Perspective the Use of Thermal Energy Lost From an Engine Cooling System to Run an Absorption Refrigerator for Vehicle Air Conditioning

Asst. Prof. Dr. Abdulwadood Salman Shihab,

Technical College of Basrah
E-mail: wadsal54@yahoo.com

Abstract:

This paper is addressed to investigate the possibility of using a waste heat to drive absorption refrigeration system as an alternative system for automobile air conditioning. LiBr-water absorption refrigerator is suggested for this application. A theoretical analysis of the system has been carried out to maximize its cooling potential. This study includes also an actual experimental test to measure the available waste heat that is rejected from water cooling system of a 6-cylinder petrol engine fitted to Toyota Land Cruiser vehicle. The test was performed directly on the vehicle using out-city high way roads. It is found that the wasted energy from the engine cooling system is ranged between 10 kW for idling engine speed and 68 kW when loading. The engine cooling water temperature is ranged between 71 ℃ and 84 ℃. The reject heat available in the cooling water system is measured and found to be adequate for producing cooling up to 34 kW, but the low level of the water temperature did not allow the refrigeration system to produce continuous cooling corresponding to engine and vehicle speeds at a refrigeration temperature of 5 to 10 ℃, especially at hot weather condition.

Keywords: Alternative Vehicle Air-conditioning, Absorption Refrigerator.

المستخلص

يتناول هذا البحث التحقق من إمكانية استخدام الطاقة الحرارية المفقودة من محركات المركبات لاستخدامها كمصدر لتشغيل منظومة تبريد امتصاصية لأغراض التكييف كنظام بديل. اقترحت منظومة التبريد الامتصاصية ماء –بروميد الليثيوم لهذا التطبيق. تم انجاز تحليل نظري للمنظومة للحصول على أفضل أداء. احتوت الدراسة أيضا على تنفيذ اختبار عملي لقياس الطاقة الحرارية الفعلية المتاحة والمطروحة من منظومة تبريد محرك بنزين ذو ستة اسطوانات لسيارة نوع تويوتا لاندكروز حيث اجري الاختبار العملي على السيارة مباشرة واختيرت الطرق الخارجية لهذا الغرض. لقد وجد أن الطاقة المطروحة من منظومة تبريد محرك السيارة تتراوح بين ١٠ كيلووات عندما يعمل المحرك بدون تحميل إلى ٦٨ كيلووات مع التحميل. تراوحت درجة حرارة ماء تبريد محرك السيارة بين ٧١ إلى ٨٤ درجة مئوية. تم حسا ب الطاقة الحرارية المفقودة من مشع منظومة تبريد المحرك وظهرت إنها وافية لإنتاج التبريد بما يعادل ٣٤ كيلووات، ولكن المستوى المنخفض لدرجة حرارة الماء قيدت عملية إنتاج التبريد بصورة مستمرة بدرجات حرارة تبريد ٥ إلى ٨ درجة مئوية وخاصة في الأجواء الحارة.

1- Introduction:

The air conditioning systems currently utilized in automobiles are the vapor compression systems. This type of system has many disadvantages: the refrigerant used is not environmentally friendly, the compressor is in competition with the engine coolant system, and the compressor uses a significant portion of the engine power. To date, almost all car air-conditioning systems are charged with R-134a. However, alternatives with lower global warming impact than R-134a are desirable.

It is a well-known fact that a large amount of heat energy associated with the automobiles engine cooling system and with the exhaust gases are wasted. A rough energy balance of the available energy in the combustion of fuel in a motor car engine shows that one third is converted into shaft work, two third is lost to atmosphere [1].

A waste heat driven absorption refrigeration system is one alternative to the current systems. The absorption refrigeration system (ARS) uses solutions for the absorbent-refrigerant pair that do not harm the environment. Recently, there has been increasing interest in the industrial and domestic use of the ARSs for meeting cooling and air conditioning demands as alternatives. The most widely used refrigerant and absorbent combinations in ARSs have been ammonia–water and lithium bromide-water. The lithium bromide-water pair is available for air-conditioning and chilling applications (over 4℃, because of the crystallization of water). Ammonia-water is used for cooling and low-temperature freezing applications (below 0℃) [2].

The main drawback of absorption systems for cars is the heating-up time needed to create the temperature level necessary in the boiler to produce refrigerant vapor. This system therefore needs a refrigerant storage as well as a minimum operating time of about half an hour before efficient cooling operation [3].

Akerman, 1969 [4], thermally analyzed three cycles of an automotive absorption air conditioning system that would use engine-rejected heat. The three cycles were based on using water-lithium bromide, ammonia-water and Refrigerant 22-Di methyl form amid of tetra ethylene glycol (R22-DMFTEG) as a refrigerant-absorbent pairs. He concluded that; due to the number of additional parts required, the large size of the required heat transfer surface of the heat rejection devices and the large amount of heat energy required, the absorption refrigeration system is not suitable for use in automotive air conditioning.

Wang, 1997 [5], showed that even for a relative small car-engine, such as for the Nissan1400, 15 kW of heat energy can be utilized from the exhaust gases. This heat is enough to power an aqua-ammonia absorption system to produce a refrigeration capacity of 5 kW.

The studies by Horuz (1998, 1999) [6], [7] utilized waste heat from the vehicle engine exhaust gas as the sole driving mechanism for his proposed absorption refrigeration systems, he demonstrated that a vapor absorption refrigeration system running on a diesel engine is indeed possible and he concluded that engine exhaust gas heat would provide sufficient power to drive his proposed systems during normal cruise conditions (engine speeds around 2000 rpm), but would not provide sufficient capacity at rest (idle) or at slow-moving traffic conditions. Horuz pointed out the limitations imposed by exhaust gas back pressure on the engine and the effects of corrosive exhaust gases that could condense within exhaust system components as a result of extracting heat from the exhaust gases.

Boatto et al. 2000 [8], conducted extensive measurements on the exhaust system of a 2.0-liter, four-cylinder, spark-ignition engine of mid-sized passenger cars. They concluded that an automotive absorption refrigeration system driven by exhaust heat recovery allows for considerable power recovery and seems feasible as long as provisions are made to store liquid refrigerant (water) for use during transient startups and when temporary exhaust gas power deficits occur.

Shannon, 2005 [9], concluded that the absorption refrigeration system with LiBr-Water mixture is a feasible alternative to the traditional vapor compression system for automotive case, a typical 3-liter, 4 stroke carbureted engine was used. In order for the system to run at the most favorable conditions, the outside air temperature needs to be below 38°C which can be a problem in places with extremely high temperatures. The minimum generator temperature should be around 93°C. Ideally, the system will work best if the motor is running at 115.5°C. Also the condenser temperature must be below 55.5°C. The absorption refrigeration system would work for temperatures out of the above mentioned ranges, but the efficiency drops off rather drastically.

Vicatos, 2008 [10], used energy from the exhaust gas of an internal combustion engine to power an absorption refrigeration system to air-condition an ordinary passenger car. The theoretical design is verified by a unit that is tested under both laboratory and road-test conditions. The unit was installed in a Nissan 1400 truck and the results indicated a successful prototype and encouraging prospects for future development. The low coefficient of performance (COP) value is an indication that improvements to the cycle are necessary.

Apart from the limitations imposed by exhaust gas, the present work proposes the reject heat that available in the vehicle engine cooling water system to be used as the driving thermal energy to operate LiBr-Water absorption refrigeration system for the purpose of air-conditioning the passenger compartment.

2- Theoretical analysis

An absorption system is a heat-operated refrigerator. It consists four basic components; an evaporator and an absorber which are located on the low pressure side of the system, and a generator and a condenser which are located on the high pressure side of the system.

The absorption cycle (Figure 1), can be divided into three distinct parts. *The water side* is a vapor/compression cycle through which the compressed water vapor out of the generator-separator assembly is cooled and condensed in the condenser. It is then expanded through the expansion valve and evaporated in the evaporator, producing cooling in the passenger compartment. Upon exiting the evaporator, the water vapor enters in to *the LiBr-Water side* of the device. When the water vapor comes in contact with the rich solution (rich in LiBr) in the absorber, the vapor is rapidly dissolved, or absorbed into the fluid producing heat that must be dissipated out of the absorber. The produced weak solution at the exit of the absorber then is driven by a small liquid pump toward the *desorption chamber* (vapor generator). Between the absorption and desorption chambers heat is exchanged to raise the temperature of the weak solution via preheating heat exchanger. In the desorption chamber, due to heat addition, the water is desorbed from the solution and librated as a vapor at a higher temperature. The rich solution out of the vapor generator is then cooled at the preheating heat exchanger and then back to its original temperature in the absorber.

The schematic diagram of the proposed LiBr-Water absorption refrigeration system is shown in Fig. (1). The corresponding points numbers of this figure are also presented on the pressure-temperature-concentration (P-T-ζ) diagram of LiBr-Water solution Fig.(2), and on the pressure-enthalpy (P-h) diagram of pure water as in Fig.(3).

The LiBr-Water solution properties and that of pure water at different pressures, temperatures and concentration can be obtained from property data and their correlations which are available in [11].

The proposed system is powered by the heat energy rejected from a variable speed internal combustion engine (the hot water of the engine cooling system). The amount of the rejected heat through the vehicle radiator Q_{rad} is;

$$Q_{rad} = \dot{m}_w \, C_p \, (T_{w,in} - T_{w,out}) \tag{1}$$

$T_{w,in}$, $T_{w,out}$ Are the engines cooling water temperatures at the inlet and outlet of the radiator,
$\dot{m}_w$, Engine cooling water flow rate.

Referring to Fig. (1) to Fig.(3), the steady flow analysis is carried out by applying the mass and energy balance across the evaporator and the generator with the following assumptions:

1- Perfectly insulated components.
2- No superheating in the evaporator, nor sub-cooling in the condenser.
3- The circulating pump of the weak solution is assumed to be of a variable capacity, then the weak mixture flow rate ($\dot{m}_{ws}$) that is pumped to the vapor generator is:

$$\dot{m}_{ws} = G * N \tag{2}$$

G = variable chosen based on the temperature level through the generator such that it satisfies the generator energy balance, equations (5) and (11).
N = Engine speed (rpm).

4- Assuming that the generator temperature ($T_g = T_4 = T_7$) is 5°C less than the engine hot water temperature (the minimum temperature difference between two fluids exchanging heat energy or "pinch point") [12];

$$T_4 = T_{w,in} - 5 \tag{3}$$

And the weak solution can be preheated through the preheating heat exchanger to a temperature (T_3) with a pinch point of 5°C (below the generator temperature) ;

$$T_3 = T_4 - 5 \tag{4}$$

5- It is assumed that the generator can absorbs at least 50% of the available radiated heat energy;

$$Q_g = 0.5 \, Q_{rad} \tag{5}$$

The effectiveness (∈) of the preheating heat exchanger is assumed to be 0.5.

When the heat exchanger cost is an important consideration, most heat exchangers are designed in the approximate value of $\in \leq 60\%$, [13]

$$\in = \frac{T_g - T_5}{T_g - T_3} \tag{6}$$

Considering the control volume of the generator and applying mass balances, then the strong solution mass flow rate $\dot{m}_{ss}$ is;

$$\dot{m}_{ss} = \dot{m}_{ws} - \dot{m}_v \tag{7}$$

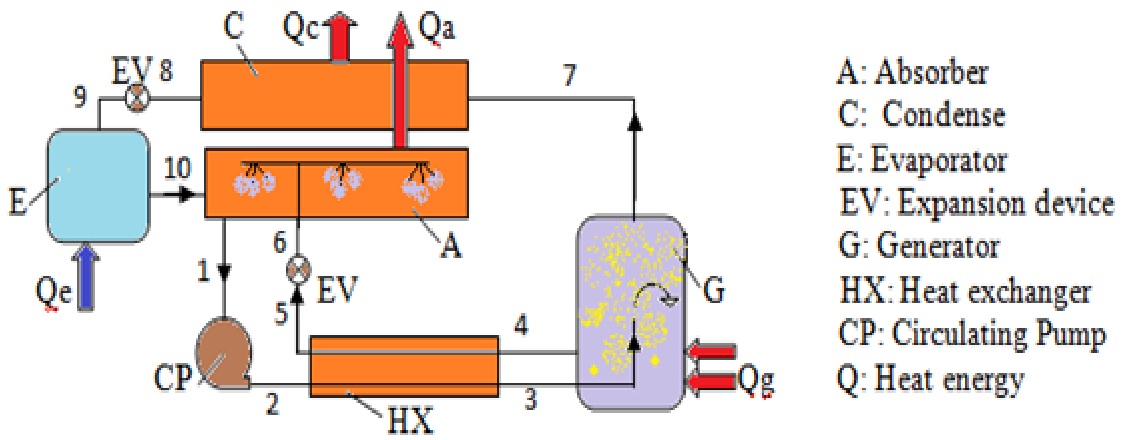

Fig. (1). The schematic diagram of the proposed LiBr-Water absorption refrigeration system

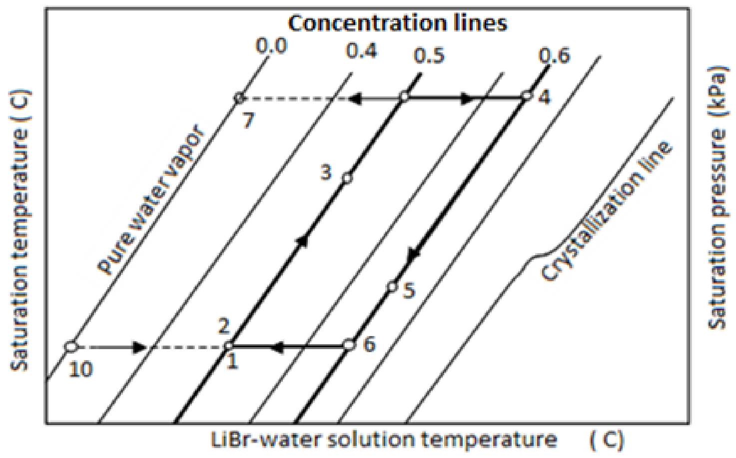

Fig. (2). Pressure-Temperature-Concentration diagram of the Proposed cycle (Duhring Chart)

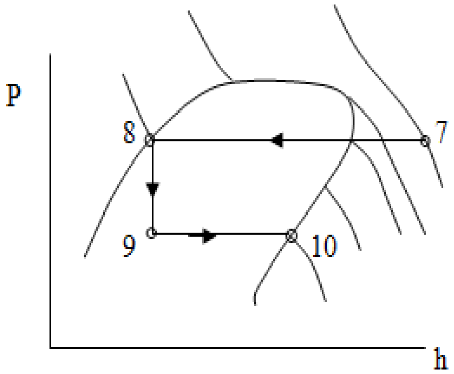

Fig. (3). the pressure-enthalpy (P-h) diagram of pure water

For the employed solution, the mass fraction (ξ) (LiBr concentration) is defined as the ratio of mass of anhydrous lithium bromide to the total mass of solution.

Applying mass balance for LiBr through the generator, the vapor mass flow rate $\dot{m}_v$ can be evaluated as;

$$\dot{m}_v = \dot{m}_{ws} \frac{\xi_{ss} - \xi_{ws}}{\xi_{ss}} \quad (8)$$

ξ_{ss}, ξ_{ws} are the strong and weak solution concentrations which are functions of temperature and pressure at the generator and the absorber respectively.

$$\xi_{ss} = \xi\,(T_g, P_c), \quad \xi_{ws} = \xi(T_a, P_e) \quad (9)$$

The dominated high pressure at the generator (P_g) is as that of the condensing pressure (P_c) and the dominated low pressure at the absorber (P_a) is as that of the evaporator pressure (P_e);

$$P_g = P_c, \quad P_a = P_e \quad (10)$$

Applying energy balance through the vapor generator, the gained heat energy (Q_g) by the solution in the generator is as follows;

$$Q_g = \dot{m}_{ss}\, h_4 + \dot{m}_v\, h_7 - \dot{m}_{ws}\, h_3 \quad (11)$$

The cooling capacity of the system is a given by;

$$Q_e = m_v (h_{10} - h_9) \quad (12)$$

Where h is the enthalpy at each corresponding point of the system cycle.

The coefficient of performance of the refrigeration cycle becomes:

$$COP = \frac{Q_e}{Q_g} \quad (13)$$

3- Experimental Test

This test is to measure the heat energy available within the engine cooling water system. This energy is rejected to the atmosphere via radiator. The water flow rate and its temperature vary considerably with the engine and vehicle speed.

Two sets of tests has to be done; one for the stationary vehicle (Idling speed), and the second for cruising condition (with engine road loading), at varying speed.

The tested vehicle is Toyota Land Cruiser car fitted with six cylinders petrol engine.

The heat energy radiated out of the circulating cooling water of the engine Q_{rad}, is due to the temperature difference between the cooling water temperatures at the exit from the engine block $T_{w,out}$(at the inlet to the radiator), and at the inlet to the engine block $T_{w,in}$(at the outlet from the radiator).

The parameters to be measured are therefore; engine speed, vehicle road speed, cooling water temperatures at the inlet and outlet of the engine block, cooling water mass flow rate and the ambient temperature. These parameters are measured as in the following:

a- Engine speed: The fitted engine speed indicator (Engine speedometer) which is an electrical measuring device indicates the engine rpm by making use of the electrical pulses occurring at the ignition coil. When the vehicle is on the road, the vehicle speed can also be found simply by direct reading on the vehicle speedometer.

b- Temperature measurements: Calibrated Chromel-Alumel thermocouple wires are used to measure the water temperatures.

c- Cooling water mass flow rate: The water circulating pump used to circulate the engine cooling water is belt driven. Therefore, its pumping capacity is a function of engine speed and is independent of the idling or road load conditions. A venturi-meter is used to calibrate the pump in the laboratory using a suitable water flow circuit. The pump is placed in its position in the vehicle while it is still belt driven by the engine.

4- Results and discussion

5-1 Experimental results

Fig. (4) Shows the relations of the cooling water temperatures at inlet and outlet of the engine block and their deference, with engine speed for idling speed. The average hot water temperature out of the engine block is 71 ℃ at 500 rpm engine speed and 88 ℃ at 1050 rpm. The obtained correlation function for temperature difference is given by:

$$\Delta T_{w,idl} = 0.0061 * N + 3.6268 \ ℃ \quad (14)$$

Fig. (5) Shows the same relations as in Fig. (4) But for the condition when the vehicle is on the road load (Cruising condition). The average hot water temperature out of the engine block is 77 ℃ at 1000 rpm engine speed (40 km/hr Vehicle speed) and 84 ℃ at 2000 rpm (80 km/hr Vehicle speed). The obtained correlation function for temperature difference is given by:

$$\Delta T_{w,cr} = 0.002 * N + 5.87 \ ℃ \quad (15)$$

Fig. (6) Shows the results of the engine cooling water flow rate ($\dot{m}_w$) at deferent engine speed (N). The relation appears to be fairly linear. The obtained correlation function for these readings is:

$$\dot{m}_w = 0.001 * N - 0.1243 \ kg/s \quad (16)$$

Fig. (7) Shows the calculated radiated heat energy (equation 1) at the vehicle radiator for both conditions, the idling and road load engine speeds. This amount of heat is ranged between minimum values of 10.5 kW at low idling speed to 68 kW at 2000 rpm cruising speed (80 km/hr).

5-2 Refrigeration system analysis results.

The present thermodynamic analytical model of LiBr-water absorption refrigeration system has been validated by comparing some of its results with the similar study that was done by Shanoon [9]. It can be seen from Table 1. that, the results of present model for reference data that were use by Shannon show the same general behavior for idling and road load speed.

Table 1. Comparing the result of the present model (A) with reference [9] (B) At Tc = 54℃, Te = 7.2℃, Ta = 37.8℃, for idling speed of 1000 RPM And road load speed of 2000 rpm.

	Speed, rpm.	Tg, ℃	Qe, kW	Qg, kW	Qc, kW	COP
A	1000	92.2	4.92	11.	5.326	0.447
	2000	92.2	13.08	27.	13.98	0.484
B	1000	93.3	4.8	10.8	5.23	0.444
	2000	93.3	13.1	27.1	14.12	0.483

The governing equations of the proposed refrigeration system are solved to obtain the refrigeration system behavior at idling and road speeds with different operating conditions. The operating variable that affect the system performance are the condenser, absorber and the evaporator temperatures. The generator temperature is limited and is corresponding to the energy source temperature. The condenser and absorber temperatures are assumed to be the same as both, the condenser and the absorber are of air cooled heat exchangers type. This temperature is ranged to be from 35 ℃ to 45℃. The evaporator temperature is ranged from 5℃ to 21℃.

Due to the limited temperature of the energy source, increasing the solution flow rate through the vapor generator restrains the vapor production, so that the solution flow rate should be controlled to ensure quite temperature level of the flowing solution in the generator to liberate vapor according to the dominated pressure. It is also assumed that only 50% of the emitted heat energy that described in equation (1) can be absorbed by the working solution at the generator. Hence, the working solution circulating pump for this purpose should be of variable capacity (equation 2), such that the value of G is chosen by iteration to conform the evaporation requirements in the generator.

Figures (8) to (13) show the refrigeration system cooling capacity and its coefficient of performance at different condensing an evaporating temperature when the vehicle engine operates with no road load (idling condition).

Fig. (8) Illustrates that at a condenser-absorber temperature of 35 ℃, cooling can be produced at any evaporating temperature above 5 ℃ even at low idling engine speed (500 rpm).

Most of the usual gasoline vehicles operate at 500 rpm engine speed or slightly more, but when using air conditioning system it needed to becomes more up to 1000 rpm.

Hence as shown in Fig.(8) an adequate cooling can be obtained (10-15 kW) with reasonably low evaporating temperature, 5 to 8 ℃, the corresponding coefficient of performance in relation to 0.86 is illustrated in Fig.(9), that is when the engine rotate at 800 to 1000 rpm and a condenser-absorber temperature is of 35 ℃. At higher evaporator temperature more cooling capacity with higher coefficient of performance can be obtained, that is because more vapor can be produced at higher evaporation temperature as the quantity $(\xi_{ss} - \xi_{ws})$ becomes more which is correspond to the amount of librated vapor (refrigerant).

Figs. (10) And (11) show how much the refrigeration performance is affected by the action of increasing the condenser-absorber temperature to 40℃.

Cooling can be produced at an evaporating temperature of 5 ℃ but at an engine speed more than 900 rpm, the effective cooling of about 6 kW is obtained at 970 rpm at which the coefficient of performance is 0.4.

To produce cooling at 8℃, the engine speed should be more than 850 rpm with an effective cooling capacity of 6 kW at 860 rpm with coefficient of performance of 0.45.

Further increase of the condenser-absorber temperature up to 45 ℃ lead to worse refrigeration performance as shown in Figs. (12) and (13). Cooling can never be produced at an evaporator temperature less than 14 ℃ even at an engine speed more than 1000 rpm.

It is evident that Tc and Ta have significant impact on the refrigeration system performance, that is due to the higher temperature level demands at the generator when increasing Tc, and the value of $(\xi_{ss} - \xi_{ws})$ becomes less at higher Ta.

Figures (14) to (18) show the refrigeration system cooling capacity and its coefficient of performance at different condensing-absorption and evaporating temperatures when the vehicle engine operates with road load (cruising condition).

Fig.(14) illustrates that at a condenser-absorber temperature of 35 ℃, cooling can be produced at any evaporating temperature, 5 ℃ and more even at low vehicle engine speed of 40 km/hr (500 rpm engine speed). An adequate cooling can be obtained (11-13 kW) with an evaporating temperature of 5-14 ℃, the corresponding coefficient of performance is in relation to 0.8 to 0.9 respectively as shown in Fig.(15).

The cooling capacity increases with vehicle speed due to an increase in the energy source temperature and so the vapor generation temperature. The cooling capacity decreases slightly with reducing the evaporator temperature.

The impact of reducing the evaporator temperature is obviously appointed on the coefficient of performance but this performance index interns improved with the vehicle speed. At higher evaporator temperature more cooling capacity with higher coefficient of performance is obtained, that is because of more amount of librated vapor (refrigerant) as $(\xi_{ss} - \xi_{ws})$ becomes more.

Figs. (16) And (17) show how much the refrigeration performance is affected by the action of increasing the condenser-absorber temperature to 40℃. Cooling can be produced at an evaporating temperature of 8 ℃ but with vehicle speed more than 68 km/hr (1700 rpm engine speed). To produce cooling at 10℃, the vehicle speed should be not less than 52 km/hr (1300 rpm engine speed). At 40℃, continuous cooling at all vehicle road load speed can't be obtained at less than 12 ℃ evaporating temperature which is reasonably still acceptable for air conditioning with high cooling capacity. At this temperature the cooling capacity is ranged from 7 kW at 40 km/hr vehicle speed to 33 kW at 80 km/hr vehicle speed. The corresponding coefficient of performance is 0.47 to 0.84 respectively.

Further increase of the condenser-absorber temperature up to 45 ℃ leads to too worse refrigeration performance as shown in Figs. (19) and (20). Continuous cooling at all vehicle speeds can never be produced at an evaporator temperature less than 21℃. The cooling capacity at this temperature is ranged between 5.5 kW with COP of 0.38 to 33.5 kW with COP of 0.86 at vehicle speed of 40 km/hr and 80 km/hr respectively.

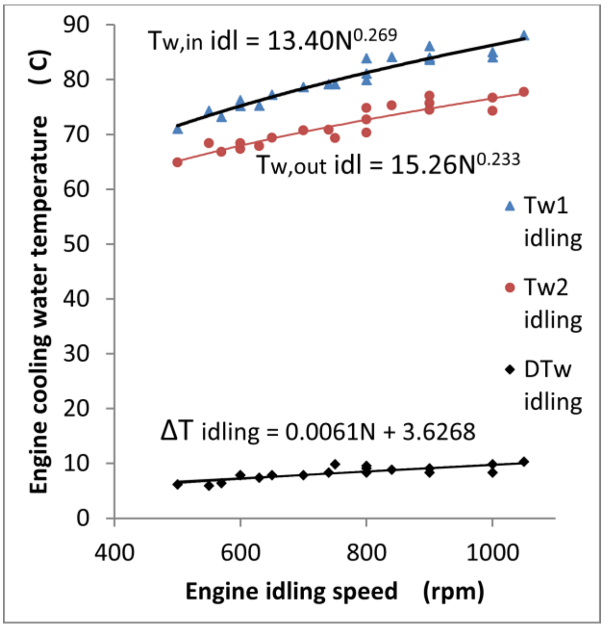

Fig. (4). The measured cooling water temperature at the inlet and outlet of the radiator verses engine idling speed.

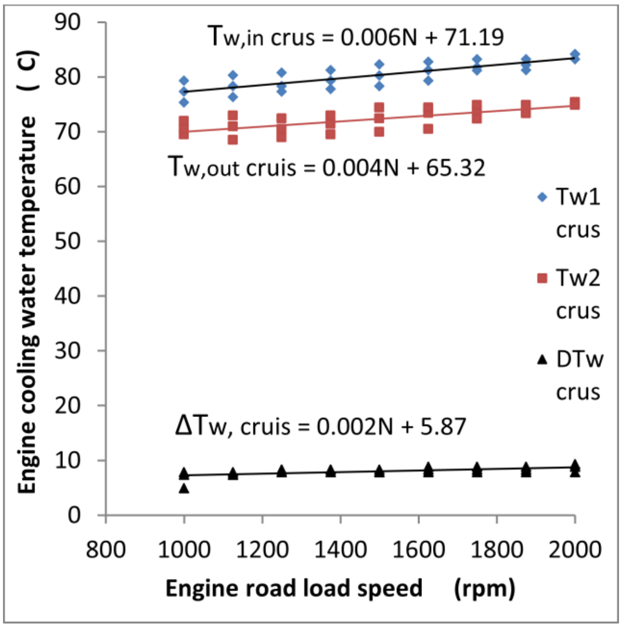

Fig. (5). The measured cooling water temperature at the inlet and outlet of the radiator verses engine road load speed.

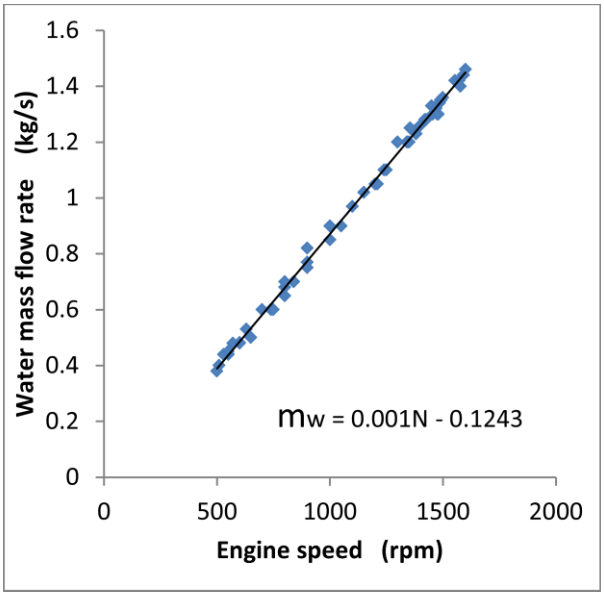

Fig. (6). The engine cooling water mass flow rate verses engine speed.

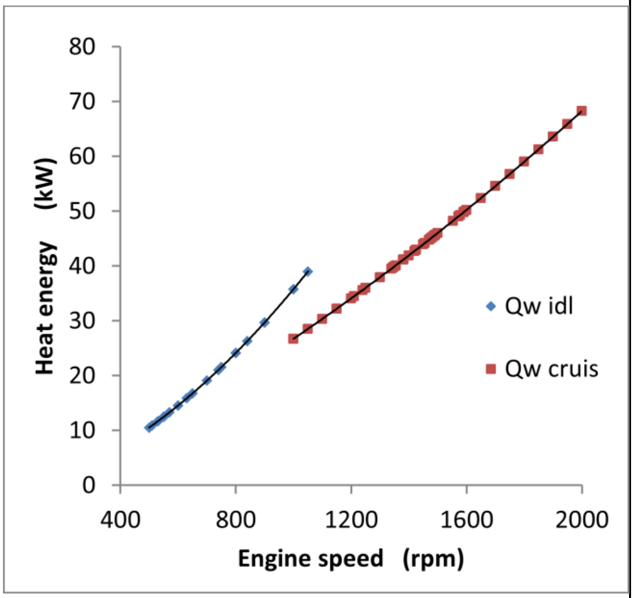

Fig. (7). The available heat energy that radiated from the engine cooling system to the surrounding for both idling and road load engine speed.

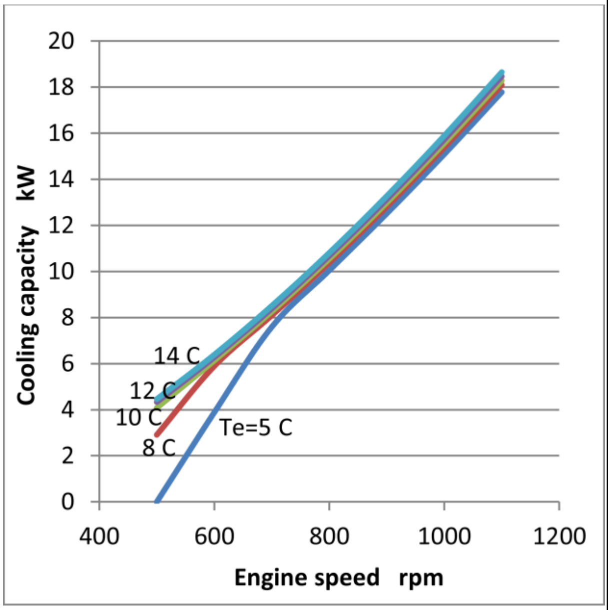

Fig. (8). Cooling capacity verses engine idling speed at Tc=Ta=35 C and different evaporating temperatures.

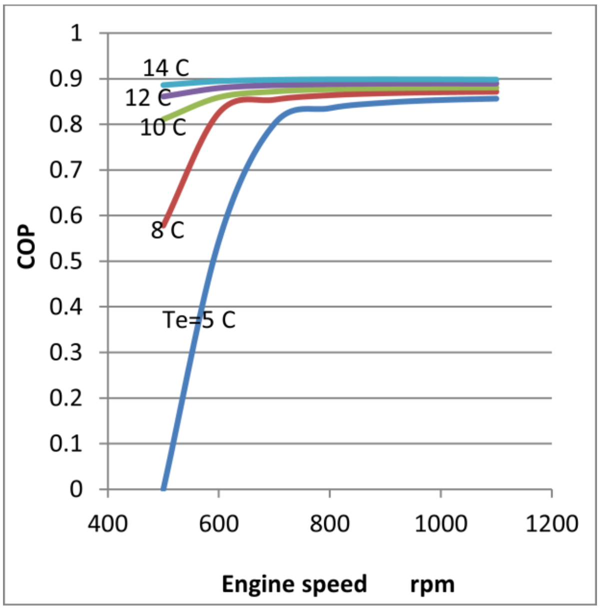

Fig. (9). Coefficient of performance verses engine idling speed at Tc=Ta=35 C and different evaporating temperatures.

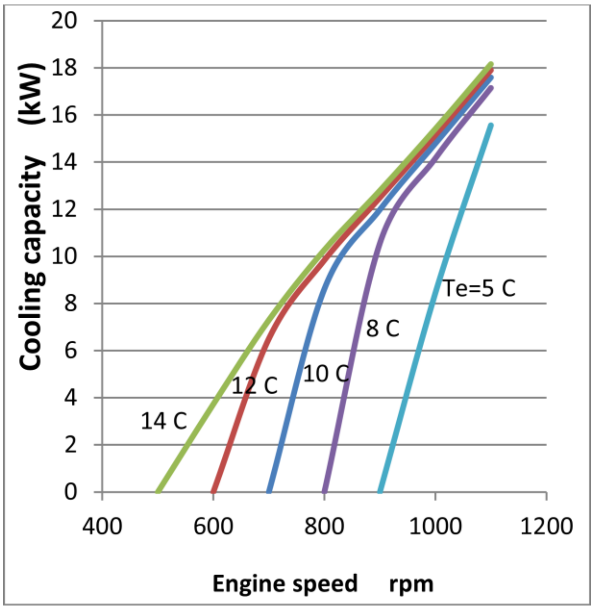

Fig. (10). Cooling capacity verses idling engine speed at Tc=Ta=40 C and different evaporating temperatures.

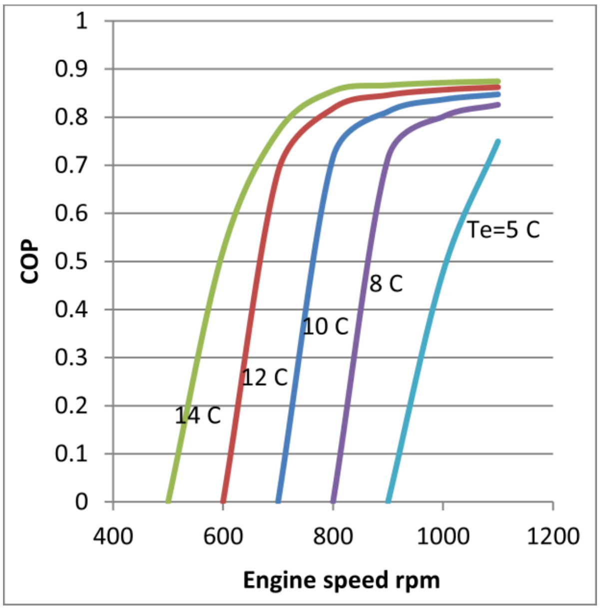

Fig. (11). Coefficient of performance verses idling engine speed at Tc=Ta=40 C and different evaporating temperatures.

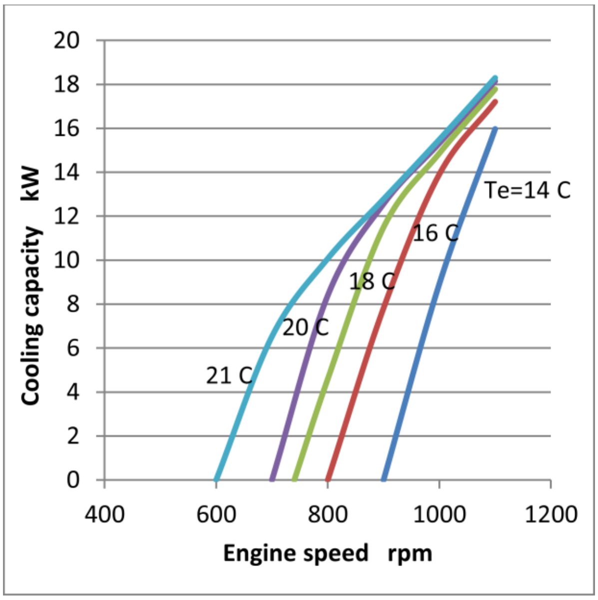

Fig. (12). Cooling capacity verses idling engine speed at Tc=Ta=40 C and different evaporating temperatures.

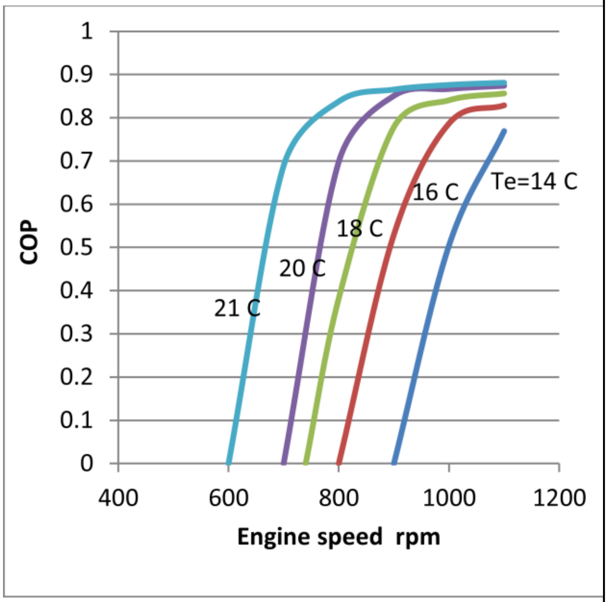

Fig. (13). Coefficient of performance verses idling engine speed at Tc=Ta=45 C and different evaporating temperatures.

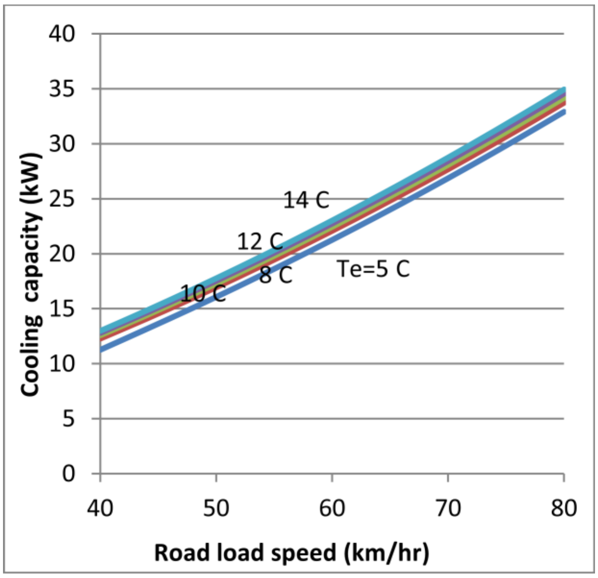

Fig. (14). Cooling capacity verses vehicle road load speed at Tc=Ta=35 C and different evaporating temperatures.

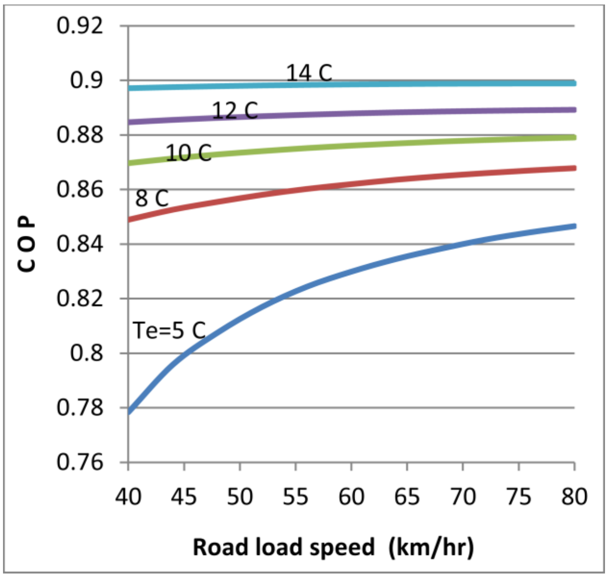

Fig. (15). Coefficient of performance verses vehicle road load speed at Tc=Ta=35 C and different evaporating temperatures.

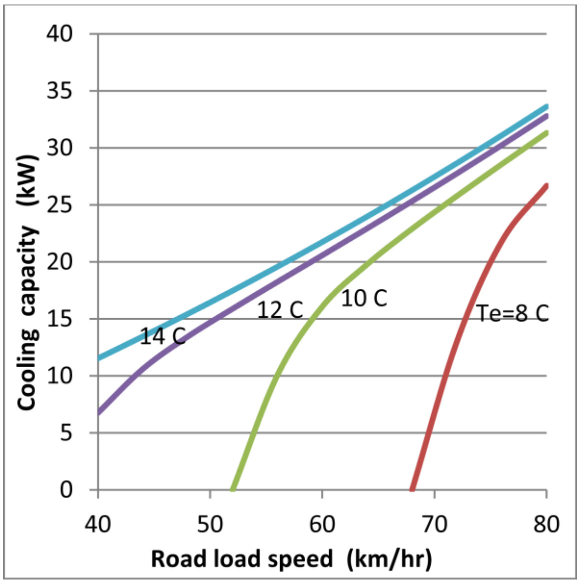

Fig. (16). Cooling capacity verses vehicle road load speed at Tc=Ta=40 C and different evaporating temperatures.

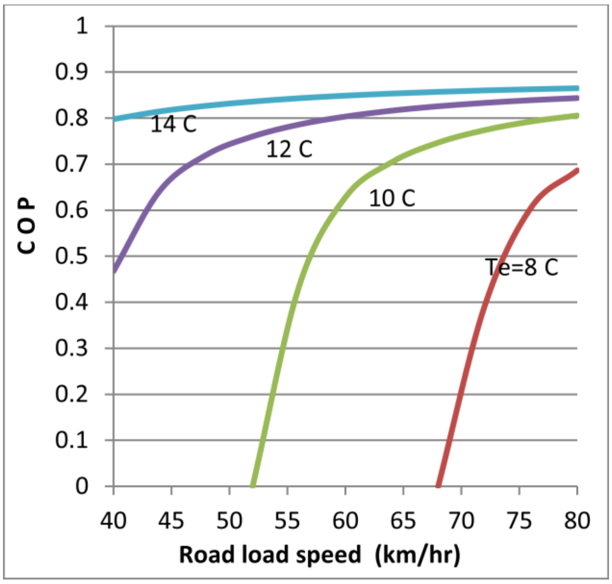

Fig. (17). Coefficient of performance verses vehicle road load speed at Tc=Ta=40 C and different evaporating temperatures.

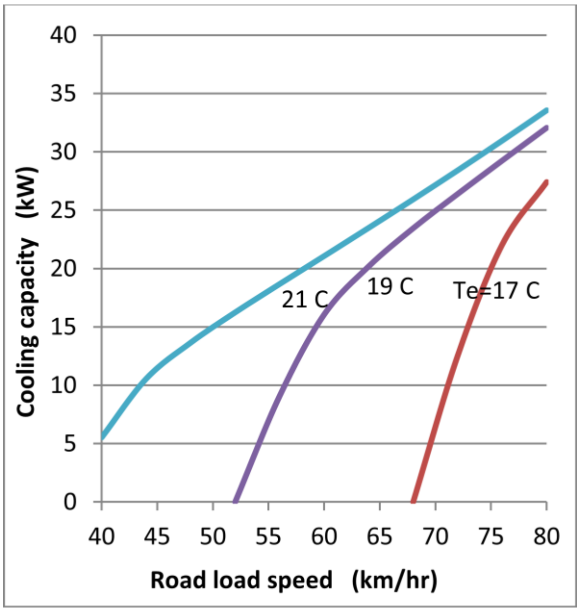

Fig. (18). Cooling capacity verses vehicle road load speed at Tc=Ta=45 C and different evaporating temperatures.

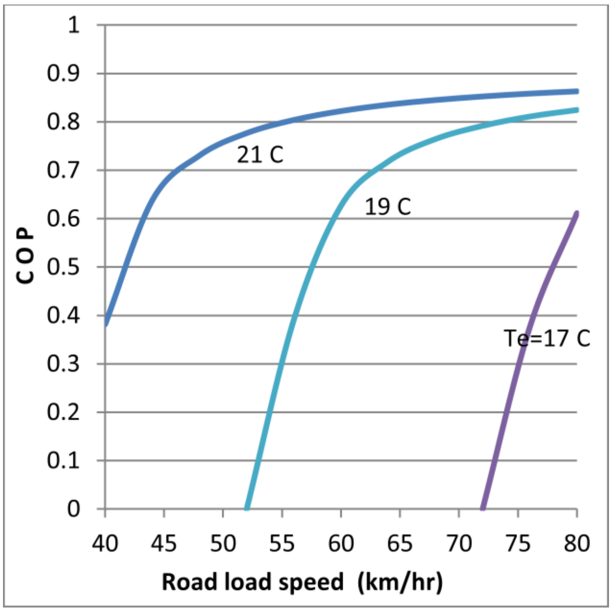

Fig. (19). Coefficient of performance verses vehicle road load speed at Tc=Ta=45 C and different evaporating temperatures.

5- Conclusion

Demonstrating the above discussed figures of LiBr-Water refrigerator performance which is driven by the wasted energy of a vehicle cooling water system, it can be concluded that;

1- At idling speed, cooling can be produced at any engine speed but with different evaporating temperatures depending on the condenser-absorber temperature.

2- Continuous cooling of 9 to 16 kW at idling speed can be produced with condenser-absorber temperatures of 35 to 45 ℃ with an evaporating temperature of 10 ℃ or more when operating the engine at 1000 rpm. The COP is ranged between 0.5 to 0.87.

3- At road load speed, continuous cooling (11-13 kW) at all vehicle speeds and an evaporating temperature of 5-14℃, can be produced at low condenser-absorber temperature (35℃), and it is never be produced at an evaporator temperature less than 12 ℃ when the condenser-absorber temperature 40 ℃ and not less than 21℃ when the condenser-absorber temperature is 45 ℃ and more which reasonably not suitable to achieve human comfort during hot days.

6- Recommendation

For the case of inadequate temperature level of the engine cooling water, it is recommended to preheat it by making use of fraction of the exhaust gas wasted energy through an indirect contact heat exchanger.

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Nomenclature

Symbols	*Definition*	*Units*	*Symbols*	*Definition*	*Units*
A	Absorber		$\bar{G}$	Variable	
C	Condenser		HX	Heat Exchanger	
CP	Circulating pump		$\dot{m}$	Mass flow rate	Kg/s
C_p	Specific het	kJ/kg.K	N	Engine speed	rpm
E	Evaporator		P	Pressure	Pa
EV	Expansion valve		$\dot{Q}$	Rate of heat energy	kW
G	Generator		T	Temperature	°C
ϵ	Heat Exchanger effectiveness		ξ	LiBr concentration in solution	

Subscripts

a	Absorber		rad	Radiation	
c	Condenser		ss	Strong solution	
e	Evaporator		v	Vapor	
g	Generator		w	water	
in	Inlet		ws	Weak solution	
out	Outlet				

Abbreviations

COP	Coefficient of performance	DMFTEG	Di methyl form amid of tetra ethylene glycol
LiBr	Lithium Bromide	R22	Refrigerant 22