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ORIGINAL PAPER

An experimental study on charge optimization of a trans-critical CO_2 cycle

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Abstract The phasing out of hydrochlorofluorocarbon and chlorofluorocarbon fluids and further environmental problems arising from new, synthetic working fluids stimulate a continuously rising interest on natural candidates. The nontoxic and nonflammable CO_2 impacts neither on ozone depletion nor on global warming if leaked to the atmosphere. The critical temperature of CO_2 (31.1 °C) is almost ambient and, therefore, it undergoes a trans-critical refrigeration cycle. In cooling mode operation, a trans-critical CO_2 system, as compared to conventional air-conditioners, has a lower performance which, contrary to conventional systems, strongly depends on the refrigerant charge. In this paper, the performance of the trans-critical system was evaluated experimentally under different refrigerant charge amounts. The influence of charge on coefficient of performance (COP), cooling capacity, compression ratio, suction line superheat was analyzed in detail. The experimental results indicate that the COP of the trans-critical cycle attains a maximum at optimal refrigerant charge. By varying the charge the cooling capacity attains a maximum, as well, that corresponds to the optimal charge. In order to understand the effect of refrigerant charge on the performance of each device of the plant, an exergetic analysis based on the experimental data was carried out. The analysis shows that the exergy flow destroyed in the compressor is one of the major causes of overall exergy destruction. At the optimal refrigerant charge, the compression losses attain a minimum.

Keywords CO_2 · Trans-critical cycle · Refrigerant charge · Coefficient of performance · Optimization

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Introduction

The traditional refrigerants, i.e., chlorofluorocarbon (CFCs) and hydrochlorofluorocarbon (HCFCs) fluids, were banned by the Montreal Protocol (Montreal Protocol 1987) because of their contribution to the disruption of the stratospheric ozone layer (Molina and Rowland 1974). Indeed, according to the Kyoto Protocol, most hydrofluorocarbons (HFCs) have large global warming potentials (GWPs) and therefore provide a non-negligible, direct contribution to global warming when leaked to the atmosphere.

As a consequence, considerable research effort was devoted to the development of environmentally safe refrigeration systems using natural working fluids (Bhatkar et al. 2013). CO_2 was considered as a promising alternative of HFCs. Indeed, carbon dioxide is a component of air, obtained from atmosphere itself by fractionation. Thus, it would have no impact on global warming, apart from for the energy consumption associated with the fractionation process.

CO_2 has many excellent advantages in engineering applications, such as no toxicity, no inflammability, higher volumetric capacity that enables compact systems, lower pressure ratio, better heat transfer properties, complete compatibility with normal lubricants, easy availability, lower price and no recycling problems (Lorentzen and Petterson 1993; Lorentzen 1994, 1995; Liao et al. 2000; Bansal 2012).

The critical temperature of CO_2 (31.1 °C) is usually lower than typical heat rejection temperatures of air-conditioning and heat pump systems. This results in a trans-critical vapor compression cycle in lieu of a conventional one in water heating and comfort cooling and heating (Rebora et al. 2006; Tagliafico et al. 2012).

Application of a trans-critical CO_2 cycle to domestic water heating systems (Neksa et al. 1998; Neksa 2002) has advantages over conventional systems in terms of power consumption and heating efficiency (Hwang et al. 1999, 2001). The performance of trans-critical CO_2 cooling system, however, is lower than that of conventional air-conditioners, due to large expansion losses and high irreversibility during the gas-cooling process (Aprea and Maiorino 2009a, b; Pérez-García et al. 2013). Therefore, many researchers analyzed the performance of the trans-critical CO_2 refrigeration cycle, in order to identify opportunities to improve the energy efficiency of the system (Kim et al. 2004).

Among the improvement methods, many researchers analyzed the effect of an internal heat exchanger (IHX) to reduce throttling losses. Boewe et al. (2001) indicated up to 25 % COP improvements, as compared to the cycle without IHX. Mu et al. (2003) showed that larger IHX capacity can lower the optimum high-side pressure. Chen and Gu (2005) derived a practical expression for IHX effectiveness with a theoretical study and found that the latter is an important factor in achieving high cycle performance with a trans-critical cycle. Aprea and Maiorino (2008) showed that using the IHX, the COP is increased by 10 %. Tao et al. (2010) proposed that the IHX can reduce the throttling loss of the trans-critical cycle. Xu et al. (2011) investigated experimentally the effect of the IHX on a trans-critical CO_2 ejector system. Experimental results showed that the IHX reduces the ejector contribution to the system performance.

Cabello et al. (2012) presented an experimental analysis of the energetic performances of a modified single-stage CO_2 trans-critical refrigeration plant with an IHX based on vapor injection in suction line. Three different injection points have been evaluated experimentally. The measurements showed that the cooling capacity and the COP can be enhanced in 9.81 and 7.01 %, respectively.

Zhang et al. (2013) reported a theoretical study based on a thermodynamic analysis to study the effect of an IHX on the performance of an ejector-expansion trans-critical cycle. They found that adding an IHX does not necessarily improve the energetic performances in the ejector-expansion cycle, contrary to what occurs in a conventional throttling valve cycle.

Further improvements to system performance in reducing expansion losses are achieved by extracting and making use of the work potentially available from the process by means of an expander. A detailed parametric analysis (Robinson and Groll 1998) revealed that COP can increase if the expander efficiency is 30 %, but decreases when the expander isoentropic efficiency is 60 %.

Heidelck and Kruse (2000) discussed a new expander design based on a modified, reciprocating machine. Nickl et al. (2002) suggested a new expander–compressor design that provides a further 10 % increase in COP and a 50 % improvement over the same system with a throttle valve.

Ma et al. (2002) investigated the performance improvement of a CO_2 cycle by applying an expander with a reciprocating, rotary or screw compressor with equivalent power consumption. Baek et al. (2005a, b: Part I and Part II) reported theoretical and experimental studies on a piston–cylinder work recovery.

In order to reduce expansion losses, an experimental investigation of the valve dynamics has been carried out by Ma et al. (2012). A semi-hermetic reciprocating compressor was developed with a test system incorporated into the compressor performance test rig, with a focus on investigating the dynamics of the discharge valves.

The performance deterioration of a basic, single-stage CO_2 cycle can be improved using a multistage compressor and intercooling of liquid and vapor refrigerant. Inagaki et al. (1997) found that, for a two-stage CO_2 air-conditioning cycle, capacity and COP are improved by 35 and 20 %, respectively, at moderate ambient temperature and by 10 and 5 %, respectively, at high ambient temperature. Halozan and Rieber (2000) and Kim and Kim (2002) studied the performance of an auto-cascade refrigeration cycle using CO_2 as a working fluid. Huff et al. (2002) compared the performance of a two-stage CO_2 cycle with that of a conventional one by means of a simplified model.

Hwang et al. (2000) carried out experimental studies on the performance of a two-stage CO_2 cycle consisting in an intercooler and a flash tank. Groll et al. (2002) performed a thermodynamic analysis, focusing on the compression ratio in a trans-critical CO_2 cycle with and without an expansion turbine.

Eskandari et al. (2012) proposed a theoretical model of a new two-stage multi-intercooling trans-critical CO_2 refrigeration cycle with ejector-expansion device. In this cycle, the vapor compression line includes two intercoolers, the first with external coolant and the second with cycle refrigerant. The overall performance of the new cycle was compared with that of the conventional ejector refrigeration cycle (15.3 % higher) and with that of an IHX ejector refrigeration cycle (19.6 % higher).

To improve the COP of the cycle, an ejector can be used instead of a throttling valve to recover some of the kinetic energy of the expansion process. Liu et al. (2002) performed a thermodynamic analysis of the trans-critical CO_2 vapor compression/ejection hybrid refrigeration cycle. Li and Groll (2005) proposed a theoretical model showing that the COP of a CO_2 cycle can be improved by more than 16 % with an ejector.

The performance of heat pumps and refrigeration systems with different refrigerant charge amounts was studied in many experimental and theoretical analyses (ASHRAE 1993; Palm 2007). Choi et al. (2002) studied the effects of improper refrigerant charge on the performance of a water-to-water heat pump using R22 as the refrigerant, with an electronic expansion valve and a capillary tube. In a further study (Choi et al. 2004), they tested the performance of R407C under various refrigerant charge and outdoor temperature conditions.

Cho et al. (2005) compared experimentally the performance of a CO_2 cycle with that of other refrigerant systems, by varying the refrigerant charge amount.

Corberan et al. (2008) presented the experimental results regarding the optimization of the charge with a reversible water-to-water heat pump using propane as a refrigerant. Poggi et al. (2008) reported a review of refrigerant charge studies in refrigerant plants.

The considerable body of literature presented clearly underlines the problems related to the low-energetic performance of CO_2 systems. Overall optimization of the plant performance is, therefore, mandatory. The performance of the trans-critical CO_2 system strongly depends on refrigerant charge, much more than that of a conventional air-conditioner. Surprisingly enough, whereas, in the literature, many papers deal with the influence of charge in traditional vapor compression plants: just one does in trans-critical ones. Therefore, the present paper intends to reduce the gap by presenting an ample database of experimental results on the performance of a CO_2 trans-critical cycle at different refrigerant charges, in order: (1) to indentify the optimum amount of refrigerant maximizing cycle COP; (2) to evaluate the effect on cycle performance of undercharging or overcharging the plant; (3) to investigate the effect of refrigerant charge on the exergetic performance of each device of the plant.

The experimental tests discussed in this study are carried out with a prototype R744 system working as a classical split system to cool air in a trans-critical cycle.

The experimental prototype has been set up at the Cooling Techniques Laboratory of the University of Salerno. The experimental tests have been carried out from June to July 2012.

Materials and methods

Experimental plant

The experimental plant is described in Fig. 1 (Aprea and Maiorino 2009a, b; Aprea et al. 2012). Basically, there are

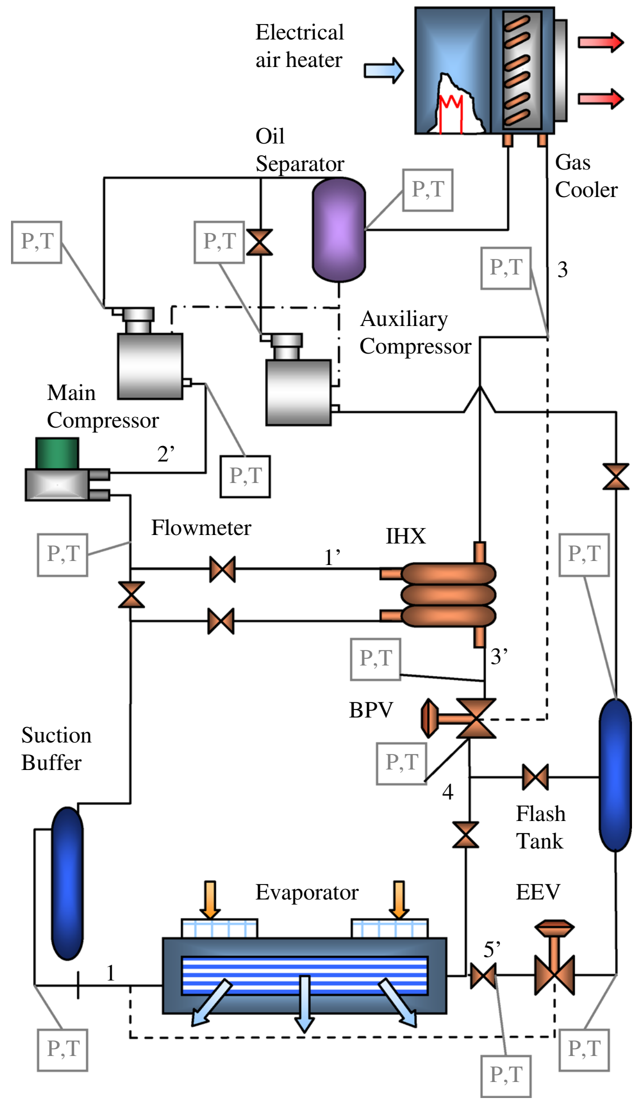

Fig. 1 A sketch of the experimental plant

two single-stage, semi-hermetic, reciprocating compressors, an oil separator, an air gas cooler, a liquid receiver, an air evaporator, an electronic expansion valve (EEV) and an electronically regulated back pressure valve (BPV).

An auxiliary compressor installed in a by-pass (Elbel and Hrnjak 2004) enables the analysis of the "flash vapor" influence. In a trans-critical cycle, the refrigerant at the inlet of the expansion device is typically a single-phase fluid with its high pressure exceeding the critical one. The refrigerant at the outlet of the expansion device is in a two-phase condition, provided that the fluid crosses the saturated liquid line during the isenthalpic expansion process. As a result, some fraction of the refrigerant flow enters the evaporator in vapor state and thus does not provide a significant cooling effect. Because of the flash gas by-pass, the evaporator is fed with pure liquid and the vapor by-passes the evaporator through the auxiliary compressor circuit. A flash tank is used downstream of the back pressure valve. The flash tank acts as an accumulator storing refrigerant in excess for test conditions requiring smaller mass flow rates. Furthermore, it also acts as a separator where the liquid phase accumulates at the bottom, the vapor phase occupying the top.

The main compressor is a semi-hermetic one, and its working pressure range is 15–120 bar. The intermediate pressure is of approximately 65 bar. Thus, the working pressure of the auxiliary, semi-hermetic compressor ranges from 55 to 120 bar. The nominal compression power for both compressors is 2.5 kW.

At evaporation temperatures of 5 and of 30 °C at the gas cooler exit, where the pressure is 80 bar, the refrigerating power is of about 3,000 W.

As discussed in (Aprea and Maiorino 2008), the use of the IHX in a carbon dioxide refrigeration split system improves the coefficient of performance (COP) by 10 %. Therefore, an IHX on the refrigerant line between compressor suction and the exit of the gas cooler was provided.

The lamination occurs thanks to the back pressure valve and to the electronic expansion one. Indeed, if a back pressure was employed as the only throttling device, it would not be possible to control the refrigerant mass flow rate, which is necessary to maintain the balance between refrigeration power and thermal load. Thus, in order to regulate the evaporation temperature, an auxiliary circuit can be used to by-pass the back pressure valve.

The air temperature on the gas cooler is modulated by some electrical resistances and an air flow driven by a blower located in a thermally insulated channel. This simulates variable external conditions, as well.

System monitoring

The plant is fully instrumented, in order to evaluate its performance as a whole, as well as that of each single component. Carbon dioxide pressure and temperature are measured at the inlet and at the outlet of each component. The working fluid mass flow rate is monitored at the main compressor suction (see Fig. 1).

Temperatures are measured by means of four-wire 100-Ω platinum resistance thermometers with an accuracy of ±0.15 K. The sensors are located outside the pipe, with a layer of heat transfer compound (aluminum oxide plus silicon) placed between the sensor and the pipe to provide good thermal contact. The whole pipe is insulated with 25-mm-thick flexible insulation tape. The temperature measurement system is checked against a sensor positioned in a pocket in a similarly insulated pipe. The difference between the two measurements was always <0.3 °C.

Pressures are measured by piezoelectric sensors whose current output is recorded directly on the data logger. They were calibrated by the manufacturer in the range of 0–100 bar with an accuracy of ±0.8 % F.S. Refrigerant mass flow rate is measured using a Micro Motion mass flow meter with an accuracy of ±0.2 %. Two-watt transducers are used to measure the electrical power supplied to the compressors with an accuracy of ±0.2 %. A balance is used to measure the refrigerant charge with an accuracy of ±0.2 %.

An analysis was carried out, according to the procedure suggested by Moffat (1985), to evaluate the accuracy of the experimental data discussed in this paper.

The refrigerant power is evaluated as:

$$\dot{Q}_{ev} = \dot{m}_{\mathrm{r}}\left(h_{\mathrm{r,out,ev}} - h_{\mathrm{r,in,ev}}\right) \tag{1}$$

with an uncertainty range of ±1.7–3.5 %.

The compression ratio is evaluated as:

$$\beta = \frac{p_{\mathrm{in.gc}}}{p_{\mathrm{out,ev}}} \tag{2}$$

with an uncertainty range of ±2.1–2.7 %.

The COP is evaluated as:

$$\mathrm{COP} = \frac{\dot{Q}_{\mathrm{ev}}}{\dot{W}} \tag{3}$$

The uncertainty of $\dot{W}$ is ±0.2 %. The COP uncertainty range is ±1.8–3.8 %.

The volumetric efficiency is defined as:

$$\eta_{\mathrm{v}} = \left(\frac{\dot{m}_{\mathrm{r}} v_{\mathrm{r,suc}}}{\dot{V}_{\mathrm{suc}}}\right) \tag{4}$$

where the displacement rate ($\dot{V}_{suc}$) is the volume swept through by the pistons in their suction strokes per unit time and is specified by the manufacturer. The uncertainty range of the latter parameter is ±2.3–3.9 %.

Temperature and pressure at key points of the plant are continuously monitored, to verify that the apparatus is indeed operating in steady-state conditions.

A software application (ISAAC) has been developed in Labview (Labview ver.7.1) using REFPROP subroutines (REFRPROP ver.9.1). It provides real-time evaluations of the COP and of the enthalpies at all points of the thermodynamic cycle. Furthermore, it depicts the whole cycle on a *p–h* diagram and checks that steady-state conditions actually hold. Figure 2 shows as an example a *p–h* diagram of the trans-critical cycle with the IHX.

Results and discussion

The heater provides a fixed ambient temperature of 30 °C.

Tests have been carried out by varying the refrigerant amount from 5.7 to 9 kg by progressively adding

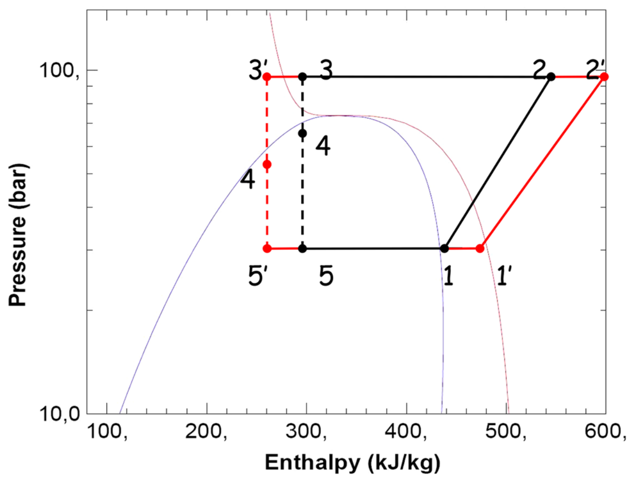

Fig. 2 A *p–h* diagram of the trans-critical cycle with the internal heat exchanger

refrigerant into the system in 100-g increments. In each test, the opening degree of the back up valve (BPV) is fixed.

CO_2 refrigerant charge (RC) amount was normalized by the maximum refrigerant charge (RC_{max}), i.e., by the maximum mass of saturated liquid at a room temperature of 25 °C:

$$\mathrm{NC} = \frac{\mathrm{RC}}{\mathrm{RC}_{\max}} = \frac{\mathrm{RC}}{V\,\rho_1(25\,^\circ\mathrm{C})} \tag{5}$$

where V is the volume of the whole plant (0.0393 m^3).

Figure 3 reports COP as a function of the normalized refrigerant charge (NC). It can be seen that COP initially increases and then decreases with increasing refrigerant charge. The optimum refrigerant charge is 0.269.

As the refrigerant charge increases, the system pressure increases in both the gas cooler and the evaporator. The pressure ratio decreases with the addition of refrigerant charge. When the system operates at low refrigerant charge, i.e., in undercharging conditions, the specific work of compression increases as a consequence of the thermodynamics properties and because the electric motor's windings are not properly cooled. As the refrigerant charge increases beyond its optimal value, COP decreases with the vaporization enthalpy.

Figure 4 reports the pressures in the gas cooler and in the evaporator, respectively, as a function of the NC. It is apparent that increasing the refrigerant charge increases both pressures. The pressure increase in the gas cooler under overcharging conditions (NC $>$ 0.269) can pose a security problem and the blowout disk might enter in action.

In Fig. 5 the compression ratio is reported as a function of refrigerant charge. The compression ratio decreases with the increasing refrigerant charge. As the refrigerant charge approaches the optimum value (NC $=$ 0.269), the slope of the compression ratio rapidly decreases.

Figure 6 shows the effect of NC on mass flow rate and indicates that the latter increases with refrigerant charge. Beyond the optimal value, the increase is quite limited. Indeed, by increasing the refrigerant charge, the compression ratio decreases, thus increasing the volumetric efficiency of the compressor.

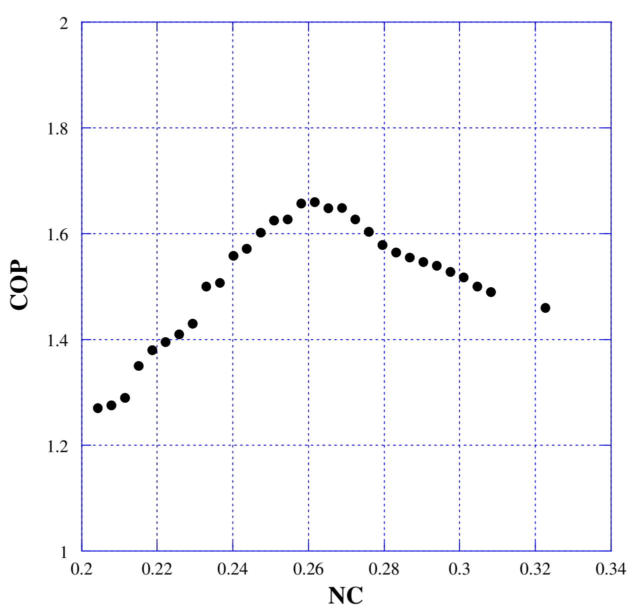

Fig. 3 COP as a function of normalized refrigerant charge

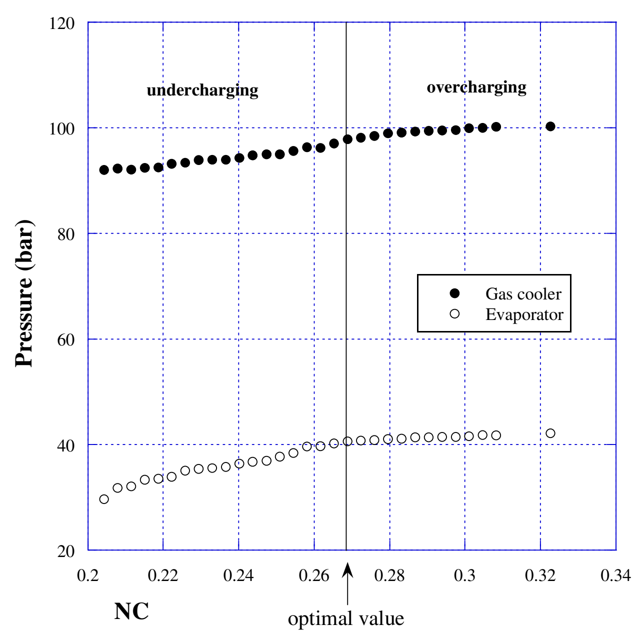

Fig. 4 Pressures in the evaporator and in the gas cooler as a function of normalized refrigerant charge

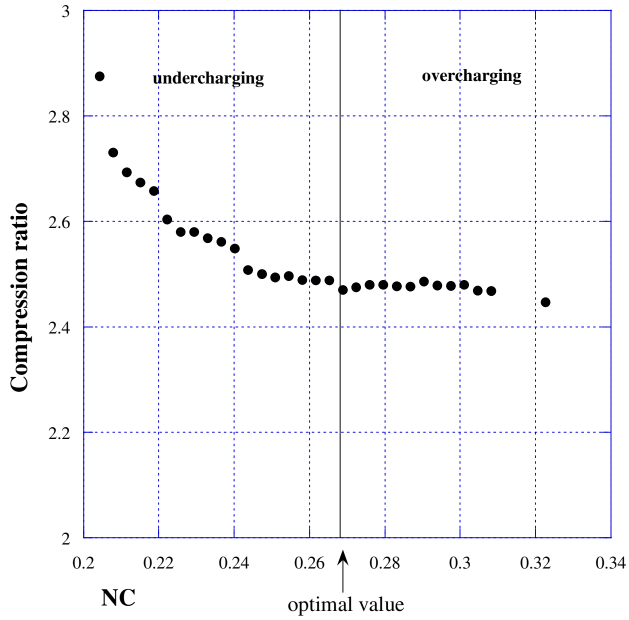

Fig. 5 Compression ratio as a function of normalized refrigerant charge

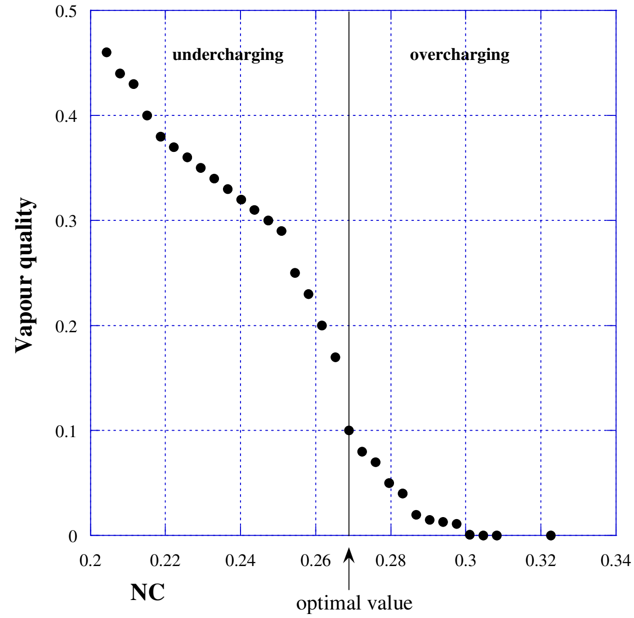

Fig. 7 Vapor quality at the outlet of the back pressure valve as a function of normalized refrigerant charge

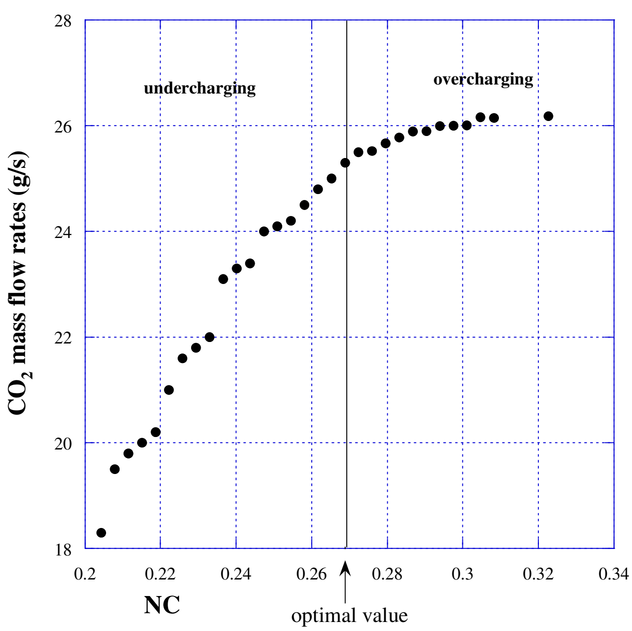

Fig. 6 CO_2 mass flow rate as a function of normalized refrigerant charge

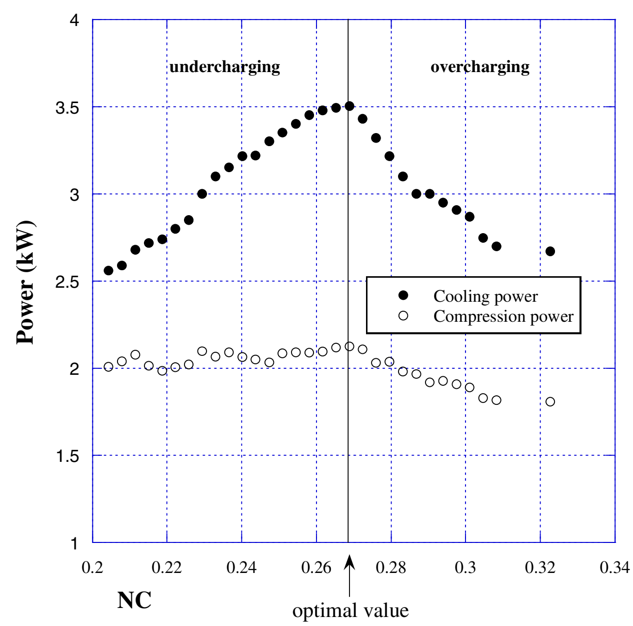

Fig. 8 Cooling capacity and compression power as a function of normalized refrigerant charge

Figure 7 indicates that vapor quality at the outlet of the back pressure valve decreases with increasing charge, eventually reaching very low values (near the conditions of saturated liquid) at overcharging. Indeed, the gas cooler pressure increases with refrigerant charge. The external temperature is fixed, and therefore, the temperature at the gas cooler exits too. This leads to a decrease in the enthalpy at the outlet of this device decreasing the vapor quality at the evaporator inlet.

Figure 8 shows the variations in both cooling capacity and compression power with increasing NC. Initially, the compression power slightly increases and then decreases. Indeed, when the system operates in undercharging conditions, the specific work of compression slightly increases as a consequence of the corresponding increase in compressor suction pressure. This yields higher specific volume and mass flow rate. This increase is also because windings of the electric motor are not properly cooled. Under overcharging conditions, the power decreases because of the compressor malfunctioning. Indeed, at an excessive charge, liquid-phase refrigerant can reach the compressor. Furthermore, it should be noted that the cooling capacity increases with refrigerant charge, reaching a maximum. The maximum in cooling capacity corresponds to maximum COP. Indeed, by increasing the charge, refrigerant mass flow rate and the evaporating pressure both increase. For undercharged conditions, the cooling capacity increases with increasing charge, due to the increase in refrigerant mass flow rate and to the decrease in vapor quality at the inlet of the evaporator. For overcharged conditions, the cooling capacity decreases with increasing refrigerant charge, since the effect of the increase in evaporating pressure prevails. Indeed, beyond the optimal value, the increase in refrigerant mass flow rate is very limited and the refrigerant enthalpy at the evaporator outlet increases with increasing the evaporating pressure. This leads to a reduction in the specific variation of the vaporization enthalpy.

Table reports gas cooler pressure, evaporator pressure, compressor volumetric efficiency, temperature at the compressor inlet, as a function of the refrigerant charge. The results reported in Table 1 clearly show that the temperature at the inlet of the compressor always decreases increasing refrigerant charge. Indeed, at overcharging conditions, the superheating degree at the compressor inlet is very low and liquid refrigerant might reach the compressor. The volumetric efficiency always increases with refrigerant charge. Indeed, by increasing refrigerant charge, the compression ratio decreases thus increasing the compressor's volumetric efficiency.

In order to understand the effect of refrigerant charge on the performance of each device, an exergetic analysis based on experimental data was carried out. This analysis indicates the total plant irreversibility distribution among the plant components, pinpointing those that contribute most to overall plant-inefficiency.

The exergy flow destroyed in the evaporator is evaluated as:

$$\dot{\mathrm{E}}\mathrm{x}_{\mathrm{des,ev}} = \dot{m}_{\mathrm{r}}\left(\mathrm{ex}_{\mathrm{r,in,ev}} - \mathrm{ex}_{\mathrm{r,out,ev}}\right) - \dot{Q}_{\mathrm{ev}}\,|\tau_{\mathrm{ev}}| \tag{6}$$

where the dimensionless exergetic temperature of the air flow in the evaporator can be defined as:

$$\tau_{\mathrm{ev}} = 1 - \frac{T_{\mathrm{o}}}{T_{\mathrm{TMT,a,ev}}} \tag{7}$$

The exergy flow destroyed in the gas cooler is evaluated:

Table 1 Gas cooler and evaporator pressures, compressor volumetric efficiency and temperature at the inlet of the compressor as a function of refrigerant charge

Charge (kg)	p_{ev} (bar)	p_{gc} (bar)	η_{v}	$T_{\mathrm{in,cp}}$ (°C)
5.7000	29.677	92.000	0.65568	13.230
5.8000	31.828	92.300	0.66780	12.800
5.9000	32.091	92.100	0.67900	12.300
6.0000	33.357	92.400	0.68000	12.000
6.1000	33.514	92.500	0.68800	11.900
6.2000	33.891	93.200	0.69300	11.700
6.3000	35.113	93.400	0.69800	11.500
6.4000	35.434	93.900	0.69900	11.400
6.5000	35.602	93.990	0.70010	11.330
6.6000	35.741	94.000	0.71200	11.000
6.7000	36.409	94.300	0.72300	10.920
6.8000	36.744	94.800	0.73100	10.410
6.9000	36.962	94.993	0.73600	9.8000
7.0000	37.698	95.000	0.74400	9.6600
7.1000	38.404	95.626	0.75500	9.3300
7.2000	39.630	96.300	0.76900	9.1200
7.3000	39.755	96.208	0.77900	8.7000
7.4000	40.251	97.005	0.79900	8.5000
7.5000	40.606	97.850	0.80300	8.3300
7.6000	40.738	98.171	0.80600	8.2200
7.7000	40.868	98.481	0.81100	8.1100
7.8000	41.076	98.982	0.81900	8.0200
7.9000	41.119	99.093	0.82200	7.9800
8.0000	41.368	99.300	0.82900	7.8200
8.1000	41.436	99.460	0.83400	7.7900
8.2000	41.460	99.520	0.83900	7.5900
8.3000	41.506	99.614	0.84200	7.4300
8.4000	41.619	99.928	0.84900	7.4200
8.5000	41.841	100.00	0.85300	7.3800
8.6000	41.785	100.20	0.85500	7.3100
9.0000	42.143	100.30	0.86200	7.2200

$$\dot{\mathrm{E}}\mathrm{x}_{\mathrm{des,gc}} = \dot{m}_{\mathrm{r}}\left(\mathrm{ex}_{\mathrm{r,in,gc}} - \mathrm{ex}_{\mathrm{r,out,gc}}\right) - \dot{Q}_{\mathrm{gc}}\,\tau_{\mathrm{gc}} \tag{8}$$

where the dimensionless exergetic temperature of the air flow in the gas cooler can be defined as:

$$\tau_{gc} = 1 - \frac{T_o}{T_{TMT,a,gc}} \tag{9}$$

The exergy flow destroyed in the compressor, neglecting the heat transfer with the environment, is evaluated as:

$$\dot{\mathrm{E}}\mathrm{x}_{\mathrm{des,cp}} = \dot{m}_{\mathrm{r}}\left(\mathrm{ex}_{\mathrm{r,in,cp}} - \mathrm{ex}_{\mathrm{r,out,cp}}\right) + \dot{W} \tag{10}$$

The exergy flow destroyed in each valve (BP and EEV) is evaluated as:

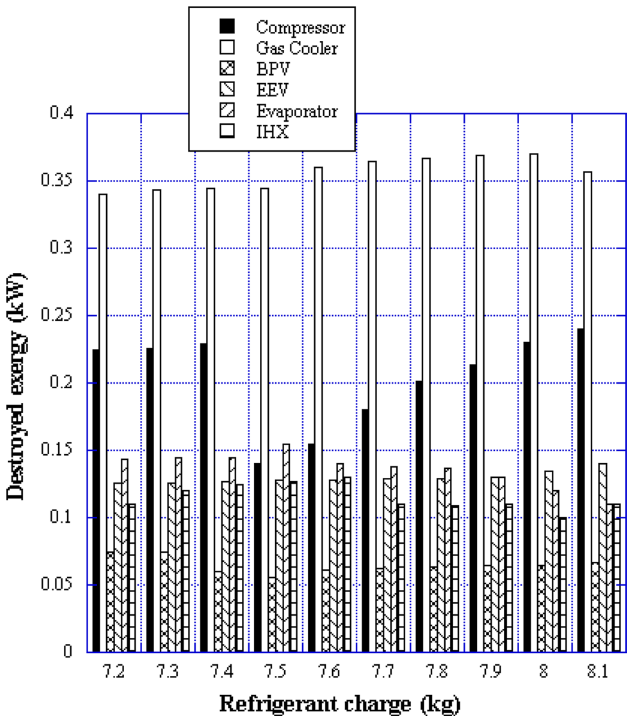

Fig. 9 Destroyed energy in the devices of the experimental plant as a function of refrigerant charge

$$\dot{E}x_{des,v} = \dot{m}_r\left(ex_{r,in,v} - ex_{r,out,v}\right) \tag{11}$$

The exergy flow destroyed in the IHX is evaluated as:

$$\dot{E}x_{des,ihx} = \dot{m}_r\left(ex_{r,in1,ihx} - ex_{r,out1,ihx}\right)_1 + \dot{m}_r\left(ex_{r,in2,ihx} - ex_{r,out2,ihx}\right) \tag{12}$$

Figure 9 reports the exergy flow destroyed in each component of the plant as a function of the refrigerant charge.

Figure 9 clearly shows that the exergy loss in the gas cooler predominates and accounts for more than 30 % of the total exergy loss. The exergy losses in this device are not affected significantly by refrigerant charge.

The exergy flow destroyed in the compressor is one of the major causes of exergy destruction in the plant. The refrigerant charge strongly affects the exergy losses in this device. Under both overcharging and undercharging conditions, the compressor losses amount to approximately 22 % of the total losses. At the optimal refrigerant charge, the compression losses are 15 % of the total. When the system operates at low refrigerant charge, the specific work of compression increases as a consequence of the thermodynamic properties and because the electric motor's windings are not properly cooled. When the refrigerant charge yielding a correct cooling of the electric motor's windings, the compression power attains a minimum. As the refrigerant charge increases beyond the optimal value, the compressor power consumption increases because the increase in suction pressure in the compressor results in higher specific volume and mass flow rate.

Conclusion

The performance of the trans-critical system is more sensitive to refrigerant charge amount than that of a conventional air-conditioner. System performance stem is critically linked to an appropriate mass charge. In the present paper, the effects of refrigerant charge were analyzed by means of an experimental prototype, working as a classical spit system to cool air in a trans-critical cycle.

The COP of the trans-critical cycle attains a maximum value corresponding to the optimal refrigerant charge (7.5 kg).

In order to characterize the effects of refrigerant charge, a detailed analysis was carried out on the major parameters affecting cycle performances by varying the charge from −32 to +17 % of the optimal value. In this range, plant COP varies between −21 and −7.5 % of the optimal value. Therefore, the COP reduced more significantly at undercharged conditions than at overcharged.

As the refrigerant charge increases, the system pressure increases in both gas cooler and evaporator, thus decreasing the compression ratio. The volumetric compressor's efficiency decreases with increasing refrigerant mass flow rate. The vapor quality at the inlet of the evaporator decreases due to a reduction in the enthalpy at the exit of the gas cooler. Initially, the compression power slightly increases and then decreases with increasing charge. Indeed, at an excessive charge, liquid refrigerant can reach the compressor. The cooling capacity increases with refrigerant charge reaching a maximum and then decreases. The maximum in cooling capacity corresponds to the maximum COP.

In order to understand the effect of refrigerant charge on the performance of each device of the plant, an exergetic analysis based on the experimental data was carried out. This analysis shows that the exergy flow destroyed in the compressor is one of the major causes of exergy destruction in the plant. The refrigerant charge strongly affects the exergy losses in this device. Under both overcharging and undercharging conditions, the compressor losses amount to about 22 % of the total. At the optimal refrigerant charge, the compression losses are 15 % of the total. Therefore, at the optimal refrigerant charge, the compression losses attain a minimum.

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List of symbols

Symbols

BPV	Back pressure valve
COP	Coefficient of performance
ex	Specific exergy (J/kg)
$\dot{\mathrm{E}}\mathrm{x}$	Exergy flow (W)
EEV	Electronic expansion valve
h	Specific enthalpy (kJ/kg)
IHX	Internal heat exchanger
$\dot{m}_{\mathrm{r}}$	Refrigerant mass flow rate (kg/s)
NC	Normalized charge
$\dot{Q}$	Heat transfer rate (kW)
RC	Refrigerant charge (kg)
SC	Specific charge (kg/kW)
T	Temperature (°C, K)
T_{o}	Ambient temperature (°C, K)
v	Specific volume (m^3/kg)
$\dot{V}$	Volumetric flow rate (m^3/s)
$\dot{W}$	Compression power (kW)

Greek symbols

β	Compression ratio
η_{v}	Volumetric efficiency
τ	Dimensionless exergetic temperature

Subscripts

a	Air
des	Destroyed
cp	Compressor
ev	Evaporative
ev	Evaporator
gc	Gas cooler
in	Inlet
IHX	Internal heat exchanger
max	Maximum
TMT	Mean thermodynamic
out	Outlet
r	Refrigerant fluid
suc	Suction
v	Expansion valve

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