COMBINED FLASH-VAPORISATION AND SOLAR EVACUATED TUBE HEAT-DRIVEN ABSORPTION AIR CONDITIONING SYSTEM FOR THE TROPIC

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ABSTRACT

A theoretical study based on the thermodynamic consideration has been conducted for a combined flash-vaporisation and solar evacuated tube, heat-driven air conditioning system for cooling of buildings under the tropic weather conditions. The initial analysis has shown that the performance of a basic (lithium bromide–water) LiBr-H_2O absorption cooling system can be enhanced with an evacuated-tube collector coupled with the addition of a 'flash vaporisation' unit. The preliminary computational results based on the thermodynamic consideration demonstrated the system efficiency increment patterns prevailed in the system associated with the evacuated tube-generator input temperature, subject to the local meteorological and the solar thermal energy characteristics. Subsequently, a parametric study was conducted to explore the influence of the principle operating parameters on the performance of the basic single stage, cooling cycle of the absorption air conditioning system. The parametric findings and the monographs thus developed are beneficial for both designer and user's guide, especially for tropical solar-absorption application consideration. The analytical results revealed that coupled with the evacuated tube, the flash vaporisation could enhance the overall system performance. The recommended operating range, based on local cooling requirements are: generator pressure, 0.08-0.12 bar with generator, condenser, absorber and evaporator temperature ranges of 85 – 100 C°; 37 - 50°C, 29 - 35°C and 5 - 14°C respectively.

Keywords: *Absorption, Air-conditioning, Solar Radiation, Evacuated Tube Solar Collector, Flash Vaporisation*

1.0 INTRODUCTION

The heart of any air-conditioning system is the refrigerating sub system, which absorbs heat from the ambient air from the space to be cooled as it passes through the chiller. Heat absorbed by the refrigerant flowing though the chiller is rejected to the environment. In general, there are two types of refrigeration cycles, which could provide the refrigerating effect for air-conditioning: (a) the conventional vapour compression cycle and (b) the vapour absorption cycle. The vapour compression air conditioning system uses a compressor, which compresses the vaporised refrigerant in order to elevate its temperature approximately 10°C higher than the condenser ambient temperature for efficient heat rejection. To accomplish the same function, the air conditioner working on the vapour absorption cycle utilises a secondary fluid, usually called absorbent, to absorb the refrigerant at a lower temperature level and then delivering the liquid to the higher temperature level usually by means of a pump.

The advantages of conventional compression cycle air conditioners are their reliability and efficiency. The disadvantage is that the refrigerants they use have a damaging effect on the ozone layer. The refrigerants are chlorofluorocarbons(CFC), for example R-22, which are known harmful to the environment as pointed out by Baxter and Fischer [1]. Vapour absorption system, on the other hand, can be operated from low-grade thermal energy, from various sources such as waste heat from plants, solar radiation, geothermal etc. They are simple as they include fewer moving parts, consequently they tend to generate less noise. The current study therefore deals only with the study of vapour absorption cycle air conditioner for the reasons mentioned above. The use of solar energy as the alternative energy, as it is available abundantly locally, is obviously attractive for driving a vapour absorption air conditioning system. Moreover, the solar heat-driven cooling air conditioner would be very beneficial especially in areas far away from electrical energy distribution grid such as that of isolated villages and petroleum rigs offshore. Also, it could reduce peak demand for electric power and could be of useful especially in emergency cases.

The solar-powered air conditioning technology for thermal comfort is not new. However, until today, these solar-powered cooling machines are mainly designed for research purposes and that there were still many areas yet to be improved in order for the chillers to be widely accepted by the public. Various studies based on the use of absorption cycle principles had been investigated by many researchers. A review on the subject can be found, for example in Auh [2] and Grossman [3]. Most of the systems investigated and reported in the literature elsewhere are for single-stage absorption application, with low-temperature solar collectors for heat input at the generator.

Sheridan [4] presented the study of a solar-air conditioning of a house, for a typical clear summer day, in Brisbane, Australia. The solar collector has an average daily efficiency of 0.32 while the average COP of the refrigerator was 0.65. The results had shown the feasibility of such design although the associated cost study conducted was reported to be only marginally economic, probably due to high initial cost in Australian condition.

The first commercial absorption air conditioner built specially for solar-operation was the 3-ton 'Solarie" unit by Arkla Industries [3, 5]. It was not cost-effective as the design was based on the earlier gas-fired machine. Ishibashi [6] described the improved solar-absorption machines by considering the low-temperature requirement of the single-stage absorption chiller. He reported the success of residential cooling by the water-heating lithium bromide-water system applied on a double-story solar house in Japan. Total solar radiation during summer days was about 2 MJ/m^2 day with collector efficiency of approximately 0.3 and a COP chiller of 0.5 was reported on the clear days. Gas was used as auxiliary fuel for the burner installed below the heat storage tank.

Grossman et al [7] described the improved solar-absorption machines by incorporating a generator design based on falling film type and an addition of a unit of solution pre-heater. The absorption system equipped with an auxiliary generator was expected to be able to operate at overall capacities up to 150% of the nominal. However the adding of the main components contributed to higher cost of the machine.

Although the use of solar energy for powering absorption was principally attractive, one of the shortcomings of these machines was their low efficiency. The single stage absorption systems are limited in COP to about 0.7 due to Carnot limitation [3]. This corresponds to about 1.45 kW of steam rate per kW of cooling. The system required a rather large collector area to supply the solar heat needed for their operation. Multiple stage absorption systems [8, 9] utilised the heat regeneration principle where heat rejected at condenser is used to power additional generators thus doubling the amount of refrigerant extract out from the solution in the generator. These multiple stage systems require high-temperature heat source and are more expensive in cost. Unfortunately, the high temperature was not easy to attain by the conventional solar collector system without an elaborated system, such as the solar parabolic concentrator. The current work, however, proposed the use of a 'flash vaporiser' coupled with the use of vacuum tube solar assisted technology to increase the rate of vaporisation of refrigerant (H_2O) from the $LiBr$-H_2O solution thus improving the system efficiency without the need of multiple stage heat requirement.

The review of the solar evacuated tube for temperature build-up application is well documented [10-12]. Balzar et al. [12] and Kumar et. al. [13] described the use of this technology for cooking. Li et al. [14] demonstrated the ability of the vacuum tube for temperature build-up as well as thermal insulation purposes. The findings of the above mentioned workers [10-14] demonstrated that temperature can be built up relatively easily by such evacuated tubes with temperature range of 80-150ºC. The applications of the generating heat technique as mentioned was many but the literature describing the use of this technology for absorption air conditioning is scarce. Pilatowsky et. al. [15] discussed the use of a monomethylamine–water absorption system for milk cooling in Mexico. The results showed that the monomethylamine– water pair is a good candidate for solar absorption refrigeration systems at moderate evaporator temperatures viz. -5 to 10°C for the Mexico's semi-tropical climatic conditions. The thermal yearly behaviour of the solar heating system was analysed using a program based on the f-chart method. It was found that the maximum COP is 0.39 in autumn and the minimum COP is 0.36 in spring. The current work however, described the use of $LiBr$-H_2O working pair for a tropical condition designed for space cooling. In this work, a study conducted has shown that the system performance of the solar-powered absorption system could be enhanced through two design considerations viz. (1) with the addition of a simple 'flash evaporation' unit and (2) coupled with the use of vacuum or evacuated solar generator for the heat source. A parametric study was subsequently conducted to explore the influence of principle variables on the performance of the basic cooling cycle of the absorption air conditioning system. Both of the thermodynamic design and computer study were based on a local condition at Kuching (latitude 01° 29' N; longitude 110° 20' E), East of Malaysia, however the findings can be useful for other similar places too.

2.0 CONSIDERATION FOR LOCAL SOLAR THERMAL CHARACTERISTICS

Kuching (latitude 01° 29' N; longitude 110° 20' E) is a typical town located in the eastern part of Malaysia. Being situated in an equatorial and maritime region, it has quite plentiful yet consistent sunshine and solar radiation. The average sunshine hours are 5–6 hours per day and mean air humidity around 85% (Appendix A). From the 6-years of the daily 24-hour solar radiation tabulation data, obtained from the local meteorological department [16], had shown that all the days were having somewhat similar solar radiation pattern. The solar radiation, θ, of Kuching can be represented by Equation (1a) and shown as in Figure 1 respectively.

$$\theta = \begin{cases} \theta_{\max} = -0.13t^2 + 3.16t - 15.09; & \text{at } 7 \le t \le 17 \\ \theta_{\max} \cong 0; & \text{at } 6 \le t \le 18 \\ \theta_{avg} = -0.069t^2 + 1.690t - 8.201; & \text{at } 7 \le t \le 17 \\ \theta_{avg} \cong 0; & \text{at } 6 \le t \le 18 \end{cases} \tag{1a}$$

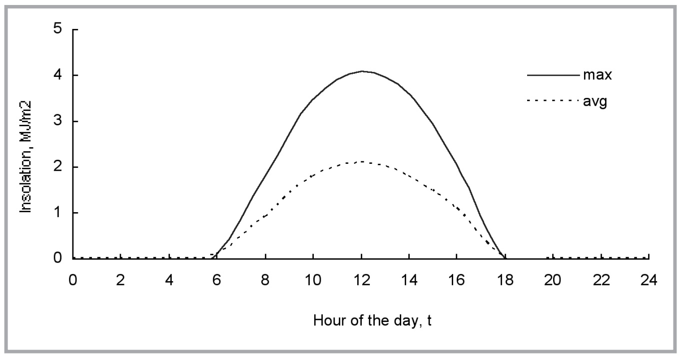


Figure 1: Solar radiation model for Kuching

As shown from Figure1, the maximum significant solar radiation occurs around 10:00 am until 04:00 pm. This is the time where the ambient temperature is highest and therefore cool air is most needed. It is to be noted that fortunately, this is also the time where solar heat-powered absorption air conditioning systems could be operated most effectively. Solar radiation characteristic in the form of time series data had been plotted for the period from 03-07 May as depicted in Figure 2.

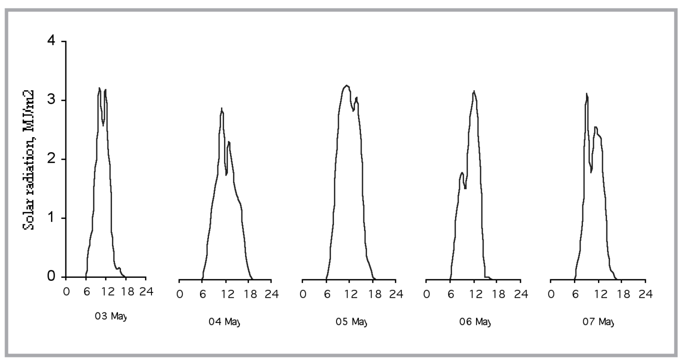


Figure 2: Time series data of solar radiation 03 May – 07 May

3.0 PERFORMANCE OF THE EVACUATED TUBE SOLAR COLLECTOR AND HEAT INPUT FOR THE SOLAR GENERATOR

Evacuated tube solar collectors are designed to achieve high thermal energy gain while minimise heat loss potential. It minimises convective heat loss by placing the solar absorbing surface in a vacuum. The radiation heat loss is minimised by using a low emissivity absorber surface [10, 14, 17]. The performance of such heat source system can be evaluated by the standard pre-heater test method specified in the International Standard for solar heating [18]. The useful energy gain (Q_u) is calculated using the ISO 9459-2 performance model given in Equation (1b).

$$Q_u = a_1 + a_2\theta + a_3(T_{wi} - T_a) \quad (1b)$$

Based on Equations (1a) and (1b) above, the temperature attainable for running the evacuated tube solar collector – absorption air conditioning system can well be estimated.

4.0 PARAMETRIC STUDY

4.0.1 Solar thermal energy and solar-generator input temperature

Input of the solar generator temperature can be calculated by Equation (2)

$$T_{gen} = \begin{cases} 12.9 \cdot \varepsilon \cdot A \cdot (1 + 0.43(t+2.3)) & \text{at } 5 \le t \le 7 \\ 13.5 \cdot \varepsilon \cdot A \cdot (\theta + (1 + 0.37(t+2.5))) & \text{at } 8 \le t \le 17 \\ 17.5 \cdot \varepsilon \cdot A \cdot (1 + 0.7(2.6 - t)) & \text{at } 18 \le t \le 21 \end{cases} \quad (2)$$

Therefore, the generator temperature level of well above 80°C could be achieved for driving the solar-driven air conditioning system, as far as local weather condition is concerned.

Such operating range is corresponds to around 09:00 morning until 19:00 afternoon, for the month of May as shown in Figure 3. Higher level of temperature is expected in the month of June – September.

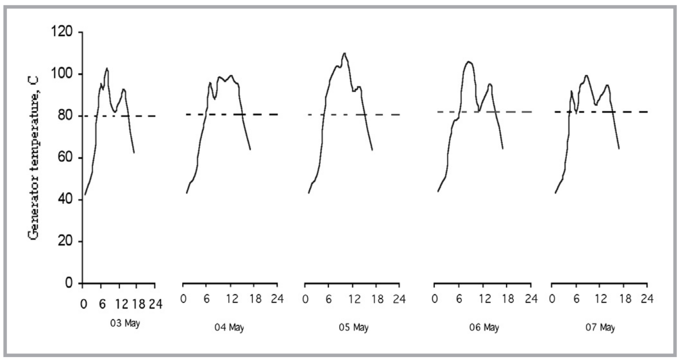


Figure 3: Time series data of solar generator temperature

4.0.2 Thermodynamic consideration and modelling of the solar-driven absorption cycle

Figure 4 is a schematic showing alignment of the absorption cycle cooling system. As can be seen from the figure, the flash vaporisation unit add in an extra unit of energy output in the form of vapour. Here, the object of using flash vaporisation technique is to enhance the generation of vapour to quickly boil off the working fluid i.e. lithium bromide-water solution. The numbers refer to the state point at specific location as indicated in the figure.

In designing an absorption air conditioning plant, for heat rejection, the condenser temperature must be greater than the ambient temperature, around 10°C for local consideration.

The pressure exerted in the condenser would be the saturation refrigerant i.e. saturated water pressure, P_c

$$P_c = f(t_c, \text{ saturated water}) \quad (3)$$

The state points 1 through 20 and the associated properties such as temperature, t and enthalpy, h, as stated in Equation (4) through Equation (17) below are referring to Figure 4.

The generator exit temperature, t_{19} would be close to the generator temperature, so that $t_{19} \approx t_g$. The evaporator pressure, P_e corresponds to the saturated vapour condition at t_e, i.e..

$$Pe = f(t_e, \text{ saturated vapour}) \quad (4)$$

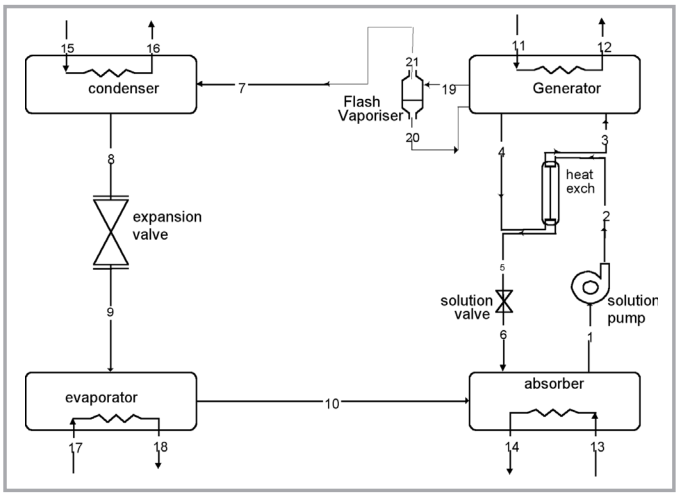


Figure 4: Schematic of a conceptual design set-up of a flash vaporiser in an absorption cycle

At solution valve with throttling process, the enthalpy at points 6 and 5 is assumed the same,

$$h_6 = h_5 \quad (5)$$

The temperature at heat exchanger exit, t_5, is

$$t_5 = t_4 - \varepsilon_{hx}(t_4 - t_2) \quad (6)$$

Consider a dimensionless parameter- mass flow ratio, MFR. Here, it can be defined by mass flow of solution from generator / mass flow of refrigerant (water) i.e.

$$MFR = \frac{M_{4weak}}{m_{10}} = \frac{X_{gen}}{X_{gen} - X_{abs}} \quad (7)$$

where, $m_{10} = \dfrac{Q_e}{h_{10} - h_9}$

Assuming conservation of mass of each component, the following equations are obtained:

The rate of heat transfer at generator with flash unit is,

$$Q_{g,flash} = M_{4weak} \times h_4 - M_{3strong} \times h_3 + m_{19} \times h_{19} - m_{20} \times h_{20} \quad (8)$$

The following conditions are assumed for above equation: (a) $t_{20} = t_{21}$ since Points 20 and 21 are within same internal of chamber; *(b)* $t_7 = t_{21}$ ignoring friction loss and *(c)* $t_{19} = t_g$ assuming immediate temperature exit equal component temperature.

The rates of heat transfer at absorber Q_a, and condenser, Q_c are:

$$Q_a = -(M_{4weak} \times h_1) + (M_{3strong} \times h_6) + (m10 \times h_{10}) \quad (9)$$

$$Q_c = m_7 \times (h_7 - h_g) \quad (10)$$

The coefficient of performance, COP with flash evaporation is evaluated by the following equations:

$$COP_{flash} = \frac{Q_e}{Q_{g\,flash}} = \left(\frac{Q_e}{M_{4weak} \times h_4 - M_{3strong} \times h_3 + m_{19} \times h_{19} - m_{20} \times h_{20}} \right) \quad (11)$$

It follows that heat flow balance at the flash evaporation can be defined as, Equation (12),

$$m_{21} \times h_{21} = Q_{flash} + m_{19} \times h_{19} - m_{20} \times h_{20} \quad (12)$$

Such that its rate of heat transfer is,

$$Q_{flash} = m_{21} \times h_{21} - m_{19} \times h_{19} + m_{20} \times h_{20} \tag{13}$$

The above are the equations governing the cycle with the addition of the flash evaporation unit. Without one such unit, the COP of the basic cycle however, would be

$$COP = \frac{Q_e}{Q_g} = \left[\frac{Q_e}{M_{4weak} \times h_4 - M_{3strong} \times h_3 + m_7 \times h_7} \right] \tag{14}$$

The evaporation rate is increase due to the flow restriction effect such that,

$$m_{19} \times h_{19} - m_{20} \times h_{20} \leq m_7 \times h_7 \tag{15}$$

And therefore, $COP_{flash} \geq$ COP as expected.

The fraction of water converted by flash vaporisation into steam, y, can be estimated from enthalpy balance by,

$$y = \frac{(h_{initial} - h_f)}{(h_g - h_f)} \tag{16}$$

i.e.

$$y = \frac{(h_{19} - h_{f,20})}{(h_{g,21} - h_{f,20})} \tag{17}$$

The LiBr-H_2O solution concentration could be modelled as described in an earlier work [19].

The flow chart summarised the computation steps for the parametric study of the thermodynamic cycle is given in Figure 5. The purpose of the study was to explore the influence of the principle design variables on the performance of the system. It would also serve as a simple optimisation method to observe the critical performance parameters of the absorption cycle.

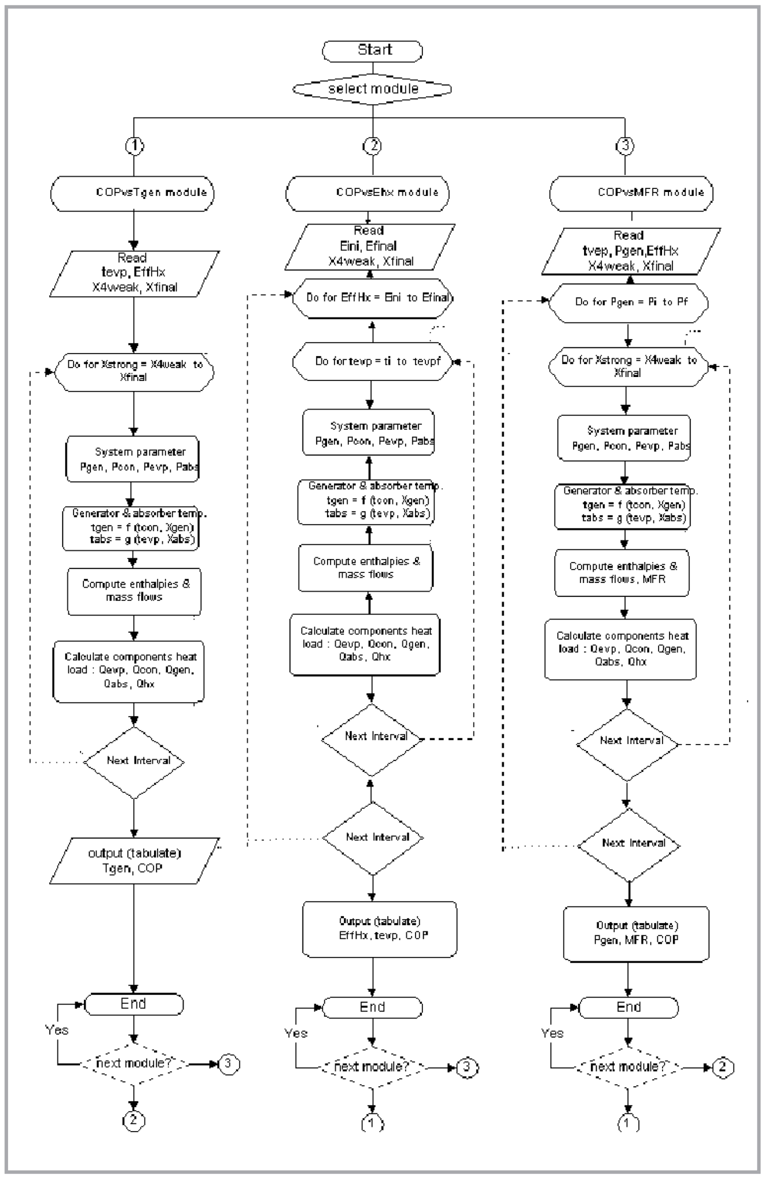


Figure 5: Flow chart for the parametric study

5.0 RESULTS AND DISCUSSIONS

The performance i.e. coefficient of performance (COP) of the absorption air-conditioning system is influence by various parameters such as generator temperature, heat exchanger effectiveness, mass flow ratio and generator pressure as shown in Table 1. The figures or diagrams related to the above influencing parameters are also tabulated for ease of reference as well as a summary of result obtained from the theoretical investigations.

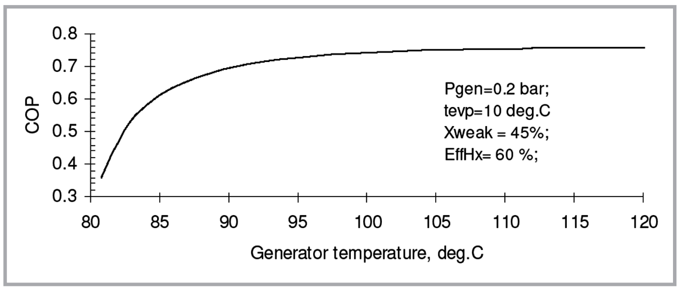
Table 1: Influencing parameters and their related diagrams

Generator temperature	Figure 6
Heat exchanger effectiveness	Figure 7
Mass flow ratio (MFR)	Figure 8
Generator pressure	Figures 6, 8

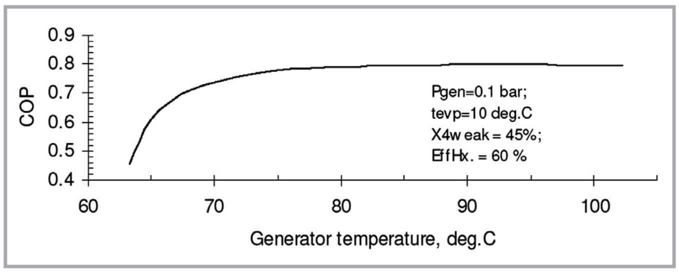
The effect of generator temperature on system performance is shown in Figure 6. All the three graphs, at three different generator operating pressures, show a general trend of COP verses generator temperature. At constant pressures, the values of COP initially increase with generator temperatures, reach their maximums and finally decrease very slowly (almost stable) with generator temperature. Therefore, it seemed that there was a point of optimum evacuated tube (generator) temperature. This point occurred when maximum of vapour successfully boiled-off from the LiBr–H_2O solution. However, after some time, the condensation of parts of the vapour contributed to a slight, remarkably small of COP gradual reduction. Also, it can be seen that generally COP increases with lower pressures i.e. higher vacuum. For instance, at 0.2 bar, the system COP would be around 0.6 correspond to generator input of 85°C. At 0.04 bar, a COP of 0.8 could be obtained under similar condition.

Figure 7 shows the influence of heat exchanger effectiveness, EffHx on the performance of the absorption system. It can be seen clearly from the graph that COP increases with EffHx. The increase in COP with EffHx appears to be exponential. This is because the heat exchanger pre-heat the cold stream from the absorber going to the generator. And it also pre-cool the hot stream from the generator to the absorber. Both processes providing heat recovery consequently the heat input required to the system will be reduced.

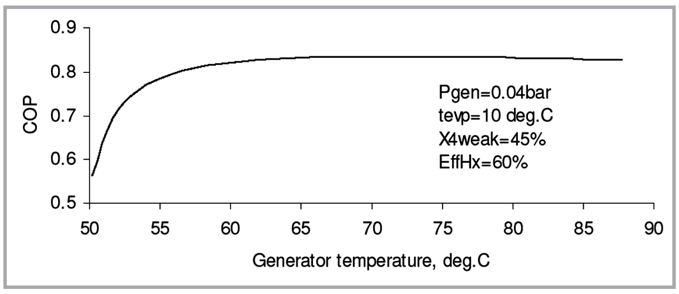
Figure 8 shows that generally, COP decreases with increasing mass flow ratio (MFR). This general trend confirmed to the study of absorption heat pump (AHP) by Abraham et. al. [20]. For the current study, the effect of COP



a) 0.04 Bar



b) 0.1 Bar



c) 0.2 Bar

Figure 6: The effect of generator temperature on COP at various constant generator pressures

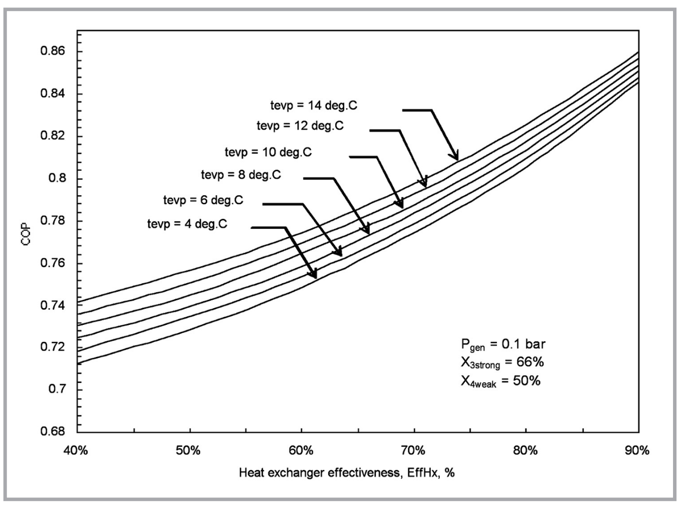


Figure 7: COP as a function of heat exchanger effectiveness at various evaporator cooling temperature

reduction is more profound at high operating generator pressure. At MFR=5 and MFR=50, the system would have COP range of around 0.7-0.8 to only about 0.4-0.6 respectively. This is because increasing MFR causes mass flow rate of all streams to increase and more water must be vaporised in the generator. Therefore, for the current application, it may be deduced that the good MFR operating range is less or between 5 through 13 associated with the condenser rejection range requirement. Smaller dimensions, simpler and less expensive pump could be used for circulating the LiBr-H_2O solution.

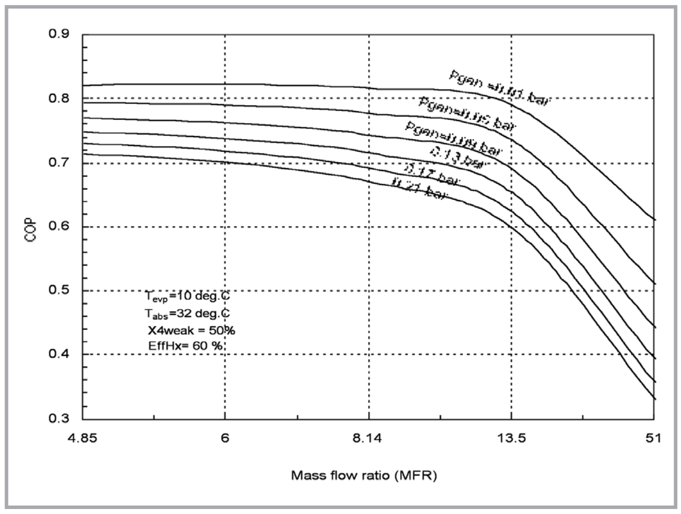


Figure 8: COP as a function of mass flow ratio

From the parametric study conducted, it can be deduced that, in as far as the present study and the operating conditions are concerned, for absorption cooling application in Kuching, the optimum operating conditions could be a combination of the followings operating conditions:

- P_{gen} = 0.1 bar,
- T_{gen} = 100°C,
- T_{evp} = 10°C, and
- T_{abs} = 32°C.

And under these operating conditions, hypothetically, a COP of around 0.7-0.8 could be possibly attained. And, the effectiveness of the flash vaporiser, i.e. the increment of fraction of water converted by flash vaporisation into steam, Δy is estimated to be in the range of 0.1-0.4. The rate of vaporisation of the solution due to the restriction effect must be kept as high as possible. Such effectiveness however depends on the design of the actual construction unit.

The present study showed that the combined system could be practical for use in the tropical condition. The COP range predicted is generally superior i.e. around 0.6-0.8 depends on the operating conditions. A comparison with that reported by Pilatowsky et al.[15] where the COP range reported is from 0.39 to 0.36. Firstly, this is because the solar radiation level is higher in Kuching, ranging around 2.3 – 3.2 MJ m^{-2} while under the Mexico's condition, the solar radiation intensity level is lower, fluctuated from 1.7 in autumn to 2.5 MJ m^{-2} in spring. Secondly, the increment of vaporisation help in the quickly boiled-off of water vapour from the LiBr–H_2O solution thus increasing system performance.

Also, a simple compare with that of Ishibishi [6], the current system could achieve higher system performance due to the advantage of location and the associated solar thermal characteristics. This is understandable as total solar radiation during summer days in Japan was only about 2 MJ/m^2 day. Similar discussion can be presented to other basic single-stage absorption system operating at similar working conditions.

6.0 CONCLUSIONS

The present paper provides some technical performance of an absorption system powered by solar evacuated tube coupled with a flash evaporation unit based on the local meteorological behaviour. Generally, the analytical results revealed that coupled with the evacuated tube, the flash vaporisation could enhance the overall system performance. Other conclusions that can be drawn are given in the following paragraphs.

First, based on the parametric study conducted, the operating range for local cooling requirement for solar-absorption air conditioning system can be recommended i.e. generator pressure: 0.08-0.12 bar; generator temperature: 85–100°; condenser temperature: 37-50°C (for effective heat rejection); absorber temperature: 29-35°C and evaporator temperature: 5-14°C. Detail design of the flash evaporator and further experimental works, however, must be carried out to verify the findings associated with the cooling requirements described herein.

Second, the performance of the absorption system is somewhat complex, depending on many operating conditions. Generally, COP increases with evacuated tube (generator) input temperature, evaporator temperature and heat exchanger effectiveness. However, the COP decreases with operating pressure (high vacuum) and mass flow ratio.

Finally, nomographs for design of the solar evacuated tube - absorption system coupled with the flash vaporisation were developed for a wide range of EffHx and MFR under some operating conditions as presented in Figures 7 and 8 respectively. The monographs thus developed are beneficial for both designer and user's guide, especially for tropical solar-absorption application consideration.

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Appendix A

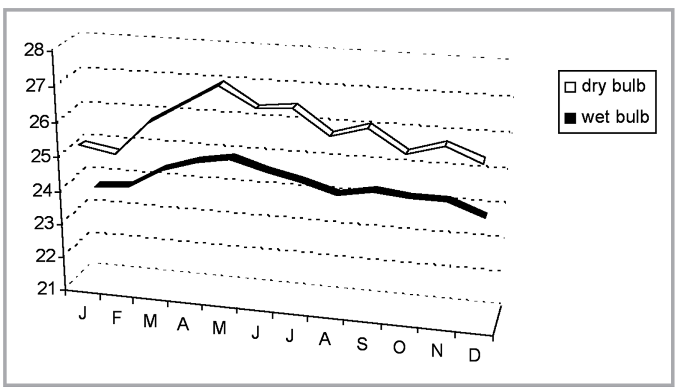

Figure A1: Monthly mean dry- and wet-bulb temperature profile in 1996(unit:°C)

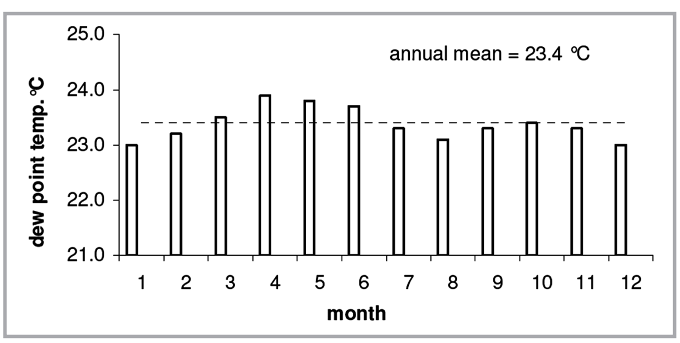

Figure A2: Monthly 24-hour mean dew point temperature in 1996 (unit:°C)

Table A1: Records of monthly mean relative humidity (%)

Month / year	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Ann. Mean
1991	88.6	87.1	85.8	85.5	86.8	82.7	81.8	83.0	82.4	85.0	87.4	88.4	**85.4**
1992	88.9	87.8	86.7	85.2	85.8	83.7	83.3	81.8	84.6	86.6	89.0	88.7	**86.0**
1993	86.6	84.9	86.5	85.4	86.6	84.0	83.9	82.5	85.5	85.7	87.3	87.1	**85.5**
1994	88.5	86.3	88.0	86.3	86.5	85.5	78.3	83.3	83.0	86.1	87.6	86.5	**85.5**
1995	88.0	88.3	87.0	86.0	84.5	83.8	83.6	86.7	85.5	86.7	86.9	87.8	**86.2**
1996	87.6	90.0	86.6	85.5	82.9	84.9	82.8	84.6	84.2	87.5	85.2	86.0	**85.7**
1997	86.5	86.3	84.6	85.2	84.5	82.6	81.4	77.4	85.5	-	-	-	-

Station: Kuching Airport (Lat: 01Deg.29'; Long.: 110° 20'E; Height Above M.S.L.: 21.7 m)

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GREEK

θ	= insolation (MJ/m^2) as in Equation 1a and 2
θmax	= maximim insolation as in Equation 1a
θavg	= average insolation as in Equation 1a
ε	= overall efficiency of the solar collecting system (=0.4) as in Equation 2
ε_{hx}	= heat exchanger efficiency as in Equation 6

NOMENCLATURE

a_1	= 0.597 as in Equation 1b
a_2	= 1.066 as in Equation 1b
a_3	= –0.208 as in Equation 1b
A	= area (m^2) as in Equation 2
COP	= coefficient of performance of the absorption cycle system
EffHx	= heat exchanger effectiveness
f	= function
h	= enthalpy, kJ/kg
t	= hour of the day as in Equations 1a and 2
T_a	= ambient temperature as in Equation 1b
t_c	= condenser temperature. °C
te	= evaporator temperature. °C
T_{gn}	= output of solar generator temperature(°C) as in Equation 2
tg	= generator temperature, °C
T_{wi}	= the temperature of the tank at the start of the day
M_{weak}	= the mass flow of weak (in LiBr) solition from the generator, as in Equation 7 & 8
M_{strong}	= the mass flow ofstrong (in LiBr) solution from the absorber, as in Equation 8
Q	= heat transfer rate, kJ/kg
X	= the solution concentration (%)
y	= the fraction of water converted by flash vaporisation into steam, as defined by Equation 16

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