# ON THE VALIDATY OF A DESIGN METHOD TO ESTIMATE THE SOLAR FRACTION FOR AN EJECTOR COOLING SYSTEM

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ABSTRACT

The present paper is concerned with the simulation of an ejector cooling system in order to investigate the validity of a design method to estimate the solar fraction. The cooling capacity of the ejector cycle is assumed to be constant during day periods. The ejector is assumed to steadily operate at its optimum efficiency point. The solar fraction derived from hourly simulation of the system is compared with estimates obtained by the $f - \overline{\phi} - chart$ method based on the utilizability concept. An equivalent minimum temperature for the utilizability of the solar system is found, which is proved to be different but close to the vapor generator temperature of the ejector cycle.

## 1. INTRODUCTION

Global effort has insofar been devoted to develop renewable energy systems in favor of $CO_2$ emission reduction. Solar energy has been considered worldwide as an effective alternative, to reduce fossil fuel and electric energy consumption in domestic water heating application. Flat plate collectors and evacuated collectors are proven to be cost effective for many applications in domestic and industrial process heat, for temperatures less than 100$^{o}$C. On the other hand, solar driven cooling cycles are hardly competitive with mechanical compression cycles. There are few real situations where solar driven absorption cooling systems can be competitive with mechanical compression. Capital cost of solar collectors and barriers arising from architecture constraints contribute to reduce the economical advantages in favor of absorption cooling cycles. Furthermore, mechanical compressors have become cheaper and more efficient in the past ten years. The situation is not better for ejector cooling cycles. The coefficient of performance (COP) of a single stage absorption chiller of lithium bromide – water can easily reach 0.66 while the COP of an ejector cycle, under the same operation temperatures hardly reaches 0.6. The lower the COP the greater the optimum collector area needed to meet the load requirements. Therefore the potential advantages arising from the lower cost of an ejector cooling system is impaired by the requirement of additional collector area.

The ejector system has usually to be simulated in the hourly basis, with the help of Typical Meteorological Year (TMY) database. TMY database are usually available in the meteorological services of any developed country. However, qualified TMY database are hardly available in developing and undeveloped countries. On the other hand, monthly averages of global and beam solar radiation incident on horizontal surface became accessible by most of the countries, thanks to the well succeed techniques to estimate incoming solar radiation derived from satellites, as reported in [1]. The incoming radiation can presently be estimated with uncertainty around 5% against ground truth data [2]. For the above reasons, the design $f-\overline{\phi}-chart$ method, as proposed in [3], based on monthly average solar radiation, is still useful to design and optimize solar cooling systems, as well as to analyze the economical feasibility of theses systems for given economical scenarios. These methods have been successfully used in designing systems to provide process heat, as well as for cooling applications, as reported in [4]. In [4] is presented an analysis of an optimized ejector cooling system, reporting the results of simulation based on hourly data, with comparison with the predictions given from the $f-\overline{\phi}-chart$ method.

The present paper reports simulation results to show that the $f-\overline{\phi}-chart$ method of can be validated in terms of the monthly and annual solar fraction. The validation was carried out for the city of Florianopolis – Brazil (latitude 27,6 South) for which a TMY database is available. The database was built from a fourteen years long solar radiation data series collected in a BSRN surface station [5]. The $f-\overline{\phi}-chart$ method is applicable to cases for which the heat is supplied to the load only when the heating fluid temperature is above some minimum temperature $T_{min}$. It means that the method is expected not to be valid in the circumstance the process heat depends on the temperature of the loading system. In the case of an ejector cooling cycle, the process heat depends not only of the condenser temperature, but also on the vapor generator temperature. As is shown in this paper, the $f-\overline{\phi}-chart$ method can be validated for ejector cooling systems, once a minimum temperature around the temperature of the vapor generator is properly chosen.

## 2. THE EJECTOR SOLAR COOLING SYSTEM

The ejector solar cooling system is conceived as a solar heating system which supplies heat to an ejector cooling cycle as shown in Fig. 1.

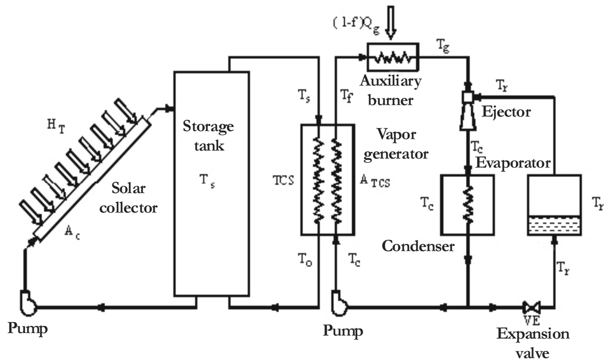


Fig 1: Solar assisted ejector cooling system.

The working fluid evaporates in the vapor generator at the saturation temperature $T_g$ to provide the primary stream to the ejector nozzle. The mixture of the primary stream with the secondary stream coming from the evaporator at $T_r$, condenses in the condenser at temperature $T_c$. The condensed liquid stream leaves the condenser and splits into two streams, one that flows back to the evaporator though the expansion value, and the other that flows back to the vapor generator. The ratio of the primary to the secondary nozzle cross section areas of the ejector are designed in order to achieve the maximum flow ratio in the evaporator, for a given flow ratio of the primary stream. Algorithms for simulation and optimization of the ejector nozzle operation can be found in [6,7,8]. It is assumed here that auxiliary heat is provided to the ejector system through a burner, whenever the heat from the solar collector is not enough in order to meet the load requirements, so that the steady state flow rate of the refrigerant working fluid is assured.

## 3. GOVERNING EQUATIONS

The full mixing model is assumed here in order to simplify the energy balance of the solar system considered. In other words, it is assumed there is no temperature gradient in the storage tank, as well as no pressure and temperature drop in the system pipes.

The energy balance in the solar system conjugated to the vapor generator of the ejector cycle leads to the following ordinary differential equation

$$(mc)_s \frac{dT_s}{dt} = A_c\left[F_R(\tau\alpha)_n K_{\tau\alpha} G_T - F_R U_L (T_s - T_{ae})\right] - (UA)_s (T_s - T_{ai}) - \alpha_s Q_s \tag{1}$$

where $T_s$ is the temperature of the heating fluid in the storage tank, $(mc_p)_s$ is the thermal capacity of the heating fluid in the storage tank, $F_R(\tau\alpha)_n$ and $F_R U_L$ are respectively the energy gain and loss coefficient of the straight line correlation of the flat plate collector efficiency, $A_c$ is the useful collector area, $(UA)_s$ is the heat loss coefficient of the storage tank $(kW/K)$, $T_{ai}$ and $T_{ae}$ are the ambient temperatures for the storage tank and solar collector, respectively, $G_T(W/m^2)$ is the solar radiation incident on the tilted collector, $K_{\tau\alpha}$ is the incidence angle modifier of the collector, $Q_g = Q_r / COP$ is the heat input of the cooling cycle $(kW)$, where $Q_r$ is the cooling capacity, and COP is the coefficient of performance of the ejector, which is a function of $T_c$, $T_g$, and $T_r$. Here $\alpha_s$ is a flag set to vanish for $T_s \le T_c$ and set equal to the unity for $T_s > T_c$. The vapor generator is considered to be a two-phase heat exchanger, so that two distinct cases should be considered as follows.

The hourly solar fraction is defined by $f = Q_s / Q_g$. The annual solar fraction $f_a$ is the average of the hourly solar fraction $f$ based on the sum of the hours of the day duration in the year.

Case I: Single-phase heat supply $(T_f < T_g)$.

In this case the heat input to equation (1) is given by

$$Q_s = W_{min}\varepsilon_s (T_s - T_c) = \omega_{ej} c_{rl} (T_f - T_c) \tag{2}$$

where $\varepsilon_s$ the single-phase heat exchanger effectivity defined as

$$\varepsilon_s = \omega_{ej} c_{rl} (T_f - T_c) / W_{min} (T_s - T_c) \tag{3}$$

where $\omega_{ej}$ is the mass rate of the refrigerant, $c_{rl}$ is the specific heat of the subcooled refrigerant, $W_{\min} = \min\{(\omega c_p)_s, \omega_{ej} c_{rl}\}$, where $(\omega c_p)_s$ is the hourly thermal capacity of the heating fluid. The effectivity $\varepsilon_s$ is a function of $(UA)_s$ and $W_{\max}/W_{\min}$, where $(UA)_s$ is the product of the global heat transfer coefficient and the single-phase heat exchanger area $A_s$, which must be equal to the total heat exchanger area $A_{TCS}$. The heating fluid temperature $T_{sl}$ corresponding to the situation for which the refrigerant temperature reaches $T_g$, by equation (2) is given by

$$T_{sl} = T_c + \omega_{ej} c_{rl} (T_g - T_c) / W_{min} \varepsilon_s \tag{4}$$

While $T_f$ remains lower than $T_g$ (and therefore $T_s$ remains lower than $T_{sl}$) the heat input is given by equation (2).

Equation (1) with the input $Q_s$ given by equation (2), can numerically be solved up to the time for which $T_s$ reaches $T_{sl}$.

Case II: Two-phase heat supply $(T_f = T_g)$.

In this case part of the heat exchanger area $A_{TCS}$ is occupied by liquid and part is occupied by vapor. The heat input is than a function of the saturated liquid and vapor enthalpies $h_l(T_g)$ and $h_v(T_g)$ according to the following equation

$$Q_s = \omega_{ej} \left(h_l - h_c + h_{lv} x_f\right) \tag{5}$$

where $h_{lv} = h_v - h_l$, $x_f$ is the vapor quality and $h_c$ is the enthalpy of the subcooled liquid at temperature $T_c(h_c = h_l(T_c))$. The maximum value of $Q_s$ is the ejector load $Q_g = \omega_{ej}(h_v - h_c)$. The liquid phase heat exchanger area is given by $A_s = A_{TCS} - A_{ev}$. For the present case the affectivities for the single-phase and the two-phase sections are given by

$$\varepsilon_s\left(\frac{U_s A_s}{W_{min}}, \frac{W_{max}}{W_{min}}\right) = \frac{\omega_{ej} c_{rl} (T_g - T_c)}{W_{min} (T_i - T_c)} \tag{6}$$

and

$$\varepsilon_{ev} = 1 - exp\left(-\frac{U_{ev} A_{ev}}{(\omega\, c_p)_s}\right) = \frac{(T_i - T_s)}{(T_g - T_s)} \tag{7}$$

It is shown in (8) that

$$x_f \varepsilon_s \omega_{ej} h_{lv}\, exp\left(\frac{-U_{ev}A_{ev}}{(\omega\, c_p)_s}\right) = \left(T_g - T_c\right)\left(\omega\, c_p\right)_s \times \left(\frac{\omega_{ej} c_{rl}}{W_{min}} - \varepsilon_s\right)\left[1 - exp\left(\frac{-U_{ev}A_{ev}}{(\omega\, c_p)_s}\right)\right] \quad (8)$$

As shown in (8), if $x_f$ reaches the unity, $T_s$ reaches the limiting-temperature $T_{sv}$ given by

$$T_{sv} = \left[T_c + \frac{\omega_{ej} c_{rl}}{\varepsilon_s W_{min}}\left(T_g - T_c\right) - \varepsilon_{ev} T_g\right] \Big/ \left(1 - \varepsilon_{ev}\right) \quad (9)$$

From energy balance in the evaporator the following equation for $x_f$ is derived

$$x_f = \varepsilon_{ev}\left(T_s - T_g\right)(\omega\, c_p)_s / \omega_{ej} h_{lv} \quad (10)$$

Equation (1) can be solved together with equations (5), (10), and (8) in terms of $T_s$, $x_f$, $A_{ev}$ and $A_s$. It should be point out that for each numerical value of $x_f$, the unknown areas $A_{ev}$ and $A_s$ can be calculated from equation (8), for a given heat exchanger area $A_{TCS}$.

## 4. NUMERICAL RESULTS

Equation (1) is solved for the case of and optimized ejector cooling cycle, for $Q_r = 10.55$kW (3 tons of refrigeration), $T_g$ = 80°C, $T_c$ = 35°C, $T_r$ = 8°C, $(\omega c_p)_s / \omega_{ej} c_{rl} = 50$, $T_{ae} = 25$ °C, $T_{ai} = 30$°C, $F_R(\tau\alpha)_n = 0.78$, $F_R U_L = 0.003$kW/m$^2$K, $COP = 0.6$ , $(\omega c_p)_s = 4.334$kW/K , and $\omega_{ej} c_{rl}$ = , 0.08667kW/K where $(mc)_s$ is chosen in order to have 75kg of heating water per squared meter of collector area $A_c$. For the present numerical example, $U_s = 1$kW/m$^2$K , $U_{cv} = 2$kW/m$^2$K . The heat exchanger chosen is of shell and tube type.

The present analysis is carried out for the particular ideal case of $A_{TCS} = 3$m$^2$ , which is considered to be a relatively large area, and for the ideal condition of $\varepsilon\, C_{min} = \infty$ for the $f - \overline{\phi} - chart$ method. The heat loss in the storage tank is neglected. Calculations were carried out for different vapor generator temperatures $T_g$. However, only the results for $T_g = 80$°C will be presented in terms of $f_a$.

The $f - \overline{\phi} - chart$ prediction for the annual solar fraction is compared with the numerical predictions of the present simulation. The best fit for $f_a$, for collector areas up to 80m$^2$ is found for $T_{\min} = 77$°C. It should be pointed out that 80m$^2$ is around the optimum economical area for the solar assisted ejector cooling system as reported in [4]. The numerical results for $f_a$ is shown in Table 1 while the plots of $f_a$ for the best fit is shown in Fig. 2.

TABLE 1

| $A_c$(m$^2$) | $f_a$ (present work) | $f - \overline{\phi} - chart$ $T_{\min} = 77$°C | $\left\lvert\frac{f_a - f_\phi}{f_a}\right\rvert \times 100$ |
|---|---|---|---|
| 10 | 0.1029 | 0.07881 | 23.4110 |
| 20 | 0.1783 | 0.1606 | 9.92709 |
| 30 | 0.257 | 0.2424 | 5.68093 |
| 40 | 0.3352 | 0.3239 | 3.37112 |
| 50 | 0.4088 | 0.4013 | 1.83464 |
| 60 | 0.4788 | 0.4771 | 0.35505 |
| 70 | 0.5471 | 0.5496 | 0.45695 |
| 80 | 0.6113 | 0.6146 | 0.53983 |

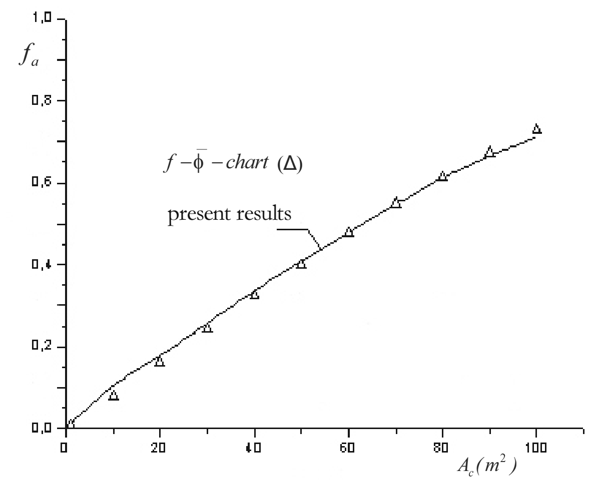


Fig. 2: Plots of $f - \overline{\phi} - chart$ results for $T_{\min}$ = 77°C against results for the annual fraction $f_a$ for $A_{TCS} = 3$m$^2$ .

## 5. CONCLUSIONS

The comparison of the $f-\overline{\phi}-chart$ method with hourly simulation of solar assisted ejector cooling cycle has been carried out, for a particular vapor generator temperature of 80°C.

The numerical results show that the $f-\overline{\phi}-chart$ prediction for the annual solar fraction is in agreement with the simulation results, for a minimum equivalent utilizability temperature of 77°C. The present results are far from being conclusive. The present analysis should be made for other values of the vapor generator temperature. It should also to be extended to other refrigerant working fluids, in order to find a correlation between vapor generation temperature and the respective equivalent minimum temperatures for the $f-\overline{\phi}-chart$ correlation. It should be pointed out here that the present results are valid for other values of $COP$, since the specific load $Q_g / \omega_{ej}$ depends only on the vapor generator temperature $T_g$ .

## 6. ACKNOWLEGMENT

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