DIRECT EVAPORATION STEAM EJECTOR REFRIGERATION PLANT

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ABSTRACT

A numerical optimisation of a steam ejector refrigeration plant have been carried out, based on the Complex optimisation method. The plant is treated as an open system exchanging thermal power with three water flow rate. Ejector is a two stage with annular primary in the second stage. The heat exchangers are shell and tube with external water in the tubes. For the evaporator the solution with shell and tube is compared to that with flash chamber. Better performances are obtained for the plant with flash evaporation. In this plant configuration, a very little evaporation area and, therefore, short heat exchanger dimensions are obtained. However, the numerical flash evaporation model used requires high solving time and gives worse statistical distribution of simulation results.

1.INTRODUCTION

Jet-refrigeration cycles are alternative to the conventional mechanical vapor compression cycles and seem to provide an interesting solution to the environment protection, for their features of simplicity, low plant costs, reliability and their skill to deal with two-phase fluids during compression, which makes it possible to work with water as operating fluid. Several efforts have been made to improve the efficiency of this cycle. Toward this direction and being the cycle performance highly influenced by the evaporator efficiency [1], in this work a flash evaporator is introduced into the plant. In a previous author's work [1] a numerical optimisation of the refrigeration steam ejector plant has been made using shell and tube heat exchangers. In this work a numerical optimisation of the refrigeration steam ejector plant with flash evaporator is compared with the previous one.

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2.EJECTOR PLANT AND NUMERICAL OPTIMISATION DESCRIPTIONS

The ejector refrigeration cycle is a three-thermal refrigeration machine that can be considered as an overlapping of a motive and a refrigeration cycle. A schematic illustration of the ejector refrigeration plant is shown in Figure 1. The thermal power Q_G, supplied to the generator from the highest temperature thermal source, is partially converted into work giving thermal power Q'_C to the intermediate temperature thermal source at the condenser. This work produces the refrigerating effect at the evaporator, transferring thermal power Q_E from lower temperature to the intermediate one. The overlapping appears at condenser and ejector, that is the device that transfers the work to the refrigerating fluid.

The plant is considered as an open system that exchanges heat with the three thermal sources and needs external work to overcame the friction losses on the water side of the heat exchangers, to pump fluid from the condenser to the generator and to bring the refrigerated water, outgoing from the flash evaporator, from the evaporator pressure to the atmospheric one. The internal pressure drops of refrigerating fluid are neglected. The inlet temperatures of the external water streams for hot, intermediate and cold source are fixed.

The ejector is a compact two stage with annular primary at the second stage. This configuration was previously investigated from the authors [1, 2].

Direct search methods must be used when the gradient of the objective function is a complex vector of the design variables, which appreciably complicate the analytic expression. The procedure we used, the well-known complex method proposed by Box et al. [3], begins by randomly and sequentially generating a set of trial points in the space of the independent variables and evaluating the 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 optimisation function is based on the plant COP defined as:

$$COP = \frac{Q_E}{Q_G + \left| W_{\lambda E} + W_{\lambda C} + W_{\lambda G} + W_{Pu} \right|} \tag{1}$$

The fixed input data are the inlet external water temperature at the heat exchangers and the evaporator heat power. The independent variables are the external water flow rates, the number and the inner diameter of the tubes in heat exchangers, the two steam flow rates at the ejector primaries, the boiling temperature at the evaporator and generator, the superheating at the generator and the condensing

temperature. These are chosen randomly by the numerical code in previously defined ranges. In the case

of shell and tube evaporator the W_{Pu} represent the power needed to pump the fluid from the condenser to

the generator; in the case of flash evaporator the power of the pump bringing the water from the flash

chamber pressure to the atmospheric value, is added.

This optimisation method had been previously used by the authors showing a good statistical distribution

of the results and an acceptable solving time [1, 4].

3. FLASH EVAPORATION MODEL AND EVAPORATOR SIMULATION

Flash evaporator is well known device used in seawater desalination process for industrial, domestic and

shipboard purposes. Recently, flash evaporators have been applied to energy saving systems and various

flashing methods have been investigated [5, 6, 7, 8, 9].

The flash evaporation occurs when a liquid is exposed to a sudden pressure drop below the saturation

vapor pressure (for pure liquids) or the equilibrium vapor pressure (for solutions) corresponding to the

liquid temperature. In an adiabatic process, part of the liquid evaporates to regain equilibrium drawing

latent heat of vaporisation from the liquid, whose temperature drops towards the saturation temperature

(pure liquids) or the equilibrium temperature (solutions), corresponding to the lowered pressure.

In this work a superheated flashing liquid method had been used to simulate the flash evaporation

process. Figure 2 illustrates the schematic layout of the flash evaporator. The water stream coming

throughout the lamination to the separator is divided in steam, that is sent to the evaporator; and in

saturated liquid that is added to the water cooled outgoing from the evaporator. The water coming from

cooled loop at atmospheric pressure is laminated to the pressure P_E. In the evaporator the suction port of

the ejector controls the pressure value of the saturated steam coming from the separator.

The flash evaporation chamber width is fixed; the unknown length is calculated summing the steps Δx. At

each step the code evaluates the steam flow rate m by the following relation [10]:

$$m = \frac{2\sigma}{2-\sigma} \Delta A \left(\frac{\overline{M}}{2\pi\overline{R}} \right)^{1/2} \left(\frac{P_E}{T_v^{1/2}} - \frac{P_{sat}[T_{liq}]}{T_{liq}^{1/2}} \right) \tag{2}$$

where the accommodation coefficient σ has a value of 0.03 [10]

The thermal power q subtracted to the liquid is:

$$q = m \cdot r[P_E] \tag{3}$$

and the residual flow rate and the temperature of the liquid at the step outlet:

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$$M_{out} = M_{in} - m \tag{4}$$

$$T_{out\,liq} = \dfrac{\dfrac{M_{in} \cdot i_{in} - m \cdot q}{M_{out}}}{c_{p\,liq}} \tag{5}$$

These two parameters are assigned to the inlet liquid at the next step. The code iterates the procedure until the total thermal power q_{tot} , subtracted to the liquid is equal to the required thermal power at the evaporator.

The steam from the flash chamber and that from the separator go to the suction port of the ejector. The cooled water outgoing from the evaporator is mixed with the water coming from the separator and pumped to the atmospheric pressure. The mechanical power needed by this pump is also considered.

4.SHELL AND TUBE HEAT EXCHANGERS MODEL

Shell and tube heat exchangers, with external water inside the tubes, were chosen in this study to assume constant pressure heat exchanges for the vapor of the ejector cycle. Each heat exchanger is numerically simulated in independent way to consider different heat transfer conditions. The evaporator, in the shell and tube case, is flowed (without superheating section); the condenser has a first rank cooling the vapor to the saturation line and a condensing section, with the same tubes number. The generator has three different sections, to preheating, boiling and superheating the working fluid.

Forced convection heat exchange factor h_w for external water side is calculated by the Petukhov relation $(2300 < Re < 5 \cdot 10^6)$[11]:

$$Nu = \left[\dfrac{(Re - 1000) \cdot Pr \cdot z}{1 + 12.7 \cdot \left(Pr^{2/3} - 1\right) \cdot z^{1/2}} \right] \tag{6}$$

where $z = \dfrac{0.5}{(1.58 \cdot \ln(Re) - 3.28)^2}$

In the laminar field we assumed Nu = 3.66, considering fully developed condition with constant wall temperature.

Since the evaporator and the evaporating section of the generator was considered to be flowed, the mean boiling heat transfer factor is evaluated by [10]:

$$h_b = 0.62 \cdot \left[\dfrac{k_v^3 \cdot g \cdot \rho_v \cdot (\rho_1 - \rho_v) \cdot r'}{\mu_v \cdot (T_w - T_{f\,sat}) \cdot D} \right]^{1/4} \tag{7}$$

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where r'=[r+0.40 c_{pv} (T_w- $T_{f\ sat}$)] and T_w is the mean temperature of the external water at the heat exchanger boiling section.

The mean condensing heat exchange factor is evaluated from [10]:

$$h_{co} = \frac{0.728 \cdot k_1}{n_r D} \cdot \left[\frac{g \cdot (\rho_1 - \rho_v) \cdot (n_r D)^3 \cdot r}{k_1 \cdot v_1 \cdot (T_{f\,sat} - T_w)} \right]^{1/4} \tag{8}$$

where T_w is the mean temperature of the external water at the heat exchanger condensing section.

External heat exchange factor $h_{f\,x}$ for first rank cooling vapor to the saturation line is evaluated in the condenser by [12]:

$$Nu = 0.33 \, Re^{0.6} \, Pr^{1/3} \tag{9}$$

The same relation allows to evaluate the external heat exchange factor $h_{f\,y}$ for the steam superheating at the generator when the Reynolds number of this stream is higher than 1000. Otherwise this factor is evaluated by [12]:

$$Nu = 0.32 + 0.43 \, Re^{0.52} \tag{10}$$

The heat exchange factors used when water is heated to the saturation line and when is super-heated, are obtained by the Raithby and Hollands [13] correlation for natural convection:

$$h_z = \left(\left(\frac{2\left(1 - \frac{0.13}{F^{0.16}}\right)}{Log\left(1 + 2 * \frac{1 - \frac{0.13}{F^{0.16}}}{F}\right)} \right)^{10} + \left(0.103 Ra^{1/3}\right)^{10} \right)^{0.1} \tag{11}$$

where $F = 0.426916 Ra^{1/4}$

The thermal power exchanged during phase-change is evaluated by

$$Q = U \cdot \Delta TML \tag{12}$$

where $U = \dfrac{\pi \, D \cdot n \cdot L}{\dfrac{1}{h_w} + \dfrac{1}{\left(1 + 2\dfrac{s}{D}\right) h_f}}$ is the mean global heat transfer coefficient ignoring thermal

resistance of the tube. The heat exchange coefficient h_f is given by eq. (7) for evaporator and generator

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boiling sections, by eq.(11) for preheating sections at the generator and by eq. (9) or (10) for superheating section, by eq. (8) for condensing section, and by eq. (9) for the first cooling rank of the condenser. The heat exchange coefficient h_w is obtained from eq. (6).

The pressure losses are calculated using the explicit Moody relation, which Haaland [14] proposed again for the friction factor:

$$\lambda = 0.0055\left[1+\left(2\cdot10^4\ \frac{\varepsilon}{D}+\frac{10^6}{Re}\right)\right] \tag{13}$$

5.RESULTS AND CONCLUSIONS

Table 1 shows some results and data input and the admissible ranges of the independent variables. The upper limit of the saturation temperature at the evaporator made by shell and tube heat exchanger is two degree less the external water inlet temperature to avoid very high area of the heat exchanger. When the flash evaporator is considered simulation the same temperature is only one degree less the external water inlet temperature.

In the same table the results of four runs are presented for each simulation. Results are in agreement with the literature data [15-18]. The COP values obtained using flash evaporator are, on the averaged value, about 13% highest than those achieved with the shell and tube evaporator. The Second Law efficiency, considering the plant as an open system, is calculated with following equations:

$$\eta_{II} = \frac{COP}{COP_{Ca}} \tag{14}$$

$$\text{where} \quad COP_{Ca} = \frac{T_{w\,Gin} - T_{w\,Cin}}{T_{w\,Gin}}\ \frac{T_{w\,Eout}}{T_{w\,Cin} - T_{w\,Eout}} \tag{15}$$

In any case the saturation temperatures at the evaporator and condenser clearly tend to the upper limit of the admissible range. Table 2 shows the dimensional and operating parameters for the comparison between the flash and the shell and tube evaporator. The flash evaporator presents a small liquid-vapor interface area. In the simulation a fixed width of 1 meter and steps of 0.01 m are chosen. The shell and tube evaporator shows a high number of tubes and a very large area. The pressure losses, that are included in the optimisation function, are very low. The mechanical power (W_{Pu}) needed to pump water from the flash evaporator to the atmospheric pressure, is not negligible and has to be considered in the optimisation function.

The numerical simulation of the flash evaporation process is very heavy. The optimisation process become very slow and few reliable for a statistical approach of the results. The good results obtained with flash evaporator push to improve the numerical optimisation model.

ACKNOWLEDGEMENTS

REFERENCES

[1]   Grazzini G., Rocchetti A., Numerical optimization of a two-stage ejector refrigeration plant. Poster Session at International Onsager- Workshop, Optimisation in Industrial and Living Systems, March 11-14, 2000, Oud Poelgeest, Leiden, The Netherlands.

[2]   Grazzini G., Mariani A., A Simple Program to design a Multi-Stage Jet Pump for Refrigeration Cycles; Energy Convers. Mgmt Vol.39, No.16-18, pp.1827-1834, 1998.

[3]   Box M.J., Davies D., Swann W.H., Non-Linear Optimisation Techniques, ICI Monograph, n. 5, Oliver & Boyd, London UK (1969)

[4]   Grazzini G., Rinaldi R. Thermodynamic optimal design of heat exchangers for an irreversible refrigerator Accepted for Int. J. of Thermal Sci.

[5]   Miyatake O., Comparative Study of Flash Evaporation Rates, Desalination, n.96, pp 163-171, (1994)

[6]   Aoki I., Water Flash Evaporation under Low Pressure Conditions, Heat Transfer – Japanese Research, n. 23, pp.544 – 555, (1994)

[7]   Aoki I., Analysis of Characteristics of Water Flash Evaporation under Low-Pressure Condition, Heat Transfer – Asian Research, n. 29, pp. 22-33, (2000)

[8]   Jin W.X., Low S.C. and Yu S.C.M., Some Experiment Observations On the Single And Multi-Phase Flow Patterns in a Model Flash Evaporation Chamber, Int. Comm. Heat Mass Transfer, vol. 26, n.6, pp. 839-848, (1999)

[9]   Miyatake O., Kabir M.E., Noda H. and Sugihara K., Transient Characteristics and Performance of Hybrid Latent Heat Storage and Spray Flash Evaporation System, J. of Chemical Engineering of Japan, vol. 30, n.6, pp. 1076-1082, (1997)

[10]   Carey V.P., Liquid-vapour phase-change phenomena; Hemisphere, Washington, USA, (1992)

[11] Kays W.M., Perkins H.C., Forced Convection Internal Flow in Ducts, in Handbook of Heat Transfer Fundamentals; Ed. W.M. Rohsenow, J.P. Hartnett, E.N. Ganic, McGraw-Hill, (1985)

[12] ASHRAE handbook, Fundamentals, ASHRAE, Atlanta, USA, (1997)

[13] Raithby G.D., Hollands K.G.T., Natural Convection, in Handbook of Heat Transfer Fundamentals; Ed. Rohsenow W.M., Hartnett J.P., Ganic E.N., McGraw-Hill, (1985)

[14] Haaland S.E., Simple and explicit formulas for the friction factor in turbulent pipe flow, Jour. Fluids Eng., vol. 105, pp. 89-90, (1983)

[15] Champoussin J.C., Hostache G., Performances d'un systeme trytherme a ejecteur a basse temperature de boilleur, 16° Cong. Int. du Froid IIF, Paris, B1, 26-31, (1983)

[16] Boufmaraf L., Lallemand A., Performances d'une machine trytherme à éjecteur utilisant des mélanges de fluides frigorigènes, Int. Journ. of Refrigeration, vol. 22, 580-589, (1999)

[17] Huang B.J., Chang J.M., Empirical correlation for ejector design, Int. Journ. of Refrigeration, vol. 22, 379-388, (1998)

[18] Munday J.T., Bagster D.F., A New Ejector Theory Applied to Steam Jet Refrigeration, Ind. Eng. Chem. Preocess Des. Dev., vol. 16, n. 4, (1997)

NOMENCLATURE

A	Heat exchange area (m)
c_p	Specific heat (J/(kg K))
COP	Coefficient of performance
D	Inner diameter of the heat exchanger tubes (m)
ΔA	Flashing step area (m^2)
Δx	Flashing step length (m)
g	Gravity factor (m/s^2)
h	Heat exchange factor (W/(m^2 K))
i	Specific enthalpy (J/(kg K))
k	Thermal conductivity (W/(m K))
L	Tube length of the heat exchanger (m)
m	Flashing step flow rate (kg/s)
M	Water flow rate (kg/s)
$\overline{M}$	Molecular weight
n	Number of tubes of the heat exchanger
Nu	Nusselt number
P	Pressure [Pa]
Pr	Prandtl number
q	Flashing step thermal power [W]

Q	Thermal power (W)
r	Latent heat of vaporisation (J/(kg K))
$\overline{R}$	Universal gas constant (=8.3144 kJ/(kg mol K))
Ra	Rayleigh number
Re	Reynolds number
T	Temperature (K)
U	Global thermal exchange factor (W/K)
W	Power (W)
ΔTML	Logarithmic mean temperature difference (K)

GREEK

ε	Absolute roughness of the heat exchanger tubes (m)
η	Efficiency
λ	Friction factor
μ	Dynamic viscosity ((N s)/m^2)
ν	Kinematic viscosity (m^2/s)
ρ	Density (kg/m^3)

| σ | Flash evaporation accommodation coefficient | liq | Liquid |
		out	Outlet
SUBSCRIPTS		Pu	Pump
II	Referred to Second Law of Thermodynamic	r	Rank
		sat	Saturation
b	Boiling section	tot	Referred to the global flash process
C	Condenser	v	Saturated steam
Ca	Referred to Carnot cycle	x	First cooling rank section of the condenser
co	Condensing section		
E	Evaporator	y	Superheating section of the generator
f	Working cycle fluid	z	Preheating section of the generator
G	Generator	w	Water
in	Inlet	λ	Related to pressure losses
l	Saturated liquid		
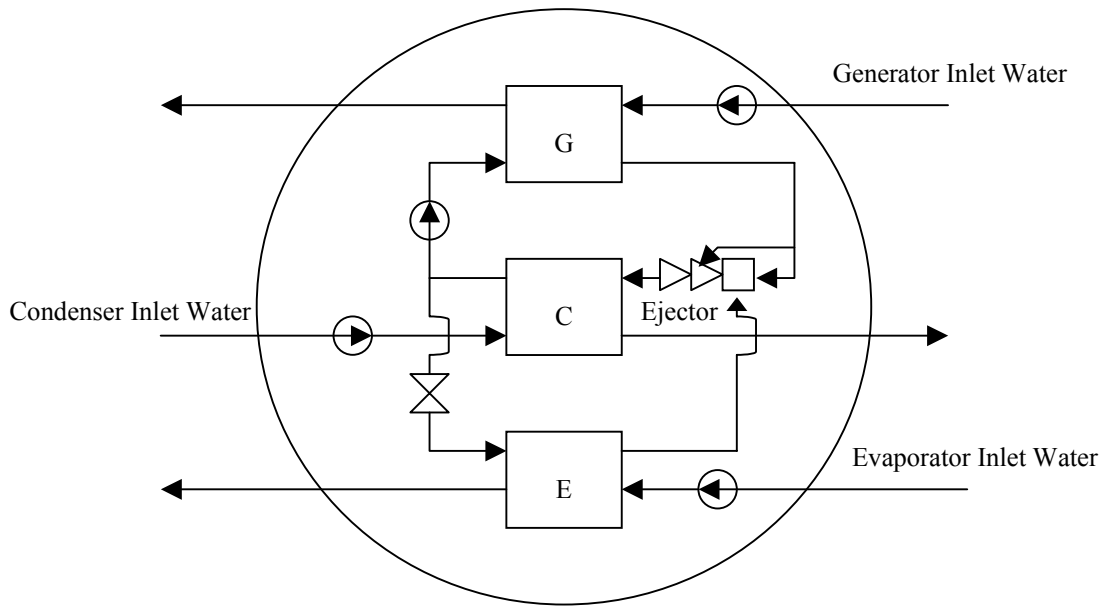
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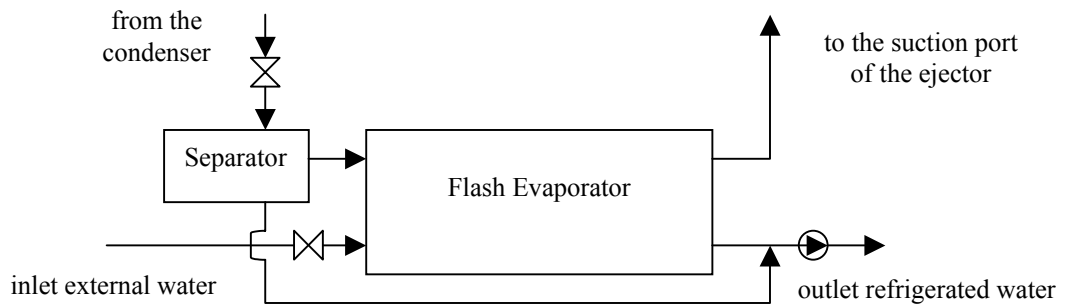
Figure 2

INPUT DATA			
Q_E (W)	$T_{w\ in\ E}$ (K)	$T_{w\ in\ C}$ (K)	$T_{w\ in\ G}$ (K)
5000	285.15	303.15	393.15

	range values	COP	h_{II}	M_{wE} (kg/s)	M_{wC} (kg/s)	M_{wG} (kg/s)	T_{satE} (K)	T_{satC} (K)	T_{satG} (K)	ΔT_G (K)
flash evaporator	lower			0.05	0.50	0.50	275.15	313.15	358.15	10
	upper			5.00	20.00	20.00	284.15	321.15	383.15	35
	run 1	0.1821	0.1068	2.68	4.47	18.37	283.82	313.15	380.16	10.53
	run 2	0.1597	0.1073	5.00	11.23	1.07	284.00	313.78	371.78	14.31
	run 3	0.1810	0.1080	4.10	16.25	19.91	284.03	313.15	377.95	12.84
	run 4	0.1783	0.1101	3.53	18.93	8.76	284.15	313.15	375.18	11.79
shell and tube evaporator	lower			0.05	0.50	0.50	275.15	313.15	358.15	10
	upper			5.00	20.00	20.00	283.15	321.15	383.15	35
	run 1	0.1509	0.1064	3.32	11.58	3.83	283.13	313.25	368.87	19.19
	run 2	0.1559	0.1047	4.96	19.95	19.91	283.13	313.41	372.77	12.11
	run 3	0.1602	0.1051	0.99	0.97	17.50	283.08	313.21	373.89	15.12
	run 4	0.1534	0.1068	2.83	13.45	10.03	283.14	313.18	369.56	10.12

Table 1

	run	M_{in} (kg/s)	M_{out} (kg/s)	m_{tot} (kg/s)	$T_{w\ in}$ (K)	$T_{w\ out}$ (K)	interface area (m^2)	width (m)	W_{pu} (W)
flash evaporator	1	2.68	2.68	2.03E-03	285.15	284.74	0.62	1	268.23
	2	5.00	5.00	2.03E-03	285.15	284.94	0.66	1	499.77
	3	4.10	4.10	2.03E-03	285.15	284.89	0.70	1	410.02
	4	3.53	3.53	2.03E-03	285.15	284.84	0.82	1	352.79
	run	M (kg/s)	Cycle flow rate (kg/s)	$T_{w\ in}$ (K)	$T_{w\ out}$ (K)	number of tubes	L (m)	A (m^2)	W_{lE} (W)
shell and tube evaporator	1	3.32	2.13E-03	285.15	284.79	16	43.96	32.04	1.43E+01
	2	4.96	2.13E-03	285.15	284.91	78	8.49	124.79	1.42E-04
	3	0.98	2.13E-03	285.15	283.94	194	8.33	165.60	9.78E-06
	4	2.83	2.13E-03	285.15	284.73	72	14.48	167.02	1.71E-04

Table 2

FIGURE CAPTIONS

Figure 1: The steam ejector refrigeration plant as a open system

Figure 2: Schematic layout of the flash evaporator

TABLE CAPTIONS

Table 1: Comparison of the cycle performances and operational parameters between the flash and shell and tube evaporators

Table 2: Comparison of the geometrical and operational parameter between the flash and shell and tube evaporators