ISSN 1313 - 8820
Volume 2, Number 3
September 2010

AGRICULTURAL SCIENCE AND TECHNOLOGY

2010

An International Journal Published by Faculty of Agriculture,
Trakia University, Stara Zagora, Bulgaria

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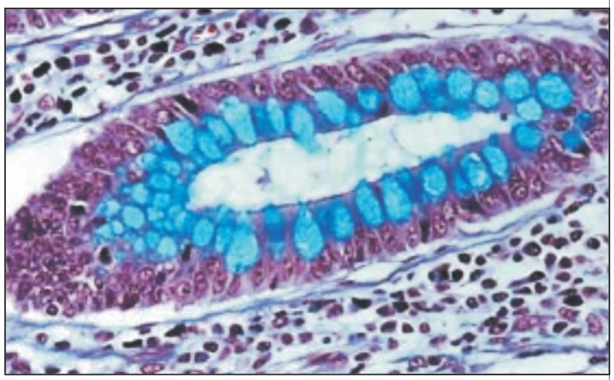

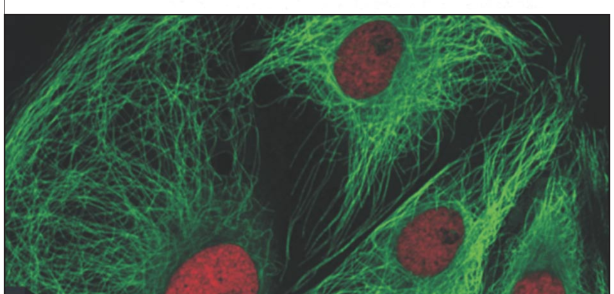

AGRICULTURAL SCIENCE AND TECHNOLOGY, VOL. 2, No 3, pp 163 - 166, 2010

Theoretical study of absorption refrigeration system thermally activated by a vacuum pipe solar collector

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Abstract. *A theoretical study has been carried out of a conceptual absorption refrigeration system for air conditioning that is energy activated by a solar collector field. The dynamics of the heat ratio describing the efficiency of the unit at various coolant temperatures to the generator has been studied at various temperature condensation conditions. Mean temperature ranges are determined for the effective operation of the generator. Definitely temperature level of condensation of refrigeration unit, which over the effectiveness of the refrigeration unit sharply reduced. The proposed model is cooling purposes in pig and poultry farming.*

Keywords: absorption refrigeration system, solar collector thermal energy.

Abbreviations: RV_1 – reducing valve before evaporator, RV_2 – reducing valve generator – absorber, P_1 – circulation pump – heat carrier pump from the collector field, P_2 – centrifugal pump for feeding the binary mixture from absorber to generator, P_3 – circulation pump to TA, Q_h – energy of heat carrier from collector field with temperature T_h, °K transmitted to the generator, Q_k – condensation heat with temperature T_k, °K taken by the environment, Q_o – heat taken by the evaporator medium with temperature T_o,°K, Q_a – absorption temperature emitted in the environment with temperature T_a,°K, Q_h – heat flow from the additional heat source (if necessary).

Introduction

The steady tendency for increase of the average annual and monthly temperature necessitates redesign of the energy approach in the sphere of household, public and production air-conditioning. The used cold steam compressor chilling systems have high condensation effect but they depend on power and their maximum electrical consumption is during the peak electrical consumption periods.

A principal alternative of compressor chiller systems is the use of absorption refrigeration units in which power supply of the „direct and reverse" process of Carnot is at the expense of an external heat source providing phase conversion of the operation binary mixture. The advantages of absorption thermal pumps compared to the compressor ones widely used nowadays can be referred to three main aspects:

1. Total lack of mechanically driven operation components except for centrifugal water pump (Todorov, 1977). This means longer operational life, lack of noise pollution, etc.

2. Energy independence from electricity distribution subjects and options for use of low-potential (low-temperature) heat flows (40-200C) (Website http//www.sbras.nsc.ru/dvlp/eng/pdf/178pdf).

3. Absorption refrigeration machines are ecologically friendly since binary mixtures do not include chlorine and fluorine-containing substances. Apart from that «heat drive» of systems can be by means of waste heat or heat from alternative energy sources – solar energy, geothermal water, biogas, etc. (Guozhen, 2006; Kaynakli and Kilic, 2007; Kececiler et.al., 2000).

In view of the above advantages, the US government has started the Solar thermal cooling program (development of solar air-conditioner) and it adopts a law on energy independence and security, regulating the implementation if that program, within the 2008-2012 period (http://en.Wikipedia.otg/wiki/Absorption-refrigerator).

Taking into consideration the above tendencies, our team aimed at conducting a theoretical study on opportunities for implementing an absorption refrigeration unit, heat driven by low temperature heat from a solar collector. Conceptually the development is referred to air-conditioning issues in animal husbandry. In several subfields (pig and poultry breeding) the lack of cooling in summer is risky for the performance and reproductive parameters of animals (Georgiev, Peychev, 2002). Moreover, from a construction and architectural point of view animal farms are extremely suitable for establishing collector fields and production of hot water from solar radiation. In animal production there is a latent energy potential in the form of waste heat flows (Peychev and Georgiev, 2006) and opportunities for biogas production, which in turn necessitates the need of reconsidering a program for implementing absorption air-conditioning chilling systems and hot water production on farms.

Material and methods

The analysis in the present article results from the theoretical model on Figure 1, describing refrigeration unit with continuous mode of operation. The operation binary mixture is water solution lithium bromide ($LiBr/H_2O$) with concentration 50-60%.

The efficiency of the absorption refrigeration unit is defined by the thermal coefficient , expressing the correlation of resulting chill production and the heat utilized in the „direct" generator process (Todorov, 1977).

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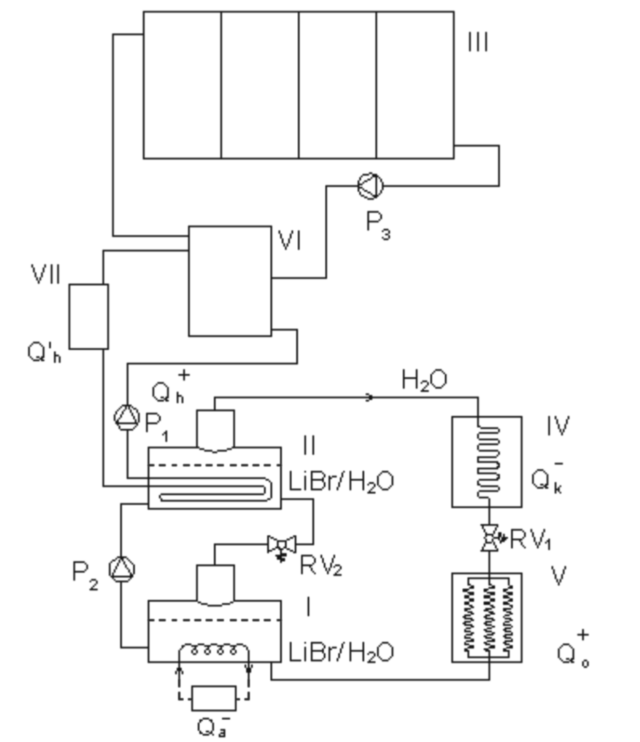

Figure 1. Principle diagram of absorption refrigeration unit supplied with thermal energy from collector field. I- absorber, II - generator, III - collector field, IV - condenser, V - evaporator, VI - thermal accumulator (TA), VII - additional heat source.

$$\zeta_{th} = \frac{T_h - T_k}{T_h} \cdot \frac{T_o}{Tk - To} \qquad (1)$$

where:

ζ_{th} – theoretical thermal coefficient value,

T_h – hear carrier absolute temperature, °K,

$T_к$ – condensation (ambient) absolute temperature, °K,

T_o – cooling area absolute temperature, °K.

In actual operation conditions all irreversible losses in the «direct and reverse» circular process of the absorption unit are accounted for by the actual thermal coefficient $_d$:

$$\zeta_d = \eta . \zeta_{th} \qquad (2)$$

where:

ζ_d – actual thermal coefficient value,

η – degree of reversibility of the actual circular process compared to the theoretical one, η = 0,70÷0,85.

For the purpose of this research we have adopted value = 0,80. The analysis has been carried out within the framework of the following temperature intervals:

T_h = 70, 75, 80, 85, 90 °C

$T_к$ = 25, 30, 35, 40, 45 °C

T_o = 5, 10, 15, 20, 25 °C

Graphic interpretation of the dynamics of the thermal coefficient ζ_d has been implemented according to the following functions

$$\zeta_d = f(T_h),\ T_к, T_o – \text{parameters} \qquad (3)$$

$$\zeta_d = f(T_к),\ T_h, T_o – \text{parameters} \qquad (4)$$

The relative variation of the thermal coefficient $\delta\zeta$ for each of the fixed values of the respective argument of functions (Guozhen, 2006; Kaynakli and Kilic, 2007) has been defined according to the expression:

$$\delta\zeta = \frac{\zeta_\iota - \zeta_1}{\zeta_1} . 100,\%$$

where:

$\delta\zeta$ - relative variation of the thermal coefficient for the "i-th" value of the argument (T_h or $T_к$),

ζ_ι - corresponding value of the thermal coefficient for the "i-th" value of the argument (T_h or $T_к$),

ζ_1 - value of the thermal coefficient for the first (initial) value of the respective argument within the accepted temperature interval.

Results and discussion

At the time of active solar radiation collector field III generates thermal energy by heating an intermediary heat carrier (water or water solution of ethylene glycol). By means of the circulation pump P_3 and the accumulating vessel VI a reserve of heat energy is provided. Circulation pump P_1 drives the heat carrier in the accumulating vessel through the generator heat exchanger II and vice versa. At the expense of the conducted heat Q_h with heat level T_h the binary mixture in the generator is activated and the heat carrier (evaporating) is released from the sorbent. At low temperature levels of the flow Q_h an additional heat source VII is activated (for example gas burner), providing a "stand-by" thermal flow Q_h. The released refrigeration agent (from the binary mixture) goes from generator II into condenser IV of the refrigerator unit.

Along with that the depleted binary mixture from generator II flows into absorber I through the reducing valve RV_2. Through the heat exchange surface of condenser IV, the heat carrier emits heat to the environment at the expense of Q_k flow and its vapour condenses. Through the heat control valve RV_1 the liquefied heat carrier is transferred to the evaporator V. At the expense of the ambient heat Q_o the liquefied refrigeration agent starts boiling and evaporating and its enthalpy increases and the ambient temperature decreases (chilling effect). Superheated steam from evaporator V enter absorber I. The depleted binary mixture absorbs them under conditions of exothermal process involving emission of the heat flow Q_a. The "recycled" binary mixture (with restored output concentration) is sucked by the pump P_2 and enters generator II under pressure, afterwards the described processes are repeated.

The idea about the adopted method is the use of solar energy as main source of "thermal drive" of the absorption unit. In this respect the temperature level of the Q_h flow fed to the generator of the refrigeration unit will be crucial for the operation heat exchange processes. Heat quantity and temperature level needed for splitting the binary mixture are indirectly related to the temperature levels of condensation of the refrigeration agent ($T_к$). Table 1 presents theoretically the specific values of the actual thermal coefficient ζ_d and its dynamics (its relative variation) $\delta\zeta_d$ under different temperature conditions of generator (T_h) and condensation environment ($T_к$).

Table 1. Values of actual thermal coefficient ($_d$) and its relative variation ($_d$) under different temperature levels of condensation ($T_к$) and temperature of heat carrier in generator (T_h).

Temperature of heat carrier in generator T_h, °C/°K *	Temperature levels of condensation $T_к$, °C/°K									
	25/298		30/303		35/308		40/313		45/318	
	ζ δ_ζ,%	δ_ζ,%	ζ δ_ζ,%	δ_ζ,%	ζ δ_ζ,%	δ_ζ,%	ζ δ_ζ,%	δ_ζ,%	ζ δ_ζ,%	δ_ζ,%
70/343	3,02	-	1,79	40,70	1,18	60,90	0,81	73,20	0,56	81,50
	-	-	-	-	-	-	-	-	-	-
75/348	3,31	-	1,97	40,50	1,32	60,10	0,93	71,90	0,66	80,10
	9,60	-	10,10	-	11,90	-	14,80	-	17,90	-
80/353	3,59	-	2,17	39,60	1,47	59,10	1,04	71,00	0,76	78,80
	18,90	-	21,20	-	24,60	-	28,40	-	35,70	-
85/358	3,86	-	2,36	38,90	1,61	58,30	1,16	69,90	0,86	77,77
	27,80	-	31,80	-	36,40	-	43,20	-	53,60	-
90/363	4,13	-	2,54	38,50	1,75	57,60	1,27	69,20	0,95	77,00
	36,80	-	41,90	-	48,30	-	56,80	-	69,60	-

* The above values in the table are defined at accepted cooling temperature To=15°C (288°K).

Within the framework of a given temperature level of condensation the increase of the temperature of heat carrier to the generator results in increase of the thermal coefficient. Practically that means increase of the operation efficiency of the air-conditioning unit. With condensation temperature of 35-40°C (typical temperature conditions during the summer months in our country) variation in the temperature of the heat carrier to the generator from 70 to 90°C shall result in increase of the thermal coefficient by more than 50%. That means at the stage of design and development of the solar unit (collector field III on Figure 1) it is necessary to look for the constructive option that guarantees temperature of the heat carrier within the range of 85-90°C (358-363°K).

Interesting is also the trend of the heat coefficient at the same temperature of the heat carrier to the generator for various temperature conditions for condensation. For example, with heat carrier temperature of 85°C (385°K) (typical temperature mode for pipe vacuum and flat absorption solar collectors with selective coating) the variation of temperature levels of condensation from 25 to 45°C (298÷318°K) results in decrease of the heat coefficient from 3.86 to 0.86, which in relative dynamics is within the range of 31-77%. At temperature levels of condensation around 45°C heat coefficient values are lower than 1.00 for all temperature options (T_h) of the heat carrier. Under the specified condensation conditions the trend in the heat coefficient dynamics reveals decrease in its values up to 82%. The above finding reveals that the operation efficiency of absorption air-conditioning units is strongly influenced by condensation temperature. Practically that means dimensioning and choice of condensers that guarantee suitable values of the heat coefficient under conditions of extremely high external

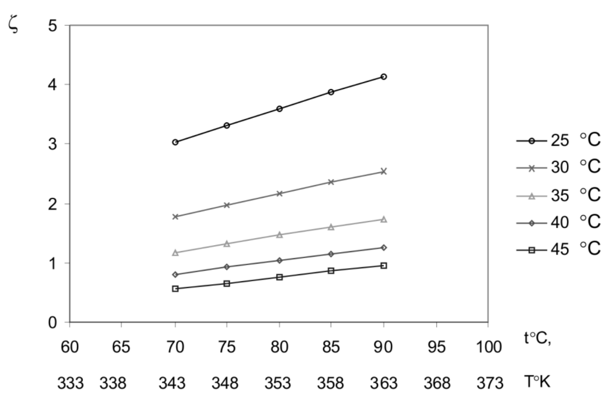

Figure 2. Correlation of the thermal coefficient of the absorption thermal unit and the temperature (T_h) of the heat carrier to the generator at various temperature levels of condensation.

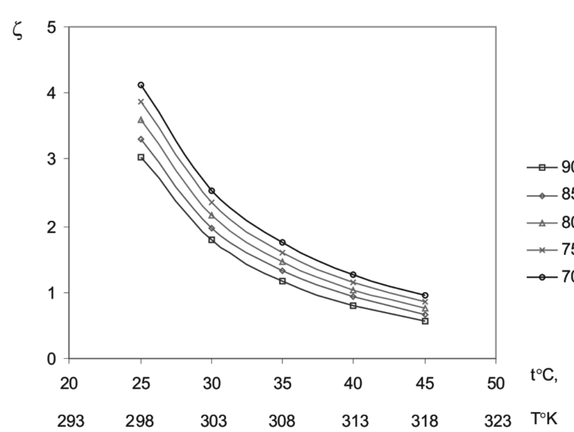

Figure 3. Correlation of the thermal coefficient of the absorption thermal unit and the condensation temperature at various temperature levels in the generator.

temperatures.

The relationship of the heat coefficient to generator temperature conditions and the condensation ones specified so far have been summarized and diagrammatically presented on the diagrams of Figures 2 and 3. The diagrams on Figure 2 interpret the relationship of the heat coefficient on the temperature of the heat carrier transferred to the generator of the air-conditioning unit. The increase in the temperature level of the heat carrier means increase in the values of the heat coefficient $_{d}$. The relationship is strictly linear and it has the same value for all temperature levels of condensation adopted in the analysis. It can be typically presented with the function y = a + bx.

The diagrams on Figure 3 describe the functional relationship of the heat coefficient on temperature conditions of condensation of the refrigeration agent. The rising temperature levels of condensation reduce the heat coefficient significantly. The reported correlation is identical for all five temperature options of the heat carrier to the generator in the air-conditioning unit. The profile of diagrams is definitely hyperbolic and these could be summarized by the function

$$y = \frac{1}{x}\ .$$

Conclusion

The heat coefficient describing the operation efficiency of the absorption refrigeration unit is strongly influenced by the temperature of the heat carrier transmitted to its generator. Within temperature range 70-90?C the dynamics of the heat coefficient rises over 50%. In design solutions it is necessary to produce options for solar collector units for production of heat energy with potential 85-90?C. The heat coefficient values depend on the temperature conditions of condensation of the refrigeration agent. Ambient temperature above 40?C reduces significantly the operation efficiency of absorption refrigeration units. The negative dynamics in the heat coefficient values for temperature levels of condensation around 45?C exceeds 80%. Practically that means dimensioning and selection of condensers that guarantee suitable values for the heat coefficient under conditions of extremely high external temperatures. Method of cooling using absorption refrigeration machine, heat-activated thermal solar module could be used in the field of livestock production to cool the premices.

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AGRICULTURAL SCIENCE AND TECHNOLOGY

Volume 2, Number 3
September 2010

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