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Motive flow calculation through ejectors for transcritical CO_2 heat pumps. Comparison between new experimental data and predictive methods

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Abstract. The revival of CO_2 as refrigerant is due to new restrictions in the use of current refrigerants in developed countries, as consequence of environmental policy agreements. An optimal design of each part is necessary to overcome the possible penalty in performance, and the use of ejectors instead of throttling valves can improve the performance. Especially for applications as CO_2 HPs for space heating, the use of ejectors has been little investigated. The data collected in a cooperation project between ENEA (C.R. Casaccia) and Federico II University of Naples have been used to experimentally characterize several ejectors in terms of motive mass flow rate, both in transcritical CO_2 conditions and not. A statistical comparison is presented in order to assess the reliability of predictive methods available in the open literature for choked flow conditions.

1. Introduction

In order to contribute to the reduction of the global warming phenomenon, the latest F-Gas Regulation [1] will limit the use of high GWP refrigerant fluids in different sectors. Carbon dioxide is a very interesting alternative since it is non-flammable, nontoxic, inexpensive and eco-friendly, having GWP=1. Neska et al. [2] showed high energy efficiency in the sanitary hot water production using CO_2 as refrigerant, although the energy performance strongly decreases when the gas-cooler water inlet temperature increases. In this case, lower performances are mainly due to the throttle losses in the expansion process.

However, it is possible to improve the performance of the overall system using an ejector system as substitute of a common expansion valve. Minetto et al. [3], through a numerical simulation, highlight that only the use of an ejector system can improve carbon dioxide heat pump performance leading to COP values similar to R410A system. Elbel et al. [4] and Lucas and Koehler [5] show COP and exergy efficiency improvements up to 17% using an ejector system in transcritical refrigeration cycle. Furthermore, Lawrence and Elbel [6] pointed out that the performance of the ejector strictly affects the ejector cycle performance.

Lucas and Koehler [5] experimental study shows a maximum in ejector efficiency with respect to high-side pressure. Maximum ejector efficiencies of 22% have been measured. Furthermore, the ejector efficiency decreases with increasing gas cooler outlet temperature and decreasing evaporation pressure. The same study shows that the driving mass flow rate increases with increasing high-side pressure and decreases with increasing gas cooler outlet temperature at fixed high-side pressure. Lee et al. [7] and Liu et al. [8] experimental studies show that an optimum design configuration exists, in terms of throat diameter, which maximizes the performance of the system, for fixed operating conditions.

A fundamental variable for the ejector sizing is the motive (or driving) mass flow rate; its theoretical calculation is particularly complicated because:

- there are often transcritical conditions at the ejector inlet;
- there are always sonic velocity conditions at the throat, so the driving mass flow rate is independent of the downstream conditions and depends only on the nozzle inlet conditions, Lucas et al. [9];
- there is often two phase flow in the throat.

The processing of the experimental tests allowed to obtain several data on flow rate conditions through two ejectors of different diameter.

In this paper, experimental data have been compared to the predictions of mass flow rate through throttling devices obtained by three methods: the first one is a recent method proposed by Lucas et al. [9]; the second one is based on an HEM developed from Leung ([10] and [11]); the last one is the HNE-DS method, proposed by Diener et al. [12] and used in ISO/DIS4126 [13], for the two-phase mass flow rate calculation in chocked conditions through nozzles and safety valves.

2. Prediction Methods

The accurate prediction of the two-phase mass flow rate is a quite tricky issue, due to incomplete knowledge of the complex thermal-fluid dynamic phenomena occurring between the two phases. In particular, the following issues should take into account:

- the close interaction between vapour quality and sharp changes in pressure;
- possible thermodynamic non-equilibrium (metastable conditions);
- the potential different velocity of the two phases;
- the sound velocity in two-phase flow that can change quickly. Its calculation is still a matter of discussion.

The equation for calculating the dischargeable mass flow rate "G_r" through a throttling device having a geometric seat area "A" is:

$$G_r = k_d\, G_t\, A \tag{1}$$

where G_t is the theoretical flux in an ideal (isentropic) nozzle and k_d is the two-phase "discharge coefficient". The Homogeneous Models calculates G_t assuming that the two-phase mixture is homogeneous and liquid and vapour phases run at the same velocity inside the valve. Consequently, all the physical parameters are defined via their averages values weighted on the vapour quality. The Homogeneous models can be divided into two main groups, the "HEM", Homogeneous Equilibrium Models, and "HNE", Homogeneous Non-Equilibrium Models, depending on whether the thermodynamic equilibrium at the throttling device outlet is assumed or not.

HEM method estimates G_t (Leung [10] and [11]) by the following equation:

$$G_t = \frac{\sqrt{-\omega \ln(\eta) - (\omega - 1)(1-\eta)}}{\omega \cdot (1/\eta - 1) + 1} \cdot \sqrt{\frac{2 p_0}{v_0}} \tag{2}$$

where η is ratio between the outlet and inlet pressures p_{out}/p_0 and ω is a characteristic parameter, function of the inlet conditions only:

$$\omega = \frac{x_0 \left(v_{g,0} - v_{l,0}\right)}{v_0} + \frac{c_{pl,0} T_0\, p_0}{v_0} \cdot \left(\frac{v_{g,0} - v_{l,0}}{\Delta h_{v,0}}\right)^2 \tag{3}$$

Here the first term denotes the fluid compressibility at inlet conditions whilst the second one expresses the compressibility depending on the evaporation due to depressurisation. In choked conditions, in eq. (2) the pressure ratio η is replaced by the choked pressure ratio $\eta_c = p_c/p_0$ calculated by

$$\eta_c^{\,2} + \left(\omega^2 - 2\omega\right)\cdot\left(1-\eta_c\right)^2 + 2\omega^2 \cdot \ln\eta_c + 2\omega^2 \cdot \left(1-\eta_c\right) = 0 \tag{4}$$

This model has limits in the case of short orifice, (Fletcher [14], Fisher et al. [15]) even if they refer mainly to tests on tubes and nozzle.

The ISO method [13], derived by the previous HEM method, introduces a parameter "N" related to the boiling delay and depending on η_c and x_0. This parameter, that multiplies the second term of eq. (3), is calculated by:

$$N = \left(x_0 + c_{pl,0} T_0\, p_0 \cdot \frac{v_{g,0} - v_{l,0}}{\Delta h^2_{v,0}} \ln(1/\eta_c) \right)^a \tag{5}$$

N may assume values in the range; for high values the method tends to the HEM (for N=1 the equations become the same) whilst for low values the non equilibrium hypothesis prevails. The exponent *a* depends on the nozzle type and is 0.6 for short nozzle and orifices. N appears in the η calculation too and, therefore, the method requires an iterative procedure of calculation. Both methods can also be used for single phase flows but not in transcritical conditions.

Moreover, the ISO method is recommended when the inlet pressure is less than 50% of the thermodynamic critical pressure of the fluid. This suggestion depends on the fact that, in this pressure range, rapid changes in the physical properties of the fluids may occur which may cause errors on the calculation method. In the case of HEM, this recommendation is less strict because the method of calculation is different.

The CO_2 mass flow rate through an ejector can also be estimated by means of a method proposed by Lucas et al [9], obtained using the driving mass flow rates measured in an experimental campaign [16] on a multi-ejector heat pump using R744 as refrigerant.

In this method η_c, is calculated by

$$\eta_c = \frac{p_{th}}{p_{in}} = 0.0871942 \cdot \frac{p_{in}}{p_{cr}}^{0.9519907} \cdot \frac{\rho_{in}}{\rho_{cr}}^{2.348013} + 0.39387 \tag{6}$$

where p_{th} corresponds to the choked pressure.

Supposing an isentropic expansion in the converging driving nozzle, the mass vapour quality, the density and the enthalpy in the throat are calculated using the inlet entropy of the driving nozzle and the pressure at the throat obtained by (6).

With this method, the driving mass flow rate through an ejector is estimated by equations

$$u_{th} = \sqrt{2 \cdot (h_{in} - h_{th})} \tag{7}$$

$$G_t = A_{th} \cdot \rho_{th} \cdot u_{th} \tag{8}$$

This method can be used to predict the driving mass flow rate for every test conditions because it does not depend on the parameter like ω or on a rapid change in the physical proprieties of fluids.

3. Experimental setup

3.1 Test facility

The experimental data used in this paper were obtained testing a 30 kW CO_2 air-water heat pump equipped with a multi-ejector expansion pack as throttling device. The tests were performed in the

experimental facility “Calorimetro Enea” at ENEA (Casaccia) research center, a climatic chamber suited to test air-to-water reversible heat pumps with thermal capacity up to 50 kW, according to UNI EN 14511/2011 [17].
The experimental set consists of an alternative semi-hermetic compressor (CP) driven by an inverter, a plate heat exchanger (GC), a finned coil (EVAP), a plate internal heat exchanger (IHE), an electronic valve (EEV) and a multi-ejector expansion pack (EJEC). Figure 1 shows the multi-ejector CO_2 system including four different ejector geometries, with throat diameters from 0.7 mm to 2.0 mm.

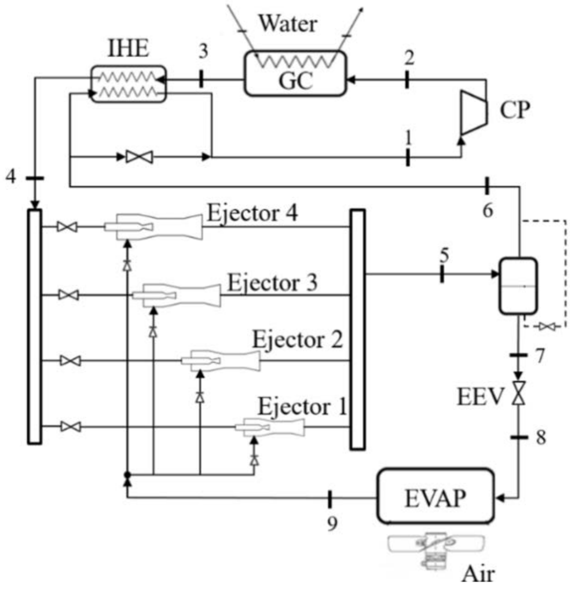

Figure 1. Layout of multi ejector CO_2 system

The heat pump control system can activate each ejector independently, depending on boundary conditions, with 16 different configurations; an ejector schematic is shown in figure 2. Temperature and pressure sensors are installed at the inlet and outlet of each component. K-type and J-type thermocouples were placed to measure the temperatures and piezoelectric transmitters to measure the pressure. The water volumetric flow rate is measured by an electromagnetic transmitter. The electrical power is measured by a wattmeter. All measurement instruments are characterized by high accuracy, according to [17].

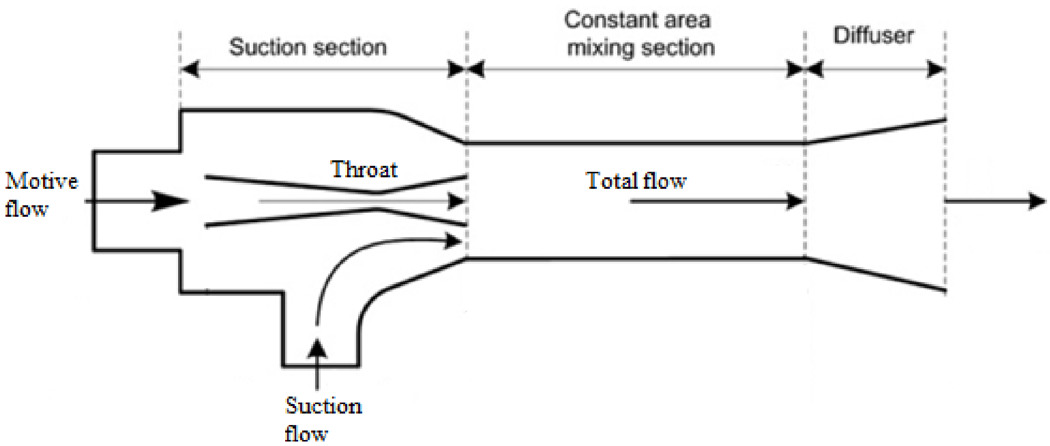

Figure 2. Schematic of the ejector system

Table 1 summarizes the instrument specifications and their uncertainties, as indicated by the manufactures.

Table 1. Measurement instrument and calibrated uncertainties.

Measurement	Range/Unit	Uncertainty
Temperature (K-type)	0/150 °C	± 1.1 K
Temperature (J-type)	-40/80 °C	± 1.1 K
Pressure	0-60/0-100/0-160 bar	0.08%
Water volumetric flow rate	0/200 l/min	0.02% of reading
Electrical power	0/25 kW	Precision class 0.5

3.2 Experimental procedure

All the tests were run according to [17]. The boundary conditions in terms of air temperature, relative humidity, water temperature and water mass flow rate were fixed and kept constant during the tests. Also, it was possible to set the outlet evaporator and inlet compressor superheating. Furthermore, it was possible to set manually the chosen ejectors configuration; for some test situations, two ejectors worked one by one. The trends of the thermodynamic measured variables were monitored via a software designed specifically. The main thermodynamic parameters were recorded and processed using Matlab software. The instabilities generated by the control system regulation are limited according to [17].

The motive mass flow rate has been calculated with an energy balance at the gas-cooler (the condenser of a CO_2 heat pump) with the equation:

$$G_{MF} = \frac{Q_{GC}}{h_{IN,GC} - h_{OUT,GC}} \tag{9}$$

The effect of variables uncertainties on the uncertainty of this calculation has been evaluated according to equation:

$$u_c(G_{PF}) = \sqrt{\left[\left(\frac{\partial G_{PF}}{\partial \dot{Q}_{GC}}\right)^2 \cdot u_c{}^2(\dot{Q}_{GC})\right] + \left[\left(\frac{\partial G_{PF}}{\partial \Delta h_{GC}}\right)^2 \cdot u_c{}^2(\Delta h_{GC})\right]} \tag{10}$$

and, for each test, it has been lower than ±7%.

4 Results and discussion

Table 2 shows the available experimental data subdivided by the inlet pressure range and the throat diameter.

Table 2. Experimental test subdivision

Throat ϕ [mm]		1.41	2
Not transcritical pressure conditions	$p<p_{cr}$	7	8
Transcritical pressure conditions	$p\geq p_{cr}$	18	12
Total		25	20

In NTr conditions, we have used Lucas correlations, HEM and ISO methods for calculating driving mass flow rate; with $p>p_{cr}$ we only able to use the Lucas correlations. For a proper comparison among the three methods, consistently to the isentropic expansion hypothesis assumed by Lucas, in (1) we set k=1.

In order to evaluate the prediction reliability of every model, a parameter *r*, defined as

$$r = \frac{G_t}{G_r} \tag{11}$$

(the ratio between the mass flow rate of the model and the actual mass flow rate), has been introduced; of course, the more *r* is close to 1, the more the prediction is good.

The figure 3 depicts *r* as a function of p_{in}; the different markers denote the predictions of three methods. The parameter *r* is better for the Lucas correlations and the values are almost constant with the p_{in}. HEM presents a similar trend with a *r* difference of almost 0.1 less. ISO predictions increase with p_{in} decreases; for the lowest p_{in}, *r* is the better.

Figure 4 shows p_c as a function of p_{in}. The p_c values are very important for mass flow rate prediction because it is present both in eq. (2) as η and in the Lucas correlations. In Figure 4 it is possible also to observe as Δp (difference between p_{in} and p_c) changes with p_c decreasing for the three methods: HEM and ISO show always Δp values between 10-15 bar, whilst Lucas correlations predict p_c values almost constant.

When p_{in} is lower than 60 bar, HEM and ISO p_c predictions are comparable with Lucas ones: in figure 3, for these p_{in} range, also *r* values are close each other. In particular, in the case of lowest p_{in}, corresponding to ISO and HEM p_c values lower than Lucas, ISO has the best *r*.

The first two figures indicate how the HEM method is good while ISO method gives acceptable results only with p_{in}< 60 bar, with a trend towards better performance with decreasing pressure. This behavior is consistent with the suggestion found in [13] even if p_{in} is far greater than to the suggested value (<50% p_{cr}, corresponding to < 36.88 bar for CO_2).

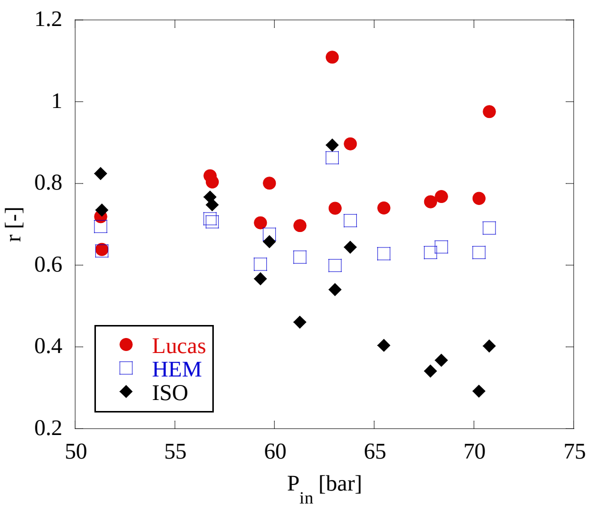

Figure 3. *r* calculated by Lucas, HEM, ISO methods vs. p_{in}

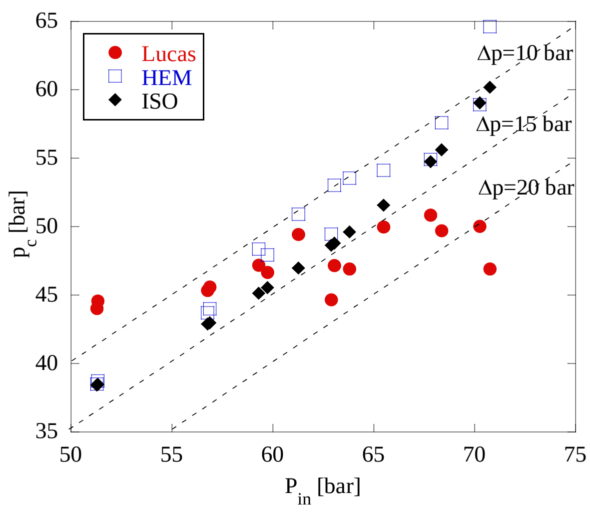

Figure 4. p_c calculated by Lucas, HEM, ISO methods vs. p_{in}

Hereafter we compare only HEM and LUCAS predictions. The vapour quality is an important parameter for evaluating the performance of a model in case of two-phase flow (§. 2). The development of the calculations with the correlations of Lucas involves x definition at the throat; even the HEM method estimates x, but it does this at inlet conditions and more generally, such as state of the fluid. Figure 5 depicts *r* as function of x calculated by Lucas; for the same points, the HEM predictions on CO_2 thermodynamic state are reported. Lucas correlations calculate x values until 0.18, and *r* increasing with x. For HEM, the flow is often subcooled, with *r* values almost constant, or saturated; in these cases, *r* increases with x.

By figure 6 we try to evaluate the difference between Lucas and HEM predictions as function of difference of the calculated p_c. The data indicate that p_c value is fundamental for method performance; the difference between *r* values decreases with p_c values difference.

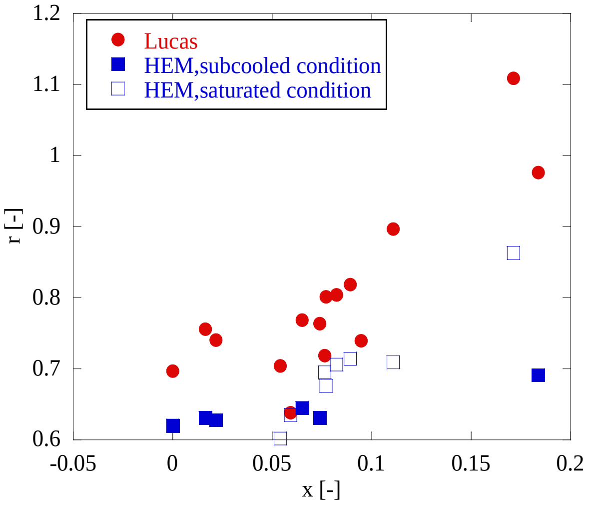

Figure 5. *r* calculated by Lucas and HEM methods vs. x

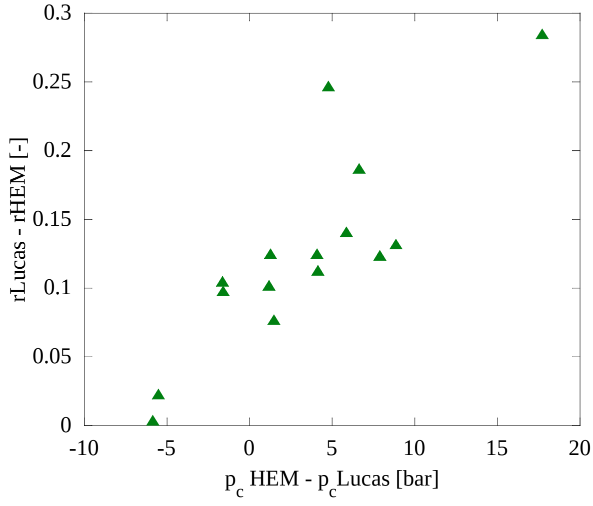

Figure 6. Difference *r* vs. difference p_c calculated by Lucas and HEM

In transcritical conditions, if we segregate the data by the throat diameters, it is possible to observe that the Lucas predictions depend on p_{in} and the predictions are as better as the throat is larger (figure 7); the *r* values are on average better than the not transcritical tests. In figure 8, it is interesting to notice the Lucas prediction behaviour in the whole range of inlet test pressure. Even when compared with HEM predictions (not transcritical tests), predictions by Lucas method do not show an evident dependence on p_{in}. This behaviour seems to be inconsistent with what previously suggested in (§.2) for the sonic conditions.

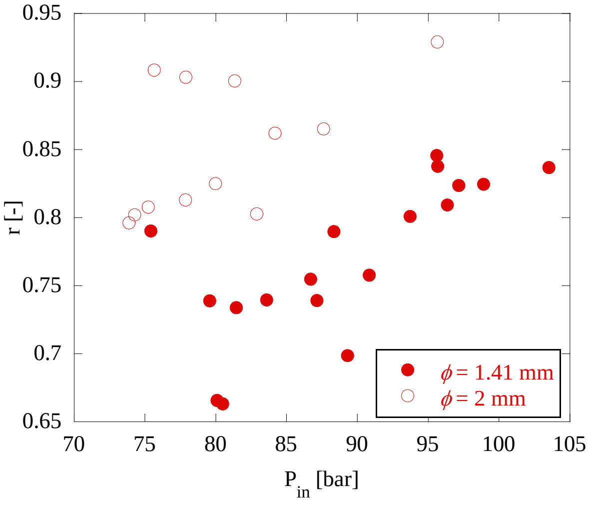

Figure 7. *r* calculated by Lucas method vs p_{in}

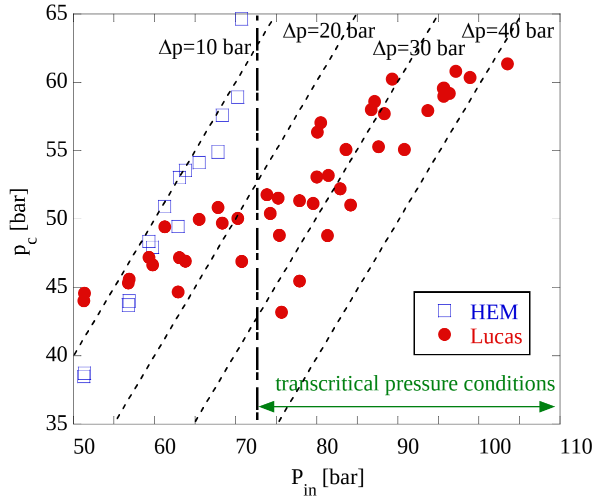

Figure 8. p_c calculated by Lucas and HEM methods vs. p_{in}

Figure 9 depicts r as function of x; we are not able to observe an r dependence on x. Instead, it is evident that tests with $\phi = 2$ mm have better predictions, as figure 7 shows.

In Figure 10, we evaluate the geometry influence (two different diameters) on the predictions of HEM models (NTr only) and Lucas as a function of the specific mass flow rate.

In the case of HEM predictions, we do not observe evident difference between the two diameters; instead, r Lucas values are better for 2 mm diameter orifice.

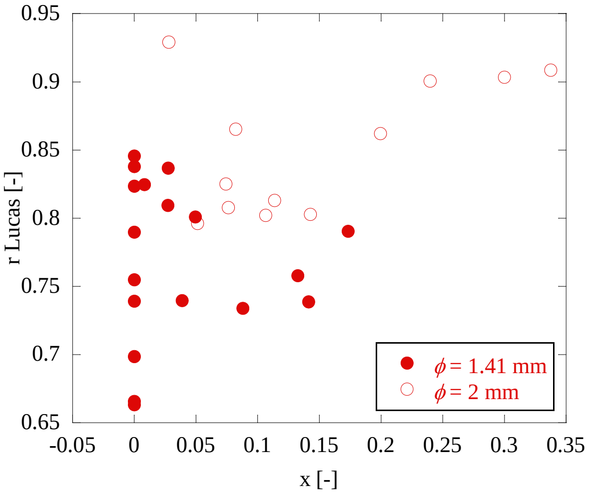

Figure 9. r calculated by Lucas method vs. x

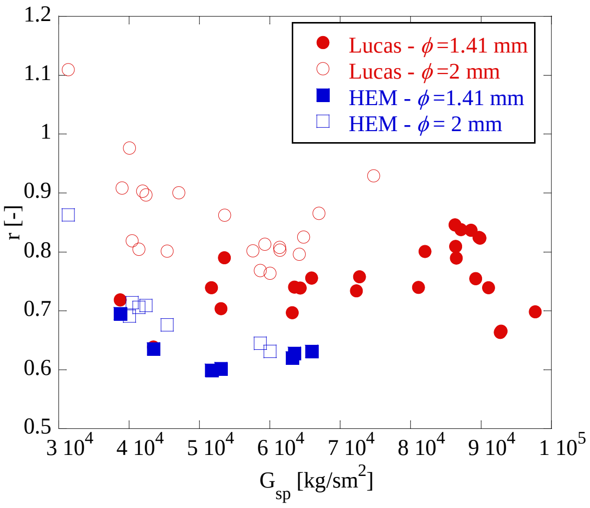

Figure 10. r calculated by Lucas and HEM methods vs. G_{sp}, parameter ϕ_{th}

5 Conclusions

In order to improve the knowledge about ejector performance, experimental tests were carried-out at the laboratory DTE-PCU-SPCT of the ENEA (Casaccia) research center. A complete heat pump system with multi-ejector pack was tested in a climatic chamber. Subcritical and transcritical motive nozzle inlet conditions, with high side pressure range between 50 bar and 105 bar, were investigated. Therefore, new experimental data in terms of motive nozzle mass flow rate were obtained. The results were elaborated using two existing prediction methods (HEM and ISO) and Lucas correlation [9], in order to find the best methods to evaluate the motive nozzle mass flow rate. We observed that:

- the calculation methods available for choked mass flow rate (HEM, ISO) cannot be used in transcritical conditions;
- Lucas correlations are useful at every pressure inlet conditions but, in same situations, its prediction reliability seems to be conditioned by the original database [9,16] (e.g. different performances for different geometries);
- HEM can be used in NTr condition and its performances are comparable with Lucas predictions
- ISO appears unusable for p>55 bar; under this pressure, it might operate better than the other methods.

Therefore, further studies and experimental tests are necessary to:

- ➢ Improve Lucas method with new experimental data
- ➢ Evaluate the possibility of application of HEM and ISO methods, both improving performance in not transcritical conditions and trying to expand the deployment in the transcritical range.

Acknowledgements

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Nomenclature

A	area	(m^2)	***greek letters***		
C	specific heat capacity	(J/kg K)	ϕ	diameter	(m)
G	mass flow rate	(kg/h)	ω	compressible flow parameter	(-)
h	specific enthalpy	(J/kg)	ρ	density	(kg/m^3)
K	discharge coefficient	(-)	$\Delta h_{v,0}$	latent heat of vaporization	(J/kg)
P	pressure	(bar)	η	pressure ratio	(-)
Q_{GC}	heating capacity	(kW)			

T	temperature	(°C)
u_c	uncertainty	(-)
u	velocity	(m/s)
v	specific volume	(m^3/kg)
x	mass vapour quality	(-)
r	prediction reliability	(-)

Abbreviations

NTr	Not transcritic pressure conditions
Tr	Transcritic pressure conditions
HEM	Homogeneous Equilibrium Model
HNE	Homogeneous Non Equilibrium Model
ISO	ISO/DIS4126-10 sizing method

subscripts

0	stagnation inlet conditions
c	choked (sonic) flow condition
cr	thermodynamic critical conditions
g	gas
GC	gas-cooler
in	inlet conditions
l	relative to the liquid phase
MF	motive mass flow
out	outlet conditions
r	actual
sp	specific
t	theoretical
th	throat conditions