A Study on the Reduction of Throttling Losses in Automotive Air Conditioning Systems Through Expansion Work Recovery

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ABSTRACT

Refrigerant flow control of the vapor compression refrigeration cycle has been largely relied upon throttling devices, such as capillary tubes, short tube orifices and expansion valves. The expansion process through these devices represents an obvious thermodynamic loss that can be mitigated by the application of different solutions such as intracycle evaporative cooling, economizer and ejector cycles or work producing expansion devices. The present paper concentrates on studying the latter, specifically applied for automotive air conditioning systems. A basic thermodynamic model was developed to simulate a vapor compression cycle for automotive air conditioning purposes employing expanders. Data and cycle conception from devices that were successfully tested and reported in the literature are briefly commented. A preliminary study was carried out taking into account some of the several characteristics that are inherent to mobile air conditioning systems, including high rejection temperature. This preliminary study concentrated in the subcritical vapor compression R134a cycle with an expander and a thermostatic expansion valve placed downstream the expander.

1. INTRODUCTION

Due to an entropy generation during isenthalpic expansion that is 3 to 4 times higher than in subcritical vapor compression refrigeration systems, transcritical CO_2 cycles are most susceptible to COP improvement with reduction of losses associated with the expansion process (Huff et al., 2003). An early study on the efficiencies of transcritical CO_2 cycles with and without expansion work recovery was presented by Robinson and Groll (1998). A thermodynamic model for the analysis of both cycles was developed. An empirical equation for the compressor efficiency was employed and expansion turbine efficiency was assumed to be either 0.3 or 0.6. Expander and compressor mechanical efficiencies were prescribed at 0.9. The effectiveness of the internal heat exchanger was taken into account in the model. An interesting finding from Robinson and Groll (1998) is that the combined use of an internal heat exchanger and the expansion turbine leads to a lower COP (a reduction to the order of 8%) if compared to those should no internal heat exchanger be used. Depending on the operating conditions and modeling assumptions, an ideal expander could improve the system COP by 45% to 75% (Huff et al., 2003) and capacity by 5% to 15% (Huff et al., 2002). This establishes an attractive paradigm, in spite of the inevitable inefficiencies that are inherent to such machines.

Zha et al. (2003) summarize the possibilities for expanders, which can be classified into two main categories: positive displacement expanders or turbines. Positive displacement devices can be either of the reciprocating, rotary or orbital types. Rotary expanders, like compressors, can make use of the sliding-vane, rolling piston or screw technologies, and orbital expanders, of the scroll type. Zha et al. (2005) list the pros and cons of the use of each technology in regard to different applications (e.g., small, medium or large capacity systems, commercial refrigerators, mobile air conditioner, heat pumps, and residential air conditioning systems). For automotive air conditioning systems, Zha et al. (2005) recommend the rolling piston expander, for its "compactness, low cost, wide application and reasonable efficiency". All other positive displacement possibilities were not discarded, though, in particular, the reciprocating expander, since, according to Zha et al. (2005), this is the type of compressor presently in use for CO_2 systems. Huff et al. (2005) alert for the additional expense for the required use of active valves (i.e., valves, like in internal combustion engines, that require a mechanical or electronic mechanism for its operation) which leads to slower operation of the expander (and, consequently, to a larger displacement volume), and, due to the presence of nearly incompressible fluids, high force peaks and accelerated wear and damage. In particular, for CO_2 expanders, Zha et al. (2003) list the main barriers to the development this technology, namely: sealing, due to high pressure differences, cavitation and liquid slugging, due to vapor liquid flow, lubrication, and vibration and noise control. Yet, the description of a number of expander prototypes can be found in the literature. The majority of them refer to the CO_2 transcritical cycle.

Quack and co-workers (Heyl and Quack, 1999; Heyl and Quack, 2000; Nickl et al., 2003; Nickl et al., 2005) report a development program of a free-piston expander–compressor for the CO_2 vapor compression refrigeration cycle. A total of 13 cycles, ranging from the conventional throttling device-compressor cycle to expander-compressor cycles, with one or two stages, with or without expansion work recovery for compression, with combination of throttling device and expander, have been proposed (Heyl and Quack, 1999). Their compressor-expander system is based on free-piston engine technology, with double acting pistons working with a common piston rod. The first generation expander operated in the so-called full-pressure mode, with the expander going through the same pressure difference as the compressor and with expander and compressor pistons moving with exactly identical strokes. Subsequent prototypes relied upon partial decoupling between compressor and expander, with the expander accounting for only part of the whole compressor pressure difference, in fact, only the second compression stage (Nickl et al., 2003). Reported improvement of the COP over the throttle cycle was of the order of 30% (Nickl et al., 2003). According to (Nickl et al., 2005), vapor compression transcritical refrigeration cycles, running on CO_2, face a relatively low COP in view of (i) a high compression discharge temperature, which can be mitigated with the use of multi-stage compression with intercooling, and (ii) large exergy losses with the throttling. The use of a three-stage expander cycle, where a vapor-liquid separator, installed between the second and the third expansion stages, supplies vapor to the third expansion stage and liquid to the electronic or thermostatic expansion valve, addresses effects (i) and (ii). In Germany, Heidelck and Kruse (2000) proposed a combined compressor-expander reciprocating machine, comprising a rotating slotted disk to determine opening and closing positions. An experimental apparatus to test the device was presented.

A piston-cylinder type work output expansion device was also developed and tested by Baek et al. (2005a). The prototype was based on a commercially available small displacement two-cylinder four-stroke spark ignition engine. The engine was highly modified to accommodate its new function. Valve opening-closing control and phase angle between pistons were two issues that led to the first substantial modifications to the original engine. Design considerations revealed that the geometry of the expander (maximum to minimum volume ratio) sets the expansion ratio of the device, and the rotational speed, the mass flow rate. Should full pressure expansion work recovery, directly to the compressor, be used, the latter conclusion imposes a new link between compressor and expander, namely, rotational speed (with or without speed reduction), besides refrigerant mass flow rate. This requires a delicate balance between compressor and expander performance curves. Tests were performed with the prototype of the piston-cylinder type expander (Baek et al., 2005a) and, in spite of expected low isentropic expansion efficiency, approximately 11%, an increment of 10.5% in the COP was found. A simulation of the device was also carried out (Baek et al., 2005b). More recently, two prototypes of free piston expanders, single and double acting types, to operate in a CO_2 transcritical cycle, were developed successfully by Liu et al. (2007). For the double acting prototype, the measured (via indicated diagram) isentropic efficiency was 62%, operating between pressures of 0.78 MPa and 0.33 MPa.

Positive displacement rotary and orbital devices have also been adapted to form work producing expanders. Zha et al. (2003), for example, provide guidelines for the design of a CO_2 rolling piston expander. Yang et al. (2007c) propose a combined CO_2 hermetic compressor-expander device, with both compression and expansion processes taking place with rolling piston devices and with two-stages. The paper reports on experimental data of the compressor, as a first step of a development program. An insight on the experimental P-V diagram of a CO_2 rolling piston expander is provided by Zeng et al. (2007). Indicated diagrams were produced and analyzed for three different operating conditions. Expander efficiency varied from, approximately, 0.3 to 0.5, within a rotational speed between 650 rpm and 1800 rpm. Under and over-expansion, typical phenomena of positive displacement machines operating with no valves, have been observed for speeds below and above the design speed, respectively.

Fukuta et al. (2003) present preliminary results of the prototype of a vane expander, adapted from a vane-type oil pump. Experimental conditions were suction and discharge pressures of 4.1 and 9.1 MPA, respectively, and inlet temperature of 40 ºC. A maximum volumetric efficiency of 0.64 and a maximum total efficiency of 0.43, were reported, with these values reducing to approximately 0.2, when rotational speed decreases from 2000 rpm to 500 rpm. Predictions of COP improvement, as a function of compression and expansion efficiency, were carried out for a CO_2 cycle with 10 MPa, 3.48 MPa and 40 ºC as operating conditions, for high and evaporating pressures and expander inlet temperature, respectively. For example, with a compressor efficiency of 0.7 and an expander efficiency of 0.6, Fukuta et al. (2003) pointed to a COP improvement of 30%. Preliminary data on a vane-type prototype is reported by Yang et al. (2007a). Theoretical models for the transcritical to two-phase expansion process of vane-type expanders were also developed (Fukuta et al., 2003; Yang et al., 2007a). An et al. (2007) address the problem of lubrication CO_2 rolling piston expanders.

Huff et al. (2003) developed two prototypes of scroll expanders adapted from two existing automotive R134a compressors. The two prototypes differed in the conversion measures from the original compressor, which affected the resulting expander displaced volume and, as a consequence, the leakage path length to pocket volume ratio. Isentropic and volumetric efficiencies were measured and plotted against expander rotational speed. The second expander prototype operated in a speed range between 1400 rpm and 220 rpm. Volumetric efficiency remained within 50% and 68%, and the isentropic efficiency reached a maximum of 42%, at approximately 1800 rpm. A simulation model, including the effects of internal leakage, heat transfer and valve losses, was also developed to predict the overall performance of vapor compression cycles with integrated compressor-expander devices (Huff et al., 2002). A numerical simulation of a two-stage compression CO_2 heat pump water heater, with a combined scroll expander-compressor unit, is also reported by Kim et al. (2007).

Brasz et al. (2000) report on the development program of a device, named the “expressor” (Brasz, 2001), which combines a twin screw compressor to a twin screw expander, forming an independent free running device. Test results, with R113, indicated an overall expansion-compression efficiency of the order of 55%, which corresponded roughly to 70% expansion efficiency and 80% compressor efficiency. The device, later tested with R134a (Brasz, 2003), was regarded as a highly cost effective means of power recovery from the expansion process in refrigeration cycles, due to its characteristics which include a high efficiency sealed unit with only one pair of rotors, low vibration, adaptability to two-phase flow, and no need for any seals or valves and timing gear (Brasz et al., 2000; Zha et al., 2003). Stosic et al. (2002) propose the twin screw compressor and expander technology for CO_2 refrigeration systems, presenting solutions to overcome the high bearing forces likely to be associated with the pressure differences that are inherent to this refrigerant. Finally, turbo expanders are also cited in the literature (Brasz, 1995a; 1995b). The use of a Pelton-type expander, with a backward-curved-blade impeller, has been recently reported in the literature (He et al., 2007). This expander was tested in a R410A refrigeration system and isolated performance data (rotational speed and specific recovery power) were obtained as a function of condensing and evaporating temperatures.

2. VAPOR COMPRESSION CYCLES WITH EXPANSION WORK RECOVERY

A review on the available literature reveals a reasonable number of proposed options for thermodynamic cycles to make the best use of the expansion work recovery from vapor compression refrigeration cycles. Cycles can be classified under many different categories, following specific criteria. One major criterion is if the heat rejection temperature is above the refrigerant critical temperature. For reasons explained by many authors (e.g., Huff et al., 2003), the CO_2 transcritical cycle is the most susceptible to COP improvements with the recovery of the expansion work. On the other hand, the high pressure differences encountered in the CO_2 transcritical cycle pose new technological challenges. Therefore, performance and technological outcomes are expected to differ significantly between transcritical and traditional vapor compression cycles with expanders, even if the same thermodynamic cycle is used (as far as component disposition in the flowchart is concerned). The existing proposed cycles are briefly outlined next.

Heyl and Quack (1999) list 14 of these cycles, starting with the basic vapor compression refrigeration cycle (compressor, condenser or gas cooler, throttle device and evaporator). The throttle device is then substituted by the expander which can either supply work directly to the compressor (direct drive type – Hiwata et al., 2003) or to another power sink (generator type – Hiwata et al., 2003). The first option results in an obvious reduction in power consumption (typically, electricity or shaft power from an internal combustion engine) but inevitably ties expander and compressor speeds. Even with a speed reduction between them, compressor and expander may find themselves operating away from their maximum efficiencies over the operational range of the system. This issue was also addressed by Huff et al. (2002) and Hiwata et al. (2003). The single stage direct drive cycle was the option proposed by Brazs et al. (2000), for the twin screw compressor-expander combination, and by Yang et al. (2007b), for a theoretical optimal heat rejection pressure study. Both recovery schemes (expansion power recovery to compressor or to a separate power shaft) can also be applied to the cycle with internal heat exchanger (liquid line-suction line heat exchanger in sub-critical vapor compression cycles). Because of refrigerant thermodynamic characteristics, the internal heat exchanger is an almost mandatory presence in the traditional CO_2 transcritical cycle without work recovery. However, Robinson and Groll (1998), in a seminal paper, pointed out that the use this exchanger in conjunction with the expander would, in fact, degrade the performance of the work recovery cycle.

Huff et al. (2002) included two other variations of the expansion work recovery transcritical cycle, by fitting an expansion valve (isenthalpic expansion) either in parallel or downstream the expander. This was done to provide room for adjustment of the optimum heat rejection pressure, in view of non-adjustable expander speed and volume ratio. Finally, Hiwata et al. (2003) proposed another variation for single-stage combined expander and expansion valve, with the latter placed upstream the expander, and a by-pass valve installed in parallel to the expansion valve/expander line.

With the adoption of multi-stage compression, the possibilities of expansion work recovery increases, as reported by several authors (Heyl and Quack, 2000; Nickl et al., 2003; Hiwata et al., 2003; Nickl et al., 2005; Kim et al. 2007; Yang et al., 2007c; Yang et al., 2007d). For the purposes of the present study, although it is recognized that work recovery with multi-stage compression may lead to an even greater COP (Nickl et al., 2005), efforts were concentrated in the most probable outcome from an adaptation of current automotive air conditioning systems, namely, the single-stage compression cycle. The single-stage vapor compression refrigeration cycle with expansion work direct recovery to compressor, with an expansion valve placed downstream the expander, as proposed by Huff et al (2002), is here analyzed. Figure 1 depicts the schematics of the cycle. Heat rejection is via a condenser (CD) or a gas cooler (GC), depending on whether sub-critical or transcritical cycle is employed. All recovered power from the expander (EX) goes to the compressor (CP). The expansion valve (XV) is located downstream the expander to control either the heat rejection pressure (transcritical cycle) or the evaporator outlet superheat (sub-critical cycle).

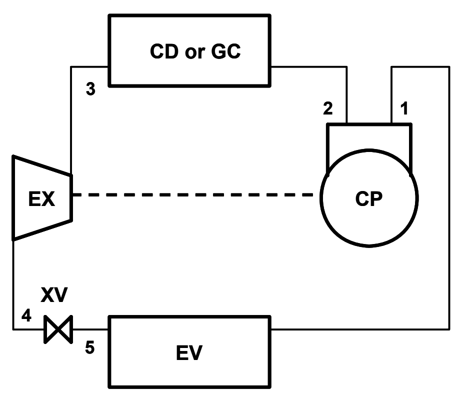

Figure 1: Single-stage compression cycle with expansion work direct recovery to compressor and downstream thermostatic expansion valve.

For this preliminary study, the sub-critical cycle, running with refrigerant R134a, was chosen. In spite of the superior improvement in COP one finds when expansion work recovery is applied to transcritical CO_2 cycles, the choice for the present analysis was based on the evidence that current automotive air conditioning systems still run on the subcritical vapor compression cycle. This study aims at presenting the potential for COP improvement with expansion work recovery applied to a cycle, Figure 1, that apparently requires the least of modifications to existing systems.

3. THERMODYNAMIC MODEL

A basic thermodynamic model was developed to evaluate the performance of the cycle under different operating conditions. Fundamental conservation mass and energy conservation equations were applied to each component of the cycle. Steady-state regime was assumed. Heat losses and pressure drops were neglected.

The work of compression, w_{cp}, is given in terms of the isentropic work of compression, $w_{cp,s}$:

$$w_{cp} = \frac{w_{cp,s}}{\eta_{cp,s}} = \frac{h_{2s} - h_1}{\eta_{cp,s}} \tag{1}$$

The volumetric and isentropic compressor efficiencies were taken from Brown et al. (2002).

$$\eta_{v,cp} = 0.8263\left[1 \text{ - } 0.09604\left(\theta_{cp}^{1/\gamma} - 1\right)\right] \tag{2}$$

$$\eta_{s,cp} = 0.9343 - 0.4478\ \theta_{cp} \tag{3}$$

where θ_{cp} is the compressor pressure ratio.

$$\theta_{cp} = \frac{P_{cd}}{P_{ev}} \tag{4}$$

and γ, the specific heat ratio, taken at suction conditions. According to Brown et al. (2002), equation (2) was obtained by curve-fitting data from different authors for CO_2 and R134a compressors. Equation (3) was used by Brown et al. (2002) for both CO_2 and R134a compressors, for the purpose of their comparison analysis.

A prescribed degree of subcooling is assumed for the condenser:

$$\Delta T_{sc} = T_{cd} - T_3 \tag{5}$$

The work of expansion, w_{ex}, is :

$$w_{ex} = w_{ex,s}\eta_{ex,s} = \left(h_3 - h_{4s}\right)\eta_{ex,s} \tag{6}$$

The literature is scarce on information about the efficiencies of positive displacement expander for subcritical vapor compression cycles, in particular, their dependence on operational conditions, such as pressure ratio or rotational speed. For transcritical cycles one could mention experimental data from: (i) Fukuta et al. (2003) and (ii) Huff et al. (2005), with volumetric and isentropic efficiencies of CO_2 vane and scroll expanders, respectively, as a function of rotational speed; from (iii) Baek et al. (2005), with isentropic efficiency and mass flow rate at three different operating conditions of a CO_2 piston-cylinder expander; (iv) from Zeng et al. (2007), who present data for a CO_2 rolling piston, with three distinct working conditions, from which the isentropic efficiency can be derived, and (v) from Brasz et al. (2000), with a combined efficiency for the compressor–expander device running on R113, as a function of shaft speed and compressor mass flow rate. Adiabatic expansion is assumed, so that:

$$w_{ex} = h_3 - h_4 \tag{7}$$

Refrigerant expansion with work recovery is from P_{cd} to P_4 :

$$\theta_{ex} = \frac{P_{cd}}{P_4} \tag{8}$$

The thermostatic expansion valve was simulated from its function, i.e., to provide a prescribed degree of superheat at the compressor inlet. And the expansion process through the valve is supposed to be adiabatic and, consequently, isenthalpic.

$$T_1 = T_{ev} + \Delta T_{sh} \tag{9}$$

$$h_4 = h_5 \tag{10}$$

The refrigerating effect, q_{ev}, is, of course:

$$q_{ev} = h_1 - h_5 \tag{11}$$

and the refrigerating coefficient of performance, taking into account the expansion work recovery and both compressor and expander mechanical losses, is:

$$COP = \frac{q_{ev}}{\left(\frac{w_{cp}}{\eta_{cp,m}}\right) - w_{ex}\,\eta_{ex,m}} \tag{12}$$

With no expansion work recovery, the coefficient of performance is:

$$COP_n = \frac{h_1 - h_3}{w_{cp}\left(\eta_{cp,m}\right)^{-1}} \tag{13}$$

4. RESULTS

A simulation program, using on the EES (Engineering Equation Solver) platform, was developed to solve the system of equations above. Figure 2 shows the predicted results for the improvement in the COP, $\delta_{COP} = COP/COP_n$, as compared to the ratio between the compression pressure ratio and the expansion ratio, $\Phi_{ex,cp} = \theta_{ex}/\theta_{cp}$. The following values were used in the simulation: $T_{cd} = 40 - 50\,^{o}C$, $T_{ev} = 0\,^{o}C$, $\Delta T_{sc} = 5\,^{o}C$, $\Delta T_{sh} = 15\,^{o}C$, $\eta_{ex,m} = \eta_{cp,m} = 0.9$. The expander pressure ratio was made to vary from the compressor pressure ratio, full pressure cycle, with no expansion valve, to a minimum of 20% of that value. The expander isentropic efficiency assumed values from 0.2 to 0.8. It can be observed, from Figure 2, that, even with the sub-critical vapor compression cycle, gains on the COP are reasonable, at least for large expander efficiencies. However, with operation of more realistic values for $\eta_{ex,s}$, theoretical COP gains lie within 0-5%, approximately. This improvement will also depend on the expansion ratio, which is a function of expander characteristics and of the coupling between compressor and expander, not covered in the present analysis. The COP improvement was more prominent for a higher condensing temperature. As expected, the observed variation on the refrigerating effect was marginal.

5. CONCLUSIONS

A theoretical analysis was carried out on the potential for COP improvement with the partial recovery of the expansion work for a vapor compression cycle running with R134a. In spite of the attention that has been primarily given for this application to transcritical CO_2 cycles, it has been proven that gains in cycle efficiency may still be relevant to traditional automotive air conditioner cycles. The present analysis also establishes conditions for future studies of such cycle running on new LGWP synthetic fluids.

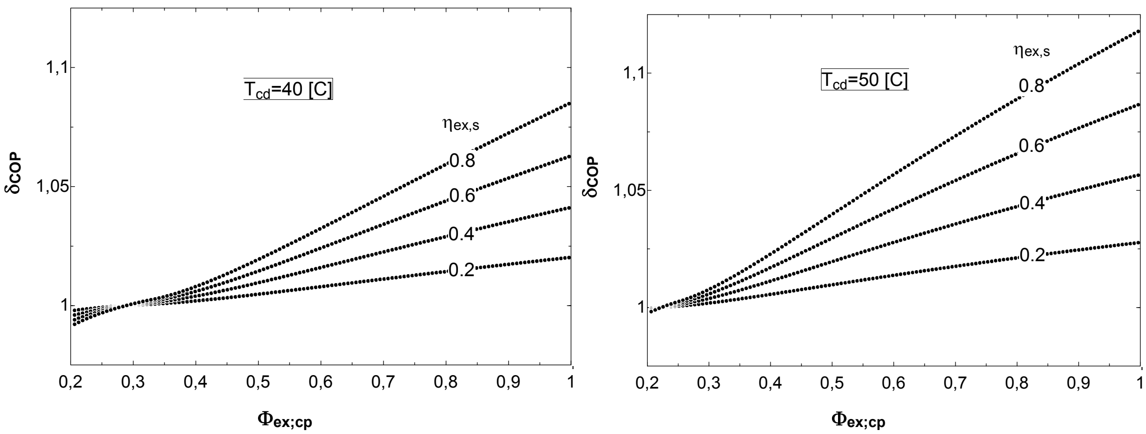

Figure 2 – Improvement on cycle COP as a function of expander pressure ratio and expander isentropic efficiency, for condensing temperatures of 40°C and 50°C (R-134a).

NOMENCLATURE

				Greek Symbols		
COP	coefficient of performance	(-)				
h	specific enthalpy	(kJ/kg)		γ	specific heat ratio	(-)
P	pressure	(kPa)		ΔT_{sc}	degree of subcooling	(°C)
q_{ev}	refrigerating effect	(kJ/kg)		ΔT_{sh}	degree of superheating	(°C)
T	temperature	(°C)		δ_{COP}	COP improvement ratio	(-)
T_{cd}	condensing temperature	(°C)		η_m	mechanical efficiency	(-)
T_{ev}	evaporating temperature	(°C)		η_s	isentropic efficiency	(-)
w_{cp}	work of compression	(kJ/kg)		η_v	volumetric efficiency	(-)
$w_{cp,s}$	isentropic work of compression	(kJ/kg)		θ	pressure ratio	(-)
w_{ex}	work of expansion	(kJ/kg)		$\Phi_{ex,cp}$	θ_{ex} to θ_{cp} ratio	(-)
$w_{ex,s}$	isentropic work of expansion	(kJ/kg)				

Subscripts

1	compressor inlet	5	evaporator inlet
2	compressor discharge	cd	condenser
2s	discharge at isentropic compression	cp	compressor
3	condenser outlet	ev	evaporator
4	expander discharge	ex	expander
4s	discharge at isentropic expansion	xv	expansion valve

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