Future perspectives in ejector refrigeration

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FUTURE PERSPECTIVES IN EJECTOR REFRIGERATION

Adriano Milazzo[1*], Federico Mazzelli[1]

[1]Department of Industrial Engineering, University of Florence, Florence, 50139, Italy

Abstract: Ejector chillers may represent a competitor for absorption chillers, as soon as their cost per unit cooling power becomes equal or lower. This target is not far from our present achievements. If input energy is waste or renewable heat, the system operating cost is mainly due to the investment in heat exchangers. Therefore, a cost reduction requires an increase of COP. This latter may be improved by a careful design of the ejector, which requires a deep insight into the thermodynamics and fluid-dynamics of its complex physics. The tools and the knowledge for an advanced design are already available and the improvement potential is significant.

Keywords: ejectors, refrigeration, CFD

Nomenclature

a	speed of sound [m s^{-1}]
c	velocity [m s^{-1}]
D	diameter
f	friction factor
h	specific enthalpy [kJ kg^{-1}]
HTC	heat transfer coefficient [W m^{-2} K^{-1}]
$\dot{m}$	mass flow rate [kg s^{-1}]
M	Mach number
Q	heating/cooling power [kW]
T	temperature [°C]
U	averaged velocity [m s^{-1}]
W	power [kW]

Greek symbols

ρ	density [kg m^{-3}]
τ	shear stress [kPa]
ω	entrainment ratio
Ω	section [m^2]

Subscripts

A	condenser exit
C	condenser entrance
$cond$	condenser saturation
E	evaporator exit
eva	evaporator saturation
f	cooling
G	generator exit
gen	generator saturation
m	motive
p	primary
s	secondary

1 Background

Supersonic ejectors are compression devices without moving parts, where a motive (“primary”) flow exchanges momentum and mixes with an entrained (“secondary”) flow. Ejectors can be used for a wide range of purposes, i.e. aeronautic propulsion, seawater desalination, suction of non-condensable gases in steam plants and compression of working fluid in refrigeration systems. This latter application, probably the most demanding in terms of efficiency, is our research topic. Any improvement in ejector efficiency would benefit the other ejector markets as well. Before any further statement, the basic constraints that affect a heat powered refrigeration system based on an ejector need to be pointed out.

1.1 Thermodynamic constraints

Any heat powered refrigeration system must comply with the efficiency limit given by a fully reversible

* **Corresponding author:** Adriano Milazzo, Email: adriano.milazzo@unifi.it

combination of heat engine and refrigerator. For example, given a hot source at T_G = 80°C, a cold source at T_E = 5°C and a heat sink at T_C = 35°C, the ideal heat engine would have an efficiency of 0.127 and the ideal refrigerator a COP of 9.27, yielding a combined COP_{max} = 1.18. Clearly, this modest performance is entirely due to the low temperature of the hot source that impairs the efficiency of the motive cycle. Therefore, any comparison between heat and electrically powered refrigerators should account for this thermodynamic limit.

Similarly, comparing the COP of heat powered refrigeration systems with different temperature levels of the heat sources may be misleading. A Second Law efficiency, i.e. the ratio between the measured COP and that of a fully reversible cycle at the same working conditions, should always be used in order to correctly evaluate the real efficiency and the improvement potential of any thermal system.

When the heat powered refrigeration system uses a working fluid that undergoes phase-changes, the thermodynamic cycle crosses the two-phase zone (Fig. 1b) and the working fluid, on the left hand side of the cycle, is liquid or low quality two-phase mixture. This makes energy recovery on the left side of the refrigeration cycle (downward from point A) unpractical. Hence, the work that could be produced on expansion is normally dissipated in a valve, as in most refrigeration systems. On the other hand, the engine cycle has a very low work input on the left side, where a pump (upward from point A) returns the working fluid from the ambient heat sink to the hot source.

Probably the ejector chiller, shown in Fig. 1a, is the simplest possible scheme of heat powered refrigerator. The corresponding thermodynamic cycle, shown in Fig. 1b, is referred to water as the working fluid. Its Coefficient Of Performance may be calculated as follows:

$$COP = \frac{Q_f}{Q_{in} + W_{pump}} = \omega \frac{h_E - h_A}{h_G - h_A} \qquad [1]$$

where Q_f is the cooling power, Q_m is the motive heat power and W_{pump} is the generator feed pump power.

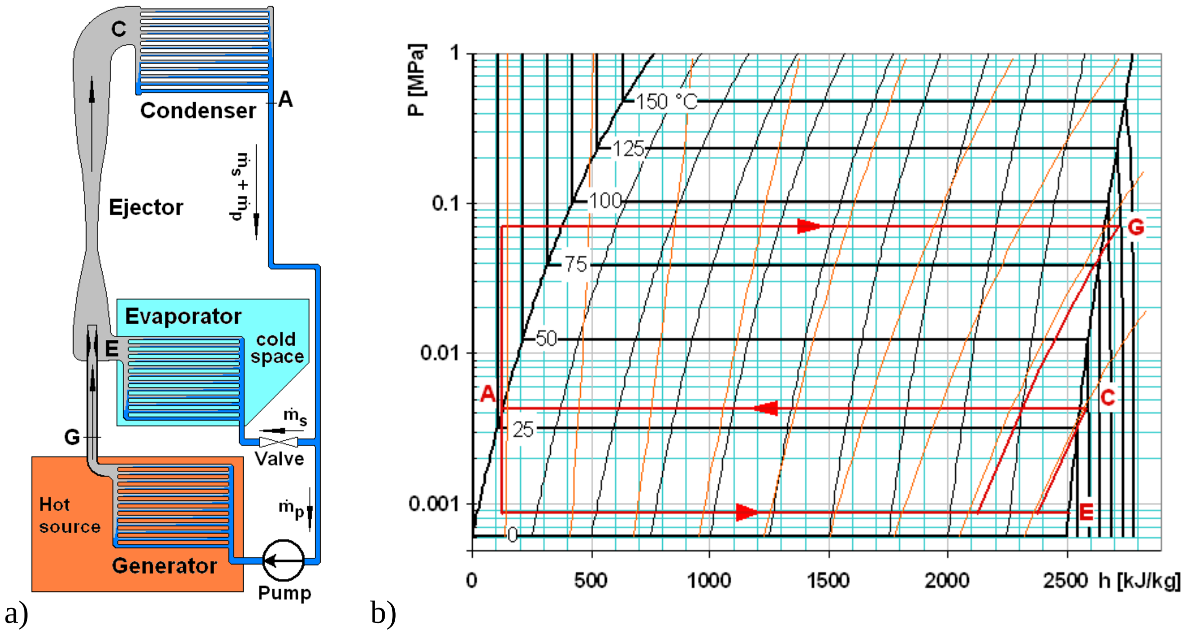

Figure 1 – Ejector chiller scheme (a) and ideal thermodynamic cycle (b) for water

The last part of Eq. 1 contains the entrainment ratio $\omega = \dot{m}_s / \dot{m}_p$ between secondary and primary mass flow rates and a further ratio between the enthalpy changes experienced by the working fluid passing through the evaporator $(h_E - h_A)$ and through generator and pump $(h_G - h_A)$.

This ratio depends on the fluid and operating conditions. Therefore, the global performance for fixed fluid and conditions depend on the ejector entrainment ratio. This latter may be evaluated from an energy balance on the ejector and turns out to be:

$$\omega = \frac{h_G - h_C}{h_C - h_E} \quad [2]$$

i.e. the ratio between motive $(h_G - h_C)$ and compression $(h_G - h_E)$ enthalpy differences. Reduced ejector losses decrease the enthalpy at condenser entrance h_C, increasing numerator and decreasing denominator in Eq. 2.

A simple calculation for an ejector cycle including the iso-enthalpic valve as the sole irreversibility gives the results shown in Fig. 2. The working fluid is water. The trends are rather obvious: performance increases with the hot and cold source temperature and decreases with the ambient temperature. The Second Law efficiency, i.e. the ratio between the COP of this cycle and that of a fully reversible heat powered refrigeration cycle (COP_c) is fairly constant and close to unity, showing that the throttling loss is scarcely harmful in the case of water. If however superheating is necessary, as could be the case for a "wet expansion" fluid if condensation within the ejector is to be avoided, the pressure at the generator, for a fixed maximum cycle temperature, must be lowered. Hence in this case, as shown in the last diagram of Fig. 2, the cycle COP and the Second Law efficiency both decrease severely.

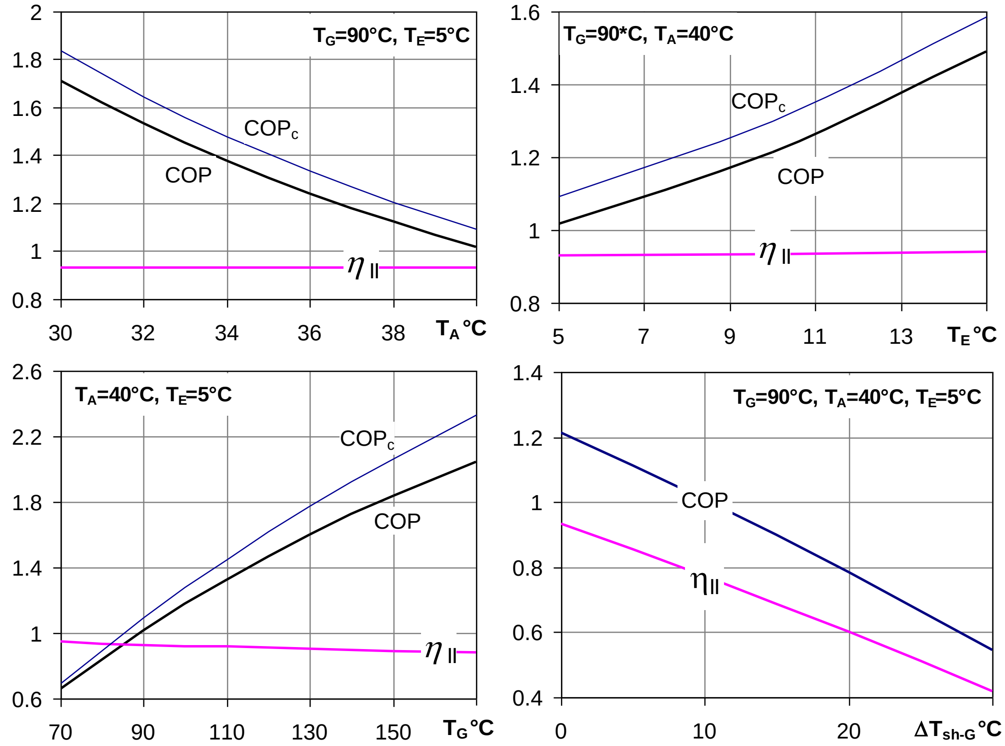

Fig. 2 – Performance of an ideal ejector cycle

1.2 Selection of the working fluid

In the past, the motive fluid in all ejectors was invariably steam, whatever the entrained flow.

Refrigeration systems, instead, use the same fluid as motive and entrained flow. The working fluid may be a refrigerant, as far as its properties allow a convenient operation at the hot source temperature and pose no environmental or safety problem. Recent works on comparative evaluation of working fluids for ejector chillers have been published by Kasperski and Gil [1], who concentrated on hydrocarbons showing that R600a yields good performance, and Varga et al. [2], who confirmed the validity of this fluid. Wang et al. [3] compared fluorocarbons, hydrocarbons and ammonia, concluding that the latter yields the highest COP. Chen et al. [4] compared R134a, R152a, R245fa, R290, R600, R600a, R1234ze, R430A and R436B, accounting for the effect of superheating of the primary flow. According to their simulation, R245fa and R600 have the highest COP. It must be stressed that ejector chillers have a relatively large fluid charge compared to vapour compression cycles of equivalent capacity, because they combine a heat engine and a refrigeration system. This exacerbates potential risks if flammable and/or toxic fluids are used.

Our experimental work with R245fa described in ref. [5] confirms that this fluid gives good results, as will be shown later. However, its GWP = 950 does not comply with F-gas regulations. Therefore, a low GWP substitute should be used, e.g. R1233zd, as shown in ref. [6]. This fluid has very low GWP, negligible ODP and is non flammable. CFD simulations described in ref. [7] have shown that an ejector working with R1233zd would have the same performance as R245fa at equal boundary conditions in terms of pressures. As the saturation curve of R1233zd is slightly below that of R245fa, a moderate adjustment of the temperatures is necessary in order to have equal pressures. Otherwise, the ejector operating on the same temperature levels would feature a slightly higher COP and a slightly lower critical condenser temperature. Hence, these results indicate that the R1233zd could be a suitable "drop in" replacement for R245fa. However, a likely drawback of this and other newly formulated fluids is the high cost and questionable availability.

A favourable feature of both R245fa and R1233zd is their dry expansion. On the contrary, wet expanding fluid condense along the expansion and the liquid droplets may cause erosion problems. Furthermore, a more refined treatment of the flow is necessary in order to correctly design the ejector and predict its performance.

This is particularly true for steam, which has a very high latent heat. Even small quantities of condensed water completely change the thermodynamic state of the expanding fluid. Neglecting this point (e.g. using an ideal gas equation of state) yields physically inconsistent results, as shown in ref. [8]. On the other hand, the high latent heat of water yields a reduced fluid charge per unit cooling power. The very high critical point is another key feature, as it reduces the extent of the throttling loss.

According to the results published by Ma et al. in ref. [9], the Second Law efficiency of a steam ejector could reach 0.25. The low cost and absolute environmental safety of water allows a relative freedom in the system design, offering significant potential for optimization and adaptation to specific needs. For example, in an industrial plant with various cooling loads at different locations, the steam generator could be centralized near the heat source (e.g. internal combustion engine or gas turbine) and the steam be distributed to several chillers, each one comprising only an ejector, an evaporator and a condenser. This configuration would be much simpler than a series of steam-powered absorption chillers.

Icing may be a problem in a steam ejector chiller, but it may also turn out to be an opportunity. As shown by Eames et al. in ref. [10], an ice storage may be easily integrated in an ejector chiller in order to decouple the ejector from the cooling load oscillations.

Last but not least, water has a high liquid density and a low saturation pressure at common generator temperatures, allowing to feed the generator by gravity, as long as the condenser is placed at sufficient height with respect to the generator level as pointed out in ref. [11]. This would eliminate the pump, i.e.

the sole moving part in the circuit, and allow a complete sealing of the latter. Alternatively, the pump could be substituted by an injector, i.e. a jet device that uses a small amount of steam to pump the liquid back to the generator. This approach was very common on old steam engines and should consume a small amount of steam. The injector would be a simple, robust and economic device, having no moving parts and guaranteeing the circuit sealing.

2 Means for improving ejector competitiveness

2.1 Fluid dynamics

As above shown, once the fluid and the operating temperatures are fixed, the ejector chiller COP is a function of the ejector entrainment ratio. Therefore, even if a careful design of the heat exchangers is fundamental, a significant effort should be placed on the improvement of this parameter. The other performance parameter is the compression ratio, that qualifies the ejector chiller in terms of capability to work in hot climates. Clearly, there is a trade-off between these two parameters.

An attempt to improve the methods normally employed in ejector design was the Constant Rate of Momentum Change (CRMC) criterion, introduced by Eames in ref. [12]. Given a rate of deceleration (dc/dx), the CRMC gives a continuous profile for the ejector. The procedure may be written in terms of the classic Hugoniot equation, which may also be modified in order to account for friction by a suitable factor f and may be extended to real gases.

$$\frac{d\Omega}{dx} = (M^2 - 1)\frac{\Omega}{c}\frac{dc}{dx} + \frac{M^2}{2}\frac{\Omega}{D} f \qquad [3]$$

In this way, the total deceleration that the fluid must undergo along the ejector may be translated in a diffuser length. Note that in the consolidated ejector design process, this latter is simply set as a fixed multiple of mixing chamber diameter.

However, the CRMC criterion is not so straightforward when applied to the mixing zone, where two flows at widely different velocities coexist in the same cross section. The ejector may be seen as a momentum exchanger between the fast stream produced by the primary nozzle and the slow stream coming from the evaporator (Fig. 3). The direct interaction between these streams, without any interposed moving device, is a remarkable feature of ejectors. On the other hand, this process is inherently irreversible and causes a substantial portion of the entropy increase occurring within the whole ejector.

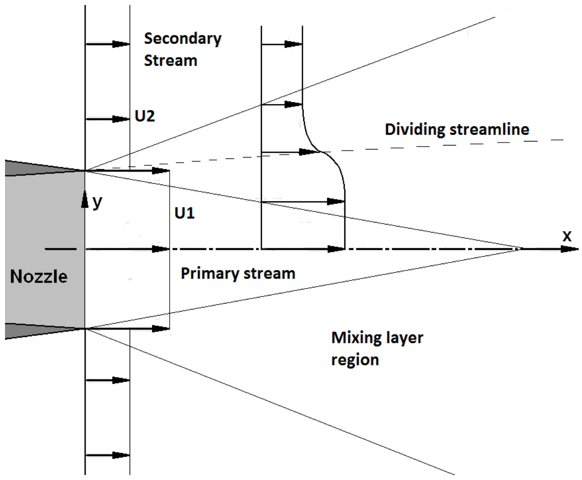

Fig. 3 – Scheme of the mixing process

In a recent work, ref. [13], we have applied previous results introduced by Papamoschou in refs. [14, 15] to devise a model of the ejector mixing layer. The model is able to compute all flow properties inside an axisymmetric or planar mixing chamber by applying the conservation equations on two control volumes that surround the primary and secondary streams.
The approach is based on the calculation of the mixing layer spreading rate and the maximum shear stress between the two streams, which is given by:

$$\tau_{max} = 0.07 \cdot (\rho_1 + \rho_2)\Delta U^2 \left[\frac{(1+\eta)(1+r)}{2(1+r\eta)} \right] \cdot \left(0.25 + 0.75 e^{-3M_c}\right) \qquad [4]$$

where $\eta = \sqrt{\rho_2/\rho_1}$, $r = U_2/U_1$, $\Delta U = U_1 - U_2$ and M_c is the convective Mach number, defined as:

$$M_c = \frac{\Delta U}{a_1 + a_2} \qquad [5]$$

Where a_1 and a_2 are the sound speeds of the primary and secondary streams.

In this way we have built a simple model of the mixing layer which has been validated against CFD results and used a design tool. A significant result was the convenience of enlarging the contact surface between the streams. For example, by splitting the primary flow rate among 4 nozzles the entrainment ratio would increase by 28%.
Further results from CFD simulations were presented in ref. [5]. CFD calculations are carried out on the commercial CFD package ANSYS Fluent v16.2, which is based on a finite volume approach. The working fluid is R245fa. The numerical scheme and computational domain adopted for the computations are summarized in Fig. 4.

Numerical scheme
2D axisymmetric domain
2° order accurate, density based
Structured grid, 80k elements
Wall resolved
Real thickness of the nozzle trailing edge
Real gas properties
Kω SST turbulence model

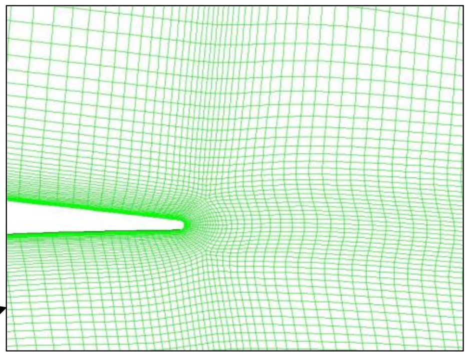

Figure 4: Numerical scheme and mesh characteristic for the CFD simulations

The main results of this analysis are condensed in Fig. 5, whose experimental uncertainties were calculated as described in ref. [5]. This chart shows the importance of the correct evaluation of the momentum and heat transfer at the ejector walls. The rightmost curve represents the numerical scheme with smooth and adiabatic walls that is commonly adopted by most of the studies in ejector research. Notably, while this scheme correctly reproduces the experimental entrainment ratio for the critical regime, the same model is far from being accurate at off-design conditions.

By introducing wall roughness, the results for the entrainment ratio change dramatically. In particular, the gold and light-blue curves correspond to roughness heights of 10 µm and 20 µm. Clearly, as the condenser pressure increases, higher values of friction cause the critical point to appear in advance. This result is indeed expected, as greater friction translates into larger amounts of total pressure losses, thus reducing the capability of the flow to withstand high values of back pressure.

The green and purple curve in Figure 5 represent two numerical schemes with 20 µm roughness height and two constant values of wall temperature. These are set equal to the condenser and ambient temperature respectively. Imposing a constant temperature along the external wall is clearly a simplification. Nonetheless, this approximate analysis shows that the lower the wall temperature, the higher becomes the critical pressure. Hence, a net heat loss toward the ambient produces a positive effect in terms of flow stability. Figure 6 illustrates the profiles of Heat Transfer Coefficient (HTC) at the ejector wall (a negative HTC represents heat flowing out of the ejector). As expected, the heat transfer increases with increasing surface roughness.

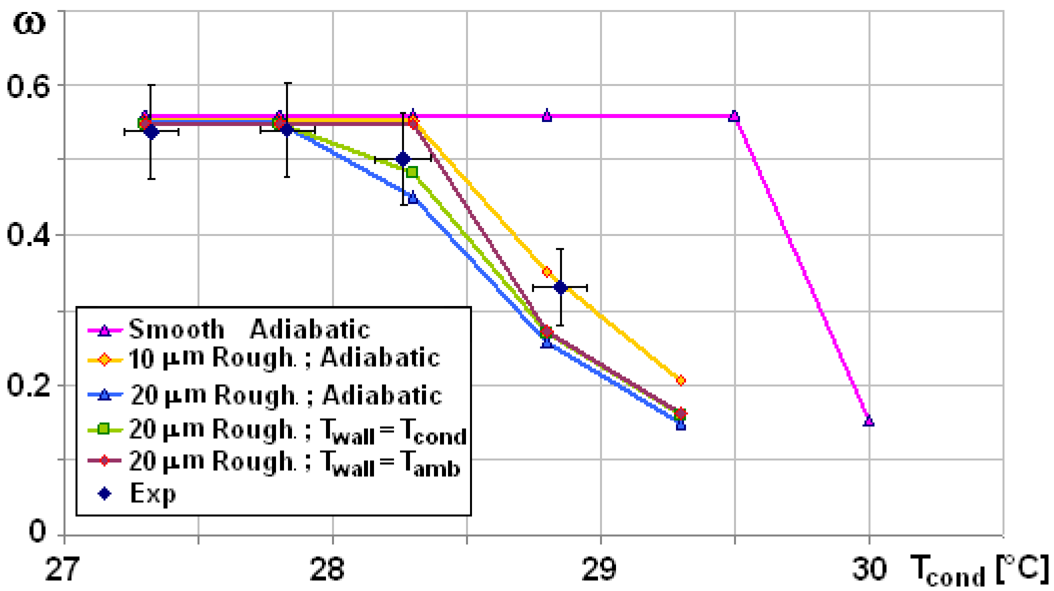

Fig. 5: Entrainment ratio profiles for different values of wall roughness and temperature; T_{gen}=89°C, T_{eva}=5°C

Figure 6 clearly shows that even in the most conservative case (i.e., smooth wall and wall temperature equal to the condenser temperature), the ejector surface cannot be considered adiabatic. Indeed, values like these are typical of liquids in forced convection. This may significantly impact the accuracy of the numerical simulations. Moreover, the heat loss toward the environment should be taken into account for a precise sizing of the condenser, being 2-4% of the total heat rejected at condenser. An experimental measurement of the wall temperature is planned for the near future.

These results suggest that CFD analysis needs some further insight. As soon as reliable CFD tools and powerful computational resources are available, a detailed optimization of the ejector geometry could be attempted and the benefits could surpass the expectations.

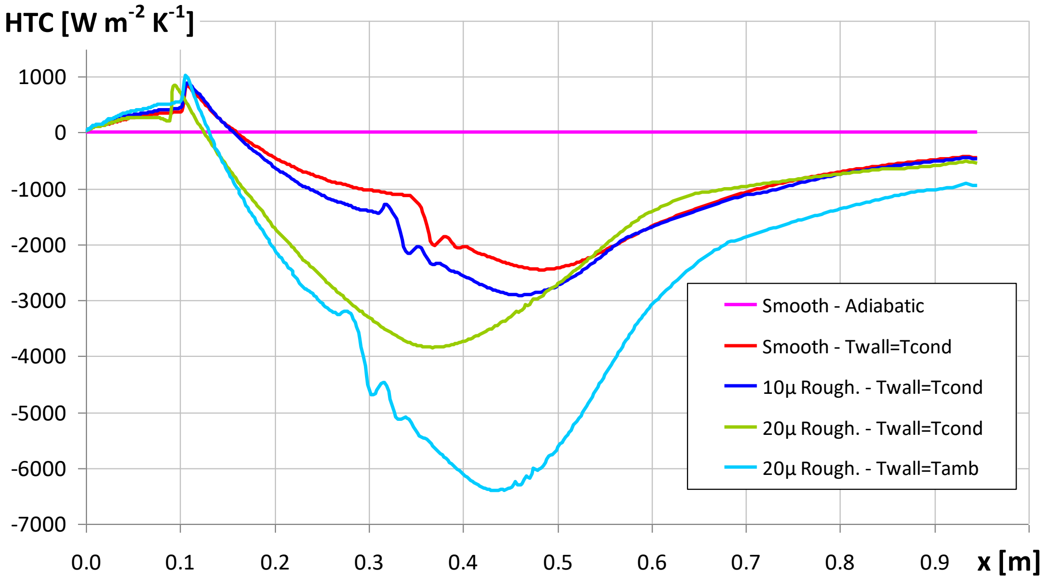

Fig. 6: Heat Transfer Coefficient profiles along the ejector for various cases with smooth and rough walls; T_{gen}=89°C, T_{eva}=5°C, T_{cond}=28.3°C

Clearly, if steam is used instead of R245fa or R1233zd, CFD analysis becomes more complex and less reliable, due to condensation phenomena that occur at supersonic speed. In this context. two-phase flow must be carefully modeled, accounting for metastable states, non-equilibrium condensation and liquid evaporation across shock waves [16, 17]. Finally, improvement of design tools and physic understanding of supersonic steam ejectors may be a good test-bench for further applications, e.g. two-phase ejectors for expansion work recovery in CO_2 vapor compression cycles [18].

2.2 Experimental activity

Other hints have been gained from the experimental activity performed in our lab and at Frigel S.p.a., an Italian Company that has teamed with our research group in the development of ejector chillers for industrial use.

For example, the prototype chiller shown in (Fig. 7) features a vertical layout, with condenser on top and generator at the bottom, that has proven very useful in order to guarantee a smooth operation of the feed pump, which could otherwise suffer from cavitation. The prototype has also shown the efficient and stable operation of an ejector designed according to CRMC criterion.

Saturation temperature at generator was set to 90°C and the following performance were reached:

- at 5°C saturation temperature at evaporator: COP above 0.4, entrainment ratio 0.5, Second Law efficiency 0.2 and critical condenser temperature around 28°C
- at 10°C °C saturation temperature at evaporator: COP above 0.5, entrainment ratio 0.7, Second Law efficiency 0.22 and critical condenser temperature around 29.5°C.

According to CFD simulations, if the ejector surface roughness were reduced from the present 20 µm to 2 µm, the critical temperature could be raised by approximately one degree.

As shown in Fig. 8, roughly half of the prototype cost was made up by the heat exchangers. Another significant share was held by the pump, which features a magnetic transmission, an electronic speed control and a multistage centrifugal design. The fluid cost is also quite significant.

Obviously, specifically designed components would decrease their cost if they were mass produced. On the contrary, heat exchangers and other minor components are already off-the-shelf products, and their size and cost may be reduced only by improving the system COP (i.e. reducing the heat input at generator and heat output at the condenser). In order to bring the system cost below 400 €/kW$_f$, the COP should reach 0.6. Meanwhile, the critical condenser temperature should be increased well above 30°C.

Fig. 7 – Detail of the prototype seen from the top

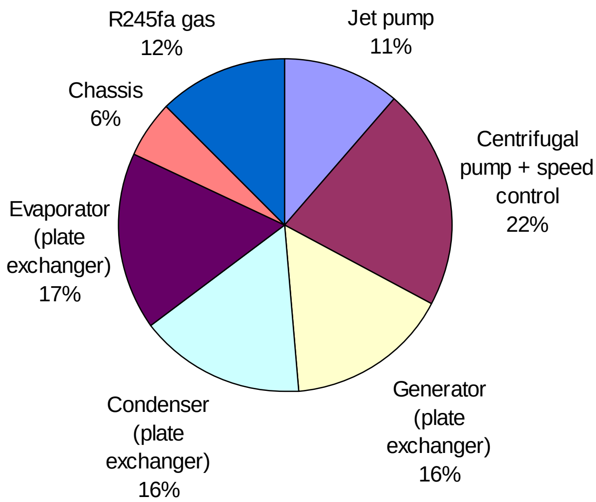

Fig. 8 – Costs of the main prototype components

If steam were used instead of R245fa, the fluid cost would be eliminated. Furthermore, the pump could be substituted by an injector, which should be very easy to manufacture (though probably rather difficult to design). On the other hand, the heat exchangers could turn out to be more expensive, given the high specific volume of steam at low temperatures. The physical size of the ejector would increase as well, but in this case a simple relation between size and cost is not available. Large size ejectors used in petrochemical industry or desalination are made of welded metal sheet and most likely have a relatively low cost per unit volume.

3. Conclusion

Ejector chillers may enter the market of heat powered refrigeration as soon as their cost per unit cooling power becomes equal or lower than that of absorption chillers. They may occupy market niches left vacant by absorption chillers and increase the market share of heat powered refrigeration systems.

However, market competitiveness of ejector chillers may be reached only after an increase of the system COP, which entails a reduction of heat exchangers size and plant costs. In this respect, ejector efficiency is key and our experimental results show that moderate improvements in ejector performance would produce a refrigeration system with low cost, robust operation and absolute environmental safety.

In this respect, the choice between steam and synthetic working fluids is still open. Steam should be preferred in the industrial environment, where fluid quantities are huge and fluid cost and safety is a major concern. Synthetic fluids could be an option for small systems powered by low temperature heat sources, e.g. for solar cooling of small buildings.

Moreover, further efforts must be addressed to develop adequate numerical and experimental tools to study details of the internal flow field and the source of thermodynamic irreversibilities. In particular, our work indicates that attention must be paid on the features governing the dynamics of mixing and boundary layers. Our results focus on these aspects and suggest that improvements may be reached by careful design of the mixing chamber, wall profiles and surface finishing.

Finally, improvement of design tools for a supersonic steam ejector with condensation and momentum exchange within a two-phase flow may be a good test-bench for further applications, e.g. two phase ejectors for expansion work recovery in vapor compression cycles.

Acknowledgements

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