# A LiBr-H$_2$O Absorption Refrigerator Incorporating a Thermally Activated Solution Pumping Mechanism

**Ian W. Eames**

Department of Mechanical Materials and Manufacturing Engineering, University of Nottingham, Nottingham NG7 2RD, UK; ianweames@aol.com; Tel.: +44-115-951-5151

**Abstract:** This paper provides an illustrated description of a proposed LiBr-H$_2$O vapour absorption refrigerator which uses a thermally activated solution pumping mechanism that combines controlled variations in generator vapour pressure with changes it produces in static-head pressure difference to circulate the absorbent solution between the generator and absorber vessels. The proposed system is different and potentially more efficient than a bubble pump system previously proposed and avoids the need for an electrically powered circulation pump found in most conventional LiBr absorption refrigerators. The paper goes on to provide a sample set of calculations that show that the coefficient of performance values of the proposed cycle are similar to those found for conventional cycles. The theoretical results compare favourably with some preliminary experimental results, which are also presented for the first time in this paper. The paper ends by proposing an outline design for an innovative steam valve, which is a key component needed to control the solution pumping mechanism.

**Keywords:** thermally activated pump; absorption refrigeration cycle; cooling; refrigerator; thermodynamics

## 1. Introduction

The use of thermally activated refrigerators has the potential to reduce CO$_2$ and other damaging emissions. Because of their relatively high coefficients of performance, LiBr-H$_2$O vapour absorption cycle refrigerators are seen as an important technology in this area. Furthermore, since the refrigerant is water, there is no danger that fugitive emissions can damage the ozone layer or contribute to global warming. However, a drawback to the wider use of LiBr-H$_2$O absorption refrigerators has been their relatively high capital cost compared to electric vapour compression machines with similar cooling capacities. A significant contribution to the capital cost of small-scale, sub-kilowatt, absorption refrigerators is thought to be that of the solution recirculation pump. The present paper describes a novel absorption refrigeration cycle which avoids the use of a solution pump in the normal sense and which it is thought will suit the design of small capacity refrigerators for applications such as small capacity chilled drink cabinets and vaccine coolers.

Because of their low energy efficiency, the electrical power consumption of small-scale absorption refrigerator circulation pumps might in some cases be better used to drive a conventional electrically powered vapour compression refrigerator. Xu and Wang [1] provide a very useful general discussion on various types of absorption cycle including combined cycles and a recent research report by Asdrubali et al. [2] summarized the thermal and electrical performance of small scale LiBr-H$_2$O absorption cycle machines. By taking into account both heat and electrical input, overall Coefficient of Performance (COP) values were shown to be in the range 0.53–0.64 for machines with cooling capacities in the range 5.2 to 11.4 kW [2]. This author believes that the problem of electrical inefficiency may be exacerbated for similar machines with cooling capacities of less than one kilowatt.

The idea of using thermal energy alone to circulate the working fluids through absorption cycle refrigerators is not new. The diffusion absorption refrigeration (DAR) cycle uses a bubble pump to circulate the solution, as described in the ASHRAE Handbook [3]. A key to high efficiency DAR cycles lies in the design of the bubble pump. These have been the subject of investigation for many years. Recently, Gao at al. [4] theoretically investigated the motion of bubbles within a vertical tube for which the vapour–liquid density ratio was over 270. They studied single and twin bubble formations along with coalescence tims of the latter in order to understand their influence of liquid entrainment; however, bubble formation (nucleation) was not considered in that study. Benhmidene et al. [5] undertook an experimental investigation of bubble pump operation in a DAR system, finding that submersion ratio and generator heat input played important roles in the performance of the pump. Zohar and Jelinek et al. [6] investigated the influence of the generator and bubble pump design on the performance of DAR systems. They showed that bubble pump performance, and therefore COP, can be improved by changing the construction of the bubble pump. Their results showed that an 86% increase in COP was possible by altering the design of the bubble pump lift tub, however, but the absolute COP value were never better than 0.34, which is low compared with values expected of typical LiBr-$H_2O$ single-effect vapour absorption refrigerator (VAR) systems. An et al. [7] described an interesting theoretical and experimental study on the lifting performance of bubble pump with a lift-tube of varying diameter. The rig design was described and experimental results were given. The efficiency results were a little confusing but otherwise the paper was most interesting. Various combinations of refrigerant, absorbent and inert gas have been tested for DAR systems. The normal working vapour pressures of the refrigerants tested so far in DAR systems have been in excess of atmospheric pressure, which aids the nucleate boiling process used by bubble pumps. However, COP values have tended to be significantly lower than that of a conventional single-effect LiBr-$H_2O$ absorption cycle refrigerator.

The bubble pump has been suggested for use in LiBr-$H_2O$ VAR systems. Saravanan and Maiya [8] investigated the influence of thermodynamic and thermophysical properties on water-based working fluids for bubble pumps for VAR systems. This paper discusses the benefits of various aqueous–salt solutions (binary, tertiary and quaternary) with regard to variations in density and vapour pressure and their effect on bubble pump operation. The results for some tertiary and quaternary solutions were particularly encouraging. Chan and McCulloch [9] provided a useful theoretical study of bubble pump performance validated with experimental evidence for water based systems. Pfaff et al. [10] described an experimental stud, which included a detailed description of the construction, operation and theory underlying typical bubble pumps. The effect of lift tube diameter and lift-height were provided. System performance was given in terms of pumping ratio—the pumping ratio being between the volumetric flows of the strong solution and refrigerant (water) vapour generated in the lifting process. The results suggested that low values solution concentration were needed to enable the pump to work. COP data for the experimental refrigerator were not provided. Gu et al. [11] described an experimental study on a new solar pump-free LiBr-$H_2O$ VAR system. The system they describe could be operated in both single and double-effect modes. The system used a lunate thermo-syphon elevation tube, described in the paper, which effectively created a bubble pump effect to circulate the weak solution to the first stage generator. One set of tabulated results for the single-effect operation gave a 2% change in concentration at the absorber with a concentration of 50% at entry to the thermo-syphon (bubble pump). This is particularly low and was possibly needed in order to encourage bubble formation. COP values reported were shown to increase when the system was operated in double-effect mode with the value rising from 0.53 to 0.78, which is encouraging. The use of a bubble pump to circulate the solution in a LiBr-$H_2O$ VAR system has also been investigated by Saravanan and Maiya [12,13], who investigated the design of bubble pumps and the effects of pressure losses on system performance. Asdrubali and Grignaffi [14] carried out experimental testing of a single-stage LiBr-$H_2O$ absorption refrigerator manufactured by the Yazaki. They reported that the COP of the machine they tested ranged between 0.2 and 0.43 with a recirculation factor of not less than 9. It is interesting to note that the highest COP

was measured at the lowest generator temperature (68 °C). At this generator temperature, the cooling duty also had the lowest value. The variation in solution concentration between the absorber and generator was found to be at best 50.3% to 56.3%. This variation in concentration is small in comparison with conventional machines and may be probably responsible for the high recycle factor (f) and the high recirculation rate required by the bubble pump. A possible reason for needing to keep the water content of the solution high may have been to assist bubble formation in the pump. A problem with nucleate boiling at low absolute pressures is well known as reported by Eames, Marr and Sabir [15]. Steady nucleate boiling, of the type needed to operate a bubble pump, is difficult if not impossible to achieve with water at very low saturation pressures. This is because, as bubbles form, they tend to be very large and grow rapidly, due to their low density at low pressures. Their formation is sometimes described as explosive. In addition, the number of potential nucleation sites tends to decrease at lower saturation pressures and therefore opportunities for small stable bubbles to form are reduced.

The idea of using a thermo-syphon effect instead of a bubble pump to circulate the solution between the generator and absorber of an absorption cycle heat transformer system was investigated by Abrahamsson, Gidner and Jernqvist [16]. They described an experimental study of a 10 kW $NH_3$-$H_2O$ absorption heat transformer incorporating a self-circulation mechanism based on the thermo-syphon principle. COP values were not provided because of what the authors felt was difficulty in calculating values due to external heat losses. The authors thought these COP values might be as much as 50% of the heat input due to poor insulation. However, the authors also suggested that COP values would be "very low".

The thermo-gravity cycle described in this article overcomes the drawbacks of a bubble pump operating at low saturation pressures that lead to high solution circulation rates and high recycle factors. In doing so, the novel cycle is able to operate with COP values much closer to those of a conventional machine. Stephens and Eames [17] experimentally tested a valve operated VAR cycle described later in this paper. The theoretical results given in Section 4 are based on Stephens and Eames' experimental data described in Section 5. Eames and Wu [18] theoretically analysed the solution circulation process by considering a heat engine. Paurine et al. [19] manufactured and tested an almost identical cycle to that described in this paper. However, for reasons which were not clear, the action of the automatic steam valve, described and discussed in Section 6, was reversed. This resulted in incurring significant sensible heating losses as the solution in the generator needed to cool to ambient conditions between each pumping cycle. A diagram of Paurine's valve is included in Section 6 of this paper.

The paper begins by describing the operation of a conventional LiBr-$H_2O$ absorption cycle refrigerator before going on to describe the innovative cycle. A set of sample calculations are then presented and the architecture and operation of a novel flat operated valve, a key component of the system, is described. The results of the research are discussed and the design of an automatic steam valve, a key component, is outlined.

## 2. The Conventional Single-Effect Vapour Absorption Refrigerator Cycle

A schematic diagram of a conventional single-effect vapour absorption cycle refrigerator is shown in Figure 1. A complete description of the cycle and its analysis is provided in the ASHRAE Handbook [3]. However, in order to assist later discussion, a short description of the conventional LiBr-$H_2O$ absorption cycle refrigerator is provided here.

Referring to Figure 1, in the evaporator, water vapour is generated absorbing latent heat and this provides the necessary cooling or refrigeration effect. The vapour flows to the absorber vessel where it is absorbed by the hygroscopic lithium-bromide-water solution. Because the absorption process is exothermic, the solution must be continuously cooled. In addition, to ensure that the solution concentration is maintained, a proportion is continuously pumped to the generator section where it is heated in order to drive out some of the water. The steam created by this process is fed to a condenser where it is liquefied and returned to the evaporator for reuse. Meanwhile, the hot concentrated solution returns under pressure to the absorber via a solution heat exchanger and expansion valve. The solution

heat exchanger transfers heat between the hot solution returning to the absorber and the cold solution. This heat transfer process improves a cycle's coefficient of performance (COP) by reducing the amount of sensible heating and cooling otherwise required at the generator and absorber, respectively.
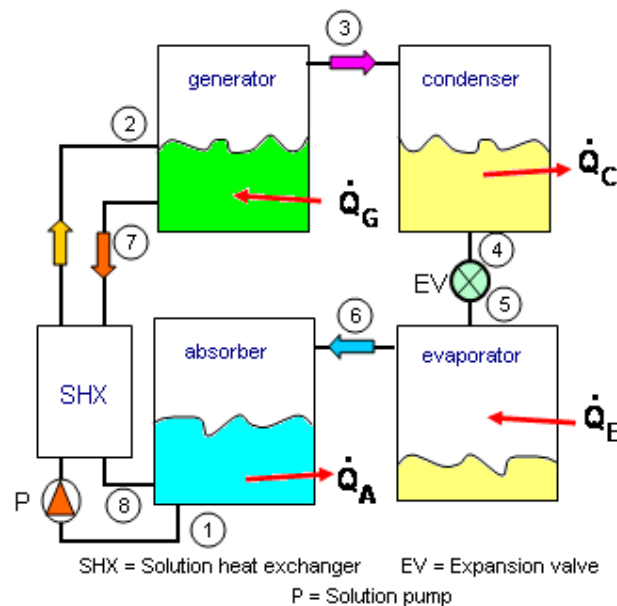


**Figure 1.** A conventional single-effect absorption cycle.

Because the refrigerant (water) freezes at 0 °C, the absolute vapour pressure within the evaporator and absorber is never likely to be less than 611 Pa. The pressure in the generator is determined by the saturation condition in the condenser. If the maximum condenser temperature was 55 °C, then the maximum generator pressure would not exceed 15.74 kPa. This relatively small absolute pressure difference raised the possibility of using the static head pressure difference alone to return the dilute solution from the absorber vessel to the generator. This was the basis of the idea described in the following section.

## 3. A Single-Effect Absorption Cycle with a Thermo-Gravity System

Figure 2 shows a schematic layout of the novel absorption refrigerator proposed here. This machine uses the same process vessels as the conventional absorption refrigerator shown in Figure 1—namely, absorber, evaporator, condenser and generator, but without the need for an electrically powered solution circulation pump (P), in Figure 1, or an expansion device or valve.

In the novel cycle, shown in Figure 2, the evaporator operates in the same way as the conventional cycle shown in Figure 1; that is, it continuously absorbs heat to create the necessary cooling effect by evaporating water at low pressure. This low pressure steam flows to the absorber vessel where it is continuously absorbed by the LiBr-$H_2O$ solution. In the absorber, a quantity of concentrated solution is held in a reservoir at the top of the vessel. The solution flows slowly downwards over cooling coils, becoming increasingly more dilute as it absorbs water vapour in order to maintain the cooling effect at the evaporator. It is pointed out that the evaporator and absorber could be contained in the same low-pressure vessel, making its manufacture less complicated. The difference between the new cycle described in this paper and the conventional cycle lies in the mechanism used for circulating the solution between the absorber vessel and generator vessel.

The pumping process occurs in two stages. In the first stage, the solution collects at the bottom of the absorber vessel from where it flows downwards via a non-return valve (NRV) and a regenerative heat exchanger (R) into the vapour generator, where the solution is concentrated by distillation in the normal way. With no solution being pumped back to the absorber, the solution level in the

generator rises until it reaches the upper (maximum) of two predetermined levels, as shown in Figure 2. Meanwhile, steam raised in the generator during this first stage of the pumping process flows via an open valve (V) to the condenser vessel, where it is liquefied before returning to the evaporator by both gravity and excess of vapour pressure at the condenser.
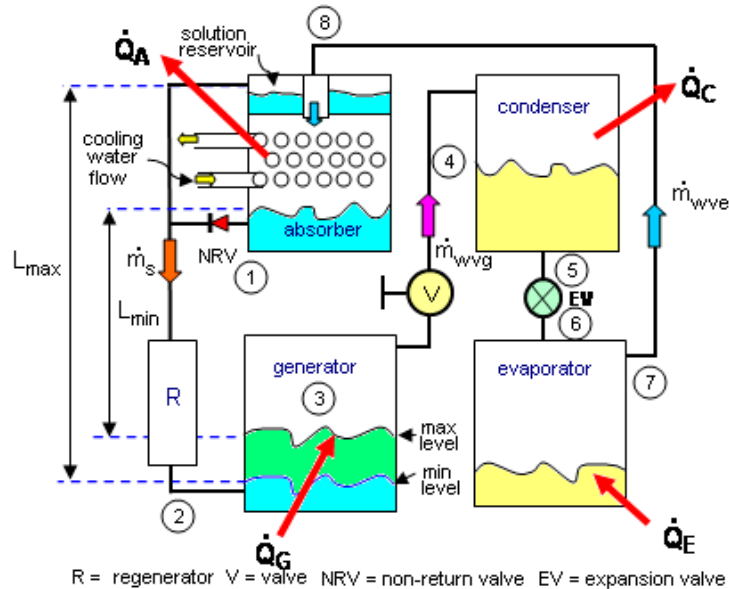


**Figure 2.** Schematic view of the novel cycle using a valve (V) to control the flow of solution.

The solution is able to flow from the absorber to the generator, which is at a higher pressure, because of the static head pressure created by the height of the liquid column between the surfaces in the absorber and generator. The system should be designed so that the flow will be maintained at the lowest level difference, $L_{min}$, shown in Figure 2. In the following sample calculation, the height difference needed for all practical purposes is shown to be less than one metre.

Stage two of the pumping cycle begins once the solution level in the generator has reached its maximum mark. At this point, the steam valve (V) is closed, trapping steam inside the generator. With the steam-valve (V) closed, the pressure in the generator will rise. This pressure is great enough to overcome the static-head difference between the absorber and generator, causing the heated and concentrated solution to flow back towards the absorber vessel. The hot solution first passes through the regenerative heat exchanger, where it would be partially cooled, and then into the reservoir located at the top of the absorber vessel. The positioning of the non-return valve in the pipeline leading to the bottom of the absorber vessel prevents flow from entering the vessel at the bottom.

When the solution level in the generator has fallen to a predetermined minimum-level, shown in Figure 2, the steam valve (V) would be re-opened and the cycle would again enter the first stage as previously described.

Throughout the second stage of the cycle, it is assumed that there is sufficient concentrated solution held in the reservoir to maintain the absorption process there.

The regenerative heat exchanger (R) in the solution pipeline between the generator and absorber is intended to reduce sensible heating and cooling effects at the generator and absorber, respectively. Such sensible heat transfer effects, if not reduced to the minimum possible within the limits of the second law of thermodynamics, can significantly lower the COP value of an absorption cycle refrigerator. Furthermore, because the generator temperature needs to be raised along with the solution temperature, when transferring liquid back to the absorber, it is particularly important that the generator is well insulated.

Although the solution flow between the absorber and generator is intermittent, the refrigeration effect will be constant. If the heat input at the generator is maintained at a constant value, then the coefficient of performance should not be too different to that of a conventional cycle machine. The following sample calculations help clarify this difference.

## 4. Simplified Theoretical Analyses

In the sample calculations below, the following general assumptions were made:

1. The heat rate to both the evaporator and generator are assumed to be constant throughout. In addition, the temperature of the absorber and condenser are assumed constant, although in practice, as shown by the experimental results, this is not the case. However, as shown by comparing the theoretical result with the experimental data, this assumption appears to provide adequate accuracy.
2. Refrigerant vapour leaving the evaporator is saturated.
3. The refrigerant flow from evaporator equals $5.5279 \times 10^{-4}$ kg/s (value measure by Stephens and Eames [17]).
4. The flow through heat exchangers, steam valve and all connecting pipelines is frictionless.
5. The regenerator has an effectiveness of 50%.
6. 1 L of solution is circulated between the generator and absorber during each pumping cycle, which is equal to 1.567 kg, as per Stephens and Eames data [17].
7. The generator contains 4.7 kg (3 L) of solution when the steam valve just is closed, as per Stephens and Eames' data [17].
8. The surface level of the solution in the generator changes by 100 mm between maximum and minimum levels during the pumping cycle.

The above data match those used by Stephens and Eames [17] in his experiments.
Input data: $T_G$ = 70 °C, $T_E$ = 10 °C, $T_A$ = 25 °C, $T_C$ = 40 °C.
From data tables [3], the relevant properties of LiBr-$H_2O$ at $T_C$ = 40 °C and $T_G$ = 100 °C are:

$$h_3 = h_{SG} = 153.4 \text{ kJ/kg}, X_G = 0.539,$$
$$h_1 = h_{SA} = 44.8 \text{ kJ/kg}, X_A = 0.453.$$

From data tables [20] the relevant properties of steam are:

$$P_G = P_C = \text{Psat @ 40 °C (sat)} = 7375 \text{ Pa},$$
$$P_A = P_E = \text{Psat @ 10 °C (sat)} = 1227 \text{ Pa},$$
$$h_7 = h\text{g @ 10 °C (sat)} = 2519.2 \text{ kJ/kg},$$
$$h_6 = h_5 = h_f \text{ @ 40 °C} = 167.5 \text{ kJ/kg},$$
$$h_4 = h\text{g} + \Delta h_{SH} \text{ @ 40 °C (sat), 60 degrees superheat}$$
$$= h\text{g} + C_p \Delta T = 2573.7 + 1.89 \times 60 = 2687.1 \text{ kJ/kg}.$$

It should be noted that the solution pressure and temperatures given above as input data apply to the part of the cycle when the generator is filling with solution as described in Section 4.1. The calculations in Section 4.2 show that the generator temperature and pressure increase during the second-part of the cycle.

### 4.1. The First Part of the Cycle—Charging the Generator

During this part of the cycle steam valve "V" in Figure 2 is open to allow steam produced at the generator to flow to the cooled condenser. During this half of the cycle, the rate of steam flow from evaporator (assuming a refrigeration effect of 1.33 kW) to absorber is given by

$$\dot{Q}_E = \dot{m}_7/(h_7 - h_6)$$
$$= 5.5279 \times 10^{-4}/(2519.2 - 167.5) = 1.3 \text{ kW}$$  \qquad (1)

For steady state operation, the steam flow from the generator to the condenser will equal that from the evaporator to the absorber. In addition, note that, $\dot{m}_4 = \dot{m}_7 = 5.5279 \times 10^{-4}$ kg/s. The flow of solution between absorber and generator is given by,

$$\dot{m}_2 = \frac{\dot{m}_4}{(XG - X_A)} = \frac{5.5279 \times 10^{-4}}{(0.539 - 0.453)} = 6.428 \times 10^{-3} \text{ kg/s.} \qquad (2)$$

Recycle factor (*f*) is an important parameter in determining the efficiency of an absorption cycle refrigerator. It measures the mass of solution that needs to be processed at the generator to produce given mass of steam at the evaporator and, by this, the refrigeration effect. In the present case

$$f = \dot{m}_2/\dot{m}_7 = 6.428 \times 10^{-3}/ 5.5279 \times 10^{-4} = 11.63 .$$

This result can be used to compare the performance of the present cycle with other absorption cycles.

Taking into account the heat-exchange effectiveness of the regenerator, the temperature of the solution entering the generator is given by

$$T_2 = T_1 + \varepsilon_R(T_3 - T_1) = 25 + 0.5(70 - 25) = 47.5 \,^\circ\text{C}. \qquad (3)$$

Noting that the solution concentration $X_2 = X_A$, then, from property data, the enthalpy of the solution entering the generator is given by

$$h_2 = 97.1 \text{ kJ/kg} = h_S \text{ @ } T_S = 47.5 \,^\circ\text{C, } X = 0.453 .$$

Applying the non-steady energy flow equation to the generator vessel gives

$$\dot{Q}_G = \dot{m}_4 h_4 - \dot{m}_2 h_2 + \frac{dU_G}{d\theta}. \qquad (4)$$

If the internal energy of the steam within the generator vessel is assumed negligible, and the specific internal energy of the solution within the vessel ($u_{S3}$) is constant, because $T_G$ and $X_G$ are both constant and because the flow-work (Pv) of the solution is small in comparison to its enthalpy, then $u_{S3} \approx h_{S3}$, and therefore

$$\frac{dU_G}{d\theta} = \frac{dM_{SG}}{d\theta} u_3 = \dot{m}_2 h_3$$

Substituting this result into Equation (4) gives

$$\dot{Q}_G = \dot{m}_4 h_4 - \dot{m}_2(h_2 - h_3)$$
$$= 5.5279 \times 10^{-4} \times 2687.1 - 6.428 \times 10^{-3}(97.1 - 153.4)$$
$$= 1.78 \text{ kW}$$

During this part of the cycle, the quantity of water needed to boil from the 1.567 kg solution to change its concentration from 0.453 to 0.539 is given approximately by

$$M_W = M_S[X_G - X_A]$$
$$= 1.567 \times (0.539 - 0.453) .$$
$$= 0.135 \text{ kg}$$

The mass flow of steam from the generator is $\dot{m}_4 = 5.5279 \times 10^{-4}$ kg/s. Therefore, the time taken to generate 0.135 kg of steam and to complete the first-part of the cycle is given by

$$Time = \frac{M_W}{\dot{m}_4} = \frac{0.135}{5.5279 \times 10^{-4}} = 243 \text{ s.} \tag{5}$$

The minimum head difference between the surface levels of the solution in the generator and absorber is given by

$$L_{MIN} = \frac{(P_G - P_A)}{\rho_S g} = \frac{(7375 - 1227)}{1567 \times 9.81} = 0.4 \text{ m.} \tag{6}$$

### 4.2. Second-Part of the Cycle—When the Steam Valve Is Closed

In this part of the cycle, the steam valve is closed and with the rate of heat addition to the generator maintained at the same value as in the first part of the cycle, i.e., 1.78 kW, the pressure in the vessel would increase, causing the solution to be flow back to the absorber. On its return, the concentrated solution would cool as it passes through the regenerator due to the temperature gradient within it.

For the purpose of this sample calculation, it is assumed that the liquid level in the generator would fall by 100 mm whilst 1.567 kg (1 L) of solution is returned into the absorber vessel. Referring to Figure 2, the height through which the solution must be raised to reach the reservoir on top of the absorber increases from 600 mm at the start of this process, increasing to 825 mm when the solution has been transferred just when the steam valve is opened.

The pumping process occurs in two-stages: during the first stage, the temperature and pressure of the solution are raised at a constant concentration value of 0.539 from 70 °C and 7375 Pa, (the values at the end of the first part of the cycle described in Section 4.1) to 74.9 °C and 9223 Pa, which is the pressure needed to raise the solution to a height of 600 mm from the maximum level in the generator to the absorber reservoir inlet. This process requires the enthalpy to change from 153.4 kJ/kg to 163.7 kJ/kg—a difference of 10.3 kJ/kg. If it is assumed that the quantity of solution leaving the generator during this process is negligible then, by the First Law of thermodynamics for a closed system, assuming no work is done, the heat input required is given by

$$Q_G = \Delta U_G = M_{SG}\Delta u_3 = 4.7 \times 10.3 = 48.41 \text{ kJ.}$$

Note it is assumed here that $u = h$ for the reason given earlier.

If the rate of heat addition to the generator is held constant throughout at 1.78 kW then the time needed to complete the initial heating process is approximately 27.2 s (=48.4/1.78).

During the second-part of the pumping cycle, it was assumed that the generator continued to absorb heat at the same rate (1.78 kW), causing the solution pressure to increase further in order to raise the solution to the maximum height in the absorber from the minimum height in the generator, equal to 825 mm. If losses are neglected, the generator pressure required to return the solution to be absorber will increase from 9223 Pa to 12,638 Pa at constant concentration of 0.539. This process requires the solution temperature and enthalpy to, respectively, increase to 82 °C and 178.6 kJ/kg.

Applying the First Law of thermodynamics to the generator gives

$$Q_G + W_G = m_{out}h_{out} - m_{in}h_{in} + M_{G2}u_{G2} - M_{G1}u_{G1} \tag{7}$$

where in this case:

$M_{G1}$ = mass of solution in the generator at the start of the second-part of the pumping cycle;
$M_{G2}$ = mass of solution at the end of the pumping cycle;
$h_1$ = enthalpy of the solution in the generator at the start of the second-part of the pumping cycle;
$h_2$ = enthalpy of the solution in the generator at the end of the pumping cycle;
$m_{out}$ = mass of solution leaving the generator;
$m_{in}$ = mass of solution entering the generator;
$h_{in}$ = enthalpy of solution entering the generator during the pumping process;

$h_{out}$ = enthalpy of solution leaving the generator.

For this process $m_{in} = 0$, and, if it is assumed that $u_{G1} = h_{G1}$ and $u_{G2} = h_{G2}$, then

$$Q_G + W_G = \frac{1}{2}[M_{G1} + M_{G2}][h_{G2} - h_{G1}]. \tag{8}$$

From the given data, $M_{G2}$ = 3.1 kg, $M_{G1}$ = 4.7 kg. From property data tables,

$$h_{G2} = 178.6 \text{ kJ/kg}, h_{G1} = 163.7 \text{ kJ/kg}.$$

Neglecting pressure losses in the regenerator and then connecting the pipeline, the work done in returning the solution to the absorber is equal to the change in its potential energy:

$$W_G = -(M_{G1} - M_{G2})gL_{AV} = 1.567 \times 9.81 \times 0.7125 = 10.95 \text{ J},$$

where $L_{AV}$ = average height through which the solution is raised.

Substituting the above values into Equation (8) and solving for heat transfer gives $Q_G$ = 58.1 kJ, which is the heat required to complete the second part of the pumping process.

If the heat rate remains at 1.78 kW, the time taken to return 1.567 kg of solution to the absorber would be approximately 32.6 s. Adding this result to the 27.2 s required to raise sufficient pressure in the generator after the steam valve was closed (for the first-stage of the pumping process), the total time required for the steam valve to be closed would be approximately 59.8 s.

During the second part of the pumping cycle, the temperature of the solution returning to the absorber from the regenerator ($T_1$) is given by

$$T_1 = T_3 - \varepsilon_R(T_3 - T_1) = 82 + 0.5(82 - 25) = 53.5 \,^{\circ}\text{C}. \tag{9}$$

Because the evaporator cooling effect and generator heat input are assumed to be constant throughout, then the coefficient of performance for the theoretical cycle is given by

$$COP = \frac{\dot{Q}_E}{\dot{Q}_G} = \frac{1.3}{1.78} = 0.73. \tag{10}$$

### 4.3. The Efficiency of the Pumping Process

The amount of energy used to pump the solution from the absorber to the generator is of interest because the process is essentially that of a heat engine in which heated is added at the generator and, after raising the solution to the height of the absorber heat, is rejected before the cycle is completed when the solution flows back to the generator. The results of the sample calculations described in this section showed, for the case being considered, that the total amount of heat input during the pumping cycle was 25.2 kJ, whilst the ideal useful work done was 10.95 J. The overall thermal efficiency of the pumping process was therefore 0.043%. Although this is inefficient, the only heat input at the generator not used to desorb the solution is the 10.95 J used to create the work of pumping and (in practice) any sensible heat loss at the generator. In addition, as the heat rate to the generator is constant, and if sufficient solution is held in the absorber reservoir to permit its continued operation whilst the steam valve is closed, then, in practice, the COP of the cycle should not be too different from that of the conventional cycle. A similar conclusion was drawn from a theoretical investigation of the novel cycle by Eames and Wu [18].

## 5. Experimental Results

Stephens and Eames experimentally tested the absorption refrigerator shown schematically in Figure 2 [17]. The steam valve was hand operated, which allowed close control of the movement of the

solution between the generator and absorber and vice versa. Figure 3 was drawn using Stephens and Eames' recorded data. These show the variations in generator, absorber and condenser temperatures over a period of 16 minutes for two pumping cycles. In his original data, Stephens and Eames recorded only the times when the flow changed direction: he did not give the times when the steam valve was opened and closed. Therefore, from Figure 3, when the flow was reported to be "up", the steam valve must have been closed and when "down", it was reported that the valve must have been open, but times at which the valve was opened and closed was not given. For example, the results in Figure 3 showed that at the 3.5 min mark, the solution was observed to be flowing from the generator back to the absorber and, perhaps not unexpectedly, this was accompanied by a sudden rise in absorber temperature due to the influx of warm solution. However, the closing of the steam valve that produced this effect must have occurred before the solution began to flow back to the absorber for the reasons given in Section 4.2, in which the generator pressure was allowed to rise in order to overcome the static head difference between it and the absorber before any upward flow would be possible.
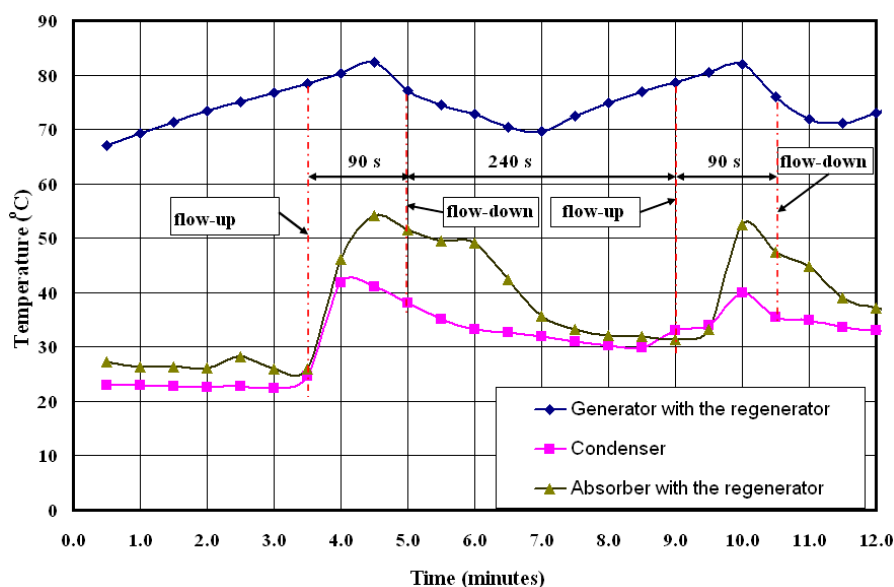


**Figure 3.** Some experimental results taken by Stephens and Eames [17].

At the 5 min mark in Figure 3, the solution was observed to be flowing from the absorber to the generator. For this to happen, the steam valve must have been open in order to allow the pressure in the generator to fall. It is likely that this occurred at the 4.5 min mark, when the generator and pressure were at their highest. Not unexpectedly, therefore, there appears to have been a transient delay of approximately 30 s between the opening of the steam valve to the response of the solution.

When the solution flows back to the absorber from the generator, i.e., when the flow is upwards in Figure 3, the condenser temperature is seen to increase (when the steam valve was closed) at the same time as the absorber temperature increased. This is due to the way in which the cooling water was piped to the condenser and absorber. During the experiment, as is usual practice, the cooling water was passed through the absorber and then the condenser—the two cooling coils being in series. Consequently, when the absorber was heated by the warm solution returning from the generator, some of this heat was transferred to the condenser, resulting in a rise in condensate temperature.

Referring to Figure 3, the overall time available for the solution to flow back to the generator was observed experimentally to be not greater than 240 s, whilst the theoretical result calculated in Section 4.1 for this part of the cycle was 243 s. This result is of interest. However, it is not clear from the experimental results at what time mark the downwards flow from the absorber to the generator actually ceased. Similarly, the experimental results show that the upward flow from the generator to

the absorber occurred in a time not greater than 90 s, which compares with a theoretical value of 60 s calculated in Section 4.2.

In the experiment, from which the data was used to produce Figure 3, Stephens and Eames stated that the electric heater used to heat the generator had a power of 2 kW and that the average measured refrigerant flow was 5.5279 kg/s—the value for refrigerant flow used in the theoretical calculations. Based on these results, the estimated coefficient of performance for the refrigerator under test was:

$$COP = \frac{\dot{Q}_E}{\dot{Q}_G} = \frac{1.3}{2.0} = 0.65. \tag{11}$$

This compares with the theoretical value of 0.73 given by Equation (9). This difference might be explained by heat loss at the generator.

## 6. An Automatic Steam Valve

In their paper, Paurine et al. [19] describe the construction and testing of a thermos-gravity VAR cycle and the operation of the automatic steam valve appears to be the reverse of what was required. Figure 4 shows a schematic view of their valve.
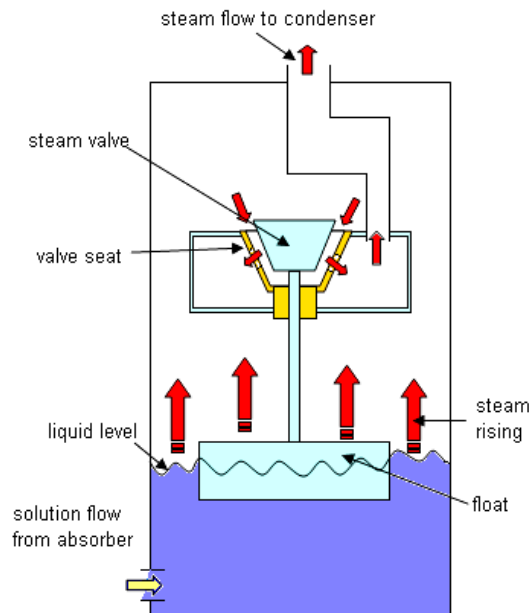


**Figure 4.** Showing a schematic view of an automatic steam valve in the generator of a VAR cycle due to Paurine et al. [19].

The operation of the automatic valve shown in Figure 4 would have resulted in either the solution in the generator needing to be cooled between pumping cycles, in order to allow the steam valve to open, or, if the machine was started with the solution in the generator vessel when heat was first applied, then the steam valve would probably not have opened: the solution contained in the vessel would be returned to the absorber without sufficient water being removed. The valve system described in the following paragraphs and diagrams aims to correct this error.

A schematic view of the automatic steam valve proposed by this author is shown contained within the generator in Figure 5. The position of the valve is controlled by a float (FV). Figure 6 illustrates the operation of the proposed steam valve. In Figure 6a, the generator half filled with solution during the first stage of the pumping cycle. In this position, the generator pressure would be constant and steam would flow past the open valve and into the condenser. As the solution level in the generator vessel rises, the float would be lifted, and once it contacts the upper stop, the valve would be raised

until it contacts its seat, as shown in Figure 6b. With the valve on its seat, steam would not be able to exhaust from the generator and the pressure within it would rise and the cycle would enter the second part of the pumping cycle. Once the pressure in the generator was sufficient, the solution would be forced back via the regenerator to the absorber as described earlier. As the level in the generator now falls, the valve would be kept on its seat by the pressure difference across it and block the steam exhaust. Eventually, the solution level will fall sufficiently for the float to contact the lower stop, shown in Figure 6c. The valve would need to be designed so that the weight of the float, unsupported by the solution, and that of the valve and valve-stem would be sufficient to pull the valve off its seat against the differential pressure across it. If this was not the case, then a pilot valve may be included to allow steam to bypass the valve in order to allow for equalising the vapour pressure across the valve to equalize and break the seal. This would allow the valve to come off its seat when the solution reached the lower level in the generator and allow the pumping cycle to re-enter its first part, shown in Figure 6d, by permitting the generator vessel to refill with solution.
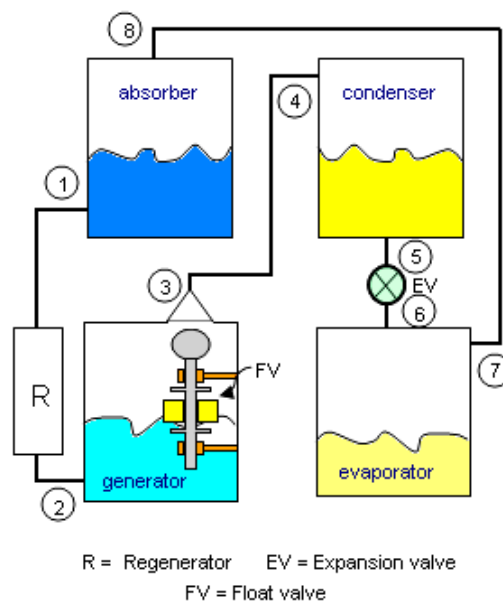


**Figure 5.** Novel automatic solution pumping system controlled by a float valve.
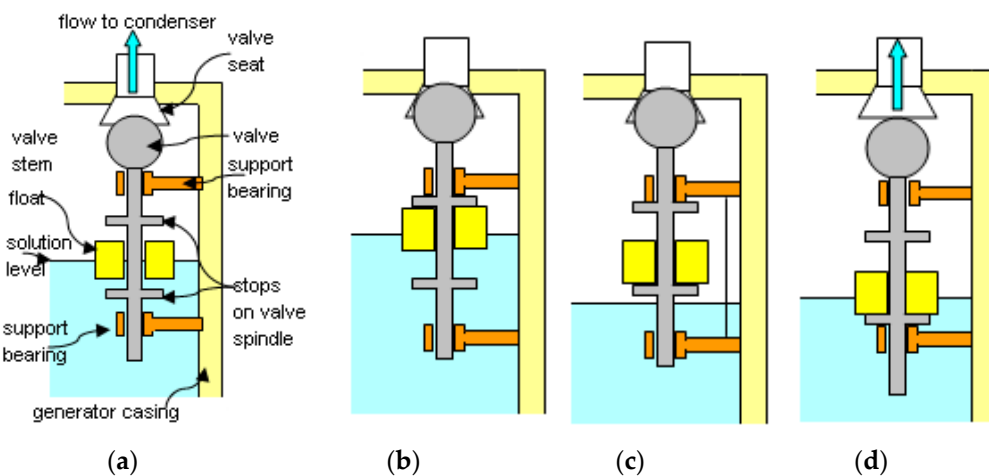


**Figure 6.** Operation of the float valve controlling the flow of solution between the absorber and generator. (**a**) Value part-open; (**b**) Value close; (**c**) Value about to open; (**d**) Value wide-open.

It should be noted that, in order to minimise pressure losses in the pipeline connecting the generator and condenser, it was recognised that the steam valve would need to be large. Proprietary steam valves of the correct volumetric flow capacity ($m^3$/s) would be too physically heavy to operate the way that is desired and also such valves may be too expensive for small capacity refrigerators. Therefore, a relatively lightweight float control was proposed.

## 7. Conclusions

(1) A novel method by which static head difference is used to control the flow of solution between the absorber and generator of an LiBr-$H_2O$ absorption refrigerator was described in Figure 2.
(2) A novel LiBr-$H_2O$ absorption cycle that does not require an electrically powered solution circulation pump was described.
(3) The addition of a steam valve positioned in the connecting pipeline between the generator and condenser provides the means by which the pressure in the generator can be controlled to determine the direction of the flow between it and the absorber, as shown in Figure 2.
(4) Calculations and preliminary test results were encouraging.
(5) A lightweight float valve with a large flow area was proposed in order to minimise pressure losses in the pipeline connecting the generator and condenser.

**Conflicts of Interest:** The author declares no conflict of interest.

## Nomenclature

| | |
|---|---|
| COP | Coefficient of performance |
| $f$ | recycle factor (-) |
| g | gravitational constant ($m/s^2$) |
| $L$ | hydrostatic height (m) |
| $M$ | mass (kg) |
| $\dot{m}$ | mass flow (kg/s) |
| $Q$ | heat (J) |
| $\dot{Q}$ | heat rate (kW) |
| $h$ | specific enthalpy (kJ/kg) |
| P | pressure (Pa) |
| T | temperature (°C) |
| $W$ | work (J) |
| X | concentration (-) |
| $U$ | internal energy (J) |
| $u$ | specific internal energy (J/kg) |
| VAR | vapour absorption refrigerator |
| Greek | |
| ρ | density (kg/$m^3$) |
| Subscript | |
| A | absorber |
| C | condenser |
| G | generator |
| E | evaporator |
| s | solution |
| W | water |
| 1, 2, 3, ... | cycle state points |

## References

1.  Xu, Z.Y.; Wang, R.Z. Absorption refrigeration cycles: Categorized based on the cycle construction. *Int. J. Refrig.* **2016**, *62*, 114–136. [CrossRef]
2.  Asdrubali, F.; Baldinelli, G.; Presciutti, A. Solar Cooling with Small-Size Absorption Chillers: Different Solutions for Summer Air Conditioning. Available online: http://www.crbnet.it/File/Pubblicazion/pdf/1439.pdf (accessed on 26 April 2013).
3.  American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. *ASHRAE Handbook, Fundamentals, S-I Edition*; American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.: Atlanta, GA, USA, 1993; ISBN:0-910110-97-2.
4.  Gao, H.; Liu, B.; Yan, Y. Numerical simulation of bubbles motion in lifting pipe of a bubble pump for lithium bromide absorption chillers. *Appl. Therm. Eng.* **2016**, in press. [CrossRef]
5.  Benhmidene, A.; Hidouri, K.; Chaouachi, B.; Gabsi, S.; Bourouis, M. Experimental investigation on the flow behaviour in a bubble pump of diffusion absorption refrigeration systems. *Case Stud. Therm. Eng.* **2016**, *8*, 1–9. [CrossRef]
6.  Zohar, A.; Jelinek, M.; Levy, A.; Borde, I. The influence of the generator and bubble pump configuration on the performance of diffusion absorption (DAR) system(s). *Int. J. Refrig.* **2008**, *31*, 962–969. [CrossRef]
7.  An, L.; Liu, D.; Chen, Y.; Yang, L.; Yang, M.; Lin, F. Theoretical and experimental study on the lifting performance of bubble pump with variable cross-section lift tube. *Appl. Therm. Eng.* **2017**, *111*, 1265–1271. [CrossRef]
8.  Saravanan, R.; Maiya, M.P. Influence of thermodynamic and thermophysical properties on water-based working fluids for bubble pump operated vapour absorption refrigerator. *Energy Convers. Manag.* **1999**, *40*, 845–860. [CrossRef]
9.  Chan, K.W.; McCulloch, M. Analysis and modelling of water based bubble pump at atmospheric pressure. *Int. J. Refrig.* **2013**, *36*, 1521–1528. [CrossRef]
10. Pfaff, M.; Saravanan, R.; Maiya, M.P.; Murthy, S.S. Studies on bubble pump for a water-lithium bromide vapour absorption refrigerator. *Int. J. Refrig.* **1998**, *21*, 452–568. [CrossRef]
11. Gu, Y.; Wu, Y.; Ke, X. Experimental research on a new solar pump-free lithium bromide absorption refrigeration system with a second generator. *Sol. Energy* **2008**, *82*, 33–42.
12. Saravanan, R.; Maiya, M.M. Experimental analysis of a bubble pump operated LiBr-H$_2$O vapour absorption cooler. *Appl. Therm. Eng.* **2003**, *23*, 2383–2397. [CrossRef]
13. Saravanan, R.; Maiya, M.P. Effect of component pressure drops in two-fluid pumpless continuous vapour absorption refrigerator. *Energy Covers. Manag.* **1997**, *38*, 1823–1832. [CrossRef]
14. Asdrubali, F.; Grignaffi, S. Experimental evaluation of the performance of a H$_2$O-LiBr absorption refrigerator under different service conditions. *Int. J. Refrig.* **2005**, *28*, 489–497. [CrossRef]
15. Eames, I.W.; Marr, N.J.; Sabir, H. Evaporation Coefficient of Water, International. *J. Heat Mass Transf.* **1997**, *40*, 2963–2974. [CrossRef]
16. Abrahamsson, K.; Gidner, A.; Jernqvist, A. Design and experimental performance of an absorption heat transformer with self circulation. *Heat Recovery CHP* **1995**, *15*, 257–272. [CrossRef]
17. Stephens, J.; Eames, I.W. *Lithium-Bromide Absorption Refrigeration*; A Project Report; School of the Built Environment, University of Nottingham: Nottingham, UK, 1999.
18. Eames, I.W.; Wu, S. A valve operated absorption refrigerator. *Appl. Therm. Eng.* **2003**, *23*, 417–429. [CrossRef]
19. Paurine, A.; Maidment, G.G.; Eames, I.W.; Missenden, J. Development of a thermo-gravity pumping mechanism for circulating the working fluids in a novel LiBr-H$_2$O vapour absorption (VAR) system. *Appl. Therm. Eng.* **2012**, *47*, 25–33. [CrossRef]
20. Rogers, G.F.C.; Mayhew, Y.R. *Thermodynamic and Transport Properties of Fluids, (SI Units)*, 3rd ed.; Basil Blackwell: Oxford, UK, 1981.