CH3-CH2-CH3	0	<20
R-744	**CO_2**	**0**	**1**

In 2006, European Parliament decided a regulation targeting some Fluor gases families to control their use, confinement and recycling aspect. In respect with the regulation, mobile air conditioning is relevant as a big issue due to large leakage ratio given by car manufacturers (Barbusse and Gagnepain 2003). From 2013, all new vehicles sold on the European market should have conditioning systems with low GWP fluids (less than 150). The other

applications using refrigerants are also targeted by the regulation for confinement and recycling purpose.
Some solutions have been proposed for several years among the use of so called natural fluids. Lorentzen (1995) focused on the suitability of such fluids to replace refrigerants with ODP effect (CFC and HCFC). Among the reliable candidates, the most employed are ammonia, carbon dioxide and hydrocarbons.
The use of carbon dioxide (CO_2) in systems such as air conditioning and heat pump induces a transcritical cycle due to the low critical temperature of the fluid (Table 2). The technical possibilities of vapour compression systems working at higher pressures than classical appliances are particularly targeted. A consequent technical and scientific literature deals with this subject. For air conditioning applications, Tamura *et al.* (2005) obtained relevant results with an automotive air conditioning system. With optimisation of the components, the cycle efficiency achieves R134a system's performance.

Table 2: Properties of propane and carbon dioxide

Fluids	***T_{sat} (1 atm)***	***T_c[1]***	***P_c[2]***	***T_p[3]***
Propane	-42,1°C	96,6°C	42,4 bar	-187,7°C
CO2	-78,4°C	31,1°C	78,3 bar	-56,5°C

The transcritical cycle with CO_2 is especially designed for heat pump at relatively high temperatures and large temperature glide (Neska 2002). With CO_2 heat pump, the shift of temperature up to 20°C is possible even with a high COP.
Compared to heat pump with HFC refrigerants, the components for CO_2 heat pump are specifically designed (Kim *et al*. 2004). The first constraint is the high working pressure in suction and discharge sections. Moreover the low viscosity of the supercritical CO_2 allows smaller diameter channels for the heat exchangers. The evaporator can also be more compact due to the higher heat transfer coefficient (Pettersen 2004). At least, lower pressure ratio and higher density in vapour phase lead to a reduction of the swept volume of the compressor that should give rise to more efficient systems (Suss 1998).
The present limitations for the development of refrigeration systems and heat pumps using CO_2 as refrigerant are mainly concerned with the design of more efficient heat exchangers, and more generally the optimization of all the components (compressors, accessories…). Indeed, with the last results expected from the Research field, the CO_2 systems could achieve higher performances than standard devices with R134a. This trend is especially observed in mobile air conditioning where the leakage rate is up to 15% per year (Liu *et al*. 2005).
Dealing with the hot water production, this application represents up to 21% of the energy demand in the residential and commercial market. The larger variation of the CO_2 temperature in the high pressure side of the transcritical cycle is rather close to the water temperature variation, leading to a decrease of the exergetic losses. As a consequence, this kind of thermodynamic cycle is typically fitted for heating water up to 60°C (Cecchianato *et al*. 2005). By combining hot water production with cooling production, the efficiency of the CO_2 cycle is intensively increased.
Japan is mainly the leader for promoting such heat pumps using CO_2 in a transcritical cycle, for hot tap water. Several projects have achieved good results and have been supported by the Japanese ministry of Industry, all of them joined in the program "ECOCUTE", based on a technology previously developed by Norsk Hydro and DENSO, patented by the Shecco Company. Thus the system designed from the collaboration of TEPCO (Tokyo Electric Power Corp.), Denso Corp. and the Central Institute of Electric Power Industry was the first transcritical CO_2 heat pump for hot tap water supply, using an air heat exchanger as an

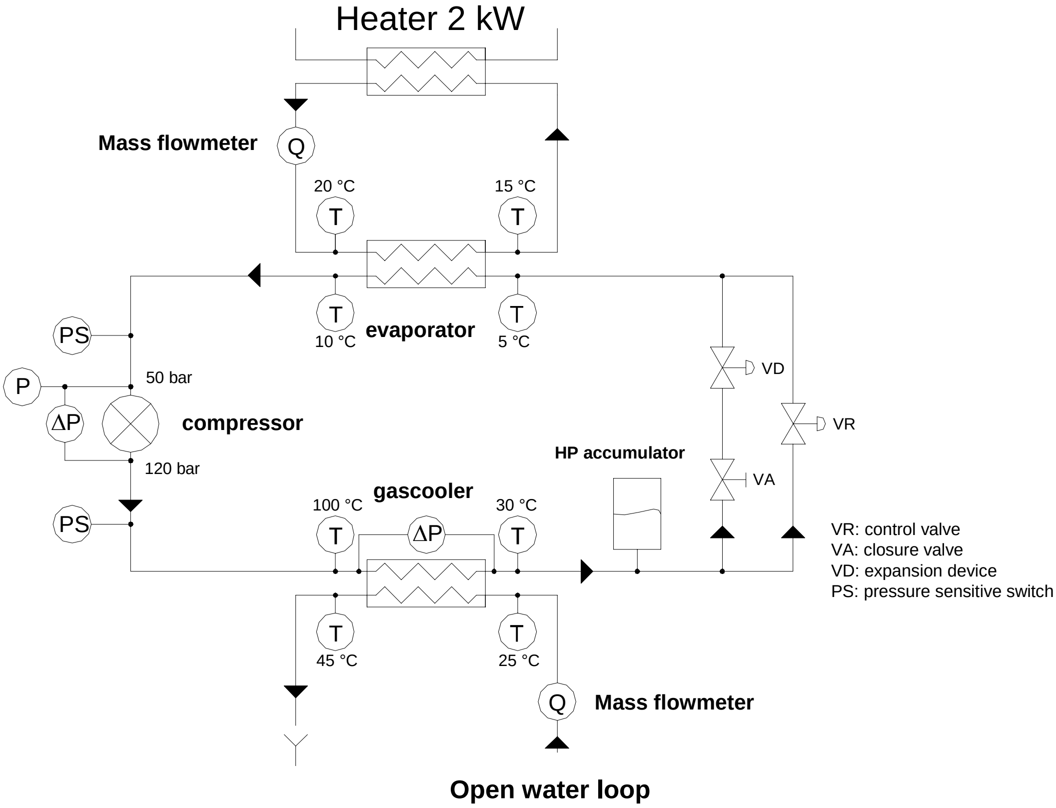
[1] Critical temperature
[2] Critical pressure
[3] Melting temperature

evaporator. Other Japanese manufacturers such as Daikin and Sanyo, are developing there own products. The Kyocera Company is also proposing a model, with the ambition to achieve a coupling system with a photovoltaic cell for the compressor's energy supply. Finally, the Hepco Company (Hokkaido Electric Corp.) has been leading a study for the characterization of a CO_2 heat pump dedicated to cold regions such as the north of Honshu and Hokkaido islands. The first tests were lead with a Sanyo model, with a heating capacity of 4.5 kW at -20°C. The water accumulator had a capacity of 370l. These field tests recently lead to an update of the Sanyo system with an increased COP of the heat pump up to 20% in these extreme conditions. Daikin also announced in the early 2008, a new heat pump achieving a maximum COP up to 5 for hot water supplying.
In 2003, about 70000 units were installed in Japan. The number increased to 220000 in 2005. The target is one million in 2010, for the Japanese market only.
In Europe, the heat pump market exhibits an increasing trend but few prototypes or field tests with the CO_2 technologies have been proposed yet (IEA 2006).

This paper presents the characterization of the performances of a CO_2 heat pump prototype, based on a small capacity compressor. Several parameters are investigated and a discussion is proposed on the design of such a system.

2 CO_2 HEAT PUMP PROTOTYPE SET-UP

The Figure 1 illustrates the CO_2 heat pump and the test loop with all the measurement devices.



Open water loop

Figure 1: Schema of the whole test loop and the heat pump prototype.

The heat pump is build around the CO_2 Danfoss compressor Model TN1416 having a electrical consumption up to 860 W at 5°C for a discharge pressure of 90 bar. With this compressor, there is the possibility to use the TBR (Thermal Back pressure Regulator) valve from Danfoss or a manual valve for the expansion device. The Figure 2 presents a general view of the prototype.

Figure 2: Overview of the CO_2 heat pump prototype.

The evaporator is a CO_2-to-water coiled heat exchanger. The CO_2 flows in 4 tubes with a diameter of 4 mm whereas the water flows in counter-current in the coil. The evaporator is stainless steel made.
For the gascooler, a previous heat exchanger prototype was used. It was a shell-and-tubes one, having 31 tubes of 2 mm inner diameter. This first heat exchanger was replaced by a spiral plate heat exchanger from Spirec Company. This heat exchanger allows a relatively high pressure constraint and a high compactness (Figure 3). The thermal characteristics of this heat exchanger in gas cooling with supercritical CO_2 were investigated previously (Bombarda *et al*. 2007).

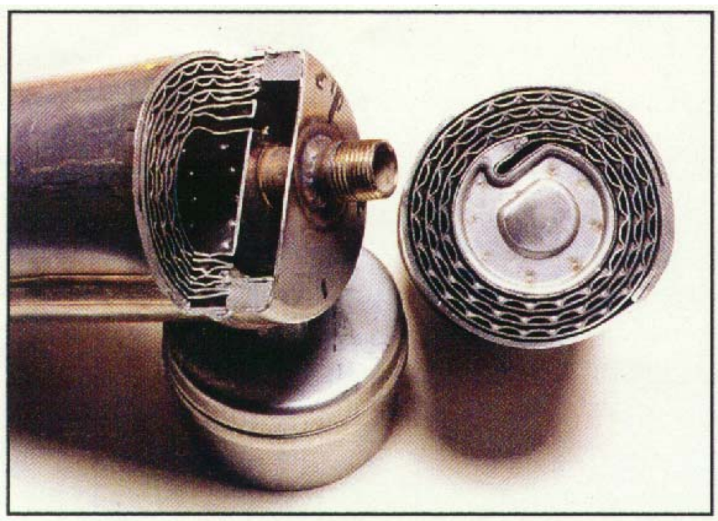


Figure 3: Spiral plate heat exchangers (Spirec).

Finally an internal heat exchanger can be used or by-passed. This heat exchanger is a double tubes "trombone", seen in Figure 2.
The heat pump is linked to a cold water loop on the evaporator side and an open water loop on the gascooler side. An electrical pre-heater is available to control the inlet temperature of the water in the gascooler.
The main measurements on the heat pump concern temperatures at the inlet and outlet of the different components with type K thermocouples. Suction and discharge pressures of the

compressor are also recorded. To permit the energy balance, the water flow is measured on both water loops as well as the electrical consumption of the compressor.

3 RESULTS AND DISCUSSION

The tests investigate different parameters on the working conditions of the heat pump. By the way, two main results are presented here, the effect of the internal heat exchanger (IHX) and the reduction of the amount of oil in the compressor on the thermal performances of the heat pump.

3.1 Internal heat exchanger

The internal heat exchanger is made by two rows of a double tube. The inner tube is dedicated to the high pressure CO_2 flow whereas the low pressure CO_2 is flowing in the annular gap (Figure 4).

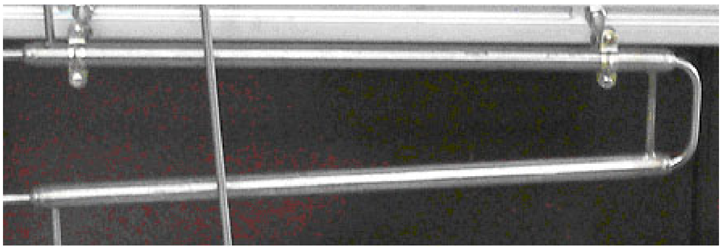


Figure 4: detailed view of the IHX.

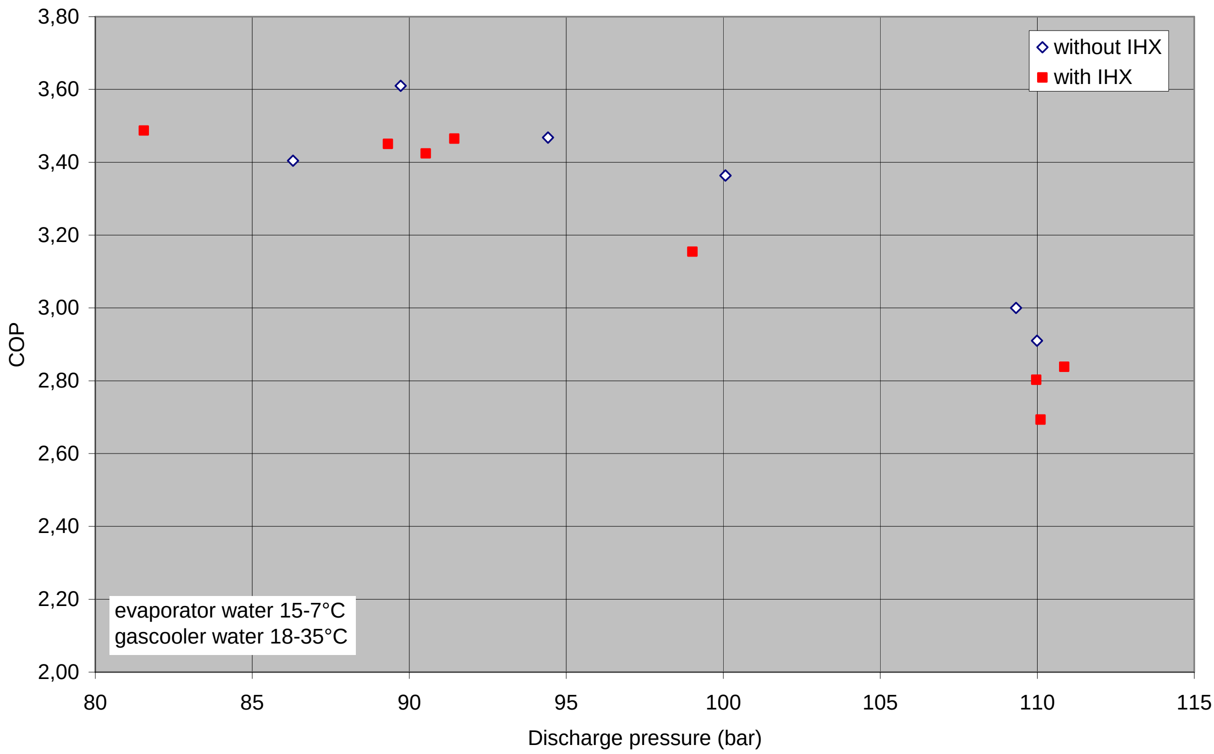


Figure 5: COP vs. Discharge pressure with and without IHX.

On Figure 5, it appears that in these test conditions, the use of the IHX decreases slightly the COP of the heat pump. This result is different that what is generally observed with CO_2 heat pump where the IHX allows a higher COP due to the low temperature at the inlet of the throttling valve, reducing the energy losses during the flashing of the refrigerant.

In the case of relatively low inlet temperature of the secondary fluid in the gascooler, the outlet temperature of CO_2 is enough low to reduce the energy losses. The IHX leads to a minimum temperature reduction (Figure 6) with a weak influence on the COP. Moreover, the Figure 6 shows also that the case without IHX induces a lower superheat. In these conditions, the compressor consumption is reduced due to colder entering vapor.

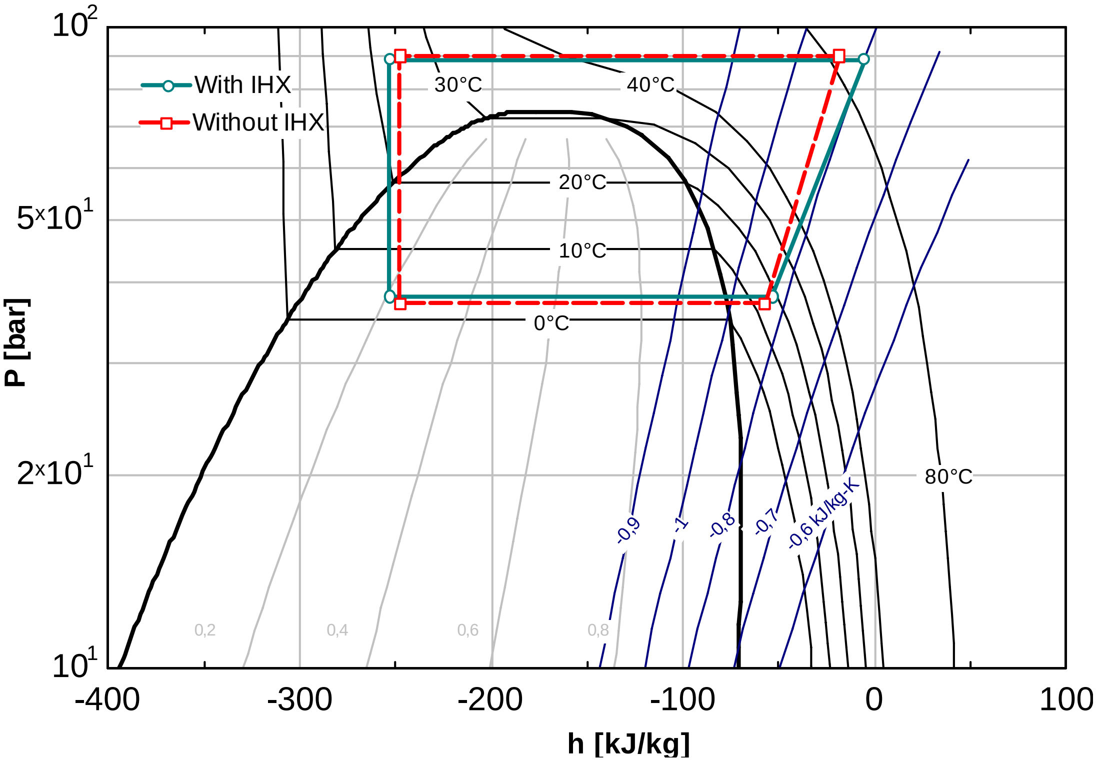


Figure 6: h-P representation for two cases with and without IHX at a discharge pressure of 90 bar.

The use of the IHX does not appear necessary in the conditions exhibited during the tests. For a CO_2 heat pump designed for direct water heating, different from the option chosen for the ECOCUTE systems with storage, with cold water feeding the gascooler, the heat pump circuit is simplified and lighter. In this case, the thermal design of the gascooler is obviously crucial to allow the best cooling of CO_2 close to the water inlet temperature.

3.2 Effect of oil amount

The second parameter investigated presented in this paper is the effect of the oil amount in the compressor. This parameter was not obvious previously until the prototype was supplied with a sight glass at the compressor inlet. The lubricant used in the Danfoss compressor is a Reniso C 85 E from Fuchs Company. The selection and the oil compatibility with CO_2 is evidently important for the performances of the components of the heat pump, among them the compressor and the heat exchangers. (Yun *et al.* 2007)

It appeared the oil circulation was too large for a reliable behavior of the prototype and suitable performances (Figure 7). In the severe conditions measured, the superheat conditions could not be achieved at the evaporator outlet, even with the IHX, and the CO_2 temperature at the outlet of the compressor was too low. In Table 3, the CO_2 temperature at the compressor outlet is indicated for several discharge pressure and lubricant amount in the crankcase.

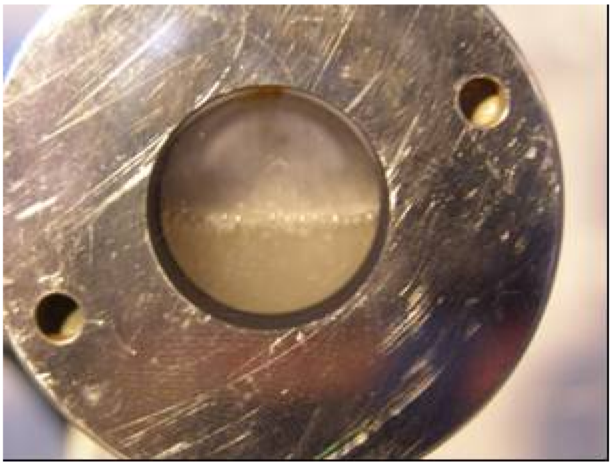


Figure 7: CO_2/oil mixture at the compressor inlet.

The initial oil charge was estimated up to 200 ml inside the compressor. After the first running tests, the amount of oil recollected into the compressor was below 150 ml. One quarter of the initial amount was spread in all the heat pump, in the pipes, the heat exchangers and the accumulator, mainly mixed with CO_2 maybe.
The heat pump loop was totally washed up and the compressor was fed with a new amount of lubricant up to 100 ml.
The first result is the behavior of the mixture CO_2/oil. With the reduced charge of oil, no more liquid level is observed at the inlet of the compressor, but a vapor flow with a light droplets of lubricant.

The second obvious effect of the reduction of the oil charge is the outlet compressor temperature of CO_2. Whereas this temperature did not exceed 100°C with 200 ml of lubricant, it achieves now values up to 130°C for the highest discharge pressures (Table 3).

Table 3: Comparison of CO_2 temperature at the outlet of the compressor for different pressures and lubricant amount.

Discharge pressure (bar)	$T_{out, comp}$ (oil amount 200 ml)	$T_{out, comp}$ (oil amount 100 ml)
90	77°C	97°C
100	90°C	104°C
110	102°C	125°C

Finally, the performances of the heat pump are modified too (cf. Figure 8).

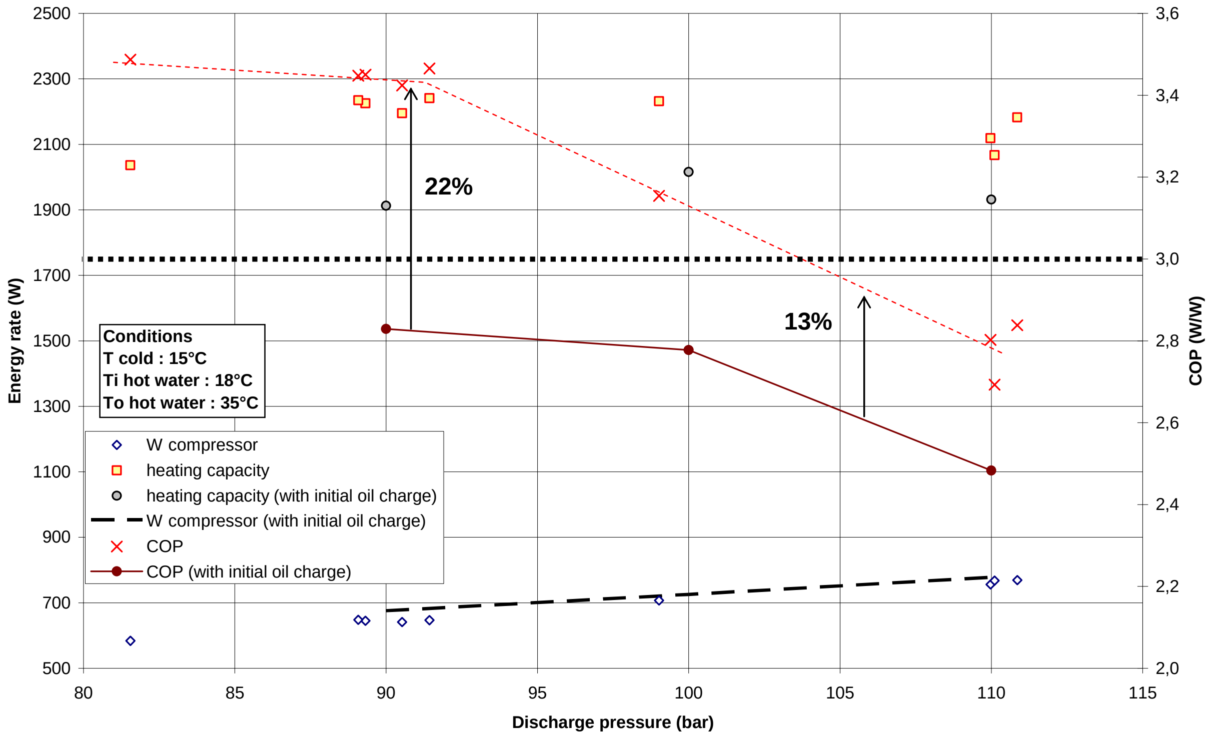


Figure 8: Effect of the amount of lubricant in the compressor on the energy performances of the prototype.

The reduction of the oil amount does not change the compressor consumption, but allows a sensitive increase of the heating capacity from 1.9 kW to 2.3 kW. Two complementary explanations can be proposed: the increase of the CO_2 temperature and the increase of the heat transfer on the refrigerant side mainly driven by the reduction of the oil circulation. The maximum heating capacity occurs at a lower discharge pressure too: about 90-95 bar versus 100 bar with the previous lubricant charge.
Consequently to the higher heating capacity, the COP of the heat pump also rises to 22% from the previous case. It clearly appears that the reduction of the oil amount allows the COP to exceed values above 3. This condition was not possible previously as the COP achieved 2.8 below 95 bar.

4 CONCLUSION

A prototype of a small capacity CO_2 heat pump was previously designed and realized to supply hot tap water. Different technological choices were made about the components, among them the gascooler, the internal heat exchanger and the expansion device. In this paper, two parameters have been investigated through the running tests: the influence of the IHX and the effect of the oil amount in the compressor. About the influence of the IHX, the low inlet temperature of water in the gascooler avoids the interest of such a heat exchanger, with a weak influence on the throttling conditions of the CO_2. Moreover the superheat induced by the IHX increases significantly the losses in the compressor and deteriorates the COP. In the case of a direct heating CO_2 heat pump, the IHX could be useless and the refrigerant loop simplified. The amount of lubricant initially feeding the compressor is also strongly modifying the energy performances of the heat pump prototype. After dividing by two the previous oil charge, a substantial increase of the COP has been measured. The oil charge is directly linked to the oil circulation as proven by visualization and should negatively change the thermodynamic properties of CO_2 as well as the thermal efficiency of the heat exchangers of the heat pump.

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