# A New Control Mechanism for Two-Phase Ejector in Vapor Compression Cycles Using Adjustable Motive Nozzle Inlet Vortex

Jingwei Zhu
*ACRC, University of Illinois at Urbana-Champaign*, jzhu50@illinois.edu

Stefan Elbel
elbel@illinois.edu

# A New Control Mechanism for Two-Phase Ejector in Vapor Compression Cycles Using Adjustable Motive Nozzle Inlet Vortex

Jingwei ZHU, Stefan ELBEL*

Department of Mechanical Science and Engineering,
University of Illinois at Urbana-Champaign,
1206 West Green Street, Urbana, IL, 61801, USA
Phone: (217) 244-1531, Fax: (217) 333-1942, Email: elbel@illinois.edu

* Corresponding Author

## 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 cycle is that the ejector cycle performance is sensitive to working condition changes which are common in real world 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. Ejector motive nozzle throat diameter (motive nozzle restrictiveness) is one of the key parameters that can significantly affect COP. This paper presents a new motive nozzle restrictiveness control mechanism for two-phase ejectors used in vapor compression cycles, which has the advantages of being simple, potentially less costly and less vulnerable to clogging. The new control mechanism can possibly avoid the additional frictional losses of previously proposed ejector control mechanisms using adjustable needle. The redesigned ejector utilizes an adjustable vortex at the motive inlet to control the nozzle restrictiveness on the flow expanded in the motive nozzle. 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 ejector in mobile or stationary systems under changing working conditions.

## 1. 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 (1931). 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.

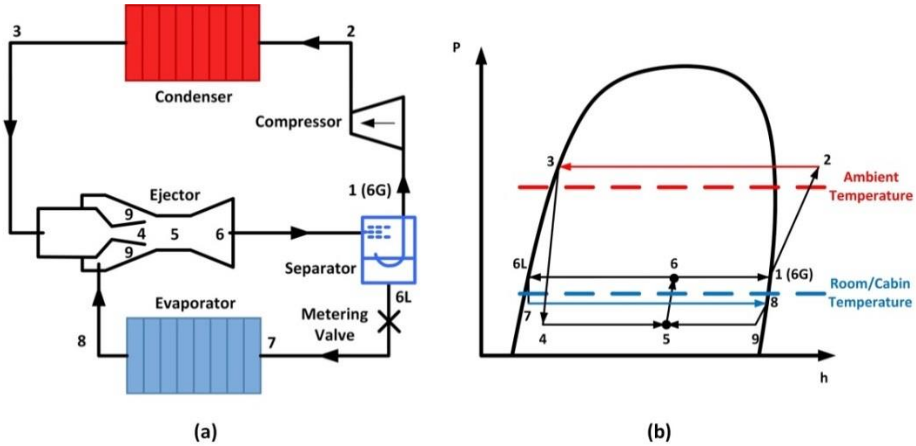


**Figure 1:** (a) Layout and (b) pressure-specific enthalpy diagram of the two-phase ejector cycle as proposed by Gay (1931)

Disawas and Wongwises (2004) 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, 2014).

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*. (2004) 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*. (2012) reported a maximum increase in COP of 8% over a baseline cycle with a conventional expansion valve.

Elbel and Hrnjak (2008) and Elbel (2011) 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 (1995) 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 (2014) 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 real world 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. 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*., 2012).

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 (2012). 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.

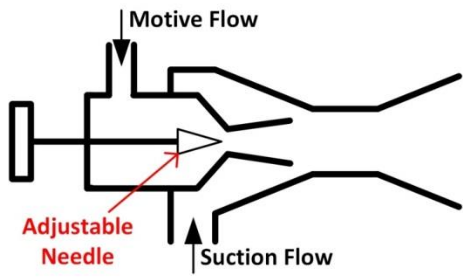


**Figure 2:** Variable ejector with adjustable needle in the motive nozzle

Elbel and Hrnjak (2008) 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*. (2014) 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 nozzle restrictiveness control mechanism, which is called vortex control, is presented. This control mechanism is possibly applicable to the control of ejector cooling cycles. It utilizes an adjustable vortex 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 vortex control mechanism are called vortex ejector and vortex 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, less vulnerable to clogging since there is no need to decrease the nozzle throat area for flow control, 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 vortex strength can be very useful for future application of ejector cooling cycles in mobile or stationary systems that experience a wide range of operating conditions.

In the following sections, the vortex ejector design and control mechanism will first be introduced in detail. A brief explanation of the influence of nozzle inlet vortex on the nozzle restrictiveness will also be provided. An experimental facility for the investigation of the inlet vortex influence on the nozzle restrictiveness will be described and the experimental results as well as preliminary visualization results will be presented and discussed.

## 2. VORTEX EJECTOR AND VORTEX CONTROL

A vortex ejector which employs the vortex control to adjust motive nozzle restrictiveness differs from a conventional ejector in that an adjustable vortex is generated at the ejector motive inlet, as is shown in Figure 3. The motive inlet vortex 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 vortex 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 vortex 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 vortex ejector cooling cycle is separated into two streams. One stream enters the vortex ejector through the motive flow tangential inlet and another enters through the motive flow axial inlet. In such a way, a vortex 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 vortex 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 vortex is introduced (downstream of the tangential inlet valve) is the same as the refrigerant state at the condenser outlet.

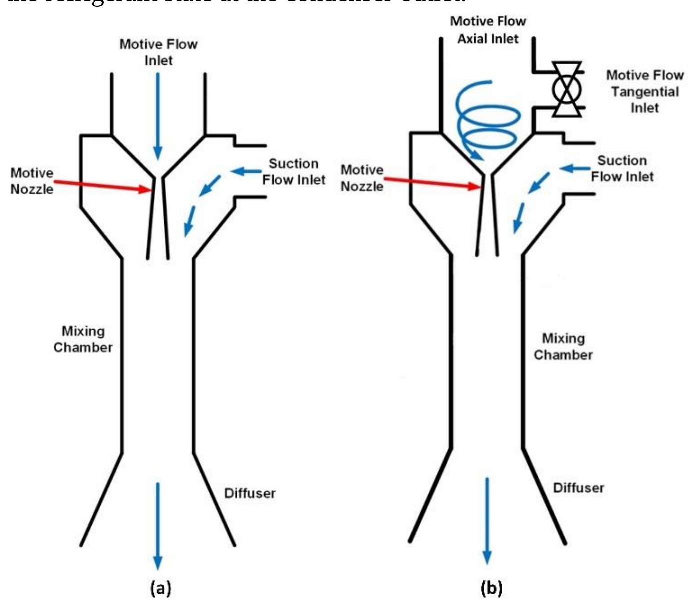


**Figure 3:** (a) Conventional ejector and (b) vortex ejector

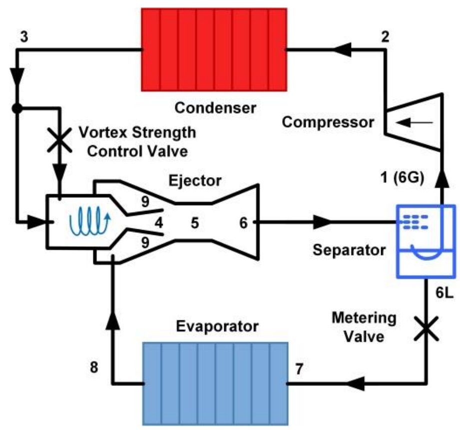


**Figure 4:** Vortex ejector cooling cycle

The influence of inlet vortex on the nozzle flow rate can be briefly explained as follows:
During the depressurization of the vortex flow in the nozzle, both the flow tangential and axial velocities increase from the nozzle inlet to the outlet. To drive the same mass flow rate through the nozzle, kinetic energy increase in the axial direction is almost the same from the nozzle inlet to the outlet regardless of the vortex strength. However, additional pressure reduction is required for the kinetic energy increase in the tangential direction when there is inlet vortex introduced. The stronger the inlet vortex is, the more kinetic energy increase in the tangential direction is and the more pressure reduction is needed. Therefore, for the same mass flow rate through the nozzle, the pressure difference between the nozzle inlet and the nozzle outlet increases as the inlet vortex becomes stronger. For the same pressure potential, less mass flow rate can be driven through the nozzle with stronger inlet vortex.

## 3. EXPERIMENTAL FACILITY AND METHODS

In order to investigate the influence of the inlet vortex on the nozzle restrictiveness and visualize the two-phase flow expanded in the convergent-divergent nozzle, a transparent nozzle with controllable vortex at the nozzle inlet has been designed and manufactured, as shown in Figure 5. Important dimensions of the vortex nozzle have been summarized in Table 1 and these dimensions are shown with corresponding letters in Figure 5. The nozzle throat has been measured with higher accuracy as small change in throat diameter may result in large difference in flow rate. It should be noted that the nozzle is only part of a vortex ejector. Experimental investigation of vortex ejectors in ejector cooling systems will be conducted in the future. The vortex nozzle is composed of three components: a tee-shaped part made of brass, a sleeve and a convergent-divergent nozzle, as shown in Figure 6, both made of an optically clear resin called Waterclear Ultra 10122 from SOMOS and manufactured with a Stereo Lithography Apparatus (SLA) from 3D SYSTEMS.

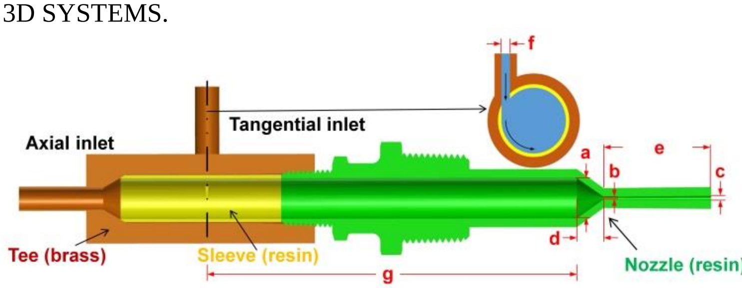


**Figure 5:** Vortex nozzle composed of tee, sleeve and convergent-divergent nozzle

**Table 1:** Vortex nozzle geometric parameters

| | |
|---|---|
| **(a) Nozzle inlet diameter (mm)** | 15.0 |
| **(b) Nozzle throat diameter (mm)** | 1.03 |
| **(c) Nozzle outlet diameter (mm)** | 1.7 |

| | |
|---|---|
| **(d) Nozzle convergent part length (mm)** | 9.9 |
| **(e) Nozzle divergent part length (mm)** | 40.0 |
| **(f) Tangential inlet inner diameter (mm)** | 2.0 |
| **(g) Vortex decay distance (mm)** | 138.0 |

**Figure 6:** 3D printed transparent convergent-divergent nozzle

The tee-shaped part serves as the vortex generator. The tangential inlet on the tee allows flow to be injected tangentially and mix with the axial flow, thus creating a vortex. The tee and the nozzle are joined by an NPT thread and sealed by epoxy adhesive. The other NPT thread on the nozzle is for connection with a visualization chamber. The sleeve is designed to provide a smooth transition between the tee part and the nozzle. There is no gap between the sleeve and the nozzle so that no additional disturbance is introduced to the flow. The inner diameter of the sleeve is the same as that of the nozzle entrance. There is a tangential inlet on the sleeve. The tangential inlet on the tee and the tangential inlet on the sleeve are coaxial and have the same inner diameter. For visualization purpose, the flow needs to travel a long distance from the tangential inlet to the starting point of the convergent part of the nozzle. This distance is called vortex decay distance, as shown in Figure 5 with the letter ‘g’. Because of the fluid viscosity and turbulence, vortex strength will decay over this distance. For actual applications, such a long distance between the tangential inlet and nozzle convergent part may not be necessary. Therefore, it is desirable to find out the vortex strength at the starting point of the nozzle convergent part so that the true relation between the nozzle inlet vortex strength (at the starting point of the convergent part) and the nozzle restrictiveness can be determined. Future work will be performed to correct for the vortex decay over the vortex decay distance. The sleeve also ensures that the vortex flow travels and decays in a pipe with constant surface properties before it reaches the convergent part of the nozzle, which simplifies future calculation of vortex decay over the vortex decay distance.

The layout of the experimental facility for the vortex nozzle tests is shown in Figure 7. A pumped-refrigerant-loop was used for adjustment of test conditions to investigate the influence of inlet vortex on the two-phase flow expanded in the convergent-divergent nozzle. The working fluid was R134a. A visualization chamber was built from clear PVC pipe. The temperature readings were all obtained from ungrounded Type-T immersion thermocouples. The measured temperatures are regarded as total temperatures. Absolute pressures were read by piezo-electric pressure transducers. Pressures and temperatures at the axial and tangential inlets of the nozzle were measured. The differences between the vortex nozzle axial inlet pressures and tangential inlet pressures were generally within 10 kPa. The axial inlet pressure is assumed to be the nozzle inlet pressure $P_{in}$. The pressure at the nozzle outlet $P_{out}$ was measured as well. The total mass flow rate $\dot{m}_{total}$ and the nozzle axial inlet mass flow rate $\dot{m}_{axial}$ were measured by Coriolis-type mass flow meters. The nozzle’s tangential inlet mass flow rate $\dot{m}_{tangential}$ can be calculated by subtracting the nozzle axial inlet mass flow rate from the total mass flow rate. The ratio of the nozzle tangential inlet mass flow rate to the total mass flow rate was adjusted by two valves. The larger the ratio is, the large the vortex strength is for the same total mass flow rate. In this paper, the vortex strength is defined as the ratio of the nozzle tangential inlet mass flow rate to the total mass flow rate, which can be expressed as shown in Equation 1:

$$Vortex\ strength = \frac{\dot{m}_{tangential}}{\dot{m}_{total}} \tag{1}$$

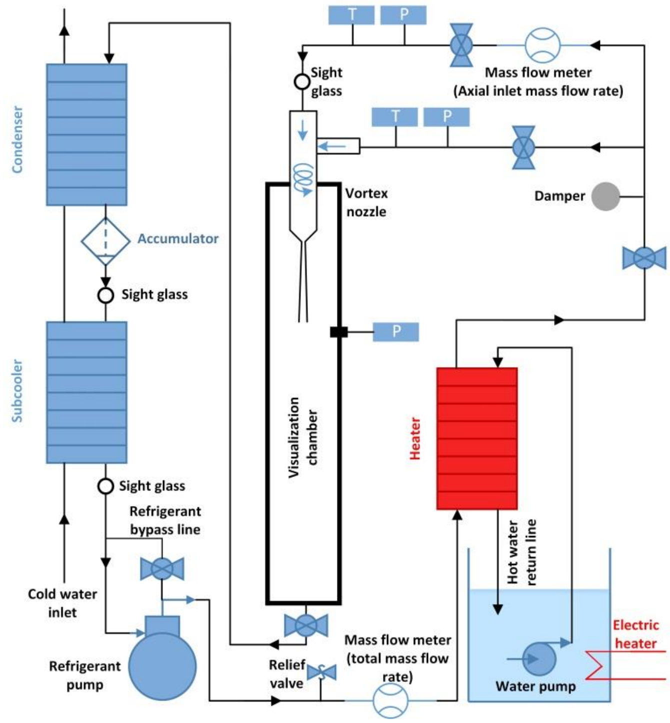


**Figure 7:** Experimental facility for investigation of the inlet vortex influence on the nozzle restrictiveness

It should be noted that in order to use the full range of vortex control from zero vortex to maximum vortex, two valves were installed in the test rig at both the nozzle's axial and tangential inlets. However, in actual applications, one valve should be sufficient to achieve nozzle restrictiveness control over a suitable range.

Different nozzle inlet pressures were achieved by adjusting the heating water temperature and pump speed which determine the saturation pressure of the refrigerant in the heater. The nozzle outlet pressure can be adjusted by a valve installed downstream of the nozzle. For all experimental results shown in this paper, the liquid flow at the nozzle inlet was subcooled by approximately 0.5 ºC. Admittedly, the calculated subcooling is close to the uncertainty of thermocouple reading and it is imprudent to claim the inlet is subcooled solely based on the thermocouple readings. The sight glass installed at the nozzle inlet allows for visual confirmation that no bubbles are present at the nozzle inlet, which provides a double check for inlet subcooling. Different nozzle inlet states with different levels of subcooling or vapor quality will be the subject of future work.

## 4. VORTEX NOZZLE TESTS WITH REFRIGERANT (R134a)

Figure 8 shows the effect of the vortex nozzle outlet pressure on the vortex nozzle total mass flow rate at constant inlet pressures. The nozzle was tested under two inlet conditions: 826 kPa and 32 ºC; 925 kPa and 36 ºC. The inlet vortex strength was varied between 0 and 1. The maximum inlet vortex can be achieved by fully closing the nozzle's axial inlet valve and fully opening the nozzle's tangential inlet valve. The Bernoulli equation has been used to theoretically calculate the incompressible single-phase liquid mass flow rate for the tested nozzle under certain inlet and outlet conditions. Nozzle inlet and outlet diameters in the calculation were 15 mm and 1.7 mm, respectively. The density of the incompressible single-phase liquid in the calculation has been assumed to be the density of the subcooled liquid at the nozzle inlet. The theoretically calculated mass flow rates for the two inlet conditions are shown in Figure 8 with solid/dashed lines. They represent the upper limits for the refrigerant mass flow rate passing through the nozzle for certain inlet and outlet conditions.

It can be observed that the two-phase flow expanded in the nozzle is choked when the nozzle outlet pressure is lower than 550 kPa. When the flow is choked, decreasing the nozzle outlet pressure does not further increase the mass flow rate. Flow choking has been observed for both inlet conditions with or without vortex when the nozzle outlet pressure is sufficiently low. Secondly, the nozzle inlet vortex reduces the total mass flow rate under the same inlet and outlet conditions, which implies larger nozzle restrictiveness.

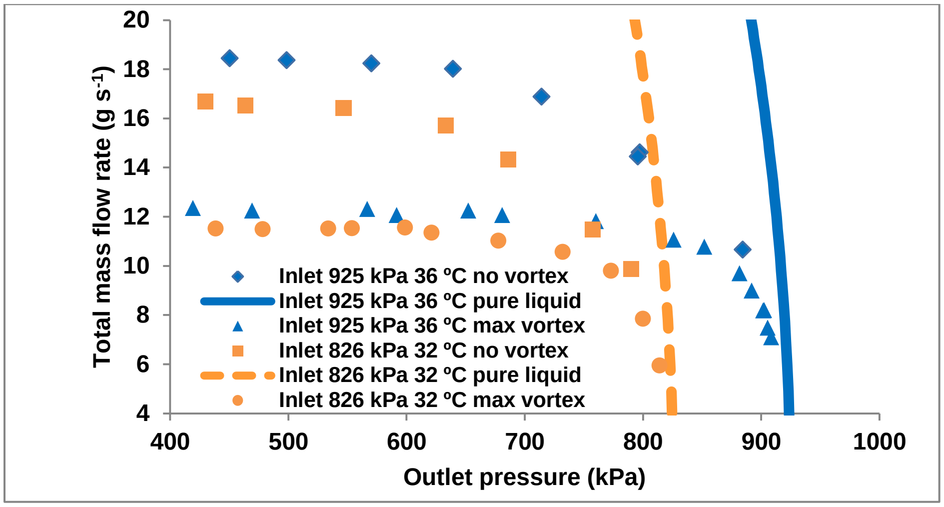


**Figure 8:** Influence of vortex nozzle outlet pressure on the vortex nozzle total mass flow rate for different constant inlet pressures

Figure 9 shows visualization of the flow expanded in the convergent-divergent nozzle. Figure 9(a) shows that the choked flow in the convergent part of the nozzle is still clear and no bubbles can be observed. It is possible that there are very small bubbles existing in the convergent part of the nozzle which are not visible to the observer without using more advanced visualization techniques. A high speed camera with high resolution and adequate lighting will be used in the future for visualization to capture more intricate flow features. The choked flow creates visible bubbles immediately after passing through the nozzle throat, which indicates that much more vapor generation takes place in the divergent part than in the convergent part of the nozzle. When the nozzle outlet pressure is close to the inlet pressure, for both the convergent and divergent part of the nozzle the flow is clear and no bubbles can be observed, as is shown in Figure 9(b).

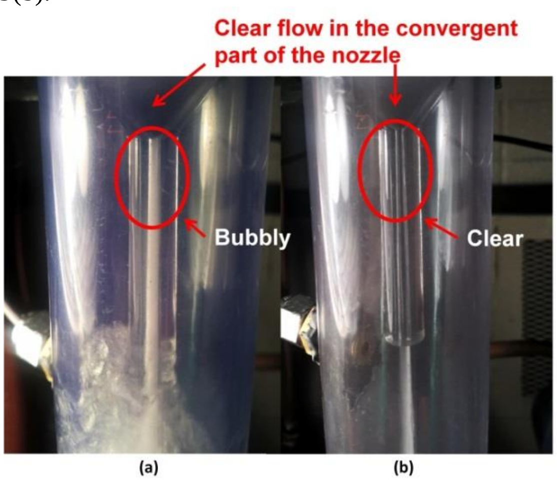


**Figure 9:** Flow expanded in the convergent-divergent nozzle: (a) choked flow ($P_{in}$ = 921 kPa, $P_{out}$ = 419 kPa, $T_{in}$ = 36.2 ºC); (b) nozzle outlet pressure close to the inlet pressure ($P_{in}$ = 930 kPa, $P_{out}$ = 885 kPa, $T_{in}$ = 35.9 ºC)

Figure 10 shows the influence of the nozzle inlet vortex strength on the total mass flow rate through the nozzle at constant inlet conditions. The nozzle outlet pressures were kept below 500 kPa so that the two-phase flow was choked. When the inlet pressure and temperature were 1034 kPa and 40 ºC, respectively, which corresponds to 0.6 ºC subcooling, by changing the inlet vortex strength the total mass flow rate varied from 20.2 g $s^{-1}$ (when there was zero vortex) to 12.9 g $s^{-1}$ (when the vortex strength was maximized). These results again show that nozzle restrictiveness can be changed by nozzle inlet vortex. The stronger the vortex is the larger the nozzle restrictiveness is, since for the same inlet conditions less mass flow rate can be driven through the nozzle. In this case, the mass flow rate can be reduced by 36% with vortex control under the same inlet and outlet conditions, which indicates substantial capacity modulation large enough to be considered for real world applications.

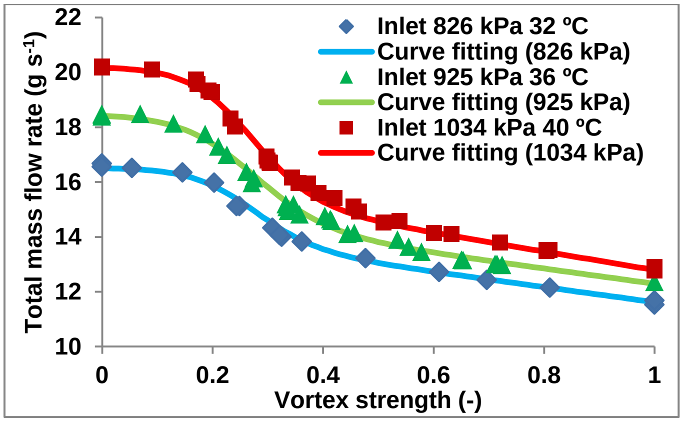


**Figure 10:** Influence of nozzle inlet vortex strength on the total mass flow rate through the nozzle at different constant inlet conditions (all data points for choked flow)

Figure 11 displays that the nozzle inlet pressure can vary in a wide range with different inlet vortex strengths at constant total mass flow rates when the flow is choked. Similarly, the nozzle outlet pressures were kept below 500 kPa to ensure choking of the two-phase flow. When the total mass flow rate was kept constant at 15 g $s^{-1}$, the nozzle inlet pressure varied from 795 kPa to 1039 kPa when the vortex strength was adjusted from 0.22 to 0.46 which again shows a large range of controllability that can be achieved with the proposed approach. Judging by the almost linear dependence of nozzle inlet pressure on vortex strength, it is reasonable to expect that the control range of the nozzle inlet pressure can be further broadened if the vortex strength is increased. At higher total mass flow rates, the required nozzle inlet pressure to achieve the same mass flow rate increases faster with the vortex strength. Data points for vortex strength between 0.65 and 1 are missing, because when the nozzle axial inlet mass flow rate is small heat losses in the tubes have significant cooling effect on the axial inlet flow even with insulation. In that case it is difficult to keep the same conditions at both nozzle inlets. When the nozzle axial inlet is fully closed, there is no such problem since only the inlet conditions at the nozzle tangential inlet need to be controlled.

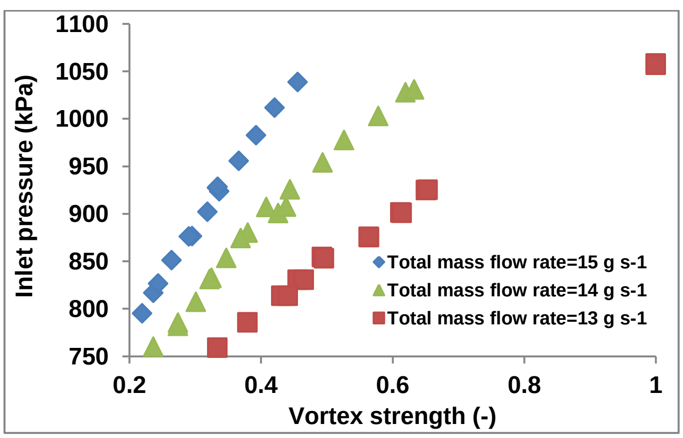


**Figure 11:** Nozzle inlet pressure variations for different inlet vortex strengths at constant total mass flow rates (all data points for choked flow)

## 5. CONCLUSIONS

In this study, a new nozzle restrictiveness control mechanism called vortex control has been proposed and verified. The approach seems suitable to be applied for the purpose of capacity modulation of two-phase ejector vapor compression cycles. This control mechanism is simple, possibly inexpensive, less vulnerable to clogging since there is no need to decrease the nozzle throat area for flow control, and can potentially yield better efficiencies than other approaches that attempt to control nozzle flow rates. A convergent-divergent nozzle utilizing vortex control was designed and manufactured for experiments with R134a. According to the experimental results, it has been shown that the strength of the nozzle inlet vortex can change the restrictiveness of the two-phase nozzle without changing the nozzle geometry. The nozzle becomes more restrictive as the strength of the vortex increases. The mass flow rate can be reduced by 36% with vortex control under the same inlet and outlet conditions. The control range of inlet pressures and mass flow rates that can be achieved by vortex control appears to be large enough to be applicable for real world applications.

## NOMENCLATURE

| | | | | |
|---|---|---|---|---|
| COP | coefficient of performance | (–) | **Subscript** | |
| h | specific enthalpy | (kJ $kg^{-1}$) | axial | axial inlet |
| $\dot{m}$ | mass flow rate | (g $s^{-1}$) | in | inlet |
| P | pressure | (kPa) | out | outlet |
| T | temperature | (ºC) | tangential | tangential inlet |
| | | | total | total mass flow rate through the nozzle |

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