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Theoretical and experimental activity on Ejector Refrigeration

Adriano Milazzo[(a)]*, Andrea Rocchetti[(a)], Ian W. Eames

[a]*Dipartimento di Ingegneria Industriale, Università di Firenze, via di S.Marta, 3, 50139 FIRENZE, ITALY*

Abstract

Ejector refrigeration has been studied at DIEF (Dipartimento di Ingegneria Industriale Firenze) since the '90s. Use of environmentally safe fluids (steam) was addressed. A two-stage prototype with cooling capacity 5 kW was optimized and built. Later, the CRMC prescription for the design of the supersonic diffuser was focused. By a gradual reduction of the fluid velocity and a continuous profile, the CRMC design promises a reduction of the normal shock that usually develops in the mixing chamber. A second 40 kW_f prototype was designed in 2010 for an industrial partner (Frigel Firenze s.p.a.). The design procedure used a thermodynamic code accounting for real gas behavior. This code gives a first design of the mixing chamber and diffuser according to the CRMC criterion. It also gives an estimation of the friction loss along the diffuser. A comparison between different operating fluids was performed and resulted in the selection of R245fa. A first design of the ejector was manufactured in carbon fiber. The primary nozzle is mounted on a movable support, in order to change its axial position with respect to the mixing chamber. In terms of COP, first results were below the values predicted by the simulation code. Meanwhile a numerical simulation was in progress with FLUENT. From the first CFD results it was decided that the diffuser throat had to be moved forward from the primary nozzle exit, in order to allow a complete mixing between the primary and secondary flows, and enlarged, the losses encountered in the mixing process being higher than expected and hence the fluid density lower. This produced a second ejector design, which was manufactured and tested in 2012, showing improved performance. These results suggested a third design, with a further lengthened diffuser, which has undergone a complete testing campaign, allowing validation of the CFD results. The activity performed till now suggests that ejector refrigeration plants have a robust operation and can be easily manufactured at relatively low cost with off-the-shelf components, a part from the ejector itself, which however represents a small fraction of the system cost. However, the COP is lower with respect to absorption refrigeration and hence needs substantial improvement through detailed thermodynamic and CFD design optimization.

Keywords: Refrigeration; Ejector

* Corresponding author. Tel.: +39-0554769333; fax: +39-0554769342.
E-mail address: adriano.milazzo@unifi.it.

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Nomenclature

COP	Coefficient Of Performance	ξ	pressure ratio P_G/P_C
d	primary nozzle throat diameter (m)	φ	area ratio $(D/d)^2$
D	mixing duct throat diameter (m)	ω	entrainment ratio $\dot{m}_s/\dot{m}_p$
h	specific enthalpy (kJ kg^{-1})		
GWP	Global Warming Potential	*Sub/superscripts*	
$\dot{m}$	mass flow rate (kg s^{-1})	*crit*	critical
NBP	Normal Boiling Point [K]	*C*	condenser
ODP	Ozone Depletion Potential	*E*	evaporator
P	pressure (MPa)	*G*	generator
Q	heat (kJ)	*mol*	molar
T	Temperature (K)	*p*	primary flow
		s	secondary flow
Greek symbols			
β	pressure ratio P_C/P_E		

1. *Introduction*

Supersonic ejectors may be seen as a modern evolution of the steam injector invented by Henry Giffard in 1858 and used thereafter for feeding water to steam boilers (jet pump) or creating vacuum in rail brake systems. In a refrigeration system the ejector may substitute the compressor and hence produce a heat driven machine with a boiler, evaporator and condenser (Fig. 1).

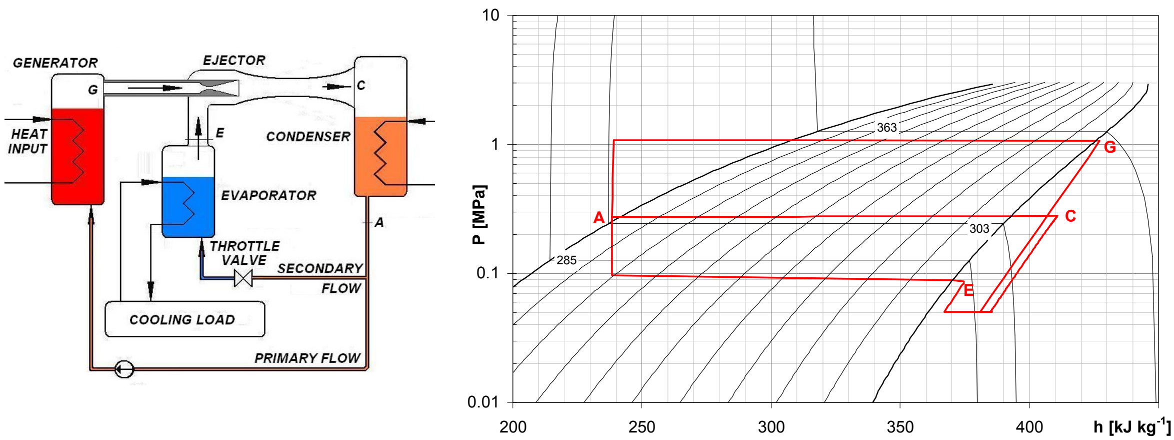

Fig. 1. General scheme of an ejector chiller and thermodynamic cycle (working fluid: R245fa)

High pressure vapor (primary flow) is produced in the generator (point G), flows through the primary nozzle of the ejector and, as it enters the mixing chamber, entrains the secondary fluid being drawn from the evaporator (point E). The combined flow is then compressed as it flows through the diffuser section of the ejector into the condenser (point C). The condensate (point A) is split into two streams: one is expanded through a throttling valve and fed back to the evaporator whilst the other is returned via a feed pump to the boiler to maintain the refrigerant level.

The system COP, pump work being usually modest, is approximately given by:

$$COP = \frac{Q_s}{Q_p} = \omega \frac{\Delta h_s}{\Delta h_p} \quad (1)$$

where $\omega = \dot{m}_s / \dot{m}_p$ is the entrainment ratio between secondary and primary mass flow rates. For a given fluid, therefore, the system performance is mainly dictated by the entrainment ratio which in turn is a function:

$$\omega = f(\beta, \xi, \varphi) \quad (2)$$

of pressure ratios $\beta = P_C / P_E$, $\xi = P_G / P_C$ and area ratio $\varphi = (D/d)^2$, subscripts *C*, *E* and *G* referring to condenser, evaporator and generator while *D* and *d* are mixer and primary nozzle throats. Hence, the fundamental problem when designing an ejector chiller is the optimization of the internal flows occurring within the ejector aiming at maximum entrainment ratio. In order to make the COP of ejector chillers competitive with absorption chillers, entrainment ratios reported in the literature [1] should be roughly doubled.

1.1. Fluid selection

The working fluid in an ejection cycle has to withstand a heat engine and a refrigeration cycle. This brings the fluid along a wide range of temperatures. Therefore, fluid selection is not easy. The safest and cheapest fluid, water, has been used in the early efforts and extensively studied thereafter [2], but is now less common, a part few works like [3], due to high specific volume of steam within the condenser and very low pressure at evaporator. Icing can be a problem for steam ejectors, but can also be an opportunity, as long as ice formation within the evaporator is controlled and used as storage of cooling capacity [4]. Favorable fluid features are:

- Low cost
- Environmental safety (zero ODP, low GWP)
- Operational safety (non toxic, non flammable – fluid charge is high with respect to vapor compression cycles)
- Critical temperature well above generator temperature (in order to increase latent heat accumulation in the boiler)
- High latent heat at evaporator temperature (in order to increase specific cooling capacity)
- Normal boiling pressure close to ambient pressure (in order to avoid vacuum condenser)
- Dry expansion (saturated vapor at boiler exit should not condense along expansion)
- Moderate pressure at boiler temperature (in order to make the operation of the feed pump less critical)

A set of fluids and their relevant features are shown in Table 1.

The first fluid in the list, R134a, has a reasonable cost and is well known in the refrigeration industry. This means for example that problems of material compatibility have been solved. A problem for R134a is the high saturation pressure at generator temperature and high compression ratio imposed on the feed pump. Critical temperature is not very high and, given that R134a is a “wet expansion” fluid, high superheating will be needed at the generator in order to obtain a dry expansion in the primary nozzle. R134a has a relatively high GWP, which has suggested its phasing out from some applications, e.g. automotive air conditioning.

R143a, R218, R227ea, R236fa and RC318 have a high GWP.

R152a and R236ea are very interesting, both in terms of critical temperature and very low GWP. Furthermore, R236ea has a dry expansion. R245ca and R245fa are quite similar: both have high critical temperatures, relatively low GWP and dry expansion.

R32 and R41 have low critical temperature, while R365mfc has the highest critical temperature, acceptable GWP and dry expansion.

Water has by far the highest critical temperature, but also a very high normal boiling point. High vacuum is required in order to reach low temperatures. Condensation problems within the ejector, or even icing, are likely to occur and most likely they will not be manageable with equilibrium models. In any case, given the low cost, environmental safety and ubiquitous availability, water deserves further investigation.

As an alternative, an interesting option in terms of environmental safety, ready availability and thermodynamic performance is R245fa. This fluid was also used by Eames et al. [5] and by Scott et al. [6].

Table 1. Relevant data for a set of non-flammable, non-toxic and zero-ODP fluids.

Fluid	M_{mol} [kg kmol^{-1}]	T_{crit} [K]	P_{crit} [MPa]	NBP* [K]	Expansion	GWP
R134a	102.03	374.21	4.059	247.08	W	1300
R143a	84.041	345.86	3.761	225.91	W	4300
R152a	66.051	386.41	4.517	249.13	W	120
R218	188.02	345.02	2.64	236.36	D	8600
R227ea	170.03	374.9	2.925	256.81	D	3500
R236ea	152.04	412.44	3.502	279.34	D	1200
R236fa	152.04	398.07	3.2	271.71	D	9400
R245ca	134.05	447.57	3.925	298.28	D	640
R245fa	134.05	427.16	3.651	288.29	D	950
R32	52.024	351.26	5.782	221.5	W	550
R365mfc	148.07	460.0	3.266	313.3	D	890
R41	34.033	317.28	5.897	194.84	W	97
RC318	200.03	388.38	2.778	267.18	D	10000
HFO1234yf	114.04	367.85	3.382	243.66	D	4
HFO1234ze	114.04	382.52	3.636	254.19	D	6
water	18.015	647.1	22.06	373.12	W	0

W=Wet, D=Dry expansion * Normal Boling Point (P = 101.3 kPa)

M_{mol}, T_{crit}, P_{crit} and N.B.P. from NIST Refprop [7]; GWP from [8]

1.2. Other components

Even if, as above stated, the ejector should be the main object of any improvement effort, other components should not be neglected. Heat exchangers represent the most of system volume and cost, but downsizing them would heavily impair the system performance. Recent prototypes feature plate heat exchangers, yielding high effectiveness, but pressure losses must be carefully accounted for.

Another crucial component is the feed pump, which must extract the fluid from the condenser and hence is prone to cavitation. Often the high available suction head is achieved at the expense of pump efficiency, which is highly undesirable, given that the pump is electrically driven, even if its power is limited.

2. Prototype steam ejector chiller

The steam ejector was designed by an optimization procedure described in [9]. The developed code allows to obtain a coherent numerical simulation of the whole ejector refrigerating plant, including thermal irreversibilities. The whole plant is considered as a steady-state open system exchanging heat with three thermal sources and consuming external work due to friction losses on the water side of the heat exchangers and to the generator feed pump. The independent variables are the external water flow rates, the number and inner diameter of the tubes within each heat exchangers, the steam flow rate at generator, the boiling and superheating temperature at evaporator and generator and the condensing temperature. The code gives as output the dimensional and operational parameters of the cycle and plant components. The code results denote high influence of the evaporator heat exchanger design on the plant COP. Moreover, better COP values are reached with relatively low heat exchanger efficiencies at condenser and generator, i.e. the best working conditions for the system are different from those corresponding to components optimization; it is necessary to model the entire system to find an optimum.

Main feature of this ejector design is the double stage structure, with an inverted second stage configuration, as presented in figure 2. The primary fluid enters from a peripheral groove and the secondary fluid flows in the center. The secondary fluid of the second stage comes out directly from the mixing chamber of the first stage, avoiding a deceleration in the intermediate diffuser and hence reducing the acceleration in the second stage. The mixing chamber of the second stage ends up in a conical diffuser and discharges the fluid in the condenser. This configuration was designed to pull down the evaporator to the water triple point and therefore produce ice.

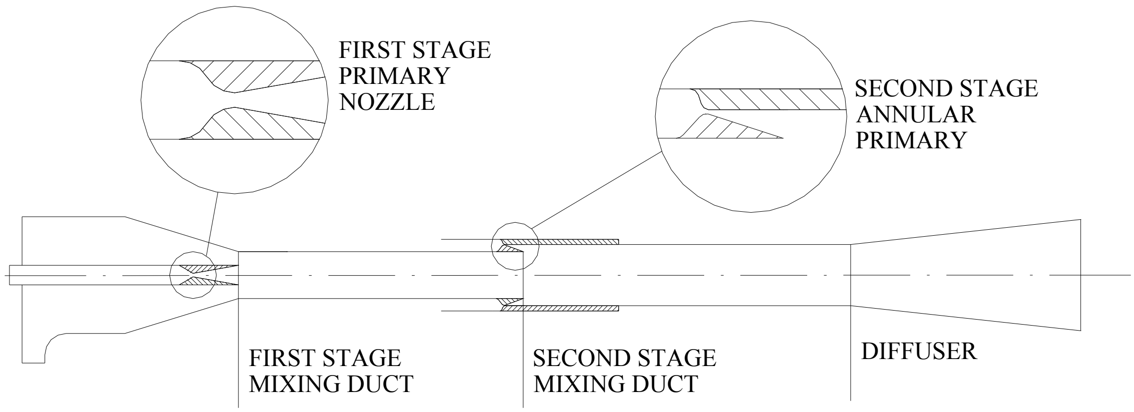

Fig. 2. Two-stages ejector scheme

Different optimization criteria have been investigated and more constrains included [10] in order to obtain the design parameters of a 5 kW cooling power refrigeration plant. An experimental apparatus has been built and tested, bringing out significant problems in the jet pump operation. A last study on this steam ejector [11] addressed the fluid behavior within the primary nozzle, showing that homogeneous condensation is likely to occur in the throat zone. This could produce pressure oscillation in the throat and unstable operation. Therefore, a new primary nozzle will be designed after a detailed two-phase modeling.

3. Industrial prototype ejector chiller

Co-operation with Frigel Firenze s.p.a. started in 2010 as an effort to give to this company new market opportunities in the field of heat driven chillers for industrial use. After an initial attempt to use R134a, which is the common choice in the power range covered by Frigel, thermodynamic reasoning summarized in section 1.1, as well as a market recognition, suggested to revert to R245fa.

3.1. Design procedure

The optimized design of the prototype was obtained through a specifically conceived simulation code, featuring complete flexibility in the selection of the operating fluid, whose thermodynamic behavior is modeled in detail with NIST Refprop routines [7]. The simulation code accepts as inputs the temperature and flow rate of the hot water feeding the generator and of the water at ambient temperature that cools the condenser. Cold water is assumed to be received at 12°C and delivered at 7°C. After an iterative procedure that calculates the entrainment ratio and the fluid properties throughout the cycle, the program delivers as output the system COP and cooling power, in terms of cold water flow rate. Furthermore, the code gives a design of the mixer/diffuser duct, according to a two zone model for the mixing zone and to the CRMC criterion for the diffuser. The CRMC (Constant Rate of Momentum Change) criterion introduced by Eames [12] assumes a constant rate of velocity decrease throughout the ejector. The approach used in our code is a bit different because a separate model is used for the mixing zone and the CRMC approach is used only on the mixed fluid.

The mixing model assumes a conical zone at the primary nozzle exit where entrainment takes place, while the secondary fluid proceeds unaltered in the surrounding annular zone. In this way the entrained flow is calculated at each step of the mixing process and the mixing duct is tailored around the remaining secondary flow. Unfortunately, no indication is given on the cone angle of the primary/entrained flow and hence some uncertainty remains about the mixing zone length.

The CRMC criterion is used in the diffuser throat area, both in the supersonic and in the subsonic zone. When the diffuser divergence angle exceeds 4°, the CRMC procedure is aborted and the diffuser is continued as a straight cone. Exit velocity towards the condenser is kept to a very low value.

A drawback of R245fa is the scarce availability in the literature of correlations for heat exchanger design. A proprietary design code was delivered by the heat exchanger manufacturer, but it has a closed structure and could not be incorporated in the system design code. Therefore the heat exchangers are dimensioned outside the simulation code in a trial and error procedure.

The details of the simulation code are given in [13]. The resulting design is shown in Fig. 3. The primary nozzle duct is mounted on a screw slid, in order to allow an experimental optimization of the nozzle position.

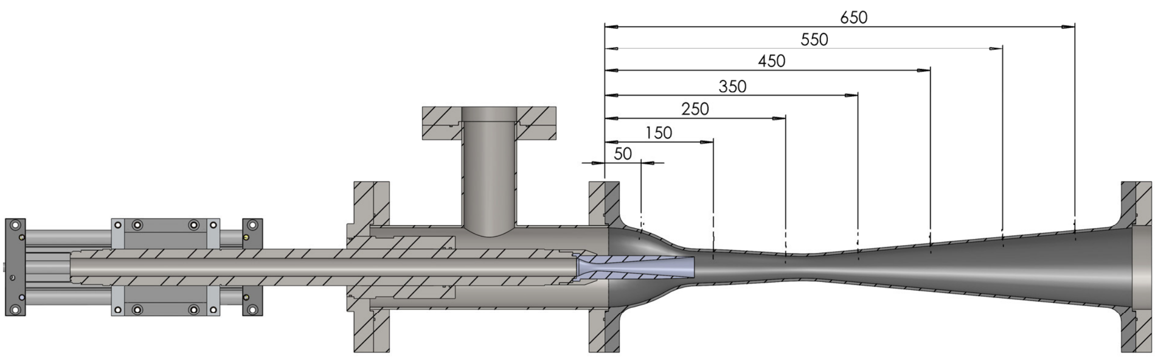

Fig. 3. R245fa Ejector – First design

3.2. Experimental results from the first design

The ejector was built and assembled with all the other components on a test bench at Frigel. The complete system, including all the pressure and temperature transducers and the data acquisition system, is shown in Fig. 4. The quoted points along the duct are the positions of 7 static pressure ports. The details of the experimental set-up and the first results are reported in [14].

Experimentally measured COP turned out to be 47% lower than the value calculated by the simulation code. Critical condenser temperature (i.e. the condensation temperature causing an abrupt fall in performance) was about 32°C when generator temperature was 100°C.

Meanwhile, a CFD model of the ejector was set up. When the first CFD results have been available, a likely explanation for the low performance of the first design was found in the incomplete mixing still clearly visible in the diffuser throat (Fig. 5). This could be due to a wrong estimation of the exit cone from the primary nozzle.

According to this conclusion, a new design was conceived with a longer and wider throat.

3.3. Second design

The second ejector (Fig. 6) shares with the first the primary nozzle and the outer dimensions, but the throat is in the zone of the intermediate coupling flange, i.e. moved about 50 mm to the right with respect to first design.

The results were encouraging: COP values of 0.25 were recorded, though evaporator temperature was 10°C instead of requested 5°C. Critical condenser temperature was reduced with respect to the first design, due to the wider diffuser throat area.

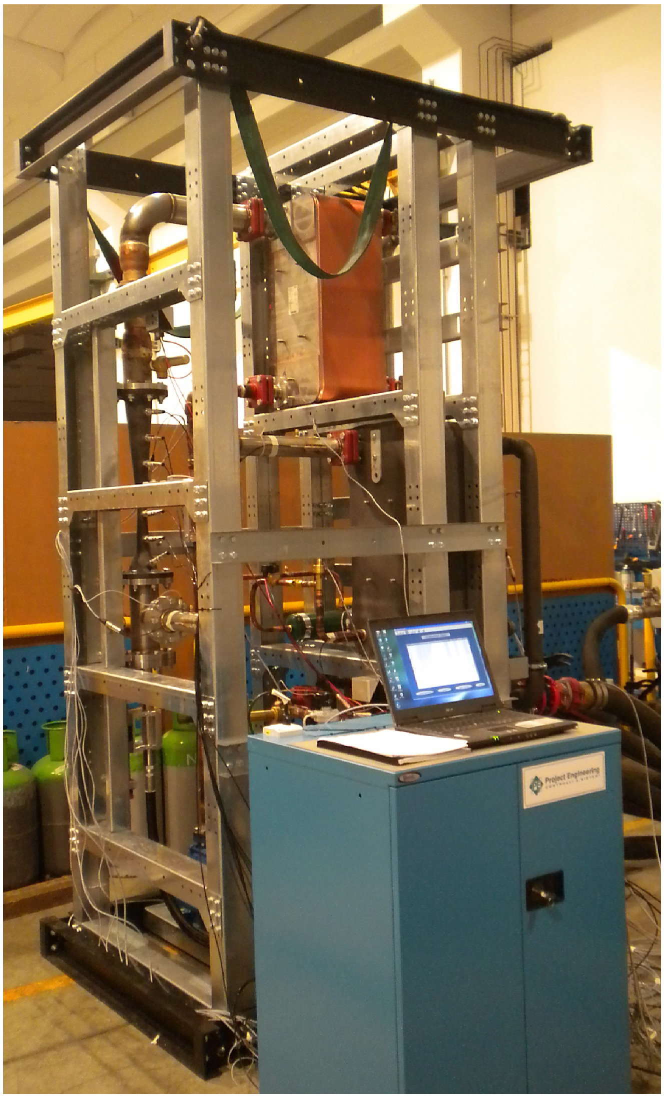

Fig. 4. Prototype R245 ejector chiller

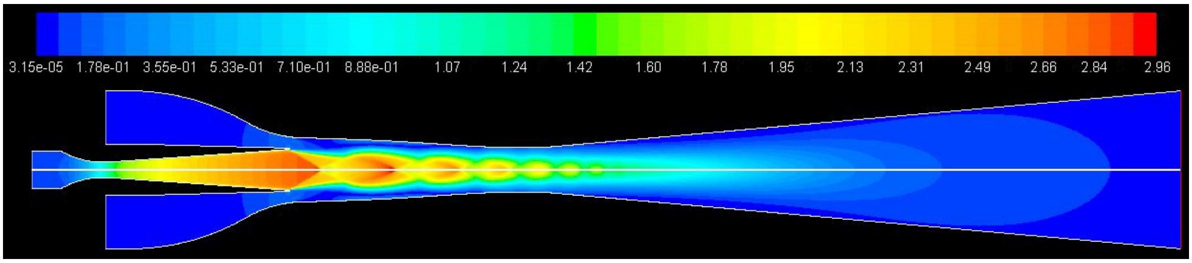

Fig. 5. R245fa Ejector – CFD analysis of the first design

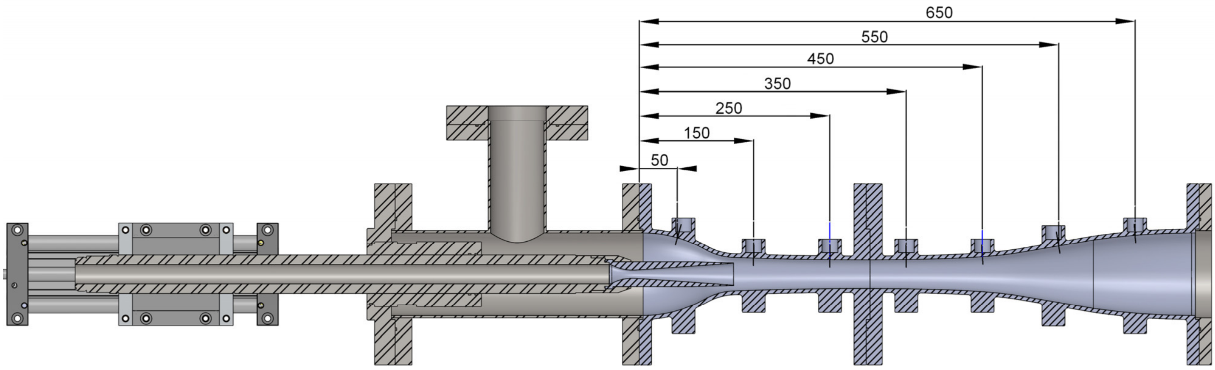

Fig. 6. R245fa Ejector – Second design

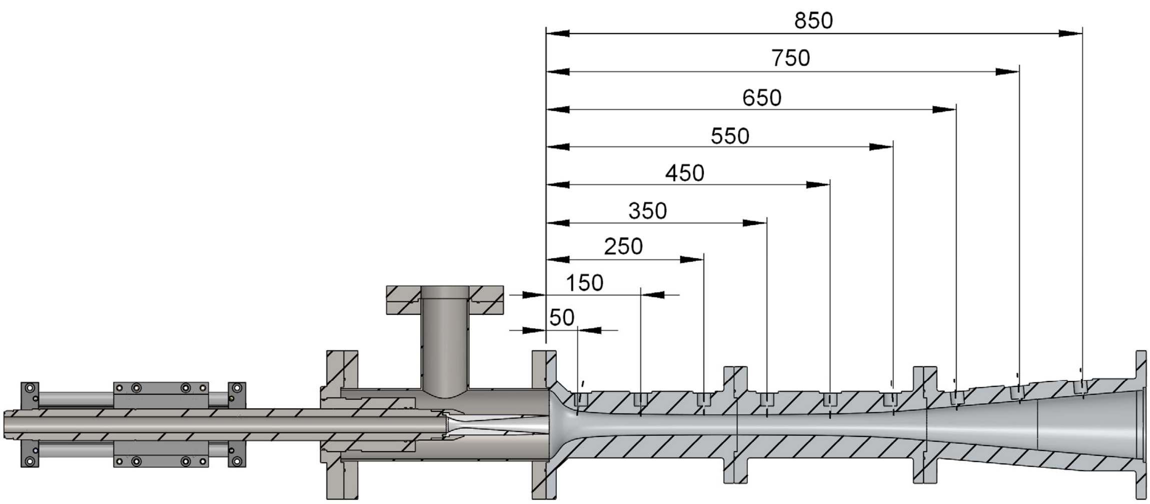

Fig. 7. R245fa Ejector – Third design

3.4. Third design

According to the results gathered with the first two ejectors, an increase in the mixing zone and diffuser length can be favorable, as long as the improvement in terms of momentum exchange between primary and secondary flow balances the increased friction loss. Hence the third ejector was designed (Fig. 7) with length between the flanges increased from 740 to 950 mm. The shape is a scaled up and modified version of a CRMC profile designed by Ian Eames. The last part is shaped as a straight cone with 4° half angle. The inlet has a fillet that allows a smooth transition from the cylindrical pipe upstream. The primary nozzle, for the moment, was left unchanged. The number of pressure ports was increased to 9.

Extensive testing has been carried out on this ejector. Stable operation is obtained with saturation temperatures of 5°C at evaporator and 100°C at generator. Primary nozzle position has been optimized and the best results have been obtained with a displacement of 8 mm to the right with reference to the position shown in Fig. 7. Fig. 8a shows the COP as a function of condenser saturation temperature. In this condition the critical condenser temperature is around 36.2°C. Fig. 8b shows how the increase in condenser temperature and pressure modifies the static pressure along the ejector. At low condenser pressure (lower curve) the flow is still supersonic at the diffuser throat and hence re-expands afterwards. This produces a normal shock somewhere in the divergent part of the diffuser. At intermediate condenser pressure the normal shock moves leftwards (intermediate curves) until, at still higher pressure the shock overcomes the throat and the ejector ceases to be supersonic. This corresponds to the last point on the right of Fig. 8a and indicates a highly unstable working condition.

When evaporator temperature is raised to 10°C, the COP increases to 0.29. This result is lower than that reported by Eames et al. in [5], but they used an higher generator temperature (110°C) and a bigger area ratio, which increases COP but lowers critical condenser temperature. On the other hand, the experimental results given in terms of entrainment ratio by Scott et al. in [6], when converted to COP, indicate a lower performance.

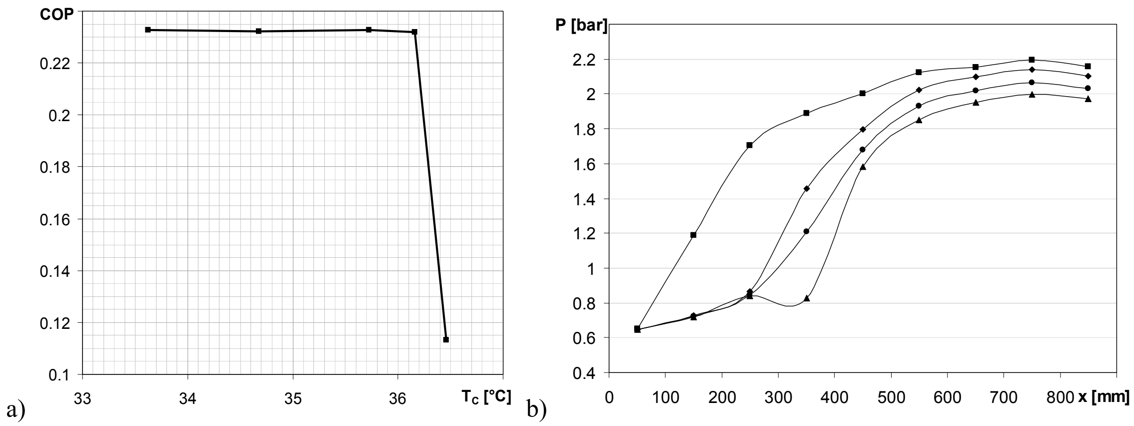

Fig. 8. R245fa Ejector – Third design – a) measured COP as a function of condenser temperature;

b) measured static pressure along the ejector

4. Conclusions

A prototype of ejector chiller for industrial use has been built by Frigel Firenze s.p.a. on a project by DIEF (Department of Industrial Engineering Florence). The performance of this prototype has been increased through improvements in the design of the ejector, reaching COP = 0.23 with evaporator temperature of 5°C and generator temperature of 100°C. These temperatures are consistent with industrial use or air conditioning applications, waste heat from an internal combustion engine being used at generator and water at 7°C being produced at evaporator. However, efficiency is still far from the target. Further improvements are expected from an extensive CFD optimization, which is in progress.

The design of an ejector chiller should be based on a thermodynamic simulation that gives the system geometry in terms of heat exchange surfaces and flow sections, as well as a first design of the ejector. This first design must be verified and further optimized by CFD simulation.

As soon as an optimized design of the industrial prototype will be reached, the design procedure will be transferred to the steam ejector prototype in the DIEF laboratory, whose ejector will be rebuilt.

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