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COP Improvement of a CO_2 Refrigeration System with an Expander-Compressor-Unit (ECU) in Subcritical and Transcritical Operation

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ABSTRACT

One of the most attractive low GWP and zero ODP refrigerants is carbon dioxide (R744). The use of CO_2 in low temperature stages of cascade systems is nowadays state of the art. If it is used in refrigeration systems with higher heat sink temperatures, e.g. supermarket systems, CO_2 has a disadvantage in terms of energy efficiency compared to other refrigerants.

One possibility to improve the system COP is the introduction of a work extracting expander linked with a compressor for a second compression stage. Such an option, an expander-compressor-unit, provides two advantages: the system COP is improved and the discharge pressure and temperature of the main compressor become reduced. Until now several expanders and expander-compressor-units have been proposed.

In this paper experimental results from several operation modes with an expander-compressor-unit (ECU) in supermarket systems will be compared with a standard throttle process.

1. INTRODUCTION

The TEWI value is one of the most popular values to compare different refrigerants and plants in respect to their environmental impact. The coolant has direct and indirect influence on this value. The direct influence depends mainly on the refrigerant itself because it is a function of the specific GWP. In most cases natural refrigerants have a low GWP e.g. CO_2 (1), NH_3 (0) in comparison to e.g. R134a (1300) but a disadvantage in energy consumption (indirect TEWI effect) or they are flammable, toxic, etc. Carbon dioxide is a common refrigerant in commercial systems. With high heat sink temperatures high pressure differences between the evaporator and condenser/gas cooler pressure occur.

A possibility to improve the system COP is to replace isenthalpic expansion in the high pressure valve with an isentropic expansion in an expander and use the extracted work in a second compression stage. In our laboratory a combined expander-compressor-unit (ECU) which is capable to expand in the two phase region has been developed and tested. Different publications about expanders and other work extracting machines presented efficiencies of their units in transcritical operation. It is known that the highest COP improvement will be archived in this operation mode. In commercial refrigeration systems like supermarkets, which use air as heat sink, different condensing/gas cooler conditions occurs during the year. That means:

- The pressure difference are decreased compared to the design point
- If the extracted work is used in the process e.g. as a second compression stage, the compressor part must be controlled
- The expander has to be controlled to maintain the high pressure, as the expansion valve will be replaced

However, refrigeration systems needs a good design and a proper control for new components like an expander, ejector etc. to achieve high COP increase during the majority of the operation conditions. In this paper measurement data from operation with and without an Expander-Compressor-Unit are presented. These two operation modes allow a comparison in terms of COP between ECU and standard process. The focus is laid on an improved control strategy for the ECU and the COP improvements.

2. COMMERCIAL REFRIGERATION SYSTEMS WITH INTEGRATED ECU

In this section a short overview of the actual systems with ECU will be given for an impression how the unit is integrate and why the unit must be controlled. Two commercial refrigeration plants with CO_2-ECU systems will be discussed.

Both plants are designed as transcritical systems and capable to run in the standard or ECU mode. The standard mode means operating with high pressure control valve and a mean pressure (flash gas bypass) control valve. This process is shown in Figure 2 and 3 in dotted lines.

Wenzel and Hesse (2012) described the actual Expander-Compressor-Unit in detail. Consequently here only a short introduction is given. The unit shown in Figure 1 is an improved design of the ECU by Nickl (2007).

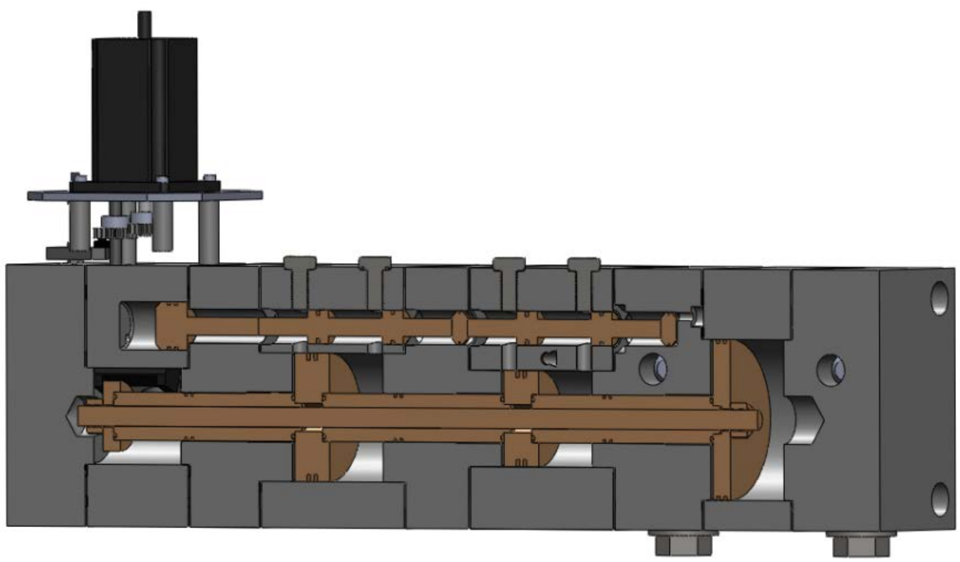

Figure 1: Expander-Compressor-Unit (ECU)

The unit has two expansion stages and one compression stage (from left to right: compressor (C-ECU), 1^{st} expansion stage (E1), 2^{nd} expansion stage (E2)). All pistons are connected to one piston rod and double acting. The first expansion stage (E1) replaces the high pressure control valve and expand liquid form the condenser / gas cooler to a liquid / vapor separator. The second expansion stage (E2) replaces the flash gas control valve and expand the flash gas to the evaporator pressure. This mass flow is guided to an internal heat exchanger (IHX) to supercool the liquid and overheat the flash gas. The small piston bound (main slider; see Figure. 1) controls the inlet and outlet valves of the expansion stages.

This parallel integration to the standard control valves (see Figure 2 and 3) allows operating with and without ECU and comparing them. The extracting work is transferred to the compressor stage. In front of this stage an intermediate cooler is installed (see Figure 2 and 3). This cooler has the same heat sink temperature as the condenser / gas cooler. On top of the unit a stepper motor also can be seen in Figure 1. The functionality will be explained in section 3.1.

2.1 Medium temperature-refrigeration unit

Gerber and Wenzel (2011) described a Swiss cash and carry market with installed ECU. In this plant the unit has been installed in the first field test. A simplified flow sheet is given in Figure 2.

This MT-refrigeration rack has an evaporating temperature of about - 8 °C. Each of the installed four single stage compressors is able to compress directly to gas cooler pressure or to an intermediate pressure, which would be the ECU's compressor part suction pressure. Two of the four compressors are frequency controlled.

For reasons of operational security in this system a variety of additional security features and various automated valves has been installed. With the automatic valves A…D and E0…E4 (see Figure. 2) it is possible to switch between standard and ECU mode automatically. One additional oil separator was installed to switch between two different control modes for the ECU.

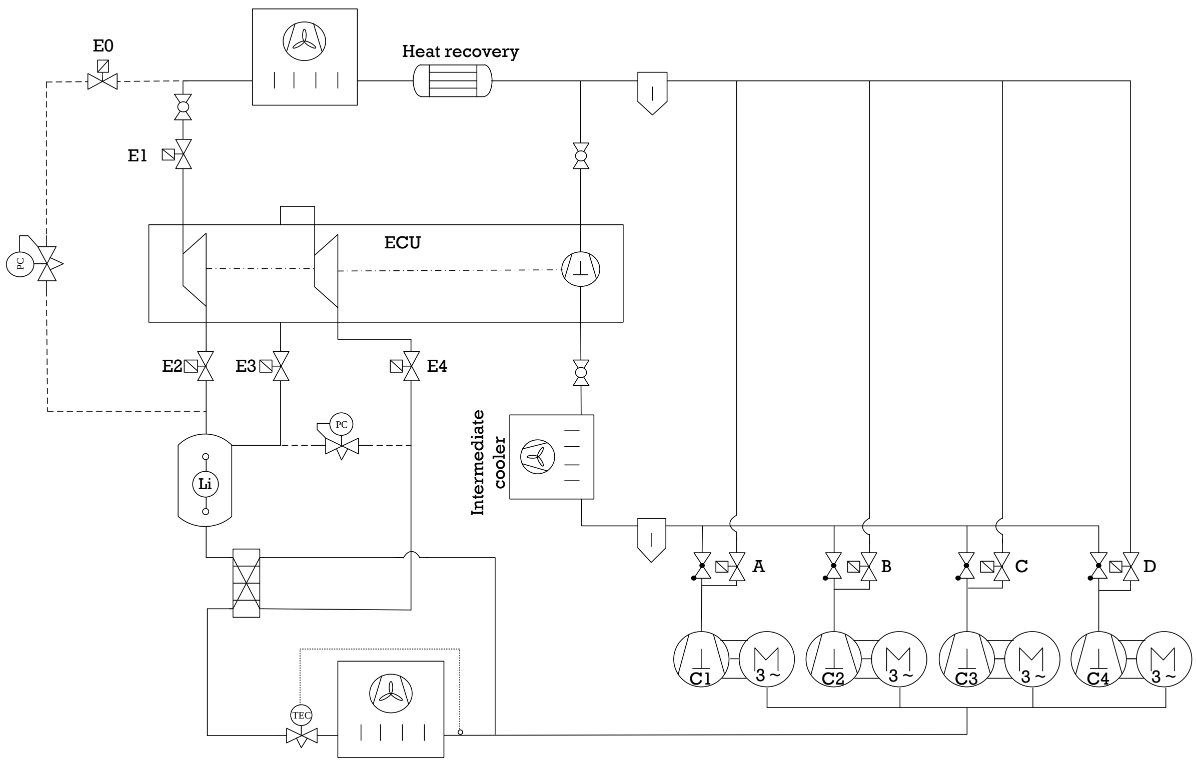

Figure 2: Field test plant; simplified flow sheet

2.2 Booster-refrigeration unit

The plant shown in Figure 3 is an enhanced system from the plant described by Wenzel and Hesse (2012) which was built as a test and demonstration plant with a connected demonstration market.

The demonstration market offers the possibility to run under real conditions. It has two evaporating temperatures: medium temperature (MT) with t_0 = - 10 °C and low temperature (LT) with t_0 = -35 °C. To investigate the behavior with a heat pump and to generate additional heat load a heat pump (HP) evaporator has been installed. This evaporator is capable to be used as MT or LT heat exchanger with ambient air. A second internal heat exchanger (IHX (2)) has been installed too. Since this heat exchanger was installed first the expanded flash gas will be overheated and therefore the liquid from the separator will be subcooled (see also section 4.2). Similar to the MT-plant the compressor rack has two frequency controlled MT compressors (C1 and C2).

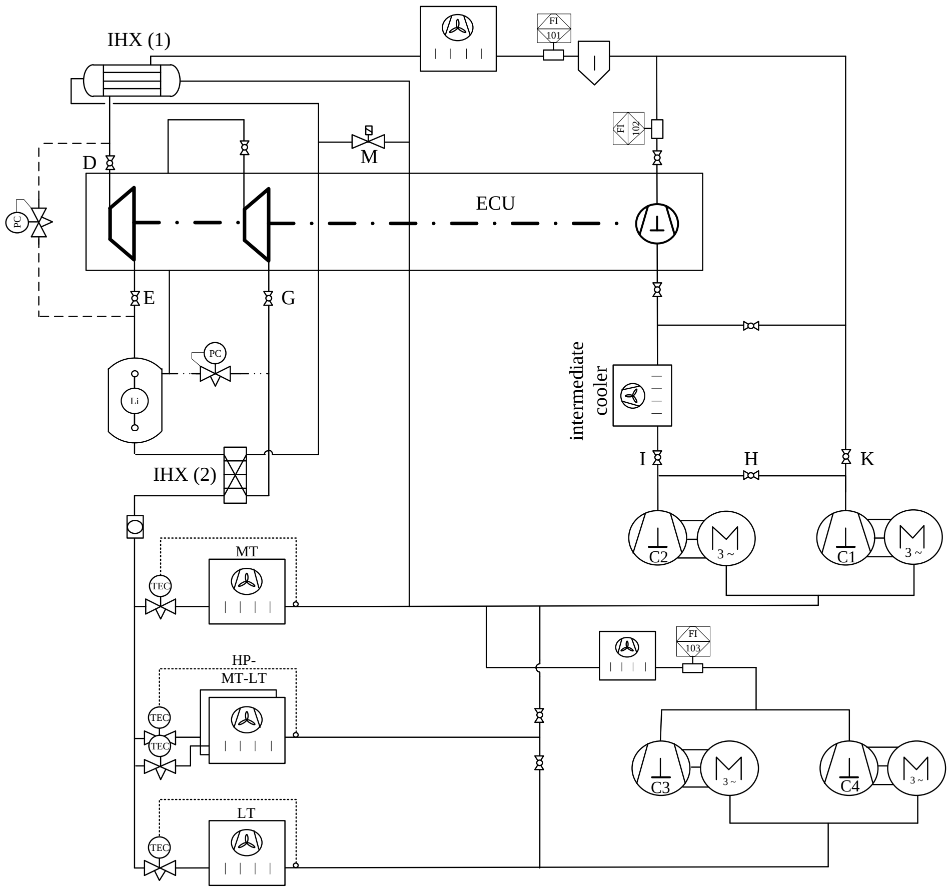

Figure 3: Booster test rig; simplified flow sheet

3. CONTROL STRATEGY

In systems where different temperature conditions and different cooling loads can occur, independent of the design, the expander and compressor must be controlled to obtain high COP's. The integration of the expansion and compression unit into the system is also important.

Heyl (1999) and Nickl (2007) described how the unit could be integrated and which COP improvement can be achieved at high heat sink temperatures and high pressure difference in a normal refrigeration circuit (evaporating, compression, condensing/gas cooling and expansion). In this section the improved control strategy is presented for processes according to Figure. 2 and 3 under operation conditions differed from design conditions.

3.1 High pressure control with integrated ECU

Cheg and Gu (2005) describe the high pressure control in a (transcritical) system that uses CO_2 as refrigerant and the high influence on the COP. In the presented plants, the high pressure will be controlled using a high pressure control valve in subcritical operation too. This is done by an industrial valve in the standard process. It uses a stepper motor with a 0 – 10 V input signal from the main control system.

Is the valve replaced by an expansion stage, the high pressure must be controlled with the expander. In the first step a stepper motor with a control unit was integrated. With this control unit (see Figure. 4) the ECU is capable to adjust the high pressure using the same input signal as the high pressure control valve through the needle valve, adjusts the delay of the main slider. The main slider controls the frequency of the piston compound and the mass flow through the unit. The desired high pressure is calculated from a linear equation in the main control unit.

Wenzel at al. (2010) describe in the first experimental experiences that they used the same controller which was controlling the high pressure valve. In the presented results in section 4 an additional controller was implemented to allow different control parameter for the standard and ECU control.

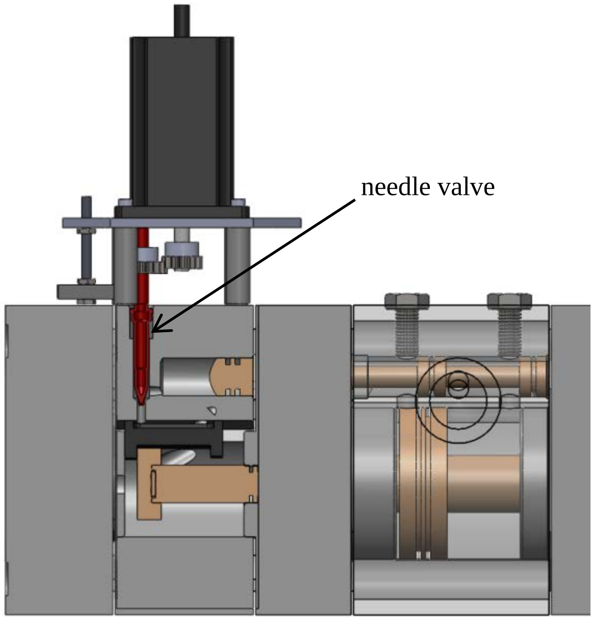

Figure 4: needle valve with stepper motor

3.2 ECU capacity control

The total mass flow enters the first expansion stage. From this flow a rate of approx. 5 % will be used to control the main slider (see Nickl 2007). The frequency of the whole unit (Expander and compressor) depends on the following points:

- Dimensions of the 1st expansion stage
- Inlet density
- High pressure control

Equation 1 shows the calculation of the mass flow through the compressor part.

$$\dot{m}_{C\text{-}ECU} = (1+\varphi)\ f\ A_{Pi}\ s\ \lambda\ \varrho_{s,C\text{-}ECU} \tag{1}$$

Using the piston rod ratio:

$$\varphi = \left(1+\frac{d_{Pir}}{d_{Pi}}\right) \tag{2}$$

Similar to other volumetric compressors the mass flow depends on the suction density too. The heat sink for the intermediate cooler (see Figure 2 and 3) is ambient air with different temperatures over the year. The intermediate pressure depends on the power equilibrium between the expander and compressor, which is shown in Equation (3).

$$\sum_{i=1}^{2} P_{Ei}\ \eta_{m,Ei} = \frac{P_{C\text{-}ECU}}{\eta_{m,C\text{-}ECU}} \tag{3}$$

With the chosen implementation in section 2 the mass flow can be adjust with two frequency controlled main compressors. Therefore a controller called frequency manipulator was developed. The set point for this controller is the superheat at the inlet of the ECU's compressor part.

4. EXPERIMENTAL RESULTS

In this section experimental data and a comparison between the standard and ECU operation will be presented. The COP increase is given by Equation (4).

$$\Delta\,COP = \frac{(COP_{ECU} - COP_{Standard})}{COP_{Standard}}\ 100\ \% \tag{4}$$

With the COP for the Standard (throttle process) and ECU operation is given by Equation (5):

$$COP = \frac{\left(\dot{Q}_{0,MT} + \dot{Q}_{0,LT}\right)}{(P_{LT} + P_{MT})} \tag{5}$$

The following diagrams are results from real measured data. Therefore in different sections in the Booster test rig pressures and therefore pressure losses were measured e.g. in the section between: MT-compressors out, Coriolis mass flow meter, oil separator, gas cooler and IHX out; between LT-compressor out, Coriolis mass flow meter, gas cooler and suction port MT compressor etc. (see Figure. 3). These pressure losses are the reason for non-horizontal lines in the P-h diagrams. The isentropic efficiency of expansion stages was assumed to be 0.8. Process points in the P-h diagrams were calculated with REFPROP 8.0 (Lemmon at. al. 2007).

4.1 MT-refrigeration unit

Gerber and Wenzel (2011) presented first transcritical test results from the MT-refrigeration system. In this operation, the Expander-Compressor-Unit capacity (see section 3.2) was manually controlled. In Figure 5 results from a transcritical operation with completely automatically controlled ECU (both high pressure and capacity control) are presented.

In terms of installation costs the plant has been equipped only with the necessary sensors (p and t) for operation. Therefore no pressure losses in the P-h diagram could be determined.

The high pressure was above the optimum calculated by the controller and can therefore be reduced. The pressure difference was determined to about 4.7 bar. That means the COP could be increased by about 9.9 %. This is a calculated value because the plant is not equipped with power meters and in comparison to the Booster plant the refrigeration system is a plant under 24 h operation in the field and could not be stopped. The comparison based on the assumption that the subcooling in the internal heat exchanger in comparison to the ECU process is not significant in the standard (throttle) process. This is confirmed by results from the booster plant.

To calculate the mass flow and outlet temperatures polynomial function for the isentropic efficiencies and linear equations for the volumetric efficiency of the two frequency controlled compressors and the two other compressors with constant speed have been provided by the manufacturer. These equations allow to calculate the mass flow through and around (dashed line) the compressor part. The outlet temperature of the two main compressors was reduced by about 6 K.

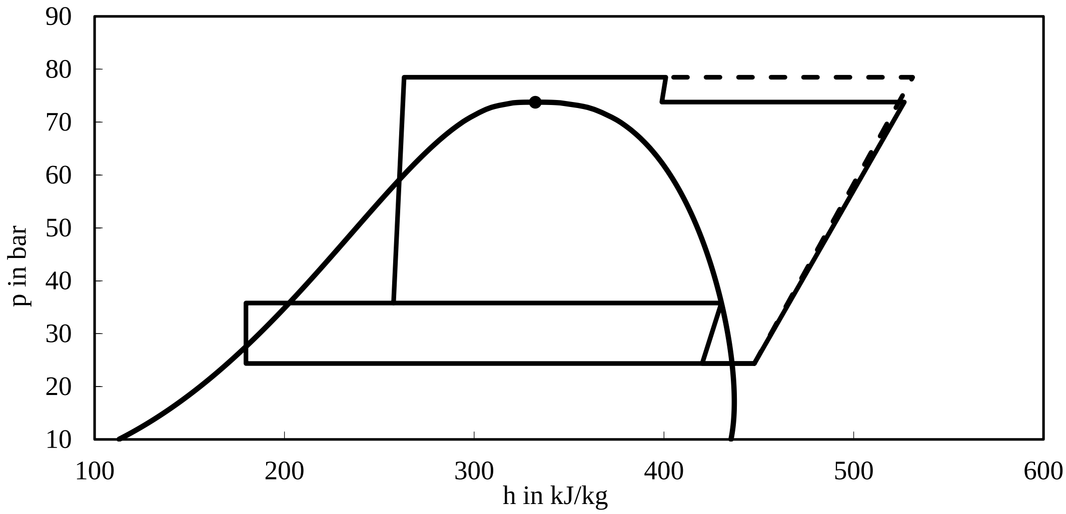

Figure 5: MT plant with ECU operation

4.2 Booster-refrigeration unit

In the next two figures test results from the operation without and with expander-compressor-unit are shown. Both tests were realized with the same process conditions like e. g. condensing temperature, high pressure. Figure 6 shows the standard process

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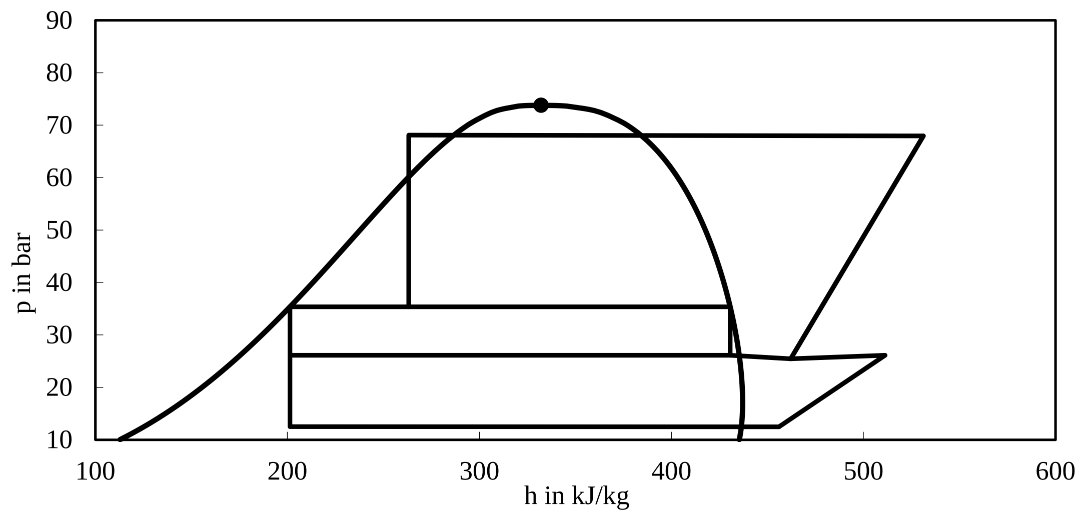

Figure 6: P-h diagram; subcritical operation - throttle process

Wenzel and Hesse (2012) show results only with one internal heat exchanger IHX (1) (see Figure 3). Here a second internal heat exchanger (IHX (2)) is used to subcooled the liquid from the separator and overheat the flash gas.

In the process without ECU the intermediate cooler is used in line with the gas cooler. This enables a smaller design for the gas cooler.

This plant is equipped with power meters one for the two LT compressors and one for each main compressor (C1 and C2). The total mass flow, the mass flows in the LT-stage and through the ECU's compressor part can be measured with Coriolis mass flow meters. The COP could be determined to 2.48.

Figure 7 shows the process with expander-compressor-unit. In this test the ECU was automatically controlled by the frequency manipulator (capacity control) and the unit controlled the high pressure with the stepper motor described in section 3.

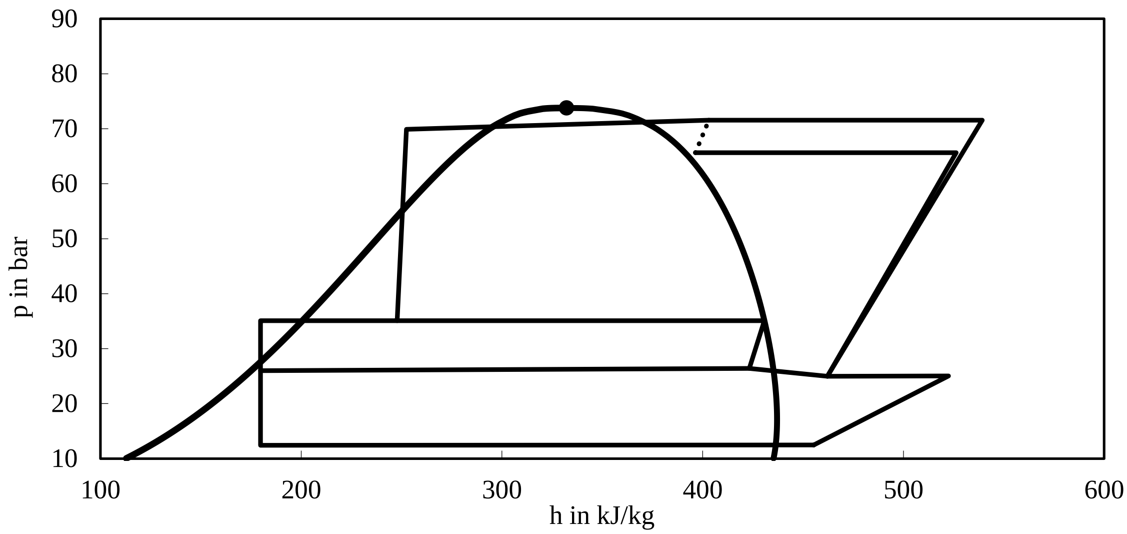

Figure 7: P-h diagram; subcritical operation with ECU

In Figure 7 the significant liquid subcooling in the ECU process assumed in section 4.1 is shown. The reason is the control mass flow for the in and outlet valves of the expansion stages. A detailed explanation is given by Wenzel and Hesse (2012). The outlet temperature from one main compressor was reduced by about 14 K and the discharge pressure from compressor C2 (see dotted line in Figure. 7) by about 6 bar. The mass flow through the ECU compressor part and was determined to about 58 % of total mass flow. The relatively low COP increase of about 9.3 % ($COP_{ECU} = 2.71$) refers to a reduced total mass flow and therefore a low speed of compressor C2. The minimal speed of C2 results in a low isentropic efficiency.

6. CONCLUSIONS

The use of CO_2 as refrigerant has increased during the last few years. The reason for this is the environmental friendliness. In comparison to other refrigerants CO_2 has a disadvantage in terms of energy efficiency at high heat sink temperatures. According to prior publications e.g. Gerber and Wenzel (2011) or Wenzel and Nickl (2009) further test results from two refrigeration systems with integrated expander-compressor-unit were presented in this publication. In a Swiss cash & carry market the ECU has been integrated in the first field test worldwide (see Gerber, R.; Wenzel, M. (2011)). The second system is a transcritical booster system with a demonstration market.
For both plants the results showed that the COP could be increased with such a unit in different process conditions. This paper shows that in the field test, the unit can be controlled completely and automatically by using the new frequency manipulator. The COP could be increased by 9.9 % and the discharge temperature could be reduced by 6 K as well as the discharge pressure of the main compressor by about 4.7 bar.

With respect to the further test in the booster plant the presented results shows the new arrangement for the necessary internal heat exchanger. In this test the COP could be increased by 9.3 %. This relatively low increase is due to the low total mass flow and therefor the very low speed of compressor C2. The discharge temperature could be reduced by 14 K and the discharge pressure by about 6 bar.

NOMENCLATURE

A	area	(m^2)
COP	coefficient of performance	(-)
ECU	expander-compressor-unit	(-)
E1	first expansion stage	(-)
E2	second expansion stage	(-)
Fi	flow indicator	(-)
f	frequency	(1/s)
GWP	global warming potential	(-)
h	specific enthalpy	(kJ/kg)
$\dot{m}$	mass flow	(kg/s)
PC	pressure control	(-)
P	power	kW
p	pressure	(bar)
s	stroke	(m)
TEWI	total equivalent warming impact	(CO_2 equivalent)
t	temperature	(°C)
HP	heat pump	(-)
IHX	internal heat exchanger	(-)
LT	low temp. (deep freezing)	(-)
MT	medium temp. (normal freezing)	(-)
ODP	ozone depleting potential	(-)
$\dot{Q}$	Heat flow	kW

Subscripts

0	Evaporating
C-ECU	ECU's compressor part
m	mechanical
Pi	piston
Pir	piston rod
s	suction

Greek symbols

η	efficiency	(-)
λ	volumetric efficiency	(-)
ϱ	Density	(kg/m^3)

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