2016

Comparative Analysis Of CO2 Cycle Enhancements: Ejector Vs. Vapor Injection

Kirill M. Ignatiev
Emerson Climate Technologies, Inc., United States of America, Kirill.Ignatiev@Emerson.com

Comparative Analysis of CO2 Cycle Enhancements: Ejector Vs. Vapor Injection

Kirill Ignatiev

Emerson Climate Technologies, Inc.,
Sidney, Ohio, USA
Phone: +1 937 498 3844, E-mail: Kirill.Ignatiev@Emerson.com

ABSTRACT

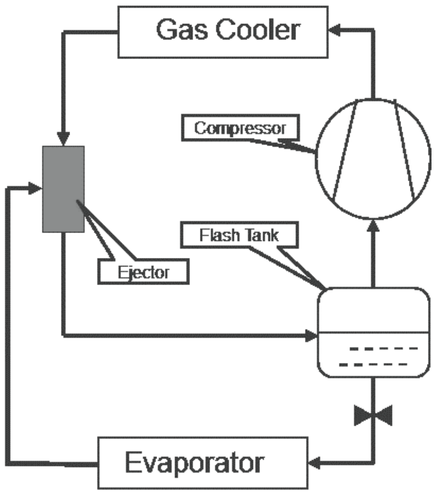
In the recent years, transcritical CO2 refrigeration is gaining popularity. However, operation above critical point at higher ambient temperatures results in significant reduction of CO2 refrigeration cycle efficiency. Vapor injection and ejector are probably the mostly widely applied transcricial cycle enhancement options.
The proposed paper includes a description of a simple transcritical one dimensional model of an ejector, based on real gas properties (accessible through Refprop7), as well as integration of this model in a refrigeration cycle.
Vapor injection cycle is analyzed based on compressor performance and its ability to digest vapor injection stream. An enhancement of traditional compressor performance approximation, enabling to include the vapor injection stream, is presented and vapor injection cycle is analyzed based on that compressor model

1. INTRODUCTION

Comparison of CO2 cycle performance between two outlined methods at different operation conditions is presented.
There has been a vast number of publications regarding application of ejector to enhance the performance of refrigeration cycle, especially in relation to Carbon Dioxide refrigeration cycle operating at transcritical conditions, for example by Liu and Groll [1], Eibel and Hrnjak [2], Li and Groll [3], and many others. They explain different level of scrutiny and complexity of the simulation models of ejectors, as well as experimental results.
At the same time, amount of publications dedicated to Vapor Injection (Economizer) cycle are also very extensive. They range from application of a specific compressor technology like scroll to vapor injected cycle for example by Jain, et.al. [4], Perevozchikov and Pham [6] and many, many others. The other vast group of publications investigates vapor injection cycle from more of the system perspective and configurations, like Roh and Kim [5], Wang et.al. [7], or Mathison et al [8].
The major objective of this paper is to apply same approach to cycle simulation for both cases, clearly state the assumptions and conduct back-to-back analysis of those two refrigeration cycles according to the assumptions, so that the performance of those cycles at the same conditions can be easily and fairly compared to each other.

2. SYSTEMS CONFIGURATION AND ANALYSIS

In this paper, as an example, direct comparison is provided for two refrigeration systems- a system with ejector, and a flash tank refrigeration system, illustrated of Fig.1 and Fig.2 respectively. For the vapor injected system, a parallel compressor model for a vapor injected compression is selected.
Below, detailed description of the modelling process of each of those configurations is provided. For the refrigerant properties calculations, Refprop7 functions were utilized.



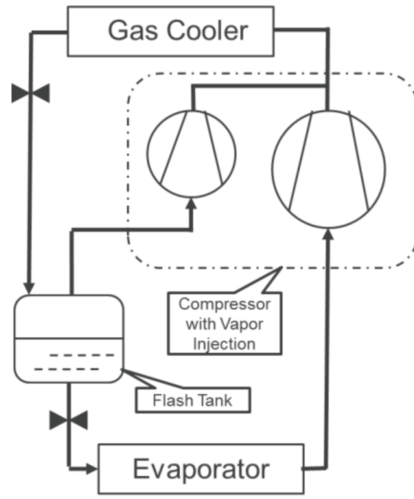


Fig. 1. Schematic for Ejector cycle

Fig.2. Schematic for vapor Injection cycle

2.1 Model of an ejector and ejector cycle

Without pretending on the novelty of describing ejectors, the author would like to have a brief description of the ejector principle of operation. Ejector is a gas dynamic device which allows to utilize the kinetic energy of expansion of one stream of fluid, to compress another stream of fluid. This device has a common outlet. It consists of the following elements: Driver nozzle, driven nozzle, mixing chamber and diffuser.

2.1.1. Driver nozzle. High pressure fluid is introduced into the driver nozzle with the initial conditions P_0, T_0, V_0, where it adiabatically expands and accelerates. During this expansion process, full enthalpy and entropy remains constant:

$$h_{0\,full} = h(P_0, T_0) + \frac{V_0^2}{2} \;;\; s_0 = s(P_0, T_0) \tag{1}$$

Assuming P as a pressure in a given cross-section of the flow, the other parameters of the fluid in this cross section can be calculated- Density, Velocity and Hydraulic Diameter:

$$\rho = \rho(P, s_0) \tag{2}$$

$$V(P) = \sqrt{2\left(h_{0\,full} - h(P, s_0)\right)} \tag{3}$$

(4)

$$\frac{D}{D_0} = \sqrt{\frac{\rho(P)V(P)}{\rho(P_0)V(P_0)}} \tag{5}$$

In case of the fluid being in 2-phase region, vapor fraction α can be calculated and then used to calculate the remaining properties- Enthalpy and Density:

$$\alpha \cdot s_V(P) + (1-\alpha)\cdot s_L(P) = s_0 \tag{6}$$

$$\alpha = \frac{s_0 - s_L(P)}{s_V(P) - s_L(P)} \tag{7}$$

$$h = \alpha \cdot h_V(P) + (1-\alpha)\cdot h_L(P) \tag{8}$$

$$\rho = \frac{1}{\alpha/\rho_V(P) + (1-\alpha)/\rho_L(P)} \tag{9}$$

To calculate the local sonic velocity, the most general equation was used:

$$c = \sqrt{\left.\frac{\partial P}{\partial \rho}\right|_{S=const}} \tag{10}$$

With the effect of the nozzle efficiency,

$$h_1^*(P) = \eta_{nozzle} h(P_1, s_0) + (1-\eta_{nozzle}) h(P_0, T_0) \tag{11}$$

$$V_1^*(P_1) = \sqrt{2(h_{0\,full} - h_1^*)} \tag{12}$$

2.1.2. Mixing Chamber. In the Mixing chamber, driver flow and driven flow streams are mixing together at a constant pressure. In the outlet, there is a uniform flow. Outlet velocity is governed my momentum conservation equation, outlet enthalpy is governed by full enthalpy conservation equation:

$$V_{mix} = \frac{V_1^* + \gamma V_2}{1+\gamma} \tag{13}$$

$$h_{mix} = \frac{h_1^* + \frac{V_1^{*2}}{2} + \gamma\, h_2 + \gamma \frac{V_2^2}{2} + \frac{V_0^2}{2}}{1+\gamma} - \frac{V_{mix}^2}{2} \tag{14}$$

2.1.3. Diffuser. In the diffuser, the mixed stream exiting the mixing chamber is slowing down, building up pressure at the same time. Diffuser efficiency is applied at the diffuser inlet:

$$V_{mix}^* = V_{mix} \cdot \sqrt{\eta_{diff}} \tag{15}$$

$$h_{mix}^* = h_{mix} + \frac{V_{mix}^2}{2}(1-\eta_{diff}) \tag{16}$$

$$s_{mix}^* = s(P_1, h_{mix}^*) \tag{17}$$

Pressure at the diffuser exit P_{slow} can be calculated from the following equation:

$$h_{slow}(P_{slow}, s_{mix}^*) = h_{mix}^* + \frac{V_{mix}^{*2}}{2} - \frac{V_{slow}^2}{2} \tag{18}$$

2.1.4. Flash Tank. The fluid exiting the diffuser is typically in a 2-phase form. It enters a flash tank, where vapor fraction is separated from the liquid fraction:

$$\beta = \frac{h_{slow} - h_L}{h_V - h_L} \tag{19}$$

Going back to the schematics of the system, the vapor fraction from the flash tank enters the compressor suction, it is being re-compressed by the compressor, cooled down in gas cooler and then enters the ejector as the driver flow. Liquid fraction from the flash tank is throttled to the evaporator and then, after the evaporator, enters the ejector as a driven flow. Therefore, at the steady-state condition, the following equation should relate the flow fractions exiting the flash tank and the flows entering the ejector:

$$\gamma = \frac{1-\beta}{\beta} \tag{20}$$

(8)

During computations, this equilibrium is achieved through iterative process.

2.1.5. Model of compressor and the rest of the system.
Conditions at the compressor suction:

$$P_S = P_{slow};\ T_S = T_{SAT}(P_{slow}) + \Delta T_{SH} \tag{21}$$

$$s_S = s(P_S, T_S)\ ; \rho_S = \rho(P_S, T_S) \tag{22}$$

(8)

Compressor isentropic delta enthalpy, power, mass flow:

$$m_s = \rho_S v_S \tag{23}$$

$$\Delta h_{IS} = h(P_0, s_S) - h(P_S, s_S) \tag{24}$$

(8)

$$N_{comp} = \frac{\rho_S v_S}{\eta_{IS}} \Delta h_{IS} \tag{25}$$

(8)

Evaporator cooling capacity:

$$\Delta h_{EVAP} = h(P_{EVAP}, T_{SAT}(P_{EVAP}) + \Delta T_{SH}) - h_L \tag{26}$$

$$Q = m_{evap} \Delta h_{EVAP} \tag{27}$$

(8)

Formulation for Capacity and COP:

$$Q = \gamma \rho_S v_S\ \Delta h_{EVAP} \tag{28}$$

(8)

$$COP = \gamma \frac{\eta_{IS} \Delta h_{EVAP}}{\Delta h_{IS}} \tag{29}$$

(8)

2.2 Model of the vapor injected cycle

For the vapor injected cycle, a traditional schematic with flash tank and parallel compressor model was chosen (Fig.2). Those systems have been described multiple times in literature, however, it makes sense to provide a few formulation describing the system:

2.1.1. Flash Tank. High pressure fluid with the initial conditions P_0, T_0 , through the first expansion valve is introduced into the flash tank, where it separates into liquid and vapor fractions. The vapor fraction β can be easily calculated:

$$h_0 = h(P_0\ , T_0) \tag{30}$$

(8)

$$\beta = \frac{h_0 - h_L}{h_V - h_L} \tag{31}$$

(8)

Liquid fraction is re-expanded in the second expansion valve into the evaporator and then enters the suction port of the compressor. Vapor fraction entering the vapor injection port of the compressor is re-compressed.
Conditions at the compressor suction:

$$P_S = P_{EVAP};\ T_S = T_{SAT}(P_{EVAP}) + \Delta T_{SH} \tag{32}$$

(8)

$$s_S = s(P_S, T_S)\ ; \rho_S = \rho(P_S, T_S) \tag{33}$$

$$s_{VI} = s(P_{VI}, T_{SAT}(P_{VI}) + \Delta T_{SH}) \tag{34}$$

(8)

Compressor isentropic delta enthalpy, power, mass flow:

$$m_s = \rho_S v_S \tag{35}$$

(8)

$$\Delta h_{IS} = \left(h(P_0, s_S) - h(P_S, s_S)\right) + \frac{\beta}{1-\beta}\left(h(P_0, s_{VI}) - h(P_{VI}, s_{VI})\right) \tag{36}$$

(8)

$$N_{comp} = \frac{\rho_S v_S}{\eta_{IS}} \Delta h_{IS} \tag{37}$$

(8)

Evaporator cooling capacity:

$$\Delta h_{EVAP} = h\left(P_{EVAP}, T_{SAT}(P_{EVAP}) + \Delta T_{SH}\right) - h_L \tag{38}$$

$$Q = m_{evap} \Delta h_{EVAP} \tag{39}$$

(8)

Formulation for Capacity and COP:

$$Q = \rho_S v_S \Delta h_{EVAP} \tag{40}$$

(8)

$$COP = \frac{\eta_{IS} \Delta h_{EVAP}}{\Delta h_{IS}} \tag{41}$$

(8)

If we compare equations (28)-(29) and (40)-(41) and their respective components, the following observations can be made:

1. Compression work may be higher for vapor injection case, since it compresses the suction stream all the way from evaporator to discharge pressure, and additionally, the vapor injection stream. Also, it is more challenging to make a compressor operating efficiently at higher pressure ratio, with vapor injection.
2. There is a factor γ in the equations (14) which, in case of being below 1, reduces both COP and capacity. However, without detailed numerical analysis, it is virtually impossible to compare both cycles.

3. MODELING AND MODELING RESULTS

3.1 Ejector cycle.

A numerical simulation model based on the equations described in section 2 of this paper has been created using object oriented VBA code and Refprop7 refrigerant properties functions. As an example of the analysis, Fig.3 provides an illustration for a flow in driver nozzle for Transcritical CO2 rating point condition (Incoming flow at 8.85 MPa/35deg C).

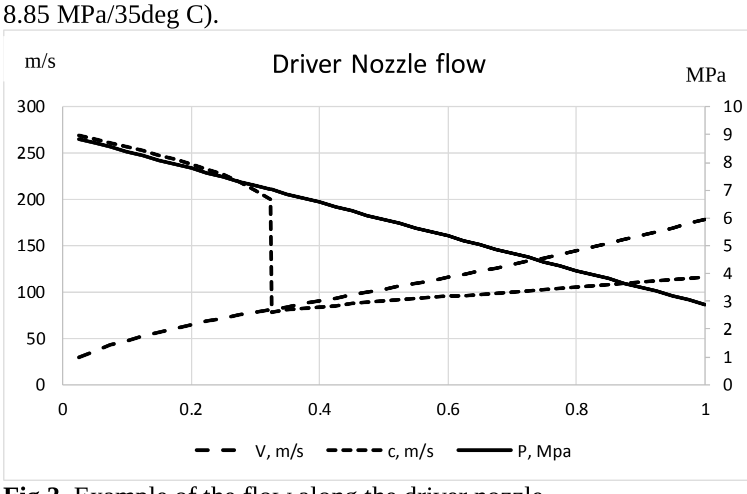


Fig.3. Example of the flow along the driver nozzle

The nozzle expansion process becomes supersonic when the flow is crossing into the dome (2-phase) curve. At the same time, the local speed of sound is dropping substantially. To utilize the expansion process, the nozzle should be of classical supersonic design, with converging subsonic and diverging supersonic sections.

3.2 Vapor Injection Cycle

Referencing the Fig.2 and equations (30)-(41), we can see that with given evaporator temperature and gas cooler outlet pressure/temperature, Flash tank pressure (and vapor fraction) can be considered as an independent parameter. For the transcritical rating point with gas cooler outlet conditions (8.85 MPa/35deg C) and typical Medium temp evaporator conditions (-5 degC) being fixed, having fixed isentropic efficiency of compressor with vapor injection (65%), and having flash tank pressure varied, COP and capacity can be calculated, which is illustrated on Fig. 4. It can clearly be seen, that, in order to achieve the peak COP, vapor injected system should operate at an optimal vapor injection pressure.

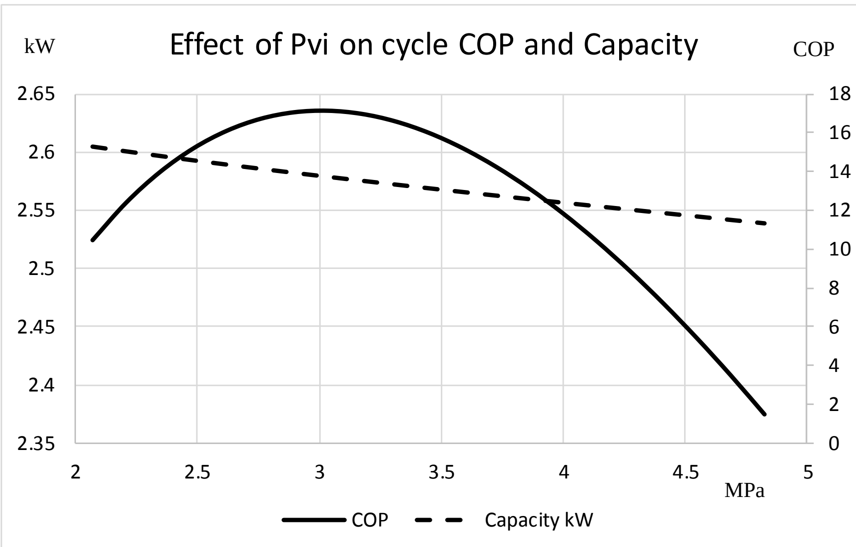


Fig. 4. Effect of Flash Tank pressure on COP and Capacity

Similar analysis can be conducted for other operating conditions. The peak COP points for each conditions can be represented as a point with dimensionless coordinates P_{VI} / P_{EVAP} , m_{VI} / m_S . A chart of those peak COP points is illustrated on Fig.5.

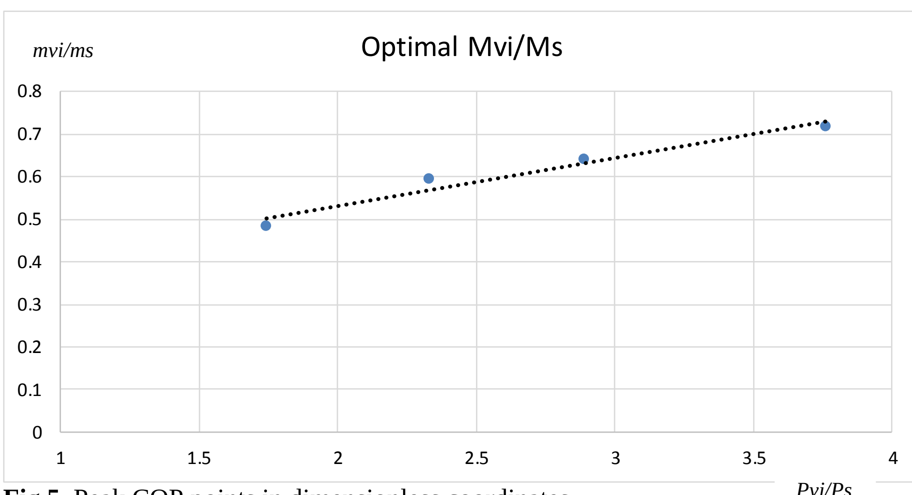


Fig.5. Peak COP points in dimensionless coordinates

Therefore, in order to operate at peak COP at multiple conditions, compressor vapor injection should be designed in such a way that it will follow the trendline illustrated on Fig.5.

3.3 Comparison between Ejector and Vapor Injected cycles.

For comparative purposes, two systems were compared side- by side with the following assumptions regarding the system components and its efficiencies (Table 1):

Table 1. Assumption for the system comparison

Comrpessor volumetric flow	*l/s*	1.47
Compressor Isentr. Efficiency		0.65
Evaporator Super-heat	*deg C*	11.1
Flash Tank Super-heat	*deg C*	2.8
Driver Nozzle Efficiency		0.96
Diffuser Efficiency		0.76

For the Vapor injected system, relationship between vapor mass flow fraction and vapor injection to suction pressure ratio was kept at the optimal value (according to trendline at Fig. 5).

Table 2. Summary of the systems performance comparisons

Parameter	*Unit*	**Ejector**	**VI**	**Ejector**	**VI**	**Ejector**	**VI**
T evap	**deg C**	-5.2	-5.2	-20.2	-20.2	-29.1	-29.1
P evap	**MPa**	3.04	3.04	1.97	1.97	1.48	1.48
T GC	**C**	35	35	35	35	35	35
PGS	**MPA**	9	9	9	9	9	9
P Flash Tank	**MPa**	3.82	5.30	2.76	4.59	2.23	4.29
GAMMA		0.51		0.47		0.45	
Mvi/ Ms			0.48		0.59		0.64
Compr PR		2.4	3.0	3.3	4.6	4.0	6.1
Capacity	**kW**	19.0	22.5	13.5	15.5	10.8	12.0
COP		2.26	2.26	1.56	1.62	1.27	1.35

Table 2 represents side-by-side the direct performance comparisons between ejector and vapor injected systems at 3 different evaporator conditions, with the gas cooler at transcritical rating point (carbon dioxide as a working fluid). Some parameters are provided for a specific system configuration only.

CONCLUSIONS

- At medium temp rating point (Tevap=-5.2 deg C), the system COPs, surprisingly, are the same with the current efficiency assumptions. At lower evaporative temperatures, however, COP of the vapor injected system is better.
- It is more challenging for Vapor injected compressor to maintain its efficiency (due to higher pressure ratio and vapor injection stream), than for ejector system. Also, vapor injected compressor should maintain vapor stream mass flow and pressure ratios at the optimal level.
- Capacity of the vapor injected system is higher at all points analyzed, assuming the same suction volumetric flows. Having parameter GAMMA substantially lower than 1 explains that. However, maintaining the same volumetric flow is more challenging for Vapor injected compressor due to higher pressure ratio.

- Ejector system provides lower flash tank pressure compared to vapor injection system. That generally provides better liquid quality in the evaporator.

NOMENCLATURE

h	Enthalpy	kJ/kg
s	Entropy	kJ/kg/C
P	Pressure	MPa
T	Temperature	deg C
V	Velocity	m/s
c	Sonic Velocity	m/s
ρ	Density	kg/m3
α	Vapor mass fraction in 2-phase mixture	-
γ	Driven flow to driving flow ratio	-
β	Vapor fraction in a flash tank	-
η	Efficiency	-
Δh	Enthalpy difference	kJ/kg
v	Compressor Volumetric flow	m3/s
m	mass flow	kg/s
N	compressor power	kW
Q	Cooling capacity	kW
COP	Coefficient of performance	-

Subscript

0	Gas cooler outlet- Driver flow inlet conditions
1	Parameters related to driver flow entering mixing chamber
2	parameters related to driven low entering mixing chamber
slow	Parameters related to diffuser outlet
L	Parameters related to saturated liquid
V	Parameters related to saturated vapor
nozzle	Parameters related to nozzle
diff	Parameters related to diffuser
mix	parameters related to mixing chamber outlet
full	related to Full enthalpy (Including kinetic energy of the flow)
is	related to isentropic process
evap	relates to fluid parameter in evaporator
comp	relates to fluid parameter in compressor
S	relates to compressor suction
SAT	relates to saturation condition
SH	super heat

Superscript

*	Parameters after consideration for efficiency loss

REFERENCES

1. Liu, F., Groll, E.A., 2008, Analysis of a two phase flow ejector for the transcritical CO2 cycle, *Proceedings of the International Ai Conditioning and Refrigeration conference at Purdue*, Purdue University, USA, paper No. 2131
2. Eibel, S., Hrnjak, P., 2008, Ejector Refrigeration: An overview of Historical and present Developments with an Emphasis on Air Conditioning Applications, *Proceedings of the International Air Conditioning and Refrigeration conference at Purdue*, Purdue University, USA, paper No. 2350
3. Li, D.Q., Groll, E.A., 2004, Transcritical refrigeration cycle with ejector-expansion device, *Proceedings of the International Air Conditioning and Refrigeration conference at Purdue*, Purdue University, USA, paper No. R153
4. Jain, S., Jain, G., Bullard, C.W., 2004, Vapor Injection in scroll compressors, *Proceedings of the International Air Conditioning and Refrigeration conference at Purdue*, Purdue University, USA, paper No. 1642
5. Roh, C.W., Kim, M.S., 2012, Performance comparison of the vapor injection cycle in terms of Branching points of the injection stream, *Proceedings of the International Air Conditioning and Refrigeration conference at Purdue*, Purdue University, USA, paper No. 1246
6. Perevozchikov, M., Pham, H., 2004 Scroll compressor for mobile HVAC/R applications, *Proceedings of the International Air Conditioning and Refrigeration conference at Purdue*, Purdue University, USA, paper No. 1656
7. Wang, X., Hwang, Y., Rademacher, R., 2008, Performance investigation of refrigerant vapor-injection technique for residential heat pump, *Proceedings of the International Air Conditioning and Refrigeration conference at Purdue*, Purdue University, USA, paper No. 2119
8. Mathison, M., Braun, J., Groll, E., 2010, Performance limit for economized cycles with continuous refrigerant injection, *Proceedings of the International Air Conditioning and Refrigeration conference at Purdue*, Purdue University, USA, paper No. 1034

ACKNOWLEDGEMENT

The author of the paper would like to express an acknowledgement for management of Emerson Climate Technologies, Inc., for their support.