Comparison between two different solution-pumping methods in absorption refrigeration machines

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Abstract

This paper presents a pumping method in absorption refrigeration where a vapour-driven pump is to replace the electricity-driven pump. The vapour pump is driven by a fraction of the generator's hot, high-pressure, vapour mixture. The refrigerator is thus exclusively heat-powered and rendered independent of the availability of electricity as the main energy source. The design and operation of the vapour pump is presented. The results obtained by a computer simulation program show a decrease in performance (COP), which is confirmed by the data obtained from a 1 kW cooling capacity refrigerator. Peak performances occur at higher generator temperatures when compared to a cycle using an electricity-driven pump.

Keywords: vapour pump, absorption refrigeration

Introduction

Absorption units, completely free of high grade energy demand, are the 3-fluid systems and these are only of domestic sizes. Medium and large scale units require electricity to power the pump.

There has been an attempt in the past to develop a solely heat operated 2-fluid system. Szucz (1963) proposed a thermal pump powered by the same heat as the generator. This pump has specially designed valves to control the flow of the weak and strong solutions. This thermal pump was further developed and tested by Vicatos (private collection). It was reported that in addition to the high capital cost and the demand of an electrical supply to the control valves, there was no improvement in the COP.

The proposed vapour pump replaces the electrically driven pump and uses the heat of the high pressure vapour generated within the system. The mechanism uses high pressure vapour to produce a net force, by the pump's differential piston, to drive the solution against the system's pressure gradient. The vapour, after leaving the pump, is then purged and it is absorbed by the solution in the absorber.

The vapour pump in the absorption refrigeration cycle

A fraction of the high pressure, hot vapour mixture, (stream 11) is channelled to the pump, (stream 16). The vapour mixture that leaves the pump, after having done work, is purged in the absorber (stream 17).

For the refrigeration capacity to remain unaffected, an additional amount of vapour mixture (the amount required for the pumping operation) has to be supplied by the generator. This vapour mixture, together with the refrigerant coming from the evaporator (stream 6), is absorbed in the absorber by the weak solution (stream 14).

Design and principle operation of the vapour pump

The vapour pump consists of essentially two parts as show in Figure 2:

- a pumping mechanism
- a control mechanism

Pumping is done by two opposed synchronised double acting pistons. Each piston has two vapour sides and a liquid side as shown in Figures 3a and 3b. The liquid side has a supply line from the absorber and a delivery line to the generator. The vapour sides of each piston are connected, via the control mechanism, to the generator and absorber lines.

Pumping is achieved by the net force due to the difference in piston area between the liquid and vapour sides. As the piston assembly moves, say to the left (Figure 3a), the left pump is in delivery mode and the right in suction mode. On the return stroke, the reverse action takes place. To reverse the direction of motion of the pistons, the high and low pressure lines are interchanged. This is the function of the control mechanism that directs the flow of the

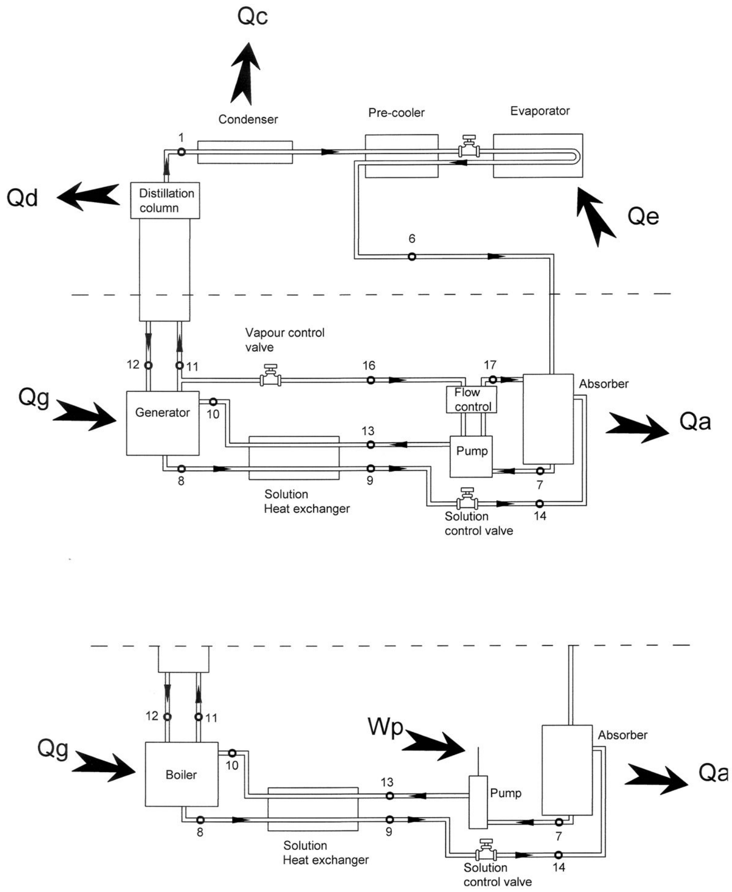

Figure 1: Schematic arrangement of the cycle with vapour driven pump (top arrangement) and the cycle with an electrically driven pump (bottom arrangement)

high pressure vapour to the appropriate side of the pistons in the pumps.

The shaft in the control mechanism has two distinct positions, one at either end. At these positions, the shaft opens and closes the respective channels for forward and return movement of the pump pistons. A link arm (Figure 2) that is rigidly fixed to the pump shaft, but spring mounted to the control shaft, provides motion to the control mechanism. This enables the control shaft to 'jump' to the distinct positions while the pump pistons undergo reciprocating motion. The control shaft is held in either of these positions by a spring plunger, for the duration of the stroke of the pump.

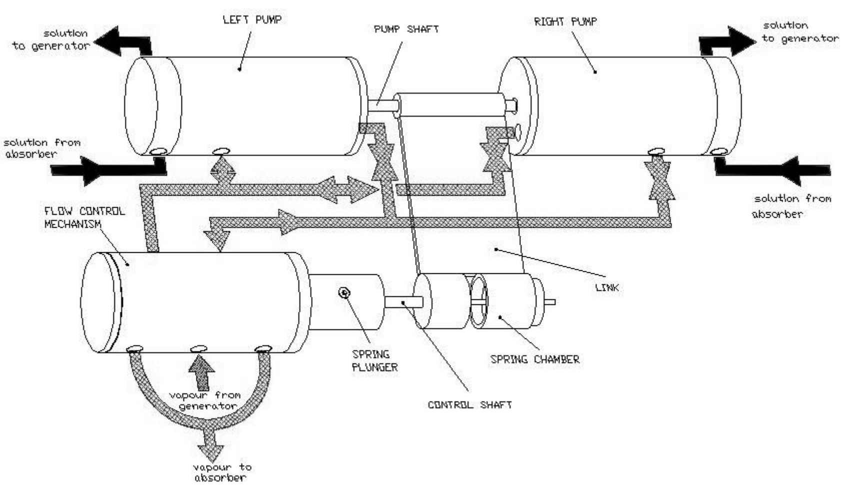

Figure 2: Assembly view of the vapour pump

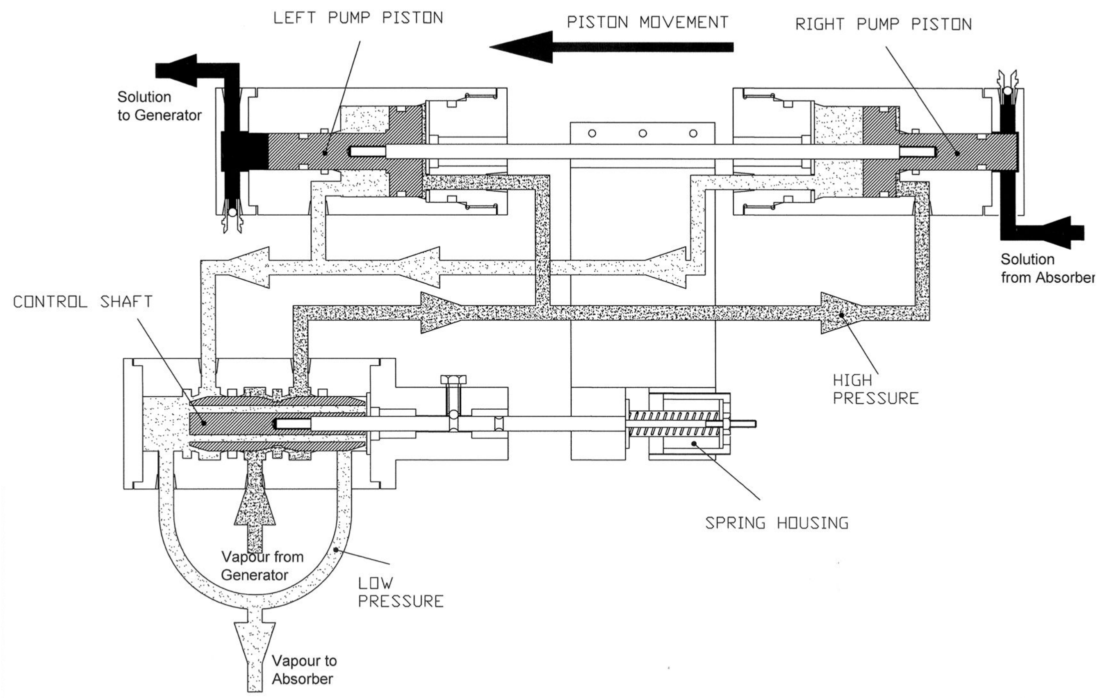

Figure 3a: Principle of operation (delivery left pump, suction right pump)

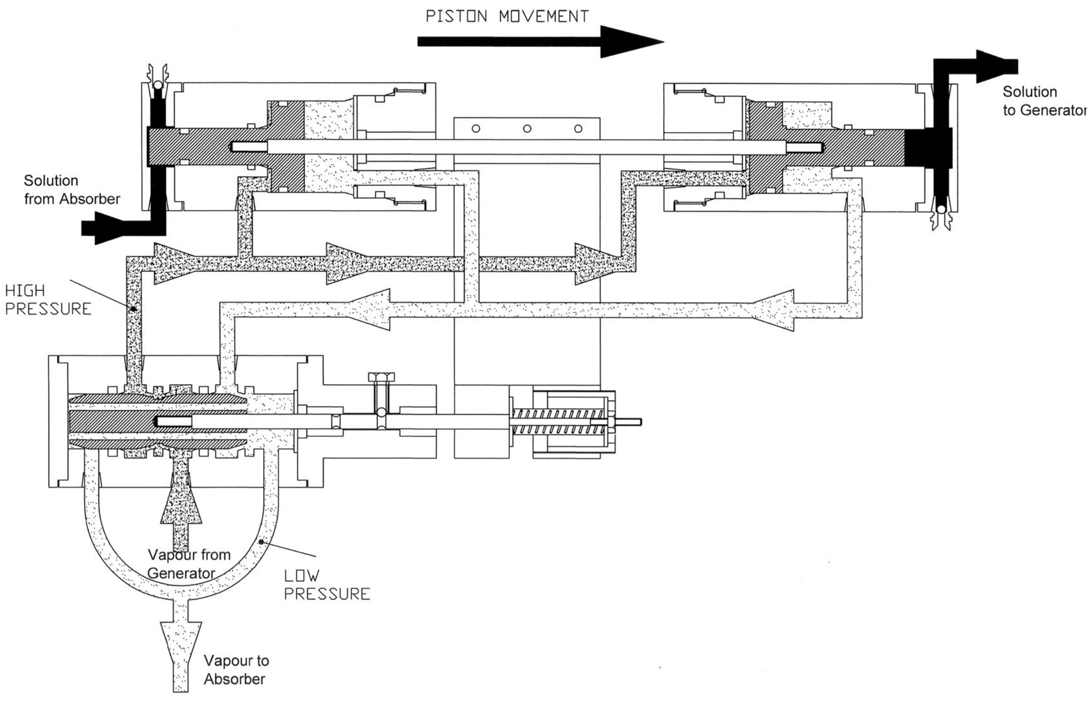

Figure 3b: Principle of operation (delivery right pump, suction left pump)

With the control shaft initially held, say in the 'right' position (Figure 3a), the high pressure vapour is channelled to push the pistons to the left. While the pistons move continuously to the left, the control shaft remains in position until the force, exerted by the increasingly compressed spring in the link, overcomes the force of the spring plunger. Then the control shaft 'jumps' to the 'left' position, switching over the high and low pressure ports, thus forcing the return movement of the pump pistons (Figure 3b).

Theoretical analysis

A mass balance around the generator/distiller of Figure 4 yields:

$$m_{16} + m_{11} + m_8 = m_{12} + m_{10} \qquad (1)$$

From Figure 1, the following mass and concentration equations (2) are derived:

$$m_{10} = m_7$$

$$m_{16} - m_{12} = m_1$$

$$m_{17} = m_{16}$$

$$m_{14} = m_{19} = m_8$$

$$x_{14} = x_9 = x_8 = x_{we}$$

$$x_{10} = x_7 = x_{st}$$

$$y_{16} = y_{11} \qquad (2)$$

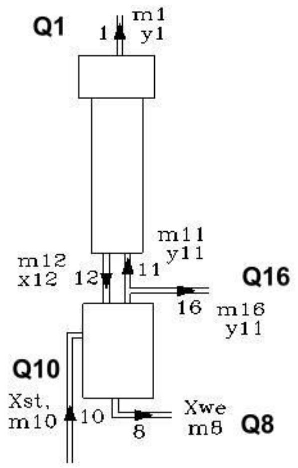

Figure 4: Generator/distillation column, during the vapour pump mode

Thus:

$$m_{16} + m_1 + m_8 = m_7 \qquad (3)$$

$$m_{16}y_{16} + m_{11}y_{11} + m_8x_8 = m_{12}x_{12} + m_{10}x_{10}$$

$$m_{11}y_{11} - m_{12}x_{12} = m_1y_1$$

Combining the above equations:

$$m_{16}y_{11} + m_1y_1 + m_8x_{we} = m_7x_{st}$$

Substituting for m_7 from equ. (3) and solving for m_8:

$$m_8 = \frac{m_1(y_1 - x_{st}) + m_{16}(y_{11} - x_{st})}{(x_{st} - x_{we})} \qquad (4)$$

A mass balance around the absorber of Figure 5 yields:

$$m_{17} + m_{14} + m_6 = m_7 \qquad (5)$$

using equations (2):

$$m_7 - m_{16} - m_1 = m_8 \qquad (6)$$

also

$$m_{17}y_{17} + m_{14}x_{14} + m_6y_6 = m_7x_7$$

$$m_{16}y_{11} + m_8x_{we} + m_1y_1 = m_7x_{st}$$

Substituting for m_8 from equation (6) and solving for m_7:

$$m_7 = \frac{m_1(y_1 - x_{we}) + m_{16}(y_{11} - x_{we})}{(x_{st} - x_{we})} \qquad (7)$$

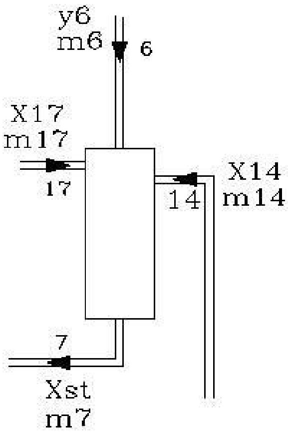

Figure 5: Absorber during the vapour pump arrangement

Equations (4) and (7) give the mass rates of the solutions in a system operating with a vapour pump.

For the system operating with an electrically-driven pump, the mass rates of the weak solution, m_8 and of the strong solution, m_7 are given respectively by Vicatos (1995):

$$m_7 = \frac{m_1(y_1 - x_{we})}{(x_{st} - x_{we})} \quad \text{and} \quad \frac{m_1(y_1 - x_{st})}{(x_{st} - x_{we})} \qquad (8)$$

Comparing equations (4) and (7) to equations (8), it shows that the circulation rates of both the weak and the strong solutions must increase in order to accommodate the surplus vapour from the generator to the absorber (stream 16). The additional amount of heat carried, Q_{16}, (supplied by the generator) is rejected in the absorber. It is therefore expected that the COP of the cycle will drop.

The COP is given by:

$$COP = \frac{R.E.}{Q_{gen}} = \frac{R.E.}{Q_1 + Q_8 + Q_{16} - Q_{10}}$$

(vapour-driven pump mode) (9)

$$COP = \frac{R.E.}{Q_1 + Q_8 - Q_{10}}$$

(electrically-driven pump mode)

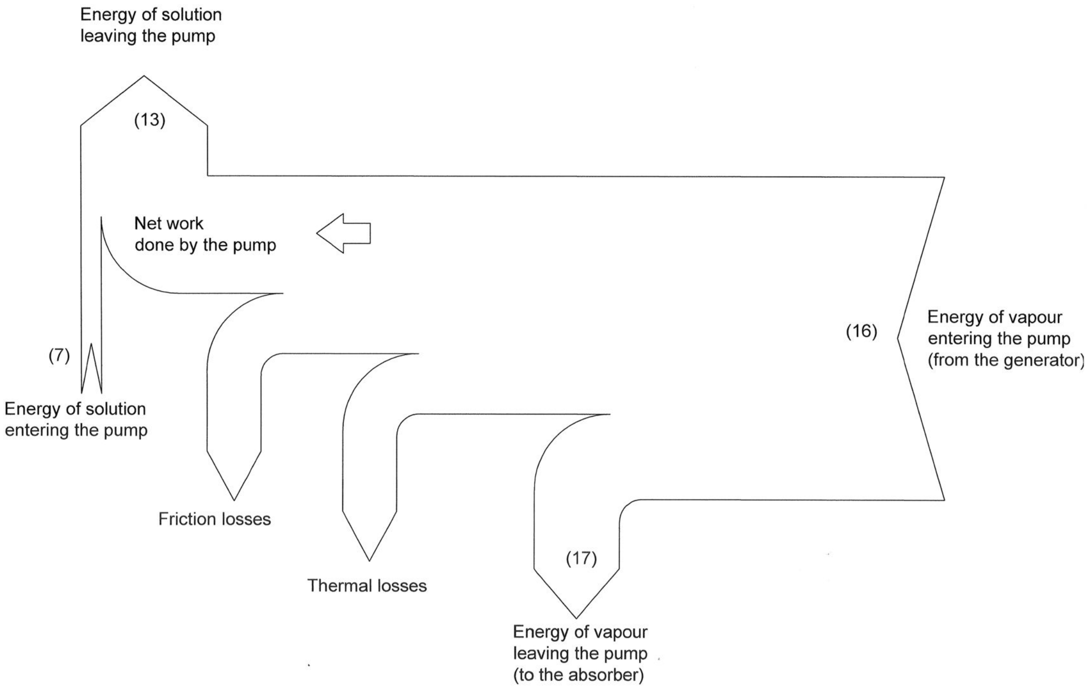

Figure 6: The energy balance around the vapour pump

Equations (4), (7) and (8) differ by the term in the numerator containing the mass rate m_{16}. This is calculated considering the energy balance around the vapour pump depicted in Figure 6.

By conservation of energy (Figure 6):

$$m_{16}h_{16} + m_7h_7 = m_{17}h_{17} + m_{13}h_{13} + Q_{friction} + Q_{thermal}$$

where

$$m_{16} = m_{17} \quad \text{and} \quad m_{13} = m_7$$

$$m_{16}(h_{16} - h_{17}) = m_7(h_{13} - h_7) + Q_{friction} + Q_{thermal}$$

which yields to:

$$m_{16}(h_{16} - h_{17}) = W_{pump} + Q_{friction} + Q_{thermal} \tag{10}$$

The specific enthalpies h_7 and h_{16} of the saturated mixtures can be evaluated, since at these states, two fluid properties are known. However, this is not the case with the vapour of stream 17 where the vapour may not be saturated at the absorber's pressure. The thermal and frictional losses are also difficult to quantify theoretically. Therefore, a percentage of the energy input to the pump can be assigned to the mechanical and thermal losses. This percentage can be translated into the vapour driving force exceeding the solution resisting force by a factor S_f.

$$F_{Vapour} = S_f.F_{Solution}$$

In this particular design, the rod and solution side piston diameters are 8 mm and 20 mm respectively. With a vapour side piston diameter of 50mm, the factor S_f is approximately 12. The mass rate of the vapour m_{16} is calculated considering the design of the pump.

From Figures 3a and 3b, it can be deduced that:

$$\frac{\pi}{4}\left(2D_V^2 - D_S^2 - D_R^2\right).\Delta P = S_f.\frac{\pi}{4}D_S^2.\Delta P$$

$$D_V = \sqrt{\frac{\left(S_f + 1\right)D_S^2 + D_R^2}{2}}$$

where D_V, D_S and D_R are the vapour, solution and rod diameters respectively.

The diameter of the solution-side piston is related to the volume of the solution being processed and the pumping frequency.

The volume of solution displaced per stroke of length L is:

$$V = \frac{1}{4}\pi.D_S^2.L$$

For the return stroke, the volumetric flow rate of the strong solution is:

$$V_{sol} = 2.\left(\frac{1}{4}\pi.D_S^2.L\right)$$

and the mass rate of the strong solution is:

$$m_7 = m_{ST} = \rho_{sol} V_{sol}$$

Similarly the volume rate of the high pressure vapour passing through the pump:

$$V_{Vap} = \frac{1}{2}\left[\left(D_V^2 - D_R^2\right) + \left(D_V^2 - D_S^2\right)\right] L$$

$$m_{16} = m_{vap} = \rho_{vap} V_{vap} \qquad (11)$$

Substituting this value in equations (4) and (7), the mass rates of the weak and strong solutions respectively can be evaluated. Subsequently the COP of the unit can be evaluated by equations 9 and by the method described in Vicatos (1995).

Results

The performance of a laboratory size 1 kW absorption refrigeration unit was tested using the vapour pump (Krafft, 1995). It was subsequently compared to the performance of the unit when operated with an electrically driven pump. The operating temperatures of the evaporator, condenser and generator during both pumping modes were kept almost constant. However, it was more difficult to match the evaporator temperatures between the two pumping modes than the generator and the condenser temperatures. The experimental results in Figure 7 compare the performance of the plant when operated in two different methods, and show a dramatic drop in performance when the vapour-driven pump was used. Due to the mismatch of temperatures, mainly the evaporator's, the difference between the performances of the two pumping modes show an unpredictable pattern.

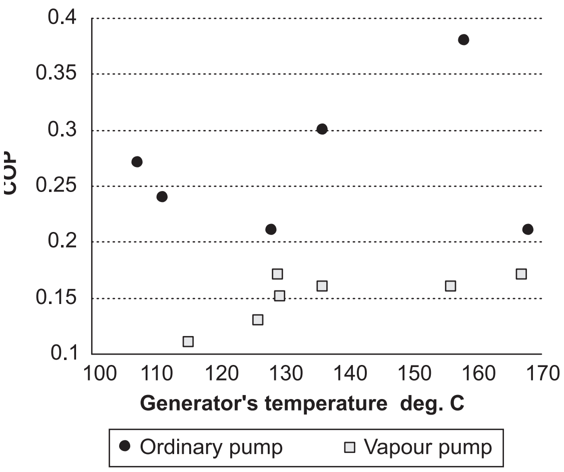

Figure 7: Experimental COP values versus generator temperatures obtained during the two methods of solution pumping

A computer simulated comparison between the performances for various generator temperatures during the two pumping modes is shown in Figure 8. These curves are produced from data predicted by computer programs developed by the authors that simulate the cycle in both pumping modes (Vicatos, 1995).

For a given set of evaporator (Te) and condenser and absorber (Ts) temperatures, the program evaluates the thermodynamic properties of the working fluids at each stage of the cycle and predicts the COP as a function of generator, evaporator and sink temperatures.

Both curves start from the same minimum generator temperature, indicating that the cycle is independent of the pumping mode but dependent on the saturation temperature of the strong solution entering the generator (Vicatos and Gryzagoridis, 1994). Thereafter both curves increase, but show two distinct differences:

- The coefficient of performance is less during the vapour pump mode for all generator temperatures.
- The maximum COP during the vapour pump mode occurs at a higher generator temperature.

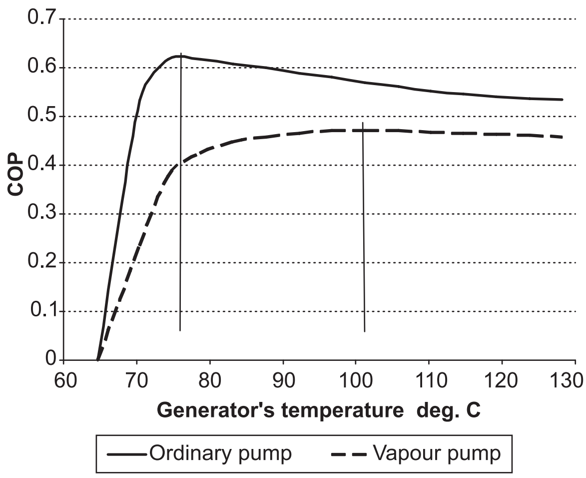

Figure 8: Simulated COP values versus Generator temperatures obtained during the two methods of solution pumping (simulated performance at Te=-10C and Ts=24C)

According to Vicatos and Gryzagoridis (1994), maximum COP is obtained at an optimum generator temperature that solely depends on an evaporator and sink temperature-pair. During the vapour pump operation, the COP-peak occurs at a higher generator temperature.

As it has been shown in Vicatos and Gryzagoridis (1994), the COP is linked to the heat carried by the strong and weak solutions, from the absorber to the generator and from the generator to the absorber respectively.

Assuming the same generator temperature during both pump modes operations, the additional vapour flow, $m_{16,}$ needed to drive the vapour-pump, will increase both the mass rates of the weak and strong solutions as indicated by equations (4) and (7) respectively. This increase in mass rates will carry an amount of heat associated with the generator's temperature, which culminates in a decrease of the COP, equations (9) . However, the total heat of the mixture is not linearly related to the temperature. As the generator temperature increases, the mass rates decrease (Vicatos, 1995). There will be a particular generator temperature at which the decrease in the heat content due to the decrease in mass flow rates will be balanced by the irreversible effects of the solution formation (Vicatos, 1995) due to the elevated temperatures. This particular generator temperature will mark the optimum generator temperature and COP-peak for the cycle, which during the vapour pump mode does not coincide with the optimum generator temperature during the electrically-driven pump mode.

Conclusions

These simulated results confirm the experimental ones; that the drop in COP is due to the generator's increased heat input, in order to supply the extra vapour.

In a 2-fluid absorption system the energy due to the pump is only a small contributor to the total energy balance. Therefore, the pump's efficiency is not of critical importance to the performance of the unit. What is important, however, is whether the pump is powered by a high energy source, or is using the potential energy of the system itself.

In the proposed pumping method, the performance loss of the cycle can be compensated by rendering the unit completely independent of any external source other than the one powering the generator. It follows that the absorption unit can be powered entirely with low grade energy. This has the obvious advantage that these units can be employed in locations where electricity is not available, and can be of sizes larger to the domestic-size units.

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