10.18462/iir.gl.2018.1392

Experimental and Numerical Investigation on an Ejector for CO_2 Vapor Compression Systems

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ABSTRACT

The use of two-phase flashing ejectors to improve the COP and capacity of CO_2 vapor compression systems has become a key research topic in recent years. Although many experimental investigations focused on both the ejector performance and overall energy efficiency have been carried out, further efforts devoted to significantly enhance this device are necessary. In the present paper, the authors experimentally investigated the performance of a two-phase ejector as its main parameters, such as pressure lift and entrainment ratio, are varied. The results are finally compared with a CFD model based on a commercial solver.

Keywords: Refrigeration, Carbon Dioxide, Ejector, CFD, Experimental

1. INTRODUCTION

During the recent years the interest of industry and scientific community in carbon-dioxide refrigeration systems has considerably grown because of the more and more strict regulations regarding environmentally safe refrigerants. CO_2 has been shown to be an interesting alternative to HFC fluids for many applications because it is non-toxic, non-flammable, low-priced and has a reduced environmental impact (GWP).
Carbon-dioxide standard vapor compression cycle can benefit of several cycle modifications to improve the system efficiency [1] and one of the most interesting is the use of ejectors as expansion device. This solution was first proposed by Lorentzen [2] and was widely tested in both numerical and experimental ways (e.g. [3-6]) showing promising efficiency improvements with low additional investment costs.
Two-phase ejectors can be used to pump liquid in the evaporator in order to obtain the fluid circulation at the evaporator [6, 4] (Fig. 1b), or to compress the low pressure vapor from the evaporator to the compressor (Figure 1a). The vapor-compression scheme is the subject of the present paper.

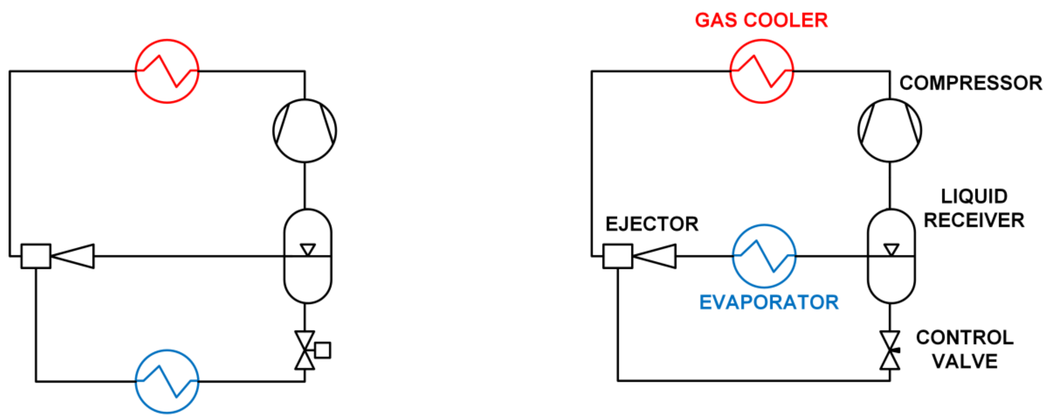

Figure 1: vapor compression cycle equipped with vapor ejector (a); vapor compression cycle equipped with liquid ejector (b)

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The main challenge with ejectors is the complex flow behavior occurring inside such devices, i.e. two-phase non-equilibrium flow with shocks and expansion-waves. Consequently, the development and validation of predictive CFD models is necessary in order to increase the performance and efficiency of the ejector. So far several CFD studies on two-phase flashing ejector have been conducted [7-9] but more work to improve the available models has still to be done. In the work of Yazdani et al. [7] the authors used a "mixture-model" approach capable to account for the non-equilibrium phase change in the ejector and implemented it in the commercial code ANSYS Fluent. In Lucas et al. [8] the authors used a homogeneous equilibrium model approach with the phase change occurring in equilibrium and the two phases sharing same pressure and temperature; in this case, the CFD code was Open-FOAM. In Smolka et al. [9] the authors used a homogeneous equilibrium model approach based on a modified enthalpy transport equation implemented in the commercial code ANSYS Fluent.
In the present work, the authors adopted an approach similar to Smolka et al. [9] by implementing a homogeneous equilibrium model in ANSYS Fluent with the additional capability of the present model to be easily and quickly adaptable to different fluids. Such feature is allowed by a combined procedure based on MATLAB [10] and in-house developed C routines. More detailed description of the model can be found in [11] and [12].
Experimental activity on ejector-equipped systems is also extremely important at both system and component level and several works are available in the literature in this field. Banasiak et al. [5] conducted a detailed experimental investigation about the influence of the ejector geometry on the efficiency of this component. Elbel and Harnjak [3] focused on the performance of a transcritical R744 system equipped with an ejector. As a final example, Banasiak et al. [13] made an extensive performance mapping of ejectors in the "multipack" device (multiple ejectors in parallel). In the experimental part of the present work the focus was on the ejector itself and the purpose was to use the obtained experimental results to benchmark and validate the CFD model.

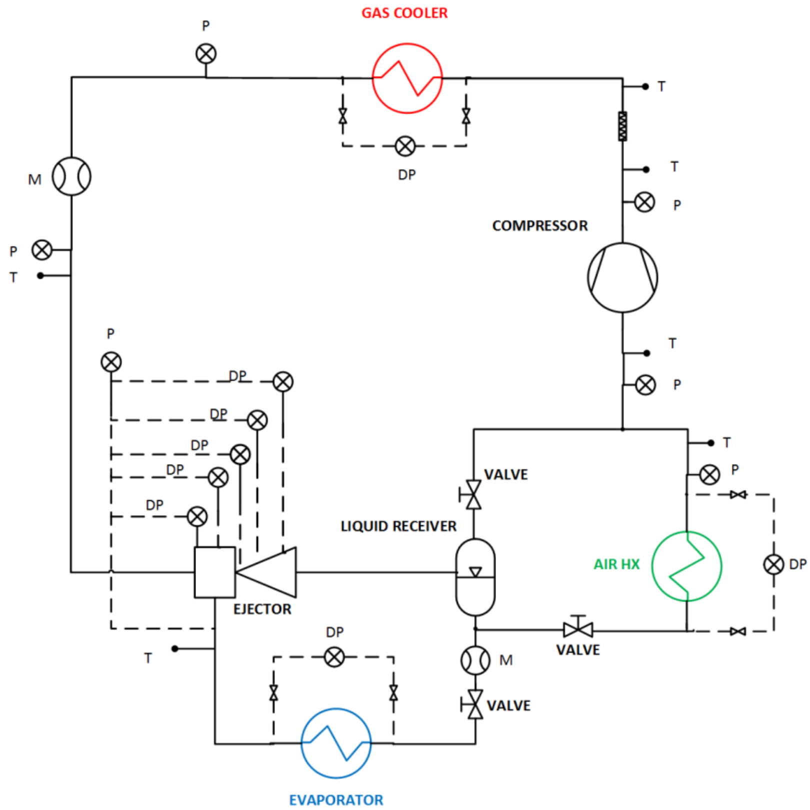

Figure 2: Experimental test-rig layout. T: T-type thermocouple P: pressure sensor DP: differential pressure sensor M: Coriolis mass flow meter

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2. EXPERIMENTAL SETUP

The experimental part of this work was conducted at the SINTEF Energy and NTNU lab. The experimental setup was described in detail in Banasiak et al. [5]. The only difference from that work is a slight modification of the ejector geometry and system layout in order to obtain a higher pressure lift of the tested ejector. The actual layout is depicted in Figure 2. Heat input and output are provided to the system via two separate glycol loops at the evaporator (blue) and the gas cooler (red) while additional heat is provided at the air heat exchanger (green) in order to achieve a higher pressure lift of the ejector.
For more detailed description of all the components of the test-rig the reader is referred to Banasiak et al. [5].
The measurement system (see Figure 2) was based on temperature sensors (calibrated T-type thermocouples), absolute and differential pressure sensors (calibrated piezoelectric elements), and mass flow meters (calibrated Coriolis-type). The uncertainties for all the measured quantities are evaluated considering both instrument and random errors with the following equation:

$$\varepsilon = \sqrt{(\sigma^2 + \varepsilon_i^2)} \qquad (1)$$

where σ is the standard deviation of the measured value and ε_i is the instrument error. For the derived quantities the following equation was used instead:

$$\varepsilon = \sqrt{\sum_i \left(\frac{\partial f(x_i)}{\partial x_i} \delta x_i\right)} \qquad (2)$$

where $f(x_i)$ is the derived quantity while x_i are the measured quantities. $\frac{\partial f(x_i)}{\partial x_i}$ is known as “sensitivity index”, which in case of absence of an analytical formulation for the derived quantity (e.g., when evaluating the enthalpy of the refrigerant through NIST libraries), can be evaluated as follows [14]:

$$\frac{\partial f(x_i)}{\partial x_i} = \frac{1}{2}\left(\left|\frac{f(\bar{x}_i + \delta x_i) - f(\bar{x}_i)}{\delta x_i}\right| + \left|\frac{f(\bar{x}_i) - f(\bar{x}_i - \delta x_i)}{\delta x_i}\right|\right) \qquad (3)$$

The mean values of the measurement uncertainties, including both sensor accuracy and the time-averaged deviations from steady state, were as follows: ε(T)= ±0.3 K, ε(P)= ±15x10^3 Pa, and ε($\dot{m}$)= ±0.5x10^{-3}kg·s^{-1}.
The main dimensions of the ejector are shown in Figure 3 and summarized in Table 1. The ejector also has a slightly swirled flow entering the secondary inlet, but, for the preliminary analyses of the present paper this feature of the device has been neglected.

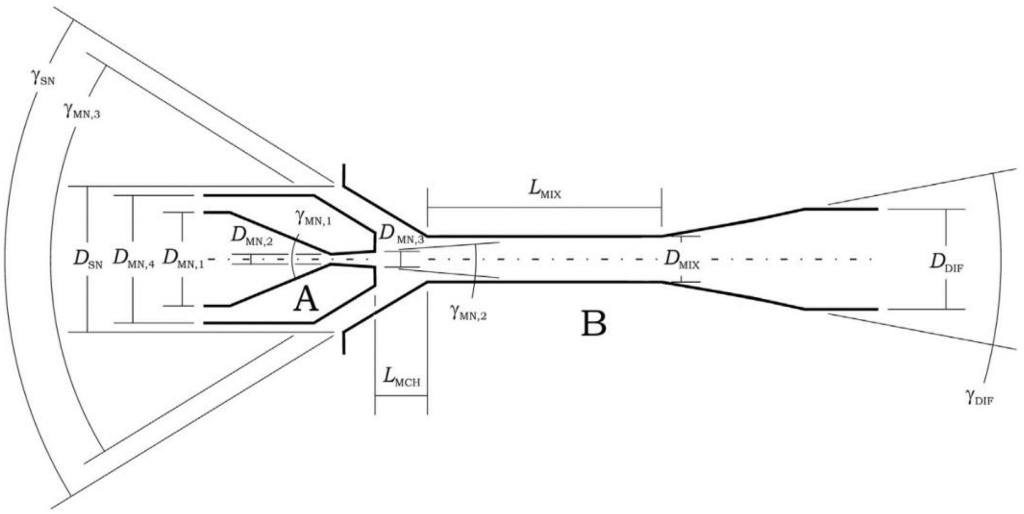

Figure 3 Basic dimensions of the ejector geometry. A e motive nozzle piece, and B e suction nozzle, mixer and diffuser piece.

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Motive Nozzle	$D_{MN,1}$, 10^{-3} m	6
	$D_{MN,2}$, 10^{-3} m	0.9
	$D_{MN,3}$, 10^{-3} m	1.03
	$D_{MN,4}$, 10^{-3} m	12
	$\gamma_{MN,1}$, °	30
	$\gamma_{MN,2}$, °	2
	$\gamma_{MN,3}$, °	42
Mixing Chamber and Diffuser	L_{MCH}, 10^{-3} m	7.5
	D_{SN}, 10^{-3} m	18.8
	γ_{SN}, °	42
	D_{MIX}, 10^{-3} m	2
	L_{MIX}, 10^{-3} m	16.9
	D_{DIF}, 10^{-3} m	10
	γ_{DIF}, °	5

Table 1 Main Ejector Dimensions

3. NUMERICAL SETUP

The computational domain consisting of 2D axis-symmetrical domain discretized in approximately 50000 quadrilateral elements is shown in Figure 4, together with details of the mesh near the motive nozzle exit. The set of computed equation has been modified by substituting the standard energy equation with a Scalar Transport Equation of enthalpy. The turbulence model used in the present work is the 2 equations *k-ε Realizable* model with turbulent kinetic energy production limiter. All the equations are solved with a second order upwind spatial discretization method and a Least Squares Cell Based method for the gradients. The solver is a pressure-based solver with pressure velocity coupling. The properties of R744 are evaluated assuming equilibrium phase change by means of lookup-tables obtained with an in-house developed MATLAB code. The code allows the user to easily change the fluid in the computational model. For a more detailed description of the developed routines and the model description the reader is referred to Giacomelli et al. [11].

Figure 4 Ejector mesh with detail view near the motive nozzle exit

4. RESULTS

3.1 Experimental Results

Table 2 summarizes the set of operating conditions maintained during the tests. Figure 5 presents mass entrainment ratio ER (suction mass flow rate to the motive mass flow rate, see Eq. (4)) as functions of pressure lift P_{lift} (difference between ejector outlet pressure and suction nozzle inlet pressure, see Eq. (5)):

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Case	P_{MN} [MPa]	T_{MN} [K]	P_{SN} [MPa]	T_{SN} [K]	P_{OUT} [MPa]
1	9.48	310.1	3.86	282.8	4.13
2	9.49	310.3	3.59	283.4	4.10
3	9.50	310.6	3.38	282.2	4.08
4	9.52	310.8	2.93	286.7	4.02
5	9.53	303.5	3.46	281.3	3.96
6	9.53	303.5	3.60	281.7	3.97
7	9.49	303.6	3.55	282.6	3.95
8	8.41	302.8	3.74	287.9	4.04
9	8.41	302.9	3.52	288.4	4.01
10	8.42	303.1	3.26	288.5	3.96

Table 2 Tested Case Boundary Conditions. MN = Motive Nozzle; SN = Suction Nozzle; OUT = Outlet

$$P_{lift} = P_{OUT}\text{-}P_{SN} \qquad (4)$$

$$ER = \frac{\dot{m}_{SN}}{\dot{m}_{MN}} \qquad (5)$$

Finally, the efficiency (η) is defined according to Elbel and Hrnjak definition [3], as the ratio between the recovered work by the ejector and the maximum recoverable work:

$$\eta = \frac{\dot{W}_{rec}}{\dot{W}_{MAX,rec}} \qquad (6)$$

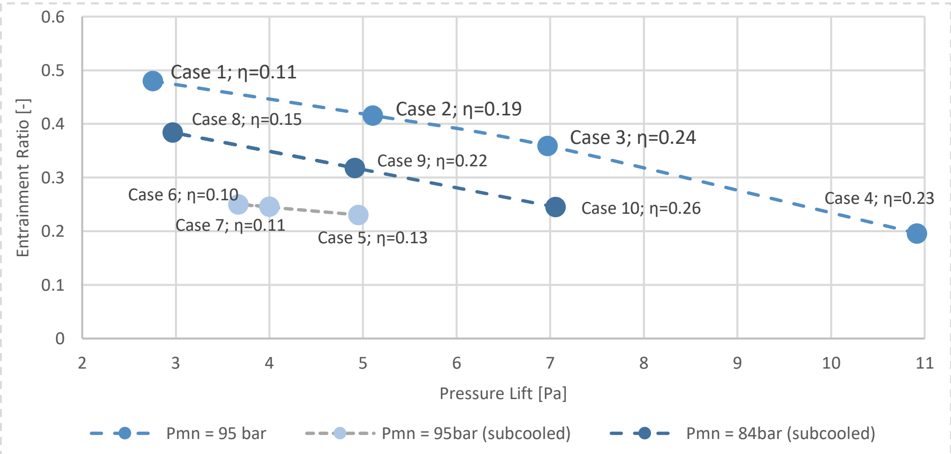

Figure 5 Experimentally obtained performance curves.

Each curve was experimentally obtained by adjusting the throttling valve before the suction nozzle of the Ejector. In facts by closing the throttling valve one can obtain a lower Entrainment Ratio, so the available energy at the motive nozzle inlet is used to increase the overall pressure lift instead of entraining the mass flow rate from the suction nozzle. These experimental results have also been compared to the numerical ones, as shown in the next section.

3.2 Numerical Results

A mesh sensitivity analysis has been conducted in order to find the independency of the computational model from the number of mesh elements. The tested meshes had respectively 25000, 50000 and 100000 elements and the resulting mass flow rates for Case 1 at both inlets are plotted in Figure 6. The mass flow rate remains almost constant by increasing the number of elements, but, in case of a 100000-element mesh the overall error on the Mass Imbalance results in a higher value (approx. $1x10^{-4}$). This is possibly due to the capability of the refined mesh to capture the flow instabilities nearby the diffuser shock leading to an increased mass imbalance error because of the steady state assumption made in the simulation, this was confirmed also by inspection of CFD results. Hence, the adopted mesh was the 50000 elements one, that showed low mass imbalance error combined to mass flow rates very similar to the most refined mesh.

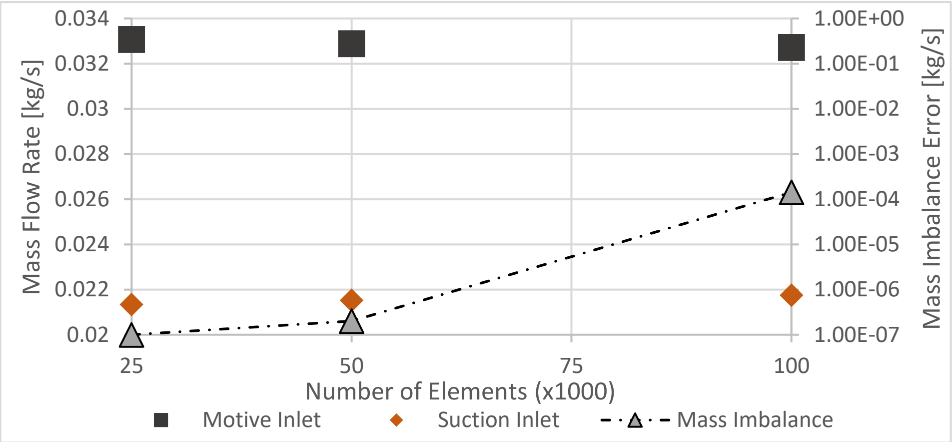

Figure 6 Mesh sensitivity for Case 1

One of the main criteria to evaluate the performance of the presented numerical model is the accuracy in the predicted mass flow rates. In Figure 77 the resulting mass flow rates at both motive and suction nozzle are shown and compared to the experimental results; the error between CFD and experiments is also plotted. The primary nozzle flow rate is better reproduced with a relative error between 12% and 19%. This discrepancy is possibly due to the inefficacy of the equilibrium model to predict the actual phase change phenomena, which resulted in an over-estimation of the vapor volume fraction and, hence, an under-prediction of the motive nozzle mass flow rate. The resulting suction mass flow rate from CFD calculations is considerably higher than experimental ones, which could be due to the aforementioned limitations of the computational model. Moreover, for these preliminary results, the swirl motion at the suction nozzle inlet has been neglected in favor of simplicity by considering an axis-symmetrical flow in the ejector. In the unchocked case (Case 4) a good agreement with experiments can be noticed.
The pressure profiles at the ejector wall are shown in Figure 8 and compared to the experimental results for Cases 1-4; the results of the remaining cases are totally similar. This comparison shows that the present model tend to place the shock position more downstream than the experimental value for all the chocked cases (Cases 1-3). However, the results at the wall are well predicted for Case 4 where the flow in the ejector mixer/diffuser is not in a chocked condition. The chocked/unchocked condition of the flow can be also seen in 9, where the pressure contours of Case 1 (chocked) and Case 4 (unchocked) are compared. As can be seen the shock-wave at the beginning of the diffuser is absent in Case 4 while the shock/expansion-waves at the motive nozzle exit have increased strength.
These results show that the present model has to be intended as a preliminary analysis tool because of its low computational demand and its applicability to different fluids (virtually any kind of pure fluid and/or mixture). A more detailed and precise flow description by means of computational fluid dynamics will require the development and validation of more advanced two-phase models for flashing R744 in supersonic ejectors.

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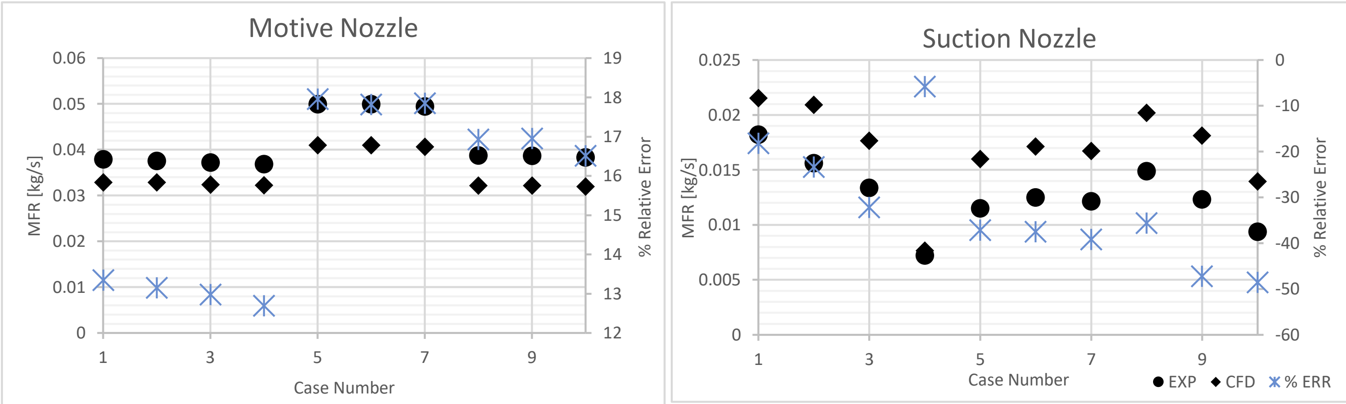

Figure 7 Mass Flow rate comparison between CFD and Experiments at both Motive and Suction Nozzles. (LEFT) Motive Nozzle; (RIGHT) Suction Nozzle.

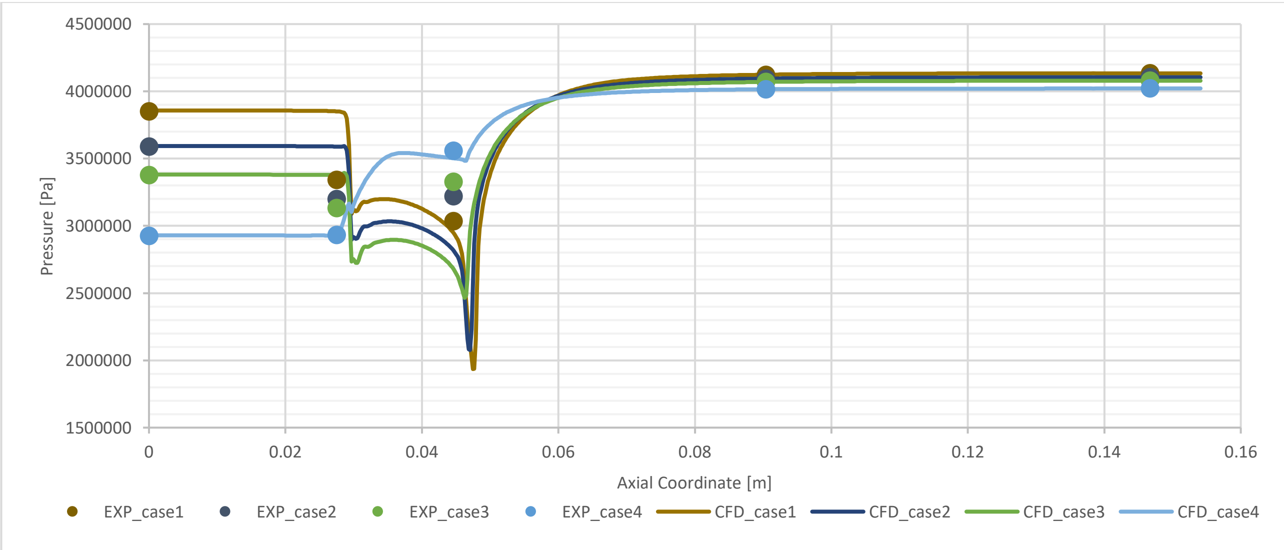

Figure 8 Pressure profile comparison at Ejector Wall

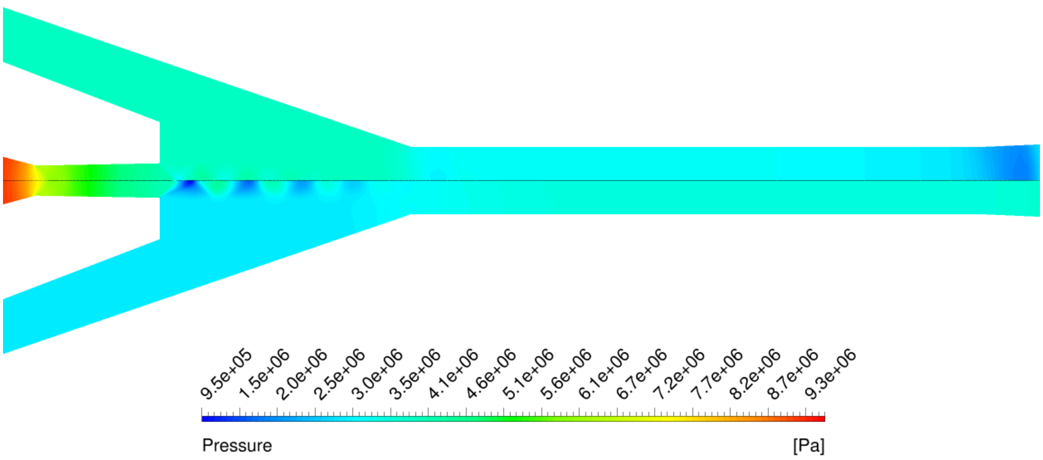

Figure 9 Pressure profile in Ejector Mixer/Diffuser. (TOP) Case 1; (BOTTOM) Case 4

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4. CONCLUSIONS

The use of two-phase flashing ejectors to improve the performance of CO_2 vapor compression systems has become an important research topic in recent years. In the present paper, the results obtained with a numerical model based on an in-house developed routines and implemented in the commercial software ANSYS Fluent are shown. These results are compared with experimental data from SINTEF/NTNU lab on a new geometry. The comparison has shown the capability of the Homogeneous Equilibrium Model to predict the main flow features of flashing ejectors, especially for the unchoked conditions, which are common in this application. The described computational approach may be a useful tool because, notwithstanding its limitations, it may be still a valid tool to quickly obtain first results, thanks to its low computational cost and its flexibility in terms of adaptability to different fluids.

ACKNOWLEDGEMENTS

The research leading to these results has received funding from the MIUR of Italy within the framework of PRIN2015 project «Clean Heating and Cooling Technologies for an Energy Efficient Smart Grid».

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