EXPERIMENTAL AND NUMERICAL INVESTIGATION ON BUBBLE PUMP OPERATED ABSORPTION-DIFFUSION MACHINE BASED ON LIGHT HYDROCARBONS

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ABSTRACT
Experimental and numerical investigations of an air cooled diffusion-absorption machine operating with a binary light hydrocarbon mixture (C_4H_{10}/C_9H_{20}) as working fluid and helium as pressure equalizing inert gas are presented in this paper. The machine made of copper -an available and very good heat conducting metal- was designed, constructed and experimentally analysed. Its cooling capacity is 30-40 W. Cold is produced at temperatures between -10 and +10°C for a driving temperature in the range 120-150°C. The maximum *COP* approaches 0.175. A thermodynamic model for the machine is then developed. The cycle performance was parametrically studied by computer simulation for 1 kW cooling capacity design. Present experimental and simulation data are expected to provide useful information for determination of the design parameters for a solar powered machine.

ETUDE NUMERIQUE ET EXPERIMENTALE D'UNE MACHINE FRIGORIFIQUE À ABSORPTION DIFFUSION À POMPE À BULLES OPÉRANT AUX HYDROCARBURES LÉGERS

RESUME
Ce papier porte sur l'étude expérimentale et numérique d'une machine frigorifique à absorption diffusion refroidie à l'air ambiant et utilisant un mélange d'hydrocarbures légers comme fluide de travail et l'hélium comme gaz égaliseur de pression. La machine est confectionnée en cuivre, un métal disponible et très bon conducteur de chaleur. Sa capacité frigorifique est de l'ordre de 30-40 W. Le froid est produit à une température entre -10 et +10°C pour une température de chaleur motrice dans l'intervalle 120-150°C. Le *COP* maximal est proche de 0.175. Un modèle thermodynamique du cycle frigorifique à diffusion absorption opérant au mélange correspondant a été ensuite développé. Une simulation numérique a été effectuée pour l'étude paramétrique des performances du cycle pour une machine de capacité frigorifique de 1kW. Les résultats expérimentaux et numériques sont de très utiles informations pour la conception et la confection d'une machine actionnée à l'énergie solaire.

NOMENCLATURE

COP	coefficient of performance	-	ξ	mass fraction	-
SRC	condenser sub-cooling	-	*T*	temperature	(°C)
SRA	absorber sub-cooling	-	*P*	Pressure	(bar)
Eff	efficiency	-	*n*	molar flow rate	(mol/s)
SHX	solution heat exchanger	-	*m*	masse flow rate	(g/s)
Pinch	pinch	-	*Q*	heat	(W)
DAR	diffusion absorption refrigerator	-	r	rich	

w	weak	ref	refreshment
Syst	system	*B*	boiler
PartMax	maximum partial	Abs	absorber
PartMin	minimum partial	evap	evaporator
cd	condensation	gen	generator
abs	absorption	pb	bubble pump

1. INTRODUCTION

The diffusion absorption technique invented by the Swedish engineers von Platen and Munters [1] in the 1920s is the best suited for solar applications. Its thermodynamic cycle is based on a refrigerant/ absorbent pair mixture as working fluids and an inert gas as pressure equalizing. A thermally driven bubble pump, which can be powered by either waste heat or solar thermal energy, is used to circle the liquid solution. The absence of any mechanical moving part allows a silent and very reliable machine [2] in addition to an economical and natural cycle.
The use of ammonia/water mixture as working fluids and hydrogen as auxiliary inert gas [3, 4] can provide cooling at very low temperature (-20 to -30°C depending on the diffusion absorption cycle configuration [2, 5]) but it requires high driving temperature (more than 150°C) which is unsuited for solar heat. Furthermore ammonia is destructively corrosive with copper.
A working fluid based on organic fluids was investigated [2, 4] and it was concluded that a complete diffusion refrigerating system that uses this mixture may be powered by solar energy.
The use of thermal solar energy is also possible with water/lithium bromide mixture fluids [6] (driving heat temperature below 100°C). Nevertheless, it presents some difficulties like to be operated under very low pressure and essentially problems related to the crystallisation of the salt.
The search of other working fluids suitable to be used with diffusion absorption machine for solar applications continues to be the focus of active research in the air conditioning and refrigeration field. In this contest, a low capacity diffusion absorption machine operating with a binary mixture of light hydrocarbons is designed, constructed and experimentally investigated. We present in this paper the results of our investigation with a computer simulation for a similar machine with 1 kW cooling capacity.

2. WORKING PRINCIPLE

Figure 1 shows a schematic assembly diagram of Platen-Munters based absorption diffusion machine. The rich solution from the absorber flows by gravity to the generator. The solution is first heated at the bottom of the pump tube by the heat flux Q_{pb}. Small vapour bubbles begin to appear. As the solution is being heated continuously, the vapour bubbles grow and merge with each others then form plugs filling the whole tube section. The rising refrigerant vapour bubbles lift the corresponding liquid slug by “bubble action” to the tube’s top. In the boiler the pumped solution is eventually further heated by additional heat flux Q_B, so an additional quantity of vapour is generated. Whereas the resulting weak solution leaves the generator and flows back (11) to the absorber via the solution heat exchanger, the vapour rises toward the condenser through the rectifier (13) where the rest of absorbent nonane vapour is removed. It condenses and falls back (14) into the boiler.

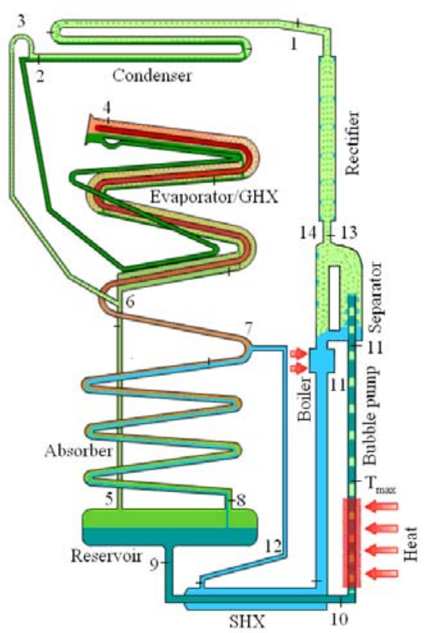

Figure1: Schematic diagram of the DAR

The resulting condensate (2) flows to the evaporator while the uncondensed gas flows to the reservoir (absorber's bottom) through a gas bypass (3). At the evaporator entrance (4), the partial pressure of the liquid butane is reduced by mixing it with the inert gas helium returning from the absorber. As a result, the liquid butane begins evaporating at law temperature. Since the liquid refrigerant is evaporating, its partial pressure in the vapour phase rises. The resulting butane-helium gas mixture leaves the evaporator (6), falls down in the reservoir (5) and flows upward the absorber in counter current to the weak solution arriving from the absorber's top (12). Along the absorber the weak solution absorbs the butane vapour. The residual helium-butane gas mixture (7) flows upward the evaporator and the rich solution at the bottom down toward the generator (9) through the regenerative solution heat exchanger.

3. EXPERIMENTAL ARRANGEMENTS

To investigate the performance of a bubble pump operated diffusion absorption refrigeration machine based on light hydrocarbons, an experimental prototype made of copper was built. The working fluids mixture is composed of n-butane (C_4H_{10}), as refrigerant, n-nonane (C_9H_{20}) as absorbent and helium (He) as pressure equalizing inert gas.

3.1 Experimental set up and procedure

Figure 1 shows also the schematic assembly illustration of the experimental system. The design of the main components allows investigating the bubble pump behaviour. After the tightness test the whole system is maintained at the required vacuum pressure. The reservoir is charged by liquid nonane followed by vapour butane and finally helium gas is added until the required total pressure is reached. The electric heater supplying heat to the generator is turned on and a series of measurements is immediately taken by varying the power input. The temperature measurements are recorded in a PC via a data acquisition system. When the steady state is established a fixed amount of water is poured in the water tank that contains the evaporator. The performance parameters, cooling capacity and *COP*, are calculated from the measured parameters. The cooling capacity Q_{evap} is estimated by calculating the rate of heat removed from the evaporating refrigerant to this constant quantity of water [7].

3.3 Results and discussion

System's performance was studied for various bubble pump heat power that varied from the minimal level maintaining flow through the machine, 170 W, to 350 W, corresponding to a driving temperature variation from 120 to 150°C.

3.3.1 Effect of the heat input on the bubble pump and machine's operating temperatures

The temperature evolution of the

- vapor/liquid mixture in the bubble pump,
- pumped solution,
- vapor at the condenser inlet and
- evaporator

with time for various bubble pump heat inputs (Q_{bp} = 170, 216, 260, 328 and 350W) was monitored and observed. Figure 2 depicts the variation of these operating temperatures with time for Q_{bp} = 260W. It corresponds to the bench heat input from witch the bubble pump temperature oscillations vanish. In fact, one can observe that the bubble pump temperature increases sharply to reach a maximum of 140 °C after approximately 15 min, drops sharply to 90°C (start up of the pumping action) and stabilizes thereafter at about 138°C. The pumped solution and the inlet condenser temperatures stabilize respectively at 100°C and 75°C. The same figure reveals also that the evaporator temperature first remains constant, starts dropping as soon as the bubble pump begins working and stabilizes at -10°C.

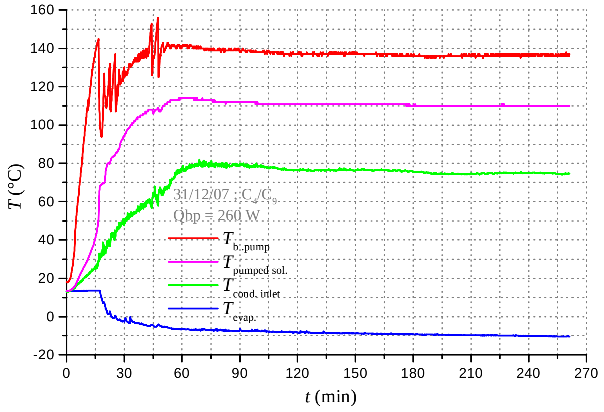

Figure2 : Variation of operating temperature with time for Q_{bp} = 260W

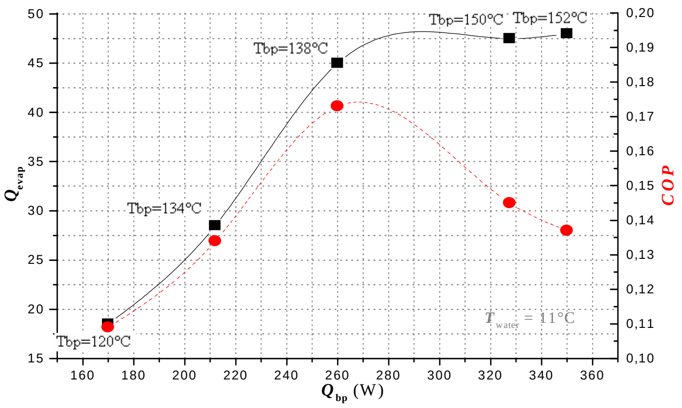

Figure3 : Variation of the machine’s performance vs. heat input

3.3.2 Effect of the heat input on the machine’s performance

Figure 3 illustrates the variation of the cooling capacity and the *COP* of the machine vs. the heat input. One can notice that the cooling capacity and the *COP* rise sharply and simultaneously while the heat input is increased from 170 to 270 W, which can be explained by the starting-up of the bubble pump. The cooling capacity reaches a maximum value of approximately 47W and the optimal *COP* is 0.175, corresponding to a heat input of Q_{bp} = 270W and at temperature near of 140°C. Heating the solution further is useless [7]; in fact the extra heat added is rejected as waste heat to the cooling air. Therefore the *COP* beyond this limit gets lower.

4. THERMODYNAMIC MODEL AND SIMULATION

The analytical model of an air cooled diffusion-absorption machine operating with C_4H_{10}/C_9H_{20} as working fluid and hydrogen as pressure equalizing inert gas is developed under the following conditions and assumptions (Eq. 1, 2, 3, 4 and 5). The basic values of the independent data are given in table 1.

$$T_{13} = T_{11} - 2 \quad (1)$$
$$T_{14} = T_{13} - 2 \quad (2)$$
$$T_{cd} = T_{abs} = T_2 - SRC = T_9 - SRA \quad (3)$$
$$T_9 = T_2 = T_{ref} + \mathrm{Pinch}_{cd} \quad (4)$$
$$T_{12} = T_9 + \mathrm{Pinch}_{SHX} \quad (5)$$

Table 1 : Independent basic data of the cycle simulation

Data	**Value**	**Data**	**Value**
Refrigeration capacity, $\dot{Q}_{Evap}$	1 kW	Absorber efficiency, Eff_{Abs}	5
Driving temperature, $T_{max}= T_{11}$ (°C)	105 – 205	*SRC* and *SRA* (°C)	1
Cooling medium temperature, $T_{air}=T_{ref}$ (°C)	25 – 35	$\mathrm{Pinch}_{cd} = \mathrm{Pinch}_{abs}$	10
Evaporator outlet temperature, T_6 (°C)	0.5 – 1	Pinch_{SHX} (°C)	7

Mass and energy balances governing the various machine’s components, illustrated here in the case of the absorber plus reservoir (Eq. 6, 7, 8 and 9), are formulated:

$$\dot{m}_{6(H2)} = \dot{m}_{7(H2)} \quad (6)$$
$$\dot{m}_{6(C4)} + \dot{m}_{12} = \dot{m}_{7(C4)} + \dot{m}_9 \quad (7)$$
$$\dot{m}_{6(C4)} + \xi_w \dot{m}_{12} = \dot{m}_{7(C4)} + \xi_r \dot{m}_9 \quad (8)$$

$$\dot{Q}_{Abs} + \dot{m}_9.H_9 + \dot{m}_{7(C4)}.H_{7(C4)} + \dot{m}_{7(H2)}.H_{7(H2)} = \dot{m}_{12}.H_{12} + \dot{m}_{6(C4)}.H_{6(C4)} + \dot{m}_{6(H2)}.H_{6(H2)} \quad (9)$$

The evaporation process is taking place between two refrigerant partial pressures defined as follows (Eq. 10 and 11):

$$P_{PartMax} = \frac{\dot{n}^g{}_{6(C4)}}{\dot{n}^g{}_{6(C4)} + \dot{n}_{6(H2)}} P_{Syst} \quad (10)$$

$$P_{PartMin} = \frac{\dot{n}^g{}_{7(C4)}}{\dot{n}^g{}_{7(C4)} + \dot{n}_{7(H2)}} P_{Syst} \quad (11)$$

4.2 Cycle performance

Diffusion-absorption refrigerator uses only the heat power to drive the cycle. The coefficient of performance, *COP*, is then given by Eq. 12:

$$COP = \frac{\dot{Q}_{evap}}{\dot{Q}_{gen}} \quad \text{with} \quad \dot{Q}_{gen} = \dot{Q}_{pb} + \dot{Q}_B \quad (12)$$

5. SIMULATION RESULTS AND DISCUSSION

The Oldham diagram of the C_4H_{10}/C_9H_{20} mixture is constructed using the basic *P-T-x–y* correlations for the liquid and the vapor phases. The simulation model is composed of a large set of non-linear equations (mass and energy balances and thermodynamic properties equations). CONLES algorithm [8] incorporated in a FORTRAN coded program is used to solve simultaneously this large set of non-linear equations. The fluid thermodynamic properties are calculated in a subroutine [9].
The thermodynamic cycle of the DAR is represented on the C_4/C_9 Oldham diagram (Figure 4) for a driving temperature T_{max} = 140°C and an air cooling temperature of T_{air} = 30°C. The transformation (4)-(6) corresponds to the n-butane evaporation process inside the evaporator.

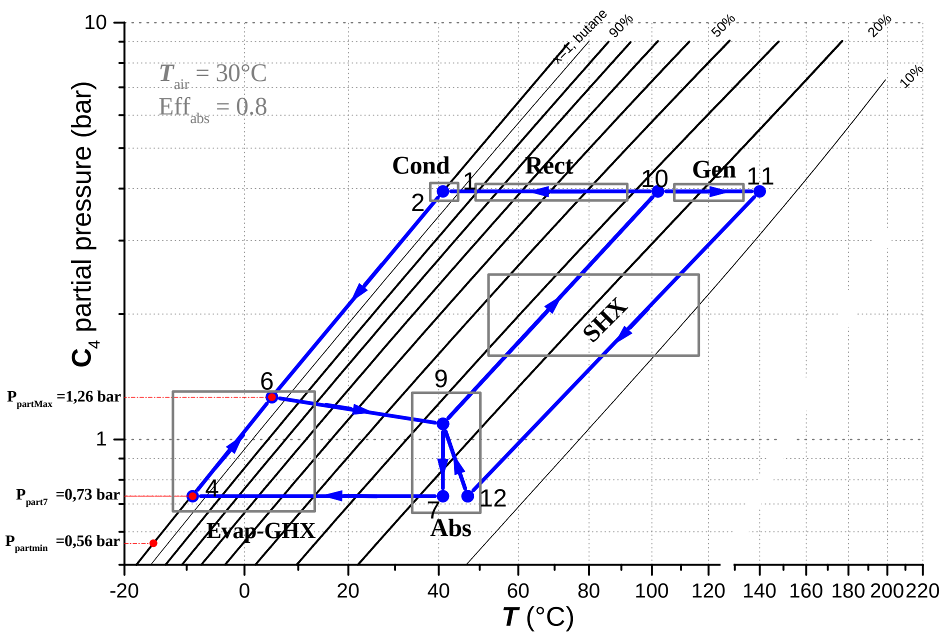

Figure 4 : DAR thermodynamic cycle processes on the C_4/C_9 OLDHAM diagram

Figure 5 shows the *COP* vs. air cooling temperature for different driving temperature values. One can see that the *COP* depends strongly on the cooling medium temperatures as well as the driving temperature (i.e. the heat supplied to the generator). Its reveals that for low cooling medium temperature (ex. 25°C) it is not necessary to increase the driving temperature beyond 120°C. But for higher cooling medium temperature (ex. 30°C and higher) it becomes necessary to increase the driving heat temperature to 140°C and more to improve the machine's performance.

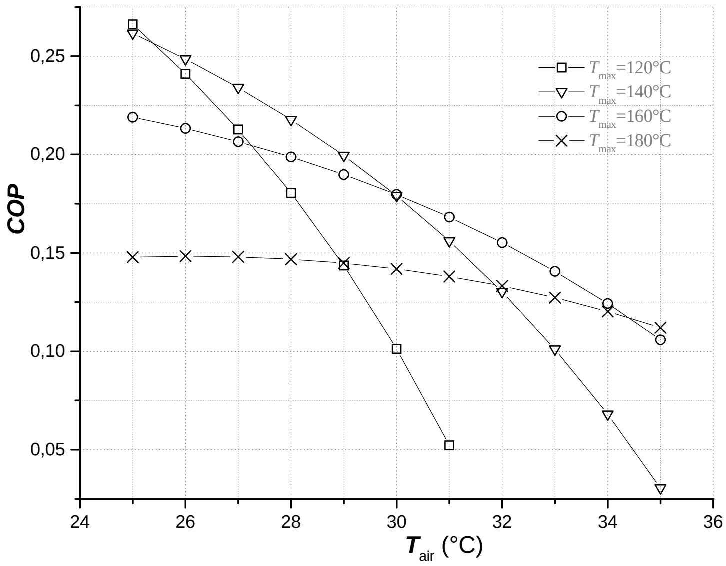

Figure 5 : COP variation vs. cooling temperature for different driving temperature values

6. CONCLUSION

A low capacity (30-40 W) thermal driving diffusion absorption machine based on light hydrocarbons is designed, fabricated and tested for various power inputs. Results show that the bubble pump temperature variation, as well as the operating temperatures variations, is strongly sensitive to the power inputs. The *COP* of the machine has reached a maximum of 0.175 provide by a power inlet Q_{bp}= 275W and a driving temperature near of 140°C. A thermodynamic model for a similar machine is developed. The cycle performances were analyzed parametrically by computer simulation. Results show that the *COP* depends strongly on the driving and the cooling medium temperatures.

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