Amelioration of Energy Efficiency for Refrigeration Cycles by Means of Ejectors

Ahrendts Fabian[1(✉)], Thoma Werner[2], and Köhler Jürgen[1]

[1] Institut für Thermodynamik, TU Braunschweig, Brunswick, Germany
fabian.ahrendts@tu-bs.de
[2] Daimler AG, Sindelfingen, Germany

Abstract. In this paper is explained how the efficiency of refrigeration cycles and in particular those with carbon dioxide (CO_2, R744) as refrigerant can be augmented by use of an ejector. Special emphasis is put on the exposure of CO_2 as a safe and environmental friendly working fluid – making it a promising candidate to replace today's standard refrigerant in the automotive field tetrafluoroethane (R134a), which is about to be banned within the European Union for mobile applications.

The working principle of ejectors as well as their operational behaviour is described. Furthermore an approach to quantify ejector efficiency is presented. Different classes of refrigeration cycles with ejectors are introduced and an automotive application example from an ongoing research project is described in detail: It is shown how an ejector can be fruitfully put into place in charge air cooling.

Keywords: CO_2 · R744 · Ejector · Refrigeration cycle · Charge air cooling

1 Introduction

Refrigeration systems are mayor contributors to both direct and indirect greenhouse gas emissions. Direct emissions are linked to leakage whereas indirect emission arise from the provision of driving energy for the system. In order to lower the greenhouse gas emissions both emission types have to be taken into account.

Direct emissions can be reduced by minimising leakage or by using refrigerants with a low GWP (Global Warming Potential). In this regard the natural refrigerant CO_2 is a good choice. It has a low GWP of one and is furthermore nontoxic as well as non-flammable in contrast to other natural refrigerants like ammonia or propane. Its rather high evaporation enthalpy makes it an interesting choice for mobile applications. Pipe cross-sectional areas can be chosen smaller in comparison to cycles with refrigerants with higher evaporation enthalpy. One mayor shortcoming of CO_2 systems is however the low exergetic efficiency at high ambient temperatures due to higher expansion losses in the throttling valve of conventional refrigeration cycles in comparison to other refrigerants. Lucas (2015) has shown that the specific expansion losses with respect to the compression work for a conventional refrigeration cycle with CO_2 as working fluid can reach 35% in dependence of the inlet temperature of the expansion valve. Those losses are about 10% lower for an R134a cycle, while the other specific

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cycle losses are more or less of the same order. Hence a reduction of the expansion losses is desirable and key to the competitiveness of CO_2 refrigeration systems. The expansion losses can be recovered and thus the system efficiency ameliorated by means of ejectors. A refrigeration cycle with an ejector is shown in Fig. 1. The ejector has a high and a low pressure inlet for the working fluid which is mixed and then decelerated in order to achieve a pressure rise at the outlet and to relief the compressor as a consequence.

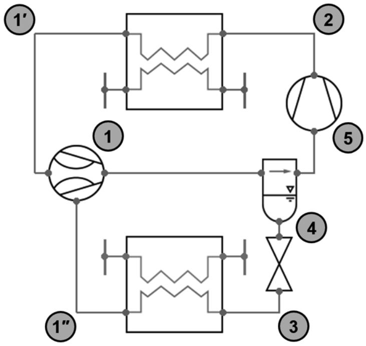

Fig. 1. Refrigeration cycle with ejector

The potential benefit of the application of ejectors in refrigeration systems has been subject to extensive research. Major improvements of the system efficiency which is characterised by the COP (Coefficient of Performance) were shown numerically among others by Fiorenzano (2011) and Jeong et al. (2004). Elbel et al. (2012) derived numerically form experimental data a COP increase of up to 7% compared to an expansion valve refrigeration system including an Internal Heat Exchanger (IHX). Lucas and Koehler (2012) were able to demonstrate experimentally a COP increases of up to 17% for an ejector refrigeration cycle compared to an expansion valve refrigeration cycle.

2 Ejector Devices

Before reviewing ejector devices it is important to notice that ejectors were initially not developed for pressure recuperation in refrigeration cycles. The first ejector invented and patented by Henry Giffard was intended to pump liquid water to the reservoir of steam engine boilers. This was achieved by condensing the high pressure steam and using the created vacuum to suck water. Other applications described by different authors were comprehensively collated by Elbel (2009) among them a gas-liquid reactor to mix two different fluid streams reported by Elgozali et al. (2002).

As a result one finds many alternative expressions for the term ejector such the term injector, eductor, diffusion pump, aspirator, or jet pump depending on the application. If one restricts oneself to single fluid ejectors, they can be classified according to a proposal of Elbel (2007) like in Table 1.

Table 1. Classification of ejectors according to Elbel (2007)

Type	Driving flow	Driven flow	Exit flow	Remarks
Vapor jet	Vapor	Vapor	Vapor	Two-phase flow can occur, shock waves possible
Liquid jet	Liquid	Liquid	Liquid	Single-phase flow without shock waves
Condensing	Vapor	Liquid	Liquid	Two-phase flow with condensation of driving vapor, strong shock waves
Two-phase	Liquid	Vapor	Two-phase	Two-phase flow can occur, shock waves possible

The vapour jet ejector and the two-phase ejector can be encountered in refrigeration cycles. The first ejector for refrigeration purposes was according to Elbel (2009) a vapour jet ejector proposed by Leblanc in 1910. Because of the vast availability of steam at the time the so-called steam jet refrigeration systems spread widely for the climatisation of large buildings and railroad cars. The first two-phase ejector was introduced by Gay (1931) in order to achieve indeed the aforementioned reduction of throttling losses.

The idea of using CO_2 as refrigerant goes as way back as the ejector itself as Elbel (2007) elaborates. The use of CO_2 as refrigerant was first patented by Alexander Twining in 1850. Its application in refrigeration systems for ice production was patented by Thaddeus Lowe in 1867. Refrigeration with CO_2 remained of importance in ice production, beer breweries, and in cargo ship refrigeration until the beginning of the 20th century. The refrigerant CO_2 was then outed by synthetic refrigerants such as R134a and only rediscovered in the late 1980s in the context of transcritical air-conditioning systems including ejectors along with the emerge of environmental awareness.

2.1 Working Principle

The basic four components of an ejector are equal for all ejectors types given in Table 1: They consists of a motive nozzle and a suction nozzle, a mixing chamber and a diffuser as shown in Fig. 2.

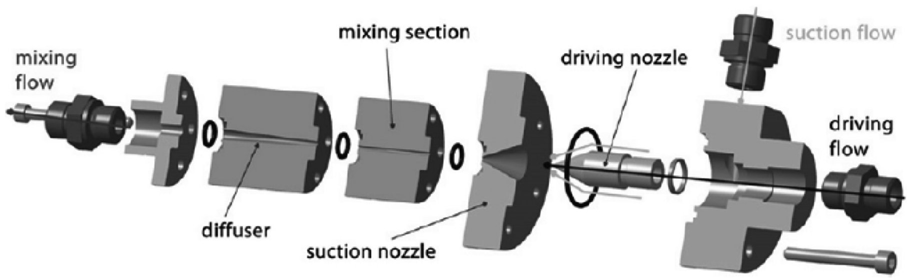

Fig. 2. Ejector components, Tischendorf et al. (2010)

In the motive nozzle the pressure energy of the driving flow is converted into kinetic energy. In case of gaseous or vaporous fluid at the inlet the motive nozzle is often realized as a converging-diverging nozzle allowing supersonic discharge velocities. As far as two-phase ejector are concerned the phase change of the primary flow inside the nozzle might be delayed due to thermodynamic and hydrodynamic non-equilibrium effects leading to flash vaporisation downstream.

The sucked fluid is accelerated and directed inside the suction nozzle which both contributes to the reduction of hydrodynamic losses. Large velocity differences between the driving and the driven flow cause shearing losses whereas a too steep angle of the sucked flow with respect to the driving flow are connected with losses due to deflexion and areas of recirculation in the mixing chamber.

When entering the mixing chamber the driving flow post-expands and entrains the sucked flow by transferring momentum. The mixing chamber can either be designed with constant cross-sectional area or conical geometry in order to achieve mixing at constant pressure. The expansion of the driving flow can be connected with the creation of a fluidic throat in which the sucked flow is further accelerated to sonic velocity. Within the mixing chamber one can usually observe shock waves that lead to a significant first pressure rise and the deceleration of the flow to subsonic velocity.

A further deceleration and pressure rise respectively is achieved by means of the subsonic diffuser. The pressure at the outlet of the diffuser lies in general in between those of the driving and the driven flow. Though there are designs and operating conditions under which the pressure at the outlet rises above the pressure of both inlets.

2.2 Operational Behaviour and Efficiency

The operational behaviour of an ejector is mainly characterized by the recuperated pressure Δp_{Rec} or the suction pressure ratio Π and the mass entrainment ratio ϕ. The recuperated pressure is the difference between the pressure at the diffuser exit and the pressure of the suction flow entering the ejector

$$\Delta p_{Rec} = p_e - p_s, \tag{1}$$

whereas the suction pressure ratio is defined as the pressure ratio at the diffuser exit pressure to the pressure of the suction flow entering the ejector

$$\Pi = \frac{p_e}{p_s} = \frac{\Delta p_{Rec}}{p_s} + 1. \tag{2}$$

As is derived in Eq. (2) both expressions (1) and (2) contain the same information and can therefore be exchanged. The recuperated pressure is dependent on the entrainment ratio which is defined as the ratio between the entrained and the driving mass flow

$$\phi = \frac{\dot{m}_s}{\dot{m}_d}. \tag{3}$$

Augmenting the driving mass flow results in lower entrainment but better pressure recuperation, while diminishing it results in low pressure change. This dependency has a linear characteristic over a wide range of operating conditions and especially around the point of maximum efficiency as has been show by Lucas (2015).

The efficiency value of an ejector is subject to its definition. In the following the definition of ejector efficiency by Köhler et al. (2007) is presented. Ebel and Harnjak (2008) presented later a similar definition which was derived in a different way.

The working principle of an ejector can be described by an equivalent model of a turbine driving a compressor as shown in Fig. 3. The maximum power in order to drive the compressor can be extracted from the driving mass flow by an isentropic expansion to the diffuser exit pressure in the turbine. The highest possible pressure rise in the compressor is achieved via an isentropic compression of the sucked mass flow to the diffuser exit pressure.

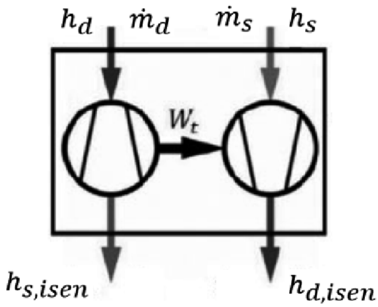

Fig. 3. Equivalent model of an ejector

The ejector efficiency has been defined by Köhler as the ratio between recuperated power and maximum recuperated power which corresponds to the product of the entrainment ratio and the ratio of the enthalpy differences of the isentropic compression and expansion

$$\eta = \frac{\dot{W}_{Rec}}{\dot{W}_{Rec,max}} = \phi \cdot \frac{h_{s,isen} - h_s}{h_d - h_{d,isen}}. \tag{4}$$

This definition has the advantage in comparison to others that it is determined with external parameters and that it is independent of the working fluid or the ambient condition. According to this definition Lucas (2015) has reported experimentally measured ejector efficiencies of up to 35%.

3 Ejector Cycles

The goal of the application of the ejector is to improve the overall efficiency of the refrigeration cycle it is integrated in. It is important to understand that the efficiency of the ejector component is only one factor among others that determine the COP. The intrinsic efficiencies of the cycle components play a role as well as their interaction for example the one of the ejector and the separator which splits liquid and gas. Due to the mass conservation of the two phases there is a dependency between the steam content at the separator inlet and the entrainment ratio. Within the cycle it is no longer a free variable. Therefore it makes sense to characterise ejectors not only at component but also at system level.

3.1 Classification of Ejector Cycles

According to a proposal by Bergander (2015) three basic types of refrigeration cycles with ejectors can be distinguished in dependence on the pressure recuperation with respect to the compressor. Typical vapour compression cycles consist of a gascooler, an expansion valve, an evaporator and a compressor. The supply of the compressor with electrical energy is effort with respect to the cycle balance. This effort can be reduced by means of an ejector as the pressure recuperation relieves the compressor. This relief can either be realized as pre- or post-compression. An ejector can also replace the compressor and take over the entire vapour compression.

In the ideal vapour refrigeration cycle also known as Evans-Perkins- or Plank-process, vaporous refrigerant is fully evaporated, isobarically superheated, isentropically compressed, isobarically cooled, condensed, isobarically supercooled and finally isenthalpically throttled. The throttling is connected with losses as the pressure energy transformed into kinetic energy is dissipated. The ejector cycle shown in Fig. 1 makes use of the device as replacement for the expansion valve and as a pre-compressor. Examples of refrigeration cycles with an ejector as post compressor and an ejector as a replacement for the compressor are shown in Fig. 4.

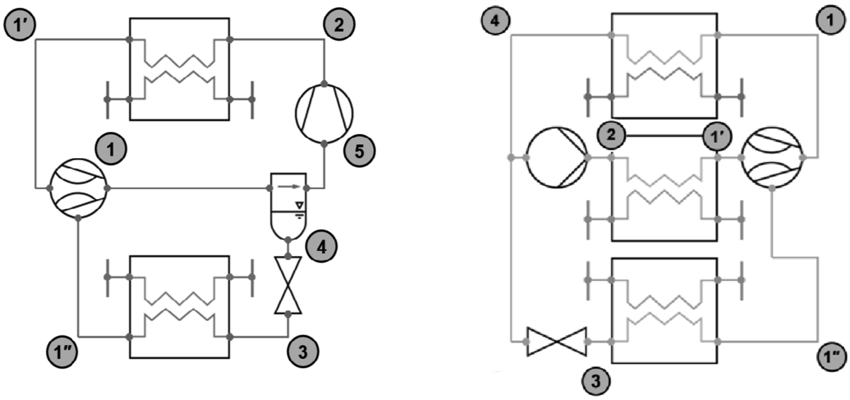

Fig. 4. Ejector as post-compressor and as replacement for the compressor

3.2 Application Example: Charge Air Cooling

A new application of ejectors is charge air cooling for automotive applications. The basic concept for this applications is outlined in Fig. 5. The hot exhaust heats up isobarically the already gaseous working fluid in heat exchanger 1 (HX 1) which is used to drive the ejector. The ejector sucks form the outlet of the charge air cooler also gaseous working fluid. The two flows are mixed inside the ejector. The mixed flow out of the ejector is isobarically cooled in HX 2 and separated. One portion of the flow delivered to the compressor which is connected to HX 1. The other portion of the working the fluid is throttled in the expansion valve (XV) and feeds the charge air cooler. With respect to the classification above the ejector is used as a pre-compressor in the cycle.

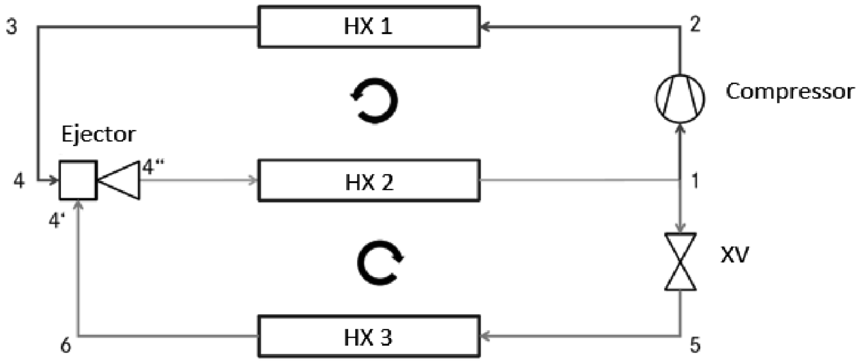

Fig. 5. Basic cycle for charge air cooling with ejector

The application of the cycle in the automotive flied is connected with particularly challenging requirements regarding performance and weight. Therefore the cycle has further been optimized as shown in Fig. 6. Two recuperators (R) have been added. The first recuperator reduces the amount of heat released while the second recuperator cools the working before entering HX 3. It reduces the mass flow necessary to achieve the same cooling effect that would have been realized without the recuperator. The cycle point 6 is defined by the charge air temperature and the dew point of the working fluid. Within this cycle the pressure and temperature levels can be chosen freely apart from cycle point 1 as the ambient is the heat sink. Cycle point 3 is dependent on the heat transfer from the exhaust.

It was possible to show that the target figures of mass flow, the pipe cross-sectional area, the heat release and the compressor power are dependent on four characteristic values. They are dependent on the values for the high and intermediate pressure as well as the temperatures at cycle points 1 and 3. An optimal operating condition was determined by optimizing the object figures with respect to the parameters named. The possible improvements by the application of the recuperators and the described optimization process led to 28% less compression power, 20% less cross-sectional area, 58% less required mass flow, 76% less released heat and 28% COP increase in comparison to the initial cycle design.

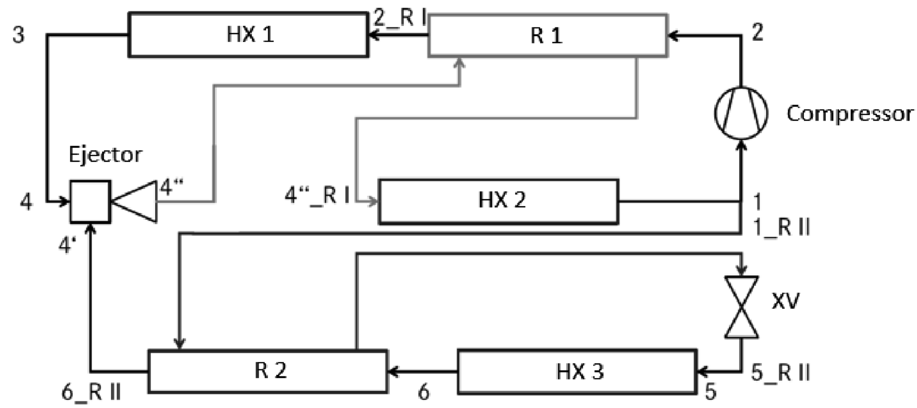

Fig. 6. Cycle for charge air cooling with ejector and recuperators

4 Summary

Due regulation new refrigerants with low GWP will be required. CO_2 is a promising option being both environmentally friendly and safe. Its relative high evaporation enthalpy makes it an interesting option especially for mobile refrigeration systems. Anyhow the low exergetic efficiency at high ambient temperatures is challenging. The component with the highest specific losses in classic refrigeration cycles with CO_2 as working fluid is the expansion valve. The application of ejectors in order to recuperate pressure and to relief the compressor is a promising solution. In dependence on the definition device efficiencies of up 35% have been reported.

A refrigeration cycle with a ejector as a pre-compressor has been presented for charge cooling. It has been shown how the basic design for the cycle can further be improved by means of recuperators and an optimization of some characteristic parameters with respect to the target figures of mass flow, pipe cross-sectional area, heat release and compressor power leading to possible COP improvements of 28% in comparison to the initial design.

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