

A New Control Mechanism for Two-Phase Ejector in Vapor Compression Cycles for Automotive Applications Using Adjustable Motive Nozzle Inlet Swirl

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ABSTRACT

Expansion work recovery by two-phase ejector is known to be beneficial to vapor compression cycle performance. However, one of the biggest challenges with ejector vapor compression cycles is that the ejector cycle performance is sensitive to working condition changes which are common in automotive applications. Different working conditions require different ejector geometries to achieve maximum performance. Slightly different geometries may result in substantially different COPs under the same conditions. The ejector motive nozzle throat diameter (motive nozzle restrictiveness) is one of the key parameters that can significantly affect ejector cycle COP. This paper presents a new two-phase nozzle restrictiveness control mechanism which is possibly applicable to two-phase ejectors used in vapor compression cycles. It utilizes an adjustable swirl at the nozzle inlet to control the nozzle restrictiveness on the two-phase flow without changing the physical dimensions of the nozzle geometry. This new control mechanism has the advantages of being simple and potentially less costly. It can possibly also avoid additional frictional losses of previously proposed ejector control mechanisms using an adjustable needle. An adjustable nozzle based on this new control mechanism was designed and manufactured for experiments with R134a. The experimental results showed that, without changing the nozzle geometry, the nozzle restrictiveness on the two-phase flow can be adjusted over a wide range. Under the same inlet and outlet conditions, the mass flow rate through the nozzle can be reduced by 36% of the full load. This feature could be very useful for the future application of ejectors in automotive systems under changing working conditions.

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INTRODUCTION

Vapor compression cooling cycles deviate from the Carnot refrigeration cycle in several ways, such as isenthalpic expansion of saturated liquid at the condenser outlet and desuperheating of refrigerant vapor at the compressor outlet. Therefore, COPs of vapor compression cooling cycles are always lower than those of a Carnot cycle under the same working conditions. Isenthalpic expansion imposes a two-fold penalty on cycle performance compared with isentropic expansion in the Carnot cycle: the cooling capacity is reduced and the compressor work is increased. Expansion work recovery devices such as ejectors which recover the kinetic energy released during the expansion instead of dissipating it in a throttling process are known to be beneficial to cycle performance. [Figure 1](#) shows the layout and pressure-specific enthalpy diagram of a two-phase ejector cooling cycle first proposed by Gay [1]. In this cycle, high pressure motive flow leaving the condenser enters the ejector through the motive inlet. The motive flow is expanded in the motive nozzle and creates a low pressure zone at the nozzle outlet, which entrains the suction flow from the evaporator. The two streams are mixed in the mixing chamber and kinetic energy is transferred from the motive flow to the suction flow. The mixed fluids leave the

ejector through the diffuser. The fluid velocity is reduced in the diffuser which results in recompression of the mixed fluids by converting velocity energy back into pressure energy. Therefore, the ejector diffuser outlet pressure is higher than the suction flow pressure (that is, the evaporator pressure). The two-phase flow then gets separated in the separator. Saturated vapor enters the compressor while saturated liquid gets throttled and is fed into the evaporator via a metering valve. That way, kinetic energy released during expansion is utilized to compress the fluid from the evaporator. As a result, some compressor work is saved while the cooling capacity is increased if the heat rejection capacity remains constant.

Disawas and Wongwises [2] proposed that, in addition to serving as an expansion device, the ejector can also act as a refrigerant pump for the low-pressure side of the system. The evaporator is therefore flooded with refrigerant and operates as in a liquid recirculation system. Their experimental results showed that the COP of the two-phase ejector refrigeration cycle using R134a was higher than that of the baseline cycle using expansion valve over the whole range of experimental conditions. The maximum improvement achieved was about 13% at low heat sink and heat source

temperatures. Liquid recirculation can improve evaporator performance by sending more liquid to the evaporator than is actually evaporated so that dryout in the evaporator can be reduced. It can also improve refrigerant distribution for evaporators with inlet headers by feeding only single-phase liquid to the inlet headers instead of two-phase refrigerant which often results in non-homogeneous distribution of two-phase flow into the parallel channels. Therefore, liquid recirculation can result in higher evaporation pressure, and higher system COP compared to a direct expansion cycle (Lawrence and Elbel [3]).

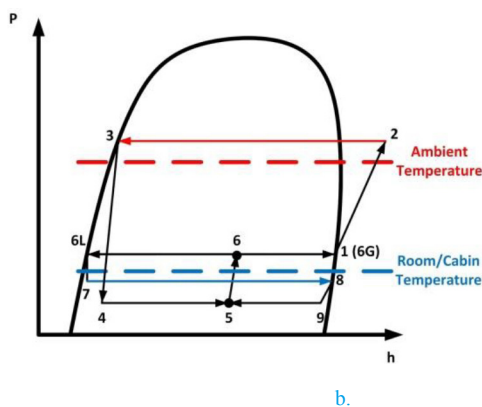
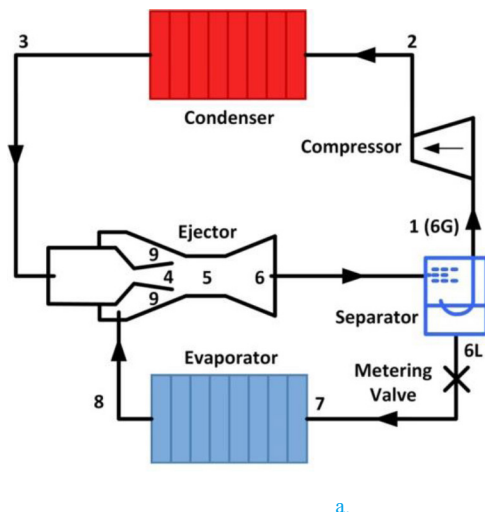


Figure 1. (a) Layout and (b) pressure-specific enthalpy diagram of the two-phase ejector cycle as proposed by Gay [1].

Many research efforts have been devoted to R744 transcritical ejector cycles. R744 transcritical cycles usually have larger expansion losses caused by throttling process than subcritical cycles under common working conditions. It is very beneficial to apply ejector to R744 transcritical cycles due to the large recovery potential.

Ozaki *et al.* [4] carried out an experiment on an automotive transcritical R744 air conditioning system using an ejector to improve system COP. The experiment showed COP improvements of 20% over a baseline cycle using a conventional expansion valve.

Banasiak *et al.* [5] reported a maximum increase in COP of 8% over a baseline cycle with a conventional expansion valve.

Elbel and Hrnjak [6] and Elbel [7] experimentally investigated a transcritical R744 system using a refrigerant ejector. They reported that for the test conditions considered the cooling capacity and COP can be simultaneously improved by up to 8% and 7%, respectively. Extrapolation was used to determine that the COP could have been improved by as much as 18% at matched cooling capacities.

Less attention has been given to low-pressure working fluids in the literature for ejector cooling cycles compared with R744 due to their lower work recovery potential. However, ejector cooling cycles using low-pressure refrigerants, such as R134a or R1234yf, can still have noticeable performance improvements.

Early investigation of a two-phase ejector cycle using R134a by Harrell and Kornhauser [8] predicted a cooling COP improvement of approximately 23% for a typical refrigerating cycle and an ideal ejector. An improvement of 12% could be achieved if the ejector performed as well as typical single-phase ejectors. Ejector performance achieved from later ejector tests corresponded to refrigeration cycle COP improvements ranging from 3.9% to 7.6%.

Lawrence and Elbel [9] experimentally investigated the performance of an alternate two-phase ejector cycle in which the pressure lift provided by the ejector was utilized in order to provide multiple evaporation temperatures. Low-pressure fluids R134a and R1234yf were used. The ejector cycle showed maximum COP improvements of 12% with R1234yf and 8% with R134a when compared to a two evaporation temperature expansion valve cycle. When compared to a single evaporation temperature expansion valve cycle, the ejector cycle showed maximum COP improvements of 6% with R1234yf and 5% with R134a.

Ejector cycle performance is usually sensitive to working condition changes which are common in automotive systems. Different working conditions require different ejector geometries to achieve maximum performance. Slightly different geometries may result in substantially different COPs under the same conditions. Therefore, it is desirable to introduce an adjustable feature to the ejector so that ejector cycle performance can be optimized under different working conditions, which could make ejector technology more suitable for real world applications (Sumeru *et al.* [10]).

The ejector motive nozzle throat diameter is one of the key dimensions that affect ejector cycle COP. It has a direct impact on motive mass flow rate. Other important ejector geometric parameters that affect ejector efficiency and ejector cycle COP include motive nozzle position, constant area diameter of the mixing chamber and suction chamber converging angle. Additional information can be found in Sarkar [11]. One way to adjust the motive nozzle throat diameter in order to optimize ejector cycle performance according to the working conditions is by using a needle which moves back and forth so that the nozzle throat diameter can be varied, as illustrated in Figure 2.

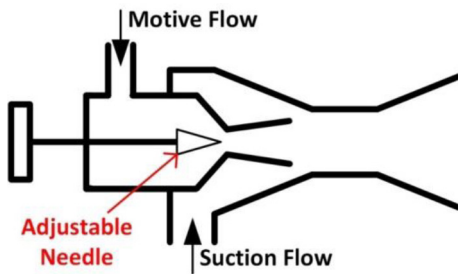


Figure 2. Variable ejector with adjustable needle in the motive nozzle.

Elbel and Hrnjak [6] were the first researchers to publish experimental results of introducing a variable two-phase ejector to a transcritical R744 system by installing a needle in the motive nozzle to control the motive nozzle throat diameter. The needle mechanism allowed control of gas cooler high-side pressure, which is an important task for a transcritical cycle to get optimum performance. However, nozzle and ejector efficiencies were impaired because of the additional frictional losses introduced by the needle. It was found that the benefits of high-side pressure control offset the losses in nozzle and ejector efficiencies.

Hu *et al.* [12] experimentally investigated the R410A ejector cooling cycle under different conditions with four different ejector motive nozzle throat diameters and a variable ejector with adjustable needle in the motive nozzle. It was shown that the ejector motive nozzle throat diameter has a significant impact on cycle performance and different conditions require different motive nozzle throat diameters. Optimal cycle performance achieved by the variable ejector under different conditions was close to the ejector cycle performance with the most suitable ejector motive nozzle throat diameter among the four. However, compared with the baseline cycle using an electronic expansion valve, the COP of the ejector cooling cycle was only increased slightly. In one condition the baseline was even better than the ejector cycle. This may be because the investigated ejector motive nozzle throat diameters did not include the diameter that would have yielded maximum COP and the variable ejector with adjustable needle has lower nozzle and ejector efficiencies because of the additional frictional losses incurred by the needle.

A variable geometry ejector with adjustable needle in the motive nozzle can optimize ejector cycle performance under different conditions, but this design is complicated and costly, and more frictional losses are incurred because of the additional surface area introduced which results in lower nozzle and ejector efficiencies. This provides motivation to develop a new technology to control the motive nozzle restrictiveness.

In this paper, a novel two-phase nozzle restrictiveness control mechanism, which is called swirl control, is presented. This control mechanism is possibly applicable to the control of ejector cooling cycles. It utilizes an adjustable swirl at the nozzle inlet to control the nozzle restrictiveness on the low vapor quality flow expanded in the nozzle without changing the physical dimensions of the nozzle geometry. The ejector and adjustable nozzle which employ this swirl control mechanism are called swirl ejector and swirl nozzle in this paper. At least one control valve is needed for the implementation of this two-phase nozzle restrictiveness control mechanism. This design has the advantages of being simple, possibly less expensive, and can potentially avoid the additional frictional losses in previously

proposed motive nozzle restrictiveness control mechanisms.

Controlling the ejector motive nozzle restrictiveness by adjusting the motive nozzle inlet swirl strength can be very useful for future application of ejector cooling cycles in automotive systems that experience a wide range of operating conditions.

In the following sections, the new ejector design and control mechanism will first be introduced in detail. A hypothesis for the influence of nozzle inlet swirl on the nozzle restrictiveness on the two-phase flow will also be provided. An experimental facility for the investigation of the inlet swirl influence on the nozzle restrictiveness will be described and the experimental results as well as preliminary visualization results will be presented and discussed.

SWIRL EJECTOR AND SWIRL CONTROL

A swirl ejector which employs the swirl control to adjust motive nozzle restrictiveness differs from a conventional ejector in that an adjustable swirl is generated at the ejector motive inlet, as is shown in Figure 3. The motive inlet swirl can be created by injecting part of the motive flow tangentially. After injection the tangential flow will be mixed with the axial motive flow. The total mass flow rate passing through the swirl motive nozzle is equal to the sum of mass flow rates entering through the motive nozzle's axial and tangential flow inlets. The ejector cooling cycle shown in Figure 4 that uses a swirl ejector is almost the same as the conventional ejector cooling cycle of Figure 1. The only difference is that the flow at the condenser outlet of the swirl ejector cooling cycle is separated into two streams. One stream enters the swirl ejector through the motive flow tangential inlet and another enters through the motive flow axial inlet. In such a way, a swirl is created at the ejector motive inlet. The ratio of mass flow rates through the two inlets can be adjusted by a valve installed at the motive flow tangential inlet, thereby changing the swirl strength. The pressure drop across the control valve is usually small. It can be assumed that the thermodynamic state at the motive nozzle inlet after the swirl is introduced (downstream of the tangential inlet valve) is the same as the refrigerant state at the condenser outlet.

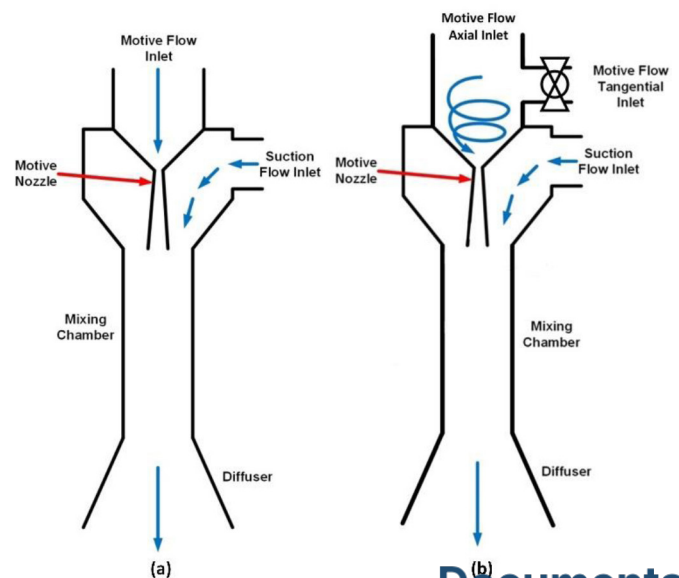


Figure 3. (a) Conventional ejector and (b) swirl ejector.



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