Multi-scale performance evaluation of ejector refrigeration systems

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Abstract. Despite the many advantages, ejector refrigeration systems have not been able to penetrate the market because of two prevailing reasons: low coefficient of performance and relevant influence of ejector operation on the performance of the whole system. Indeed, the performance of ejector refrigeration systems depends on the local flow phenomena occurring within the ejector. Thus, improving the performance of ejector refrigeration systems relies on the understanding of the fluid dynamic phenomena at the "component-scale" and on integrating such information at the "system-scale". This paper contributes to the present discussion regarding the multi-scale modeling of ejector-based systems by proposing an integrated Computational Fluid Dynamic (CFD) - Lumped Parameter Model (LPM) ejector refrigeration system. In particular, ejector performances have been obtained by a validated CFD approach, whereas a LPM approach has modeled the refrigeration cycle. The refrigeration system's performances, for different boundary conditions, have been evaluated, and the effects of the "local-scale" on the "system-scale" have been commented.

1. Introduction
The ejector device is constituted by a primary nozzle, a suction chamber, a mixing chamber, and a diffuser (Figure 1a). A "high energy" primary flow accelerates and expands through the primary nozzle creating a low-pressure region at the nozzle exit; subsequently, the secondary flow is entrained in the mixing chamber because of the vacuum-effect mentioned above and the shear action between the primary and secondary flows. The primary and secondary flows mix within the mixing chamber and are later compressed in the diffuser, converting kinetic energy into static pressure [1]. The triple effect provided by the ejector (viz., entrainment/compression/mixing) makes it suitable to be employed in refrigeration systems (ERSs - Figure 1b, the forthcoming nomenclature refers to this figure). ERSs seems a promising alternative to the traditional compressor-based technologies due to their reliability, limited maintenance, low initial and operational costs, and no limitation concerning the working fluids [2]. In an ERS, the generator employs low-grade heat energy to produce the "high energy" primary flow supplied to the ejector inlet (point#1 in Figure 1b), which entrains the low-pressure secondary flow from the evaporator (point#2 in Figure 1b); subsequently, the mixed stream is conveyed to the condenser (point#3 in Figure 1b) where condensation takes place. Finally, the liquid is split: one part is sent back to the generator by a pump (point#6 in Figure 1b), whereas the other one proceeds towards an expansion
valve (point #5 in Figure 1b) and is supplied to the evaporator, thus providing the cooling effect. Relying on low-grade energy source as input, ERSs may reduce electricity consumption; unfortunately, they suffer from a low coefficient of performance (COP - in the range of 0.1 - 0.7 [1]) and a relevant influence of ejector operation on the performance of the whole system. Indeed, ejector refrigeration systems' performance depends on the local flow phenomena occurring within the ejector (ejector is a "fluid dynamic controlled device"). Therefore, the ejector performances (viz., the "component-scale", i.e., the ratio between the secondary and the primary mass flow rates – the entrainment ratio – as well as the pressure recovery) results from the fluid-dynamics interaction at the "local-scale" (viz., boundary layers subject to adverse pressure gradients, shock waves, over-expanded and under-expanded jets, flow separation, recirculation, turbulence mixing, …). For this reason, precise modeling of COP values (viz., the "system-scale" performances) relies on a complete knowledge of the "local-scale" and the "component-scale" performances and a multi-scale coupling of such information [3].

![Ejector design](image1)

(a) Ejector design

![Layout of an ejector refrigeration system (ERS)](image2)

(b) Layout of an ejector refrigeration system (ERS)

Figure 1. Ejector component (a) and ejector refrigeration system (b).

In the broader framework of ejector research, this paper contributes to the present-day discussion regarding multi-scale modeling approaches by proposing an integrated Computational Fluid Dynamic (CFD) - Lumped Parameter Model (LPM) of the ejector refrigeration system. In particular, a validated CFD approach solves the fluid-dynamics within the ejector ("local-scale") and provides the ejector performances ("component-scale"). Conversely, the refrigeration cycle ("system-scale") has been modeled by a LPM approach using input data CFD outcomes. In conclusion, the proposed model can consider the multi-scale relationships that determine the overall ERS performances.

2. Methods: benchmark and numerical modeling
In this section, the integrated Computational Fluid Dynamic (CFD) - Lumped Parameter Model (LPM) is outlined. First, the numerical validation benchmark is presented; subsequently, CFD and LPM methods are described.

2.1. Benchmark
The numerical model has been validated against an R134a ejector, described by Del Valle et al. [4], whose geometry and boundary conditions are presented in Figure 2 and Table 1. It should be noted that a constant 10°C superheating for both primary and secondary inlets (included in Table 1) has been prescribed accordingly to the experimental boundary conditions. In this study, both global (ω, the entrainment ratio; viz., the ratio between the secondary and the primary mass flow rate) and local (wall static pressure along with the ejector) measurements are available for complete validation.

![Figure 2. Ejector design from ref. [4]; dimensions in [mm].](image)

| Code Name | \(T_1 [^\circ C]\) | \(P_1 [kPa]\) | \(T_2 [^\circ C]\) | \(P_2 [kPa]\) | \(P_3 [kPa]\) | \(\omega_{Exp}\) |
|-----------|---------------------|---------------|---------------------|---------------|---------------|-----------------|
| Case#1    | 84.2                | 2,330         | 20                  | 415           | 650           | 0.592           |
| Case#2    | 100.03              | 3,190         | 15                  | 350           | 690           | 0.361           |
| Case#3    | 100.03              | 3,190         | 20                  | 415           | 690           | 0.433           |

2.2. Numerical modeling
The finite volume code ANSYS Fluent (Release 2020 – RI) has been used to solve the steady-state Reynolds Averaged Navier-Stokes (RANS) equations for the turbulent compressible Newtonian flow, employing \(k-\omega\) SST as turbulence model (as discussed in ref. [3]). A two-dimensional axisymmetric calculation has been performed. To limit the numerical diffusion, second-order upwind numerical schemes have been used for the spatial discretization, except for the pressure equation. In this case, PRESTO! scheme has been chosen since it is designed for flows involving steep pressure gradients. Second-order upwind schemes also for the turbulence model variables have been used. Gradients are evaluated by a least-squares approach. A two-step approach has performed the initialization: (i) a hybrid initialization followed by a (ii) full multi-grid (FMG) scheme. A pressure-based solver with a Coupled algorithm has been adopted since it is described by Croquer et al. [5] as far more stable than density based-solver nevertheless sufficiently accurate and suitable for high-velocity compressible flows. A pseudo-transient option was enabled, which was found to speed up the steady-state solution. R134a properties have been evaluated with the real-gas NIST database; indeed, Del Valle et al. [4] mentioned that the implementation of a real gas model is needed to correctly model non-ideal fluid phenomena under the refrigeration system boundary conditions. Ejector inlet boundary conditions are prescribed in terms of total pressure and temperature, while the turbulence boundary conditions have been implemented as hydraulic diameter and the turbulent intensity (5% for the primary flows and 2% for the second one) as described by Besagni and Inzoli [3]. The outlet condition has been modeled as a pressure boundary condition.

2.3. Multi-scale modeling
The above-described CFD model is used to solve the "component-scale" (viz., the entrainment ratio \(\omega\)) and the fluid-dynamics ("local-scale") of the tested ejector. Once such information is derived, the ejector component is then included within ERS, modeled by a LPM approach, to estimate the "system-scale"
performances. The advantage of such an approach is to take into account the fluid dynamics phenomena within the cycle performances. In particular, LPM input data concern (i) \( P_1, T_1 \) (comprehending a 10°C superheating indicated as \( \Delta T_{sh} \)), \( P_3, T_2 \) (comprehending a 10°C superheating indicated as \( \Delta T_{sh} \)) and \( P_3 \) (Figure 1a), which act as boundary conditions for the CFD simulations and (ii) the mass flow rates \( \dot{m}_1 \) and \( \dot{m}_2 \) and \( T_1 \), which are the CFD model output. To model ERS, the prevailing assumption concerns the absence of pressure losses within the system so to relate ejector boundary conditions to the cycle pressure and temperature levels (Figure 1b). The LPM approach is presented in the forthcoming equations, where sub-scripts gen stands for generator, evap for the evaporator while cond for the condenser. The saturated vapor is indicated by subscript sv, while saturated liquid by sl. Finally, apex is stands for isentropic. First, pressure values are computed as follows:

\[
P_1=\dot{P}_6=\dot{P}_{\text{gen}} \quad (1)
\]

\[
P_2=\dot{P}_5=\dot{P}_{\text{evap}} \quad (2)
\]

\[
P_3=\dot{P}_4=\dot{P}_{\text{cond}} \quad (3)
\]

Based on the CFD boundary conditions, points #1 and #2 (Figure 1b) are characterized as follows:

\[
T_1=T_{sv}(P_1)+\Delta T_{sh} \quad (4)
\]

\[
h_1=h(P_1,T_1) \quad (5)
\]

\[
s_1=s(P_1,T_1) \quad (6)
\]

\[
T_2=T_{sv}(P_2)+\Delta T_{sh} \quad (7)
\]

\[
h_2=h(P_2,T_2) \quad (8)
\]

\[
s_2=s(P_2,T_2) \quad (9)
\]

Point #3 (Figure 1b) is computed from CFD outcomes as ejector outlet temperature \( (T_i) \) results from the mixing of primary and secondary streams.

\[
h_3=h(P_3,T_3) \quad (10)
\]

\[
s_3=s(P_3,T_3) \quad (11)
\]

Point #4 (Figure 1b) is computed based on a saturated liquid assumption:

\[
P_4=\dot{P}_{\text{cond}} \quad (12)
\]

\[
T_4=T_{\text{cond}} \quad (13)
\]

\[
h_4=h_{sl}(P_{\text{cond}}) \quad (14)
\]

\[
s_4=s_{sl}(P_{\text{cond}}) \quad (15)
\]

Point #5 (Figure 1b) is computed considering an isenthalpic transformation in the expansion valve:

\[
h_5=h_4 \quad (16)
\]

\[
T_5=T_{\text{evap}} \quad (17)
\]

\[
s_5=s(P_5,h_5) \quad (18)
\]

Point #6 (Figure 1b) is computed assuming the isentropic efficiency of the pump \( \eta^i_{\text{pump}} = 0.9 \) [6]:

\[
h_6^i=h(P_6,s_4) \quad (19)
\]

\[
h_6^i=h_4+\frac{h_5-s_4}{\eta_{\text{pump}}} \quad (20)
\]

\[
T_6=T(P_6,h_6) \quad (21)
\]

\[
s_6=s(P_6,h_6) \quad (22)
\]

After all thermodynamic points of the ERS cycle are defined, the cooling power \( \dot{Q}_{\text{evap}} \), input thermal power \( \dot{Q}_{\text{gen}} \) and electrical power required by the pump \( \dot{W}_{\text{pump}} \) are computed using as input the mass flow rates \( \dot{m}_1 \) and \( \dot{m}_2 \), which are CFD model output:

\[
\dot{Q}_{\text{evap}}=\dot{m}_2(h_2-h_3) \quad (23)
\]
\[
\dot{Q}_{\text{gen}} = \dot{m}_1 (h_1 - h_6) \\
W_{\text{pump}} = \dot{m}_1 (h_6 - h_4)
\]  

Finally, cycle COP is computed as follows:

\[
\text{COP} = \frac{\dot{Q}_{\text{evap}}}{\dot{Q}_{\text{gen}} + W_{\text{pump}}} = \frac{h_2 - h_5}{h_1 - h_4}
\]

3. Results

This section presents and discusses the numerical results. First, the CFD approach is validated against experimental data; subsequently, CFD outcomes are coupled to the LPM approach.

3.1. Validation and local-scale fluid dynamics

The CFD approach has been validated considering both global (\(\omega\), the entrainment ratio) and local (wall static pressure profiles) quantities. Indeed, as stated by Hemidi et al. [7], a validation based upon "component-scale" data only may be misleading as the same entrainment ratio could emerge from different local flow structures. From a practical point of view, CFD and experimental data are compared based on (i) the entrainment ratio relative error (Equation (29)) and (ii) the mean absolute error between computed and experimental wall static pressure data (Equation (30)).

\[
\text{Relative error} = \frac{\omega_{\text{CFD}} - \omega_{\text{EXP}}}{\omega_{\text{EXP}}}
\]

\[
\text{Mean absolute error} = \frac{1}{n} \sum \left| \frac{P_{\text{CFD}} - P_{\text{EXP}}}{P_{\text{EXP}}} \right|
\]

Where \(n\) is the number of data points. The validation procedure outcome is summarized in Table 2, whereas Figure 3 displays wall pressure profiles. CFD outcomes show a fair agreement compared with the experimental data: entrainment ratios have been predicted with a maximum relative error equal to 17.4\%, whereas the mean absolute error for the local pressure profiles is lower than 10\%. It is worth noting that the CFD approach has been able to correctly predict the sudden pressure drop within the mixing chamber, as well as the pressure rise caused by the shockwave at the diffuser inlet.

| Code Name | \(\omega_{\text{Exp}}\) | \(\omega_{\text{CFD}}\) | Relative error [%] | Mean absolute error [%] |
|-----------|----------------|----------------|------------------|-----------------|
| Case#1    | 0.592          | 0.695          | 17.4\%           | 4.80\%          |
| Case#2    | 0.361          | 0.421          | 16.6\%           | 7.95\%          |
| Case#3    | 0.433          | 0.495          | 14.3\%           | 9.13\%          |

3.2. ERS performances and multi-scale perspective

ERS performances have been computed based on the modeling approach described in Section 2.3. It is worth noting that COP is related to \(\omega\) (Equation (28)), thus linking ejector and system performances. The outcome of the ERS performances, in terms of COP and cooling power, are listed in Table 3, whereas the details of the different cases are displayed in Figure 4, Figure 5, and Figure 6.

| Code Name | \(\omega\) [-] | COP [-] | \(\dot{Q}_{\text{evap}}\) [kW] |
|-----------|---------------|---------|----------------|
| Case#1    | 0.695         | 0.594   | 3.76           |
| Case#2    | 0.421         | 0.345   | 3.05           |
| Case#3    | 0.495         | 0.413   | 3.65           |
(a) Case#1 \((T_1 = 84.2^\circ C, P_1 = 2,330 \text{ kPa}, T_2 = 20.0^\circ C, P_2 = 415 \text{ kPa}, P_3 = 650 \text{ kPa})\)

(b) Case#2 \((T_1 = 100.0^\circ C, P_1 = 3,190 \text{ kPa}, T_2 = 15.0^\circ C, P_2 = 350 \text{ kPa}, P_3 = 690 \text{ kPa})\)

(c) Case#3 \((T_1 = 100.0^\circ C, P_1 = 3,190 \text{ kPa}, T_2 = 20.0^\circ C, P_2 = 415 \text{ kPa}, P_3 = 690 \text{ kPa})\)

**Figure 3.** Wall pressure profiles for the tested cases (Table 1).

It is noted that the highest cycle performance \((COP = 0.594)\) has been obtained with Case#1, which is characterized by the lowest \(T_1\) (see the boundary conditions in Table 1). Comparing Case#2 and Case #3, the latter has higher \(COP\), owing to the higher \(P_2\). It is worth noting that CFD capability to solve the local flow phenomena (the "local-scale") allows us to understand the prevailing reasons behind these observations. In particular, the primary jet's expansion affects the available cross-section area to entrain the secondary flow (the so-called entrainment duct). An increase in \(T_1\) (viz., \(P_1\), as the superheating has been kept constant and equal to 10°C for all the three tested cases) promotes the primary flow expansion, thus resulting in the reduction of the available cross-section area to entrain the secondary flow, with the consequent decrease of \(\omega\) and thus \(COP\), in agreement with ref. [8]. Higher \(P_1\), on the other hand, would result in a higher critical pressure (viz., the outlet pressure for which ejector changes from on-design to off-design operation mode, causing a rapid decrease of \(\omega\) [1]), extending the ejector operating curve. Increasing \(T_2\) reduces the primary flow expansion and, for fixed primary flow conditions, it improves \(\omega\) and \(COP\); this is made clear by comparing Case#2 and Case#3 (\(\omega\) increased by 17.6%, \(COP\) increased by 19.7%).
(a) $P-h$ representation  
(b) $T-s$ representation  
(c) Mach contour  
(d) Pressure contour  

**Figure 4.** Case#1 ($T_1 = 84.2^\circ$C, $P_1 = 2,330$ kPa, $T_2 = 20.0^\circ$C, $P_2 = 415$ kPa, $P_3 = 650$ kPa).

(a) $P-h$ representation  
(b) $T-s$ representation  
(c) Mach contour  
(d) Pressure contour  

**Figure 5.** Case#2 ($T_1 = 100.0^\circ$C, $P_1 = 3,190$ kPa, $T_2 = 15.0^\circ$C, $P_2 = 350$ kPa, $P_3 = 690$ kPa).
4. Conclusions
A multi-scale integrated Computational Fluid Dynamic (CFD) - Lumped Parameter Model (LPM) of the ERS has been presented. The integrated model relies on a CFD approach to solve the ejector fluid-dynamics (viz., the "local-scale") and the ejector performance (viz., the "component-scale"). Conversely, the ERS system is instead modeled by an LPM that evaluates the cycle performances (viz., the "system-scale") based on CFD outcomes. The advantage of such an approach is to relate the "local-scale" to the "system-scale", which has been proved by the herein application of the proposed multi-scale model to a R134a driven ejector.

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6. References
[1] Aidoun Z, Ameur K, Falsafioon M and Badache M 2019 Inventions 4 15
[2] Besagni G, Mereu R and Inzoli F 2016 Renew. Sust. Energ. Rev. 53 373-407
[3] Besagni G and Inzoli F 2017 Appl. Therm. Eng. 117 122-44
[4] García Del Valle J, Sierra-Pallares J, García Carrascal P and Castro Ruiz F 2015 Appl. Therm. Eng. 89 795-811
[5] Croquer S, Poncet S and Aidoun Z 2016 Int. J. Refrig. 61 140-52
[6] Fang Y, Croquer S, Poncet S, Aidoun Z and Bartosiewicz Y 2017 Int. J. Refrig. 77 87-98
[7] Hemidi A, Henry F, Leclaire S, Seynhaeve J M and Bartosiewicz Y Appl. Therm. Eng. 2009 29 1523-31
[8] Chunnanond K and Aphornratana S 2004 Appl. Therm. Eng 24 311-22