A NEW EJECTOR DESIGN METHOD DISCLOSES POTENTIAL IMPROVEMENTS TO THE PERFORMANCE OF JET-PUMP CYCLE REFRIGERATORS

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ABSTRACT

So far the jet-pump refrigeration cycle has always been characterised by low COP values, due in the main to the low performance of the ejector in fulfilling its role as cycle's compressor. The recently suggested Constant Rate of Momentum Change (CRMC) method seems to offer a real possibility to significantly increase the performances of the ejector and in so doing, open up new prospects for the jet-pump refrigeration cycle. Using a numerical simulation code, developed by the authors, this paper offers a comparison between jet-pump refrigerator cycles operating with ejectors designed using both conventional and CRMC methods. In both cases the ejectors are designed with supersonic primary nozzles. The numerical optimisation code devised by the authors covers the whole jet-pump refrigerator cycle. This is considered as a open system exchanging heat with three thermal sources. The optimisation function in each case is assumed to be the system COP and the work needed at the pumps to win the pressure losses incurred by the external water flows are included. This theoretical paper shows that the COP of the jet-pump refrigerator is improved when the ejector is designed with CRCM method. This opens up prospects for the commercial utilisation of the jet-pump refrigeration system, particularly when low-grade heat is available to produce cooling.

INTRODUCTION

Several interests are producing gradual changes in the field of refrigeration. These include growing interest in environmental protection, rational use of energy, research into environmentally friendly refrigerant fluids. The ability of jet-pump refrigerators to operate with natural fluids, such as water, with simplicity and low cost plant, combined with a potential for utilising low-grade heat, represents a real opportunity for the further development and wide spread application of this type of refrigerator.

Steam ejectors are well-known devices, which were first developed during the first few years of the twentieth century. Keenan et al. (1950) are believed to be the first researchers to develop the first theoretical formulation for the design of supersonic ejectors, in particular for the interpretation of the mixing phenomena between the primary and secondary stream. At this time the low COP values experienced with jet-pump refrigerator systems has been the main reason of for their rare use. This low efficiency is caused mainly by the irreversibilities in the ejector. To increase the COP values it is necessary to improve the performance of the ejector. Towards this aim the new Constant Rate of Momentum Change (CRMC) design method was developed, Eames (2002).

In this paper a theoretical comparison is made between the performance of a steam jet-pump refrigerator using a two-stage ejector with traditional mixing section and one using a CRMC design.

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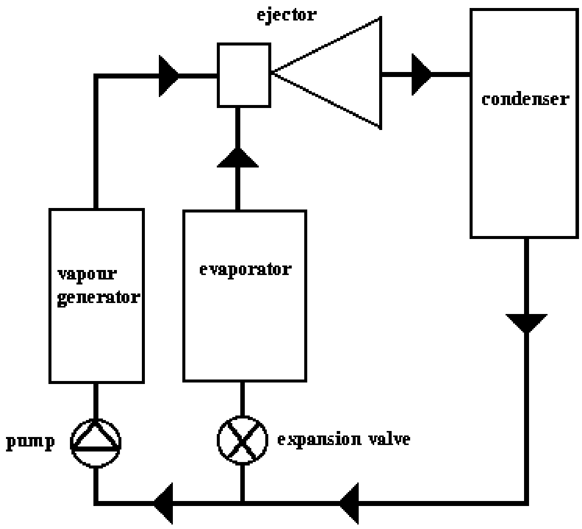
THE JET-PUMP REFRIGERATOR CYCLE

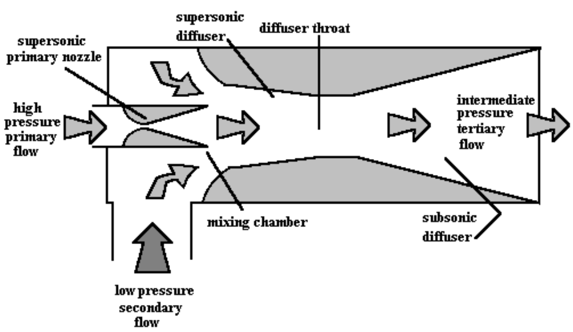
Jet-pump refrigerators are a class of thermally activated devices that exchange thermal energy with three sources at different temperatures. Figure 1 shows the construction of a jet-pump refrigerator.

Figure 1. Schematic view of a typical jet-pump refrigerator



The cycle is similar to the conventional vapour compression system except the compressor is replaced by a liquid feed-pump, vapour generator and ejector. Briefly, liquid refrigerant is vaporised at high pressure in a vapour generator and fed to an ejector where it entrains a low pressure vapour coming from the evaporator and compresses it to some intermediate pressure equal to that in the condenser. A proportion of the condensate collected in the condenser is then returned to the evaporator via an expansion valve whilst the remainder is returned to the generator via a liquid feed-pump.

Figure 2: Illustration of an ejector



The ejector is the heart of the jet-pump cycle. A schematic diagram showing the construction of a traditional single-stage supersonic ejector is described in Figure 2. In operation high pressure vapour, coming from a vapour generator, is accelerated to supersonic velocity through the convergent-divergent passage of the primary (de Laval) nozzle. As this high velocity jet emerges from the nozzle and it entrains a secondary vapour stream, (from the evaporator) which enters the conical mixing section through the suction manifold. The primary and secondary flow streams combine within the convergent passage of the mixing section to form a single stream at entry to the parallel section of a diffuser throat. As the flow enters the divergent diffuser section it undergoes a thermodynamic shock process that causes in a sudden rise in static pressure and a reduction in stagnation pressure. The location of the shock wave within the diffuser varies with condenser back-pressure. The flow emerges from this shock process with subsonic velocity and is compressed until its static pressure equals the saturation pressure in the condenser.

In the traditional ejector this supersonic stream entering the diffuser must undergo a thermodynamic shock process to enable its static pressure to rise to equal the back-pressure at the diffuser exit. This shock process causes a sudden fall in Mach number as the flow changes from supersonic to subsonic conditions and this results in a loss of total pressure which is detrimental to the pressure lift-ratio and entrainment ratio of the ejector. These parameters are defined along with motive pressure ratio in Equations (1) to (3).

$$\text{Pressure Lift Ratio, } \beta = \frac{P_{out}}{P_s} \tag{1}$$

$$\text{Entrainment Ratio, } \omega = \frac{m_s}{m_p} \tag{2}$$

$$\text{Motive Pressure Ratio, } \xi = \frac{P_p}{P_s} \tag{3}$$

THE 2- STAGE TRADITIONAL EJECTOR AND THE CRMC EJECTOR

The two-stage traditional ejector has been investigated by one of the authors, Grazzini , D'Albero (1998) and Grazzini, Mariani (1998). This type of ejector, shown in Figure 3, consists of a traditional first-stage without a diffuser. The combined flow stream from the first-stage becomes the secondary flow for the second-stage. A diffuser is then positioned at the outlet of the second-stage and this is usually a conventional straight sided divergent duct. The design in this case imposes a constant-area mixing process. The ejector defined by the CRMC method, Eames (2002), has a diffuser with a convergent-divergent shape which permits the flow to decelerate from supersonic to subsonic conditions ideally without thermodynamic shock process. The variation in cross-section with distance is obtained imposing a constant variation of the momentum of the flow stream within the diffuser. At this time the entrainment and mixing process is assumed to occur at constant static pressure. A simple scheme of the CRMC arrangement is presented in Figure 4.

The mathematical model used for the ejector's design is based on the typical relations for an isoentropic, one-dimensional steady flow within adiabatic walls where the entry kinetic energy at the primary and secondary ports is assumed to be negligible.

Some geometrical constrains are imposed for the two ejectors;

Figure 3: Scheme of the two stage traditional ejector

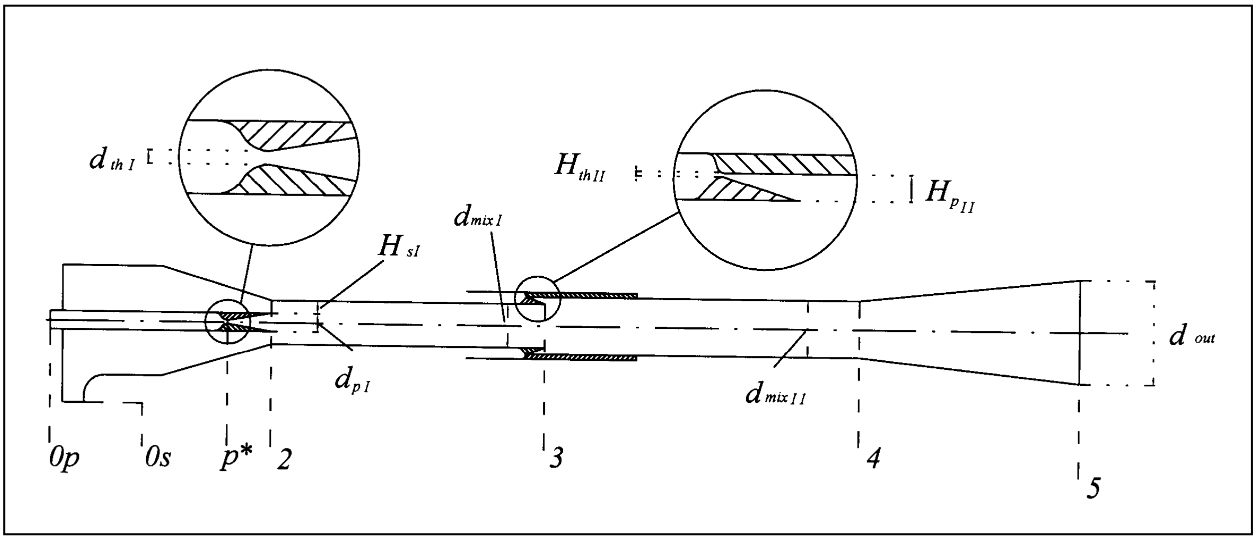
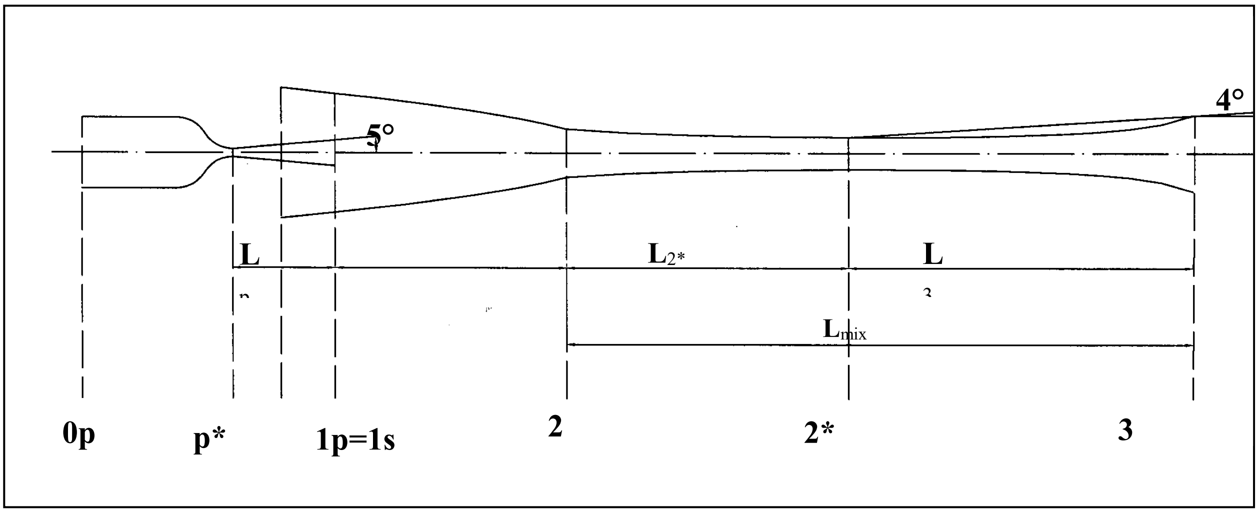


Figure 4: Scheme of the CRMC ejector



Two stage traditional ejector:

- The mixing chamber length is equal to seven time the respective diameter
- The outlet area of the diffuser is four time the inner area
- The angle for the nozzle diverging ducts is 5°
- The angle for the diffuser is 3.5°

CRMC ejector:

- The mixing chamber length is equal to seven time the diameter of the throat section of the diffuser
- At the design condition the local Mach number value at the diffuser throat is unity
- The angle for the nozzle diverging ducts is 5°
- The effective angle for the diffuser is 4°

The CRMC design method calculates the ejector diffuser geometry in such a way that ensures a constant decrease of the momentum equation to avoid the thermodynamic shock that commonly appears in the traditional supersonic ejector. The relation used is:

$$\frac{dK}{dx} = m_p(1+\omega)\frac{dw}{dx} = constant \tag{4}$$

A complete description of the mathematical model for the two-stage traditional ejector is provided by Grazzini and Rocchetti (2001b) and mathematical model for the CRMC method is described by Eames (2002).

SIMULATION CODE

A simulation code, developed by the authors Grazzini and Rocchetti (2001a, 2002), was used to investigate the performance of the jet-pump refrigerator. This code uses a numerical methods to optimise the geometrical design of the ejector and for a given set of thermodynamic parameters for the cycle.

For the vapour generator and the condenser shell-and-tube heat exchangers are assumed with the hypothesis that the external water coming from the thermal sources flows through the tube-side. Heat transfer coefficients are obtained from literature and external water pressure drop are calculated, Grazzini and Rocchetti (2002). A flash-evaporation model was used for the evaporator and the external heating-water was assumed to come from a low temperature heat source (378.15 K). A mathematical model of a pump was included in the utilisation circuit when outgoing from the evaporator.

To determine the thermodynamic properties of water and steam some NIST/STEAM routines are used, Klein and Harvey (1996).

COP was selected as the optimisation function in this case and the work input required to overcome pressure losses through the condenser and generator tubes is included. The COP is defined as,

$$COP = \frac{Q_E}{Q_G + W_{PE} + W_{PC} + W_{PG} + W_{PC-G}} \tag{5}$$

The optimisation code uses a COMPLEX search method, Box et al. (1969) which randomly creates a set of solution-points, and moves through the independent variables space evaluating the optimisation function at each vertex. Each newly generated point is tested for feasibility, and, if found unfeasible, is moved back toward the centroid of the previously generated points until it becomes feasible. The search continues in this way until the pattern of points has shrunk, so that the points are sufficiently close together and/or the difference between the function values at the points becomes small enough.

The input data requested by the code include thermal power at the evaporator and the temperatures at each of the three thermal sources. The COMPLEX method requires the user to define some admissible ranges for the independent variables. In this case we have 13 variables that are:

i. Coolant and heating fluid mass flow rates at each of the three heat exchangers
ii. Inner diameter and number of the tubes in each heat exchanger
iii. Flow rates of the motive steam of the ejector
iv. Condensing and boiling temperatures at each heat exchanger
v. Superheating value at the generator

The code's outputs were the predicted operating conditions of the cycle and the geometrical parameters of the ejectors and heat exchangers.

NUMERICAL RESULTS AND DISCUSSION

To compare the performance of the jet-pump refrigerator using the two types of ejector, the simulation outputs of the five best runs for each are presented and listed in table 3 and 4 when the same jet-pump refrigerator boundary values were used for both types of ejector as listed in Table 1 and Table 2.

Table 1. Input data

Q_E [W]	$T_{w\,E\,in}$ [K]	$T_{w\,C\,in}$ [K]	$T_{w\,G\,in}$ [K]
5000	274	303.15	378.15

Table 2. Admissible ranges for the independent variables

	$M_{w\,E\,in}$ [kg s^{-1}]	$M_{w\,C\,in}$ [kg s^{-1}]	$M_{w\,G\,in}$ [kg s^{-1}]	D_C [m]	D_G [m]	n_{TC}	n_{TG}	$T_{rf\,E}$ [K]	$T_{rf\,C}$ [K]	$T_{rf\,G}$ [K]	$\Delta T_{rf\,G}$ [K]	m_p [kg s^{-1}]
Low	0.5	0.5	0.5	0.018	0.018	10	10	267	303.65	328.15	0.5	0.0005
High	10	30	30	0.103	0.103	200	200	273	323.15	373.15	5	0.01

The COP results in terms of the Second Law efficiency, are given by Equations (5), (6), (7) and (8).

$$COP_{II\,L} = \frac{COP}{COP_{Ca}} \tag{6}$$

$$COP_{Ca} = \frac{T_{rf\,G} - T_{rf\,C}}{T_{rf\,G}} \cdot \frac{T_{rf\,E}}{T_{rf\,C} - T_{rf\,E}} \tag{7}$$

$$\eta_{II\,L} = \frac{COP}{\eta_{Ca}} \tag{8}$$

$$\eta_{Ca} = \frac{T_{wG\,in} - T_{wC\,in}}{T_{wG\,in}} \cdot \frac{T_{wE\,in}}{T_{wC\,in} - T_{wE\,in}} \tag{9}$$

The phase-change temperatures at the evaporator, condenser and generator were used in the determination of the Second Law Efficiency $COP_{II\,L}$, which refers to the cycle, and the thermal sources temperatures in the $\eta_{II\,L}$. The comparison reveals that the COP and $COP_{II\,L}$ are both improved for jet-pump refrigeration cycle with the CRMC ejector.
Two different temperatures of the external water at the inlet of the condenser were imposed: 30°C and 40°C were assumed for both the systems with the two ejector type. The code determines the better operational and geometrical condition for the system. Several runs were made and figures 6 to 8 present the results of the whole numerical simulation. The scattering of results for optimum COP, comes from the numerical approximations that we can not reduce with the search method used.
Figure 7 shows the relationship between lift ratio and entrainment ratio.

Figure 6. Lift ratio β versus entrainment ratio ω

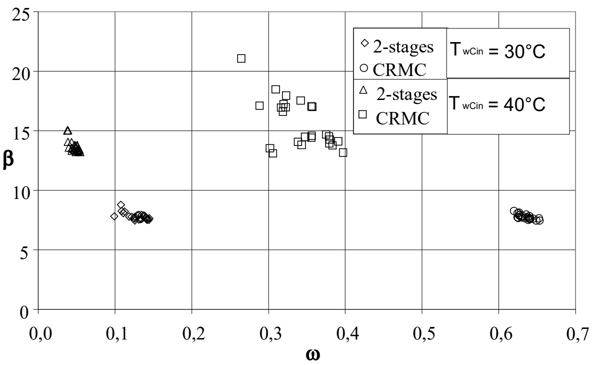


Figure 7 and 8 present the heat exchanger effectiveness values for the heat exchangers versus the cycle COP. The heat exchanger effectiveness is:

$$\varepsilon_K = \frac{C_{hot}\left(T_{hot,in} - T_{hot,out}\right)}{C_{min}\left(T_{hot,in} - T_{cold,out}\right)} = \frac{C_{cold}\left(T_{cold,in} - T_{cold,out}\right)}{C_{min}\left(T_{hot,in} - T_{cold,out}\right)} \tag{10}$$

where C_{min} = lesser of C_{hot} and C_{cold} , C is the capacity rate of the liquid water, T_{max} and T_{min} are the maximum and minimum temperature values involved in the thermal exchange and Q is the thermal power at each heat exchanger.

Figure 7. Heat exchanger effectiveness ε_K versus cycle COP; $T_{w\,C\,in}$ = 30°C

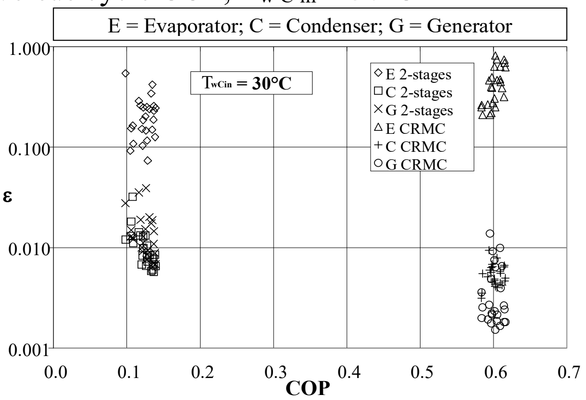
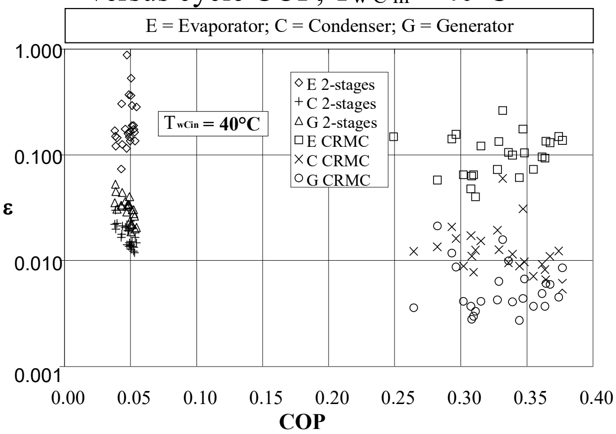


Figure 8: Heat exchanger effectiveness ε_K versus cycle COP; $T_{w\,C\,in}$ = 40°C



For the two jet-pump cycles the phase-change temperatures are consistently close to the lower limit of the condenser and the upper limit of the evaporator and generator. In our opinion that happens because the limitation of compression work at the ejector it is more efficient than thermal irreversibilities reduction. The system decreases his performances when the compression work rise: when the temperature of the external water at the condenser inlet rises from 30°C to 40 °C the needed compression ratio rises (see figure 6) and the cycle COP decrease, as the comparison between figures 7 and 8 shows.

Thermal irreversibilities at the evaporator has higher influence on the cycle COP. The low values of the heat exchanger effectiveness indicate that the optimum design of the system is not directly related to the optima of each component; then optimum of the system require to model the entire system.

Only the CRMC ejector shows a very large improvement of the optimisation function in comparison with two stage ejector.

The better performance showed by the CRMC ejector involves a considerable reduction of the primary flow rate (see figure 6) at the ejector, with consequent reduction of thermal powers, of generator and condenser heat exchange areas, of flow rates and pressure losses through the external water circuits.

The pressure lift ratio and the motive pressure ratio are bound by the phase change temperatures at the heat exchangers. Also, the entrainment ratio is better for the CRMC ejector; being about six time greater than for the two stage ejector (see figure 6).

The boundary conditions at the primary and secondary ports are given by the phase-change conditions at the heat exchangers then we have the same mixing pressure at the outlet of the primary nozzle. As a consequence the geometrical configuration and the fluid dynamic parameters for the primary nozzle and secondary duct of the CRMC ejector are very similar to those for the first-stage of the two-stage ejector.

Different mixing zones and diffuser sections for the two ejectors cause strong difference in the geometrical solution. Anyway, the overall dimensions are very similar in both cases.

CONCLUSIONS

A numerical simulation described in this paper suggests that CRMC method for the design of the ejector will provide jet-pump refrigerator with increased COP values approaching those of a typical single-effect adsorption cycle used for air conditioning. This finding is believed to open a new and exciting time for the development of jet-pump refrigerators.

Table 3 Output data for the two-stage ejector cycle

CYCLE PARAMETERS

		COP	$\eta_{II\,L}$	$COP_{II\,L}$	$T_{rf\,E}$	$T_{rf\,C}$	$T_{rf\,G}$	$\Delta T_{rf\,G}$	$m_{p\,I}$	$m_{p\,II}$
	min				267.00	303.15	328.15	0.50	0.0005	0.0005
	max				273.00	323.15	391.15	5.00	0.01	0.01
trials	a	**0.139**	1.864	1.605	272.94	303.72	370.83	4.42	0.003	0.011
	b	**0.139**	1.864	1.629	272.84	303.83	372.83	3.83	0.002	0.011
	c	**0.137**	1.864	1.564	272.96	303.71	368.67	4.21	0.002	0.012
	d	**0.137**	1.864	1.586	272.67	303.66	370.44	3.42	0.002	0.012
	e	**0.137**	1.864	1.575	272.84	303.65	369.35	2.25	0.003	0.011

EJECTOR

WORKING PARAMETERS		ω_I	ω_{II}	ω_{tot}	β_I	β_{II}	$\beta_{\tau o \tau}$	ξ	Ma_I	Ma_{II}
trials	a	0.63	0.49	0.14	2.53	2.97	7.53	158.49	3.43	2.84
	b	0.82	0.39	0.15	2.22	3.45	7.64	171.80	3.47	2.93
	c	0.93	0.35	0.14	2.03	3.70	7.51	146.28	3.40	2.90
	d	0.81	0.38	0.14	2.21	3.47	7.68	159.77	3.44	2.90
	e	0.72	0.42	0.14	2.35	3.23	7.58	151.48	3.41	2.85

GEOMETRICAL PARAMETERS		$d_{th\,I}$ (10^3m)	$d_{p\,I}$ (10^3m)	$H_{s\,I}$ (10^3m)	$d_{mix\,I}$ (10^3m)	$H_{th\,II}$ (10^3m)	$H_{p\,II}$ (10^3m)	$d_{mix\,II}$ (10^3m)	d_{out} (10^3m)	$L_{mix\,I}$ (10^3m)	$L_{mix\,II}$ (10^3m)	L_d (10^3m)
trials	a	5.2	26.2	14.6	55.4	0.4	3.1	62.3	124.5	385.1	431.1	705.6
	b	4.4	23.0	15.6	54.1	0.4	4.2	62.2	124.5	384.0	431.1	705.7
	c	4.4	21.5	15.9	53.2	0.4	4.6	62.5	124.9	379.8	431.4	706.1
	d	4.6	23.3	15.5	54.4	0.4	3.7	62.8	125.7	386.2	434.4	711.4
	e	5.0	24.6	15.0	54.7	0.4	4.3	62.5	125.0	386.4	432.2	707.5

HEAT EXCHANGERS

			Q	M	$T_{w\,out}$	n_T	n_{Tr}	D	A	$LMTD$	W_P*) ; ΔP	Re
		min		0.05		10		0.018				
		max		10.00		200		0.103				
EVAPORAT.	trials	a	5 000	4.64	273.76				1.08	0.93	467*)	
		b	5 000	8.25	273.87				0.95	1.09	831*)	
		c	5 000	5.02	273.78				1.08	0.92	506*)	
		d	5 000	3.51	273.68				0.89	1.16	354*)	
		e	5 000	5.50	273.79				0.96	1.06	554*)	
CONDENSER	trials	a	40703	24.72	303.54	111	5	0.073	391	12.73	3.06E-01	4892
		b	40563	18.52	303.67	145	7	0.103	4994	13.40	7.74E-02	1991
		c	40932	29.30	303.48	192	9	0.103	1206	12.35	4.72E-02	2374
		d	41224	24.45	303.55	32	2	0.103	309	12.28	9.80E-01	11894
		e	41021	22.52	303.59	153	7	0.103	8463	11.89	1.73E-01	2292
GENERATOR	trials	a	35730	16.90	377.65	123		0.103	69	21.93	2.18E-03	6329
		b	35577	12.96	377.50	173		0.103	122	18.50	6.66E-04	3450
		c	36006	7.95	377.08	68		0.103	56	25.85	9.92E-04	5374
		d	36279	10.72	377.35	43		0.103	47	24.41	7.00E-03	11470
		e	36082	16.65	377.64	138		0.103	48	27.82	1.32E-03	5560

Table 4. Output data for the CRMC ejector cycle

CYCLE PARAMETERS			COP	$\eta_{II\,L}$	$COP_{II\,L}$	$T_{rf\,E}$	$T_{rf\,C}$	$T_{rf\,G}$	$\Delta T_{rf\,G}$	m_p
		min				267.00	303.15	328.15	0.50	0.0005
		max				273.00	323.15	391.15	5.00	0.01
	trials	a	**0.616**	1.864	1.612	272.84	303.65	371.22	1.09	0.003
	trials	b	**0.615**	1.864	1.612	272.94	303.76	371.36	3.27	0.003
	trials	c	**0.615**	1.864	1.606	273.00	303.91	371.46	3.98	0.003
	trials	d	**0.614**	1.864	1.641	272.78	303.71	373.15	2.31	0.003
	trials	e	**0.612**	1.864	1.624	273.00	303.65	371.38	2.14	0.003

EJECTOR				ω	β	ξ	Ma_p	Ma_s
	WORKING PARAMETERS	trials	a	0.64	7.62	162.13	4.11	0.82
	WORKING PARAMETERS	trials	b	0.64	7.60	161.59	4.10	0.81
	WORKING PARAMETERS	trials	c	0.64	7.59	161.39	4.10	0.81
	WORKING PARAMETERS	trials	d	0.65	7.64	174.60	4.15	0.82
	WORKING PARAMETERS	trials	e	0.65	7.47	160.90	4.10	0.81

EJECTOR				$d_{th\,p}$ (10^3m)	d_p (10^3m)	$H_{s\,I}$ (10^3m)	d_2 (10^3m)	$d_{th\,d}$ (10^3m)	d_3 (10^3m)	L_{mix} (10^3m)	L_{2*} (10^3m)	L_3 (10^3m)
	GEOMETRICAL PARAMETERS	trials	a	5.1	22.3	15.7	44.2	37.5	257.6	262.6	730.2	783.0
	GEOMETRICAL PARAMETERS	trials	b	5.1	22.3	15.7	44.1	37.5	258.4	262.8	730.0	785.9
	GEOMETRICAL PARAMETERS	trials	c	5.1	22.2	15.6	43.9	37.4	257.7	261.7	726.9	783.6
	GEOMETRICAL PARAMETERS	trials	d	4.9	22.1	15.8	44.2	37.5	258.3	262.7	733.8	785.5
	GEOMETRICAL PARAMETERS	trials	e	5.1	22.1	15.7	43.9	37.6	258.8	263.5	726.4	786.7

HEAT EXCHANGERS				Q	M	$T_{w\,out}$	n_T	n_{Tr}	D	A	$LMTD$	W_P*) ; ΔP	Re
			min		0.05		10		0.018				
			max		10.00		200		0.103				
	EVAPORAT.	trials	a	5 000	1.66	273.32				1.18	0.78	167*)	
	EVAPORAT.	trials	b	5 000	1.65	273.31				1.33	0.66	166*)	
	EVAPORAT.	trials	c	5 000	1.62	273.31				1.43	0.59	164*)	
	EVAPORAT.	trials	d	5 000	3.11	273.64				0.98	1.03	314*)	
	EVAPORAT.	trials	e	5 000	3.10	273.63				1.19	0.80	312*)	
	CONDENSER	trials	a	12747	20.65	303.30	111	3	0.103	254	7.12	1.76E-02	2889
	CONDENSER	trials	b	12826	21.15	303.30	142	4	0.103	277	7.72	1.07E-02	2312
	CONDENSER	trials	c	12783	14.52	303.36	109	3	0.103	882	8.24	1.51E-02	2070
	CONDENSER	trials	d	12645	15.12	303.35	169	4	0.103	1236	7.59	9.41E-03	1390
	CONDENSER	trials	e	12693	23.73	303.28	100	3	0.103	183	7.22	2.53E-02	3684
	GENERATOR	trials	a	7964	14.11	378.02	192		0.103	16	26.94	2.39E-04	3391
	GENERATOR	trials	b	8019	10.64	377.97	34		0.103	12	23.19	4.90E-03	14436
	GENERATOR	trials	c	7997	14.24	378.02	75		0.103	15	21.58	2.06E-03	8765
	GENERATOR	trials	d	7854	9.51	377.95	111		0.103	21	21.56	3.10E-04	3954
	GENERATOR	trials	e	7898	3.88	377.67	45		0.103	16	25.04	1.72E-04	3970

NOMENCLATURE

A	area [m^2]
C	capacity rate (J kg^{-1})
COP	real coefficient of performance
d	diameter of the ejector sections [m]
D	inner diameter of the heat exchanger tubes [m]
H	ring width of the annular nozzle [m]
K	momentum of a stream [N]
L	length of the ejector sections [m]
LMTD	logarithmic mean temperature difference [K]
M	external water flow rate [kg s^{-1}]
m	cycle fluid flow rate [kg s^{-1}]
Ma	Mach number
n_T	number of heat exchanger tubes
n_{Tr}	number of heat exchanger tubes per rank
P	pressure [Pa]
Q	thermal power [W]
Re	Reynolds number
S	liquid-vapour separator
T	temperature [K]
w	main stream velocity [m s^{-1}]
W_P	power needed at the water pump [W]
x	ejector abscissa [m]
ΔP	pipe pressure losses [Pa]
ΔT	temperature superheating [K]

Greek

β	lift ratio
ϑ	half angle of the diverging section of the ejector nozzle and diffuser [radiant]
η	efficiency
ε_K	heat exchanger effectiveness
ω	entrainment ratio
ξ	motive ratio

Subscripts

0	total or stagnation condition
I	first stage of the ejector
II	second stage of the ejector
II L	referred to Second Law of Thermodynamic
1p	outlet section of the CRMC ejector primary nozzle
1s	outlet section of the CRMC ejector secondary nozzle
2	outlet section of the CRMC ejector mixing chamber
2*	converging section of the CRMC ejector diffuser
3	outlet section of the CRMC ejector diffuser
C	condenser
Ca	referred to the ideal Carnot cycle
cold	related to the cold fluid side on the heat exchanger
d	ejector diffuser
E	evaporator
G	generator
hot	related to the hot fluid side on the heat exchanger
in	inlet section
max	max value
min	min value
mix	ejector mixing section
out	outlet section
p	primary flow
p*	throat section on the ejector
rf	refrigerant fluid
s	secondary flow
th	throat section
w	water

REFERENCES

1. Box M.J., Davies D., Swann W,H., Non-Linear Optimisation Techniques, ICI Monograph, Oliver & Boyd, 1969, London, UK
2. Eames I. W., A New Prescription for the Design of Supersonic Jet-Pumps: the Constant Rate of Momentum Change Method, Appl. Thermal Eng., vol 22, pp 121-131, 2002
3. Grazzini G., D'Albero M., A Jet-Pump inverse cycle with water pumping column. Proceedings of Natural Working Fluids, June 2-5, Oslo, Norway, pp 27-33, 1998
4. Grazzini G., Mariani A., A Simple Program to Design a Multi-Stage Jet-Pump for Refrigeration Cycle, Energy Convers. Mgmt, vol 39, n° 16-18, pp 1827-1834, 1998
5. Grazzini G., Rocchetti A., Coefficienti correttivi per codice monodimensionale per il dimensionamento di un eiettore a due stadi, in italian, XIX UIT Congress, June 2001, Modena, Italy

6. Grazzini G., Rocchetti A., Direct Evaporation Steam Ejector Refrigeration Plant, HPC’01 conference, pp 211-216,Sept. 2001, Paris France,
7. Grazzini G., Rocchetti A., Numerical Optimisation of a Two-Stages Ejector Refrigeration Plant, Int. J. of Refrigeration 25, pp. 621-633, 2002
8. Keenan J. H., Neumann E. P., Lusterwerk F., An Investigation of jet-pumps design by theory and experiment, ASME J. Appl. Mech., pp.299-309, 1950
9. Klein S. A., Harvey A.H., NIST/ASME Steam Properties. U.S. Department of Commerce, 1996, Gaithersburg Maryland