DIRECT EVAPORATION STEAM EJECTOR REFRIGERATION PLANT

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ABSTRACT

A numerical optimisation of a steam ejector refrigeration plant has 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 were obtained for the plant with flash evaporation. In this plant configuration, a very small evaporation area and, therefore, short heat exchanger dimensions were obtained. However, the numerical flash evaporation model used requires high solving time and gives worse statistical distribution of simulation results.

KEYWORDS

Ejector/flash evaporation/refrigerator/optimisation/heat exchange irreversibilities/heat exchanger design/open system/steady-state

1. INTRODUCTION

Jet-refrigeration cycles are alternative to conventional mechanical vapor compression cycles and seem to provide an interesting solution to environment protection, owing to their features of simplicity, low plant costs, reliability and skill in dealing 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. For this purpose and since the cycle performance is highly influenced by evaporator efficiency [1], in this research a flash evaporator was introduced into the plant. In previous work by the authors [1] a numerical optimisation of the refrigeration steam ejector plant was carried out using shell and tube heat exchangers. Here a numerical optimisation of the refrigeration steam ejector plant with flash evaporator is compared with the previous one.

2. EJECTOR PLANT AND NUMERICAL OPTIMISATION DESCRIPTIONS

The ejector refrigeration cycle is a three-thermal refrigeration machine that can be considered an overlap of a motive and a refrigeration cycle. A schematic illustration of the ejector refrigeration plant is shown in Figure 1. Thermal power Q_G, supplied to the generator by 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 a lower temperature to an intermediate one. The overlap appears at condenser and ejector, that is the device that transfers the work to the refrigerating fluid.

The plant is considered 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 evaporator pressure to the atmospheric one. The internal pressure drops of refrigerating fluid are not considered. 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 has been previously investigated by 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 they 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:

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

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 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.

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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 a 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].
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 vaporisation heat from the liquid, whose temperature drops towards the saturation temperature (pure liquids) or the equilibrium temperature (solutions), corresponding to the lowered pressure.
In this research a superheated flashing liquid method was used to simulate the flash evaporation process. Figure 2 illustrates the schematic layout of the flash evaporator. The water stream coming through 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 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 by summing steps Δx. At each step the code evaluates the steam flow rate *m* by the following relation [10]:

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

where the accommodation coefficient σ has a value of 0.03 [10] and $P_{sat} = f(T_{liq})$. Because of the properties of water $P_E < P_{sat} \Rightarrow T_v < T_{liq}$ and $\left(\frac{P_E}{T_v^{\;1/2}} - \frac{P_{sat}}{T_{liq}^{\;1/2}}\right) > 0$.

The thermal power *q* subtracted from the liquid is:

$$q = m\, r \tag{3}$$

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

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

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

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 from 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 an 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 number of tubes. The generator has three different sections, for 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 < $5\cdot10^6$)[11]:

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

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

In the laminar field we assumed Nu = 3.66, considering fully developed conditions 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]:

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

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]:

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

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]:

$$Nu = 0.33\,\mathrm{Re}^{0.6}\,\mathrm{Pr}^{1/3} \qquad (9)$$

The same relation allows evaluation of 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]:

$$Nu = 0.32 + 0.43\,\mathrm{Re}^{0.52} \qquad (10)$$

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

$$\mathrm{Nu}_z = \left(\left(\frac{2\left(1 - \frac{0.13}{\mathrm{F}^{0.16}}\right)}{\mathrm{Log}\left(1 + 2 * \frac{1 - \frac{0.13}{\mathrm{F}^{0.16}}}{\mathrm{F}}\right)} \right)^{10} + \left(0.103\,\mathrm{Ra}^{1/3}\right)^{10} \right)^{0.1} \qquad (11)$$

where $F=0.427\,\mathrm{Ra}^{1/4}$

The thermal power exchanged during phase-change is evaluated by

$$Q = U\,\Delta TML \qquad (12)$$

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 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 by eq. (6).

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

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

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 degrees less than 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 than 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% higher than those achieved with the shell and tube evaporator. The Second Law efficiency, considering the plant as an open system, is calculated with the following equations:

$$\eta_{II} = \frac{COP}{COP_{Ca}} \qquad (14)$$

where

$$COP_{Ca} = \frac{T_{wGin} - T_{wCin}}{T_{wGin}}\,\frac{T_{wEin}}{T_{wCin} - T_{wEin}} \qquad (15)$$

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 was a small liquid-vapor interface area. In the simulation a fixed width of 1 meter and steps of 0.01 m were 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 becomes very slow and not very reliable for a statistical approach to the results. The good results obtained with flash evaporator encourage improvement of the numerical optimisation model.

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NOMENCLATURE

A	Heat exchange area (m^2)
c_p	Specific heat ($J.kg^{-1}.K^{-1}$)
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 ($W.m^{-2}.K^{-1}$)
i	Specific enthalpy ($J.kg^{-1}.K^{-1}$)
k	Thermal conductivity ($W.m^{-1}.K^{-1}$)
L	Tube length of the heat exchanger (m)
m	Flashing step flow rate ($kg.s^{-1}$)
M	Water flow rate ($kg.s^{-1}$)
$\overline{M}$	Molar mass ($kg.kmol^{-1}$)
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 ($J.kg^{-1}.K^{-1}$)
$\overline{R}$	Universal gas constant (=8.3144 $kJ.kmol^{-1}.K^{-1}$)
Ra	Rayleigh number
Re	Reynolds number
s	Thickness (m)
T	Temperature (K)
U	Global ther. exchange factor ($W.K^{-1}$)
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 ($m^2.s^{-1}$)
ρ	Density ($kg.m^{-3}$)
σ	Flash evaporation accommodation coefficient

SUBSCRIPTS

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

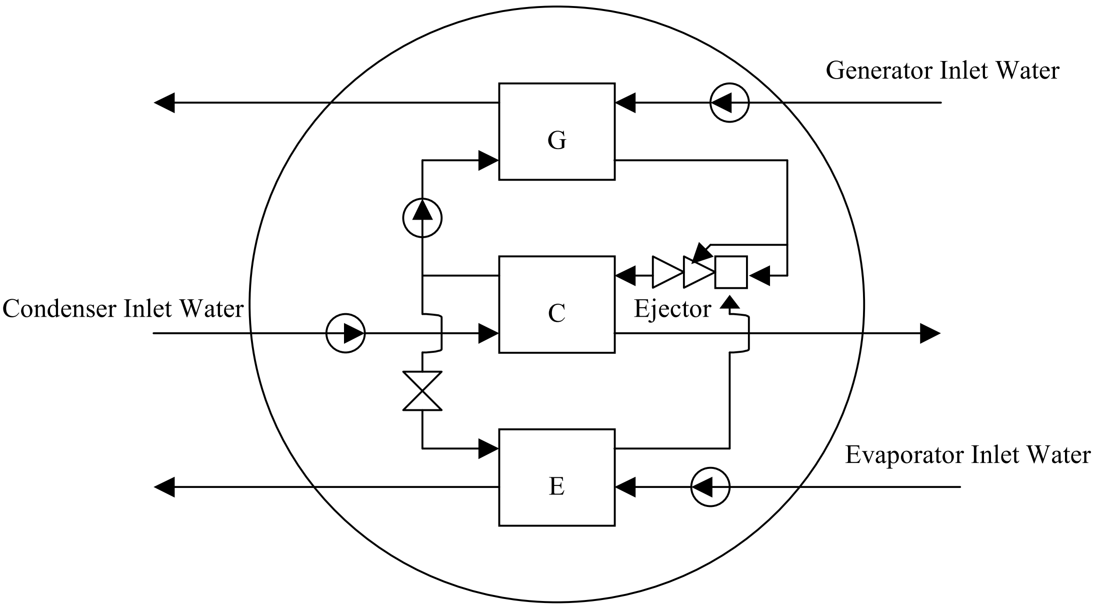

Figure 1: **The steam ejector refrigeration plant as an open system**

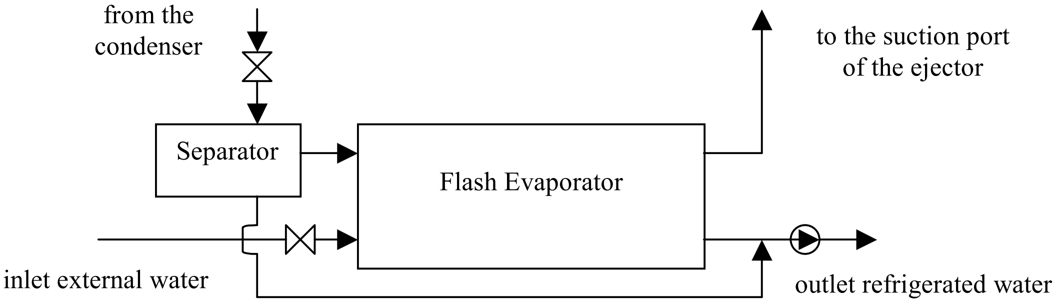

Figure 2: Schematic layout of the flash evaporator

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

			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	η_{II}	M_{wE} (kg.s^{-1})	M_{wC} (kg.s^{-1})	M_{wG} (kg.s^{-1})	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 2: Comparison of the geometrical and operational parameter between the flash and shell and tube evaporators

	run	M_{in} (kg.s^{-1})	M_{out} (kg.s^{-1})	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
shell and tube evaporator	run	M (kg.s^{-1})	Cycle flow rate (kg.s^{-1})	$T_{w\ in}$ (K)	$T_{w\ out}$ (K)	number of tubes	L (m)	A (m^2)	W_{IE} (W)
	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