Analysis of a Combined Absorption Cycle for Power and Cooling with R134a-DEGDME Working Fluid Using a Scroll Expander

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Abstract: ***This paper presents the integration of a scroll expander into a combined power and cooling absorption cycle driven by low-grade heat sources such as solar energy and waste heat. R134a-DEGDME has been selected as the working fluid mixture to replace the commonly used ammonia-water mixture.The scroll device used here is derived from a compressor. The mathematical model adopted for the expander incorporates a simplified semi-empirical model that calculates the mechanical power, exhaust temperature and supply mass flow rate of the expander at accuracy levels of ±9%, ±2 K and ±3%, respectively. Some operating parameters such as the generator (90-100°C), condenser (18-26°C) and evaporator (-4-10°C) temperatures have been varied to investigate the behaviour of the combined cycle. For the selected cycle it is suggested that increasing the generator temperature is beneficial to both power generation and cooling production. However, increasing the condenser temperature is detrimental for both power and cooling. The increase of the evaporator temperature is also favorable for both outputs.***

Keywords: Absorption cycles, Expander, Scroll, Power and cooling.R134a/DEGDME mixture

1. Introduction

The combined production of power and cooling using efficient thermally-driven systems is one of the suitable technological solutions to address the current global energy-related challenges. These challenges include energy supply uncertainty, rising price of fuels and adverse environmental impact. Air-conditioning, refrigeration and electricity are useful forms of energy products, usually produced using separate energy conversion technologies. Further, most end-users need at least dual energy products: typical example could be building's applications where space air-conditioning and electricity for various purposes are in need. In such a case the life cycle cost of the combined power and cooling system is enhanced because there is a certain number of running hours in which there is no cooling demand but a power demand is still essential.

The aim of the present study is to investigate the performance of a combined absorption power and cooling cycle, proposed by Ayou et al. [1]which adopts a scroll expander derived from a compressor as the expansion device for generating mechanical power.This novel cycle can use a single thermal energy source available at relatively low temperatures (below 200°C). Typical thermal energy sources available in this range include solar, geothermal and industrial waste heat. The first part of this article will discuss the selected combined absorption power and cooling cycle. Since the expander is a key component of these systems, the second part of the paper is devoted to the modeling of small-scale scroll expander. The accuracy of the expander model using R-134a as working fluid is presented. Finally, the scroll expander is integrated to the combined absorption power and cooling cycle and the effects of heat source sink, and chilled water temperatures on the overall performance of the cycle are investigated. Performance indicators such as the net mechanical power, cooling capacity and effective first law and exergetic efficiencies have been highlighted.

2. The combined absorption Power and cooling cycle

One of the first suggestions for a combined absorption power and refrigeration design is accredited to Goswami [2]. To generate both shaft power and refrigeration, the coupling of an absorption solution circuit and a traditional Rankine cycle was suggested as shown in Fig. 1. Initial studies on the Goswami cycle were performed with ammonia-water as working fluid however the research was extended to study the application working fluids consisting of organic fluid mixtures.

The new breeds of combined absorption cycles are derivatives of the single-stage absorption refrigeration cycle so as to facilitate latent heat transfer in the evaporator. They are developed through the coupling of a power sub-cycle with the refrigeration sub-cycle for the concurrent and/or alternating production of power and refrigeration. The modified Goswami cycle shown in Fig. 2 is essentially a Goswami cycle except the sensible refrigerant heat exchanger (cooler) is replaced by a refrigeration sub-cycle, comprising of several components, with a capability of producing latent cooling output.

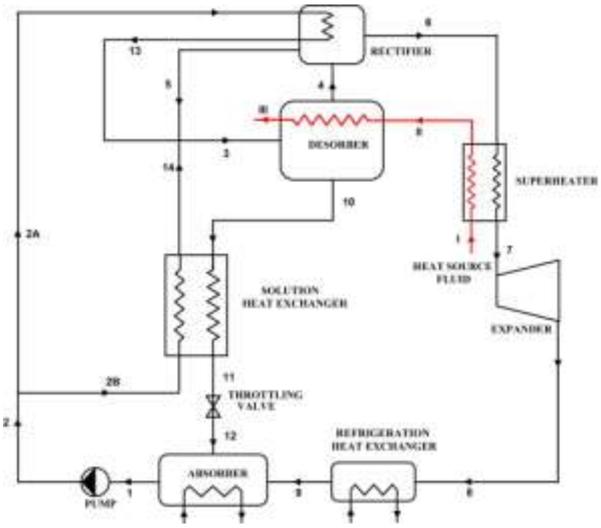

Figure 1: Schematic diagram of the Goswami cycle [2]

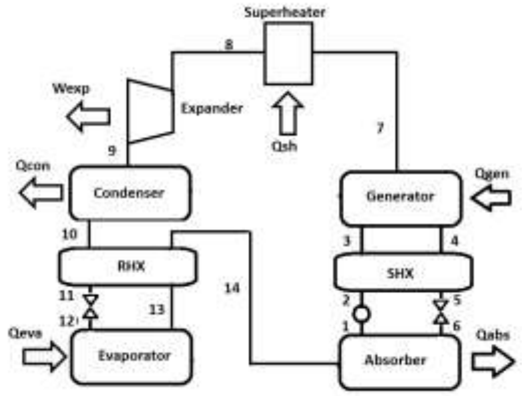

Figure 2: The modified Goswami cycle [1]

Ref. [1] concluded that the modified Goswami cycle offered the best performance for a temperature source below 200°C.Key components of the modified Goswami cycleinclude the Generator (GEN), a Super-heater (SH), a condenser (CON), an expander (EXP), an evaporator (EVA), a solution heat exchanger (SHX), a refrigerant heat exchanger (RHX) and an absorber (ABS). The cycle also incorporates a solution pump (SP) and two pressure regulation valves: a solution expansion valve (SEV) and a refrigerant expansion valve (REV). The cycle has three pressure levels; the EVA at low pressure, the CON at intermediate pressure and the GEN at high pressure. A mixture of refrigerant and absorbent (the basic solution) is pumped (states 1-2) from the low pressure ABSto the GEN for partial evaporation. The GEN yields a refrigerant-rich vapor (state 7) and a weak solution (state 4). The weak solution exchanges heat with the basic solution (states 4-5) at the SHX before being throttled by the SEV (states 5-6) into the low pressure ABS. On the other hand, the enthalpy of the refrigerant vapor from GEN is enhanced by heat addition at the SH (states 7-8) before being expanded at the EXP to intermediate pressure (state 9). From the EXP, the refrigerant is condensed at the CON (state 10) and then sub-cooled by the RHX (state 11) before being throttled at the REV to the low pressure EVA (state 12). The refrigerant is partially evaporated at the EVA (state 13), then passed through the RHX for further evaporation (through heat exchange with streams 10-11) before being fed back to the ABS (state 14).

The thermodynamic properties for the ammonia-water working fluid mixture are retrieved form the external routines database of the Engineering Equation Solver software [3]. The following assumptions were made to develop the thermodynamic model of the combined power and cooling absorption cycle:

- The study is conducted under steady state conditions.
- The only pressure drops considered are in the expander (EXP) and expansion valves (REV and SEV).
- The only heat exchange with the surroundings considered is in the GEN, SH, CON, EVA and ABS.
- The stream refrigerant at the GEN outlet is pure.
- The SP is isentropic with efficiency,is 0.8and both expansion processes in REV and SEV are isenthalpic.
- The effectiveness of the heat exchangers SHX and RHX is 0.8.
- A mass flow rate of 1 kg/s is assumed though the SP.
- A saturated stream is assumed at the outlet of the GEN, REC, CON and ABS.
- The scroll expander is simulated by a semi-empirical model.
- A pressure ratio (P_{gen}/P_{con}) of 2 is adopted.
- SH temperature is 20°C above the GEN temperature.

In order to explain and compare the behaviour of the cycle, the following performance indicators have been chosen: net mechanical power, cooling capacity and effective first law and exergetic efficiencies.

Net mechanical work output ($\dot{W}_{net}$) is the mechanical work output from the expander ($\dot{W}_{exp}$), reduced by the mechanical work supplied to the solution pump ($\dot{W}_{sp}$).According to Vijayaraghavan and Goswami[4], an accurate definition of the first law efficiency for a combined power and cooling cycle must account for the quality of the cooling output. Therefore the first law effective efficiency ($\eta_{I,eff}$) is represented as:

$$\eta_{I,eff} = \frac{\dot{W}_{net} + \frac{\dot{E}_{chill}}{\eta_{II,ref}}}{\dot{Q}_{sh} + \dot{Q}_{gen}} \quad 1.1$$

Similary, the exergetic efficiency ($\eta_{ex,eff}$) can be expressed as:

$$\eta_{ex,eff} = \frac{\dot{W}_{net} + \frac{\dot{E}_{chill}}{\eta_{II,ref}}}{\dot{E}_{sh} + \dot{E}_{gen}} \quad 1.2$$

The exergy associated with the refrigeration output ($\dot{E}_{chill}$) refers to the exergy transfer in the refrigeration heat exchanger. Depending on the way the cycle is modeled, this could refer to change in exergy of the working fluid in the refrigeration heat exchanger. Alternately, to account for irreversibilities of heat transfer in the refrigeration heat exchanger, the exergy change of the chilled fluid would be considered. The same interpretation can also be extended for the exergy transfers in the super-heater ($\dot{E}_{sh}$) and generator ($\dot{E}_{gen}$) respectively.The term$\eta_{II,ref}$ is the second law efficiency of a vapor compression refrigeration system. In this study $\eta_{II,ref}$ is taken as 0.3 [4].

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3. Working Fluids

The commonly suggested working fluid for power and cooling combined cycles is ammonia/water. Mainly because of its good properties and refrigerant, low cost, etc. However, researchers have been investigating the technical feasibility of other fluids such fluorocarbon refrigerants and organic absorbents working pairs for absorption technology. Jelinekand Borde [5] performed a comparison between different fluorocarbon refrigerants (R22, R134a, R124 and R32) and organic absorbents (DMAC, DMEU, DMETEG and NMP) as working fluids in single- and double-stage absorption heat pumps and reported that: (a) the performance of the working fluids based on R22 are superior to those based on R124 in both cycles, (b) the working fluids based on R134a cannot be used in a single-stage absorption cycle while in a double-stage cycle the performances were the worst and (c) solutions based on R32 cannot be used in either cycles. He et al. [6] carried out a theoretical analysis of the coefficient of performance to examine the efficiency characteristics of R22 + DMF, R134a + DMF, R32 + DMF as working fluids, respectively, for a single-stage and intermittent absorption refrigerator which allows the use of heat pipe evacuated tubular collectors. They concluded that considering the relative low pressure and the high coefficient of performance, R134a + DMF mixture presents interesting properties for its application in solar absorption cycles at moderate condensing and absorbing temperatures when the evaporating temperatures in the range from 278 K to 288 K.
Recently, researchers have been investigating the use of natural refrigerants, like carbon dioxide in absorption power cycles. Robbins and Garimella [7] investigated the feasibility of utilizing low-grade waste heat in a power absorption cycle using carbon dioxide-amyl acetate mixture. A desorber solution outlet temperature of 150°C was assumed, while the absorber temperature was chosen to be 25°C, with 5°C of sub-cooling. For a mass flow rate of 0.18 kg/s through the pump, a net power output of 5.04 kW was produced; the required pumping power was 2.32 kW with a 14.2 % of cycle efficiency. The cycle is essentially a Goswami type cycle without the cooler after the expander.

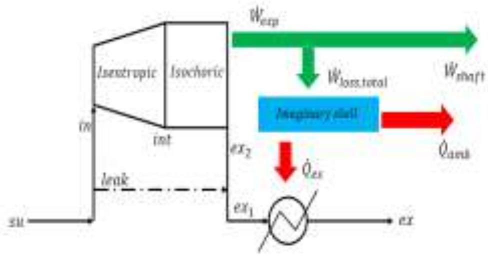
4. Semi-empirical model of the scroll expander

The expander plays an integral role in the performance of the whole system. Two types of expanders could be used in these kind of cycles: dynamic and volumetric machines. The first ones (such as axial turbines and radial flow turbines) have been successfully used for large capacity, however for small capacity (below 50 kW) their application is still a challenge since ammonia has a low molecular weight and high sound speed which considerably increases the machine rotor speed and the internal leakages thereby reducing the effective operating pressure ratio. Volumetric devices include: scroll expanders, screw expanders, piston expanders and rotary vane expanders. Volumetric technology, specifically scroll machines, has proven to have a promising future for small capacity absorption systems for cooling and power applications. Scroll compressors are simple in design, reliable, possess a compact structure, have fewer moving parts, lower level of noise and vibration. They are also easy to access because they are widely applied in the HVAC industry. Consequently, researchers have been experimenting on scroll expanders modified from scroll compressors and reporting satisfactory results [8, 9].

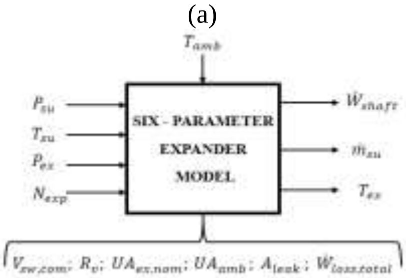
The experimental system utilized in this research is a redesigned and upgraded version of the facility which was developed by Mendoza [10] between 2011 and 2013 to characterize a scroll expander working with ammonia at the Applied Thermal Engineering Group - CREVER, Universitat Rovira i Virgili, Tarragona, Spain.

A Sanden, model TRSA05, open-drive scroll compressor (normally used in automotive refrigeration systems) was modified and operated in expander mode. According to the manufacturer, the volumetric displacement per revolution of the scroll device in compressor mode is 53.9 cm^3, and it can withstand a maximum of 3500 kPa and 166 Hz of pressure and rotational speed respectively. The built-in volume ratio is 1.9, calculated by measuring the cross section area at the suction and exhaust chambers [10]. Since the expander is adapted from a scroll compressor, performance and eventual characterization of such an expander can only be determined experimentally. On the other hand, it is very important to understand the performance of the scroll expander at different operating conditions before it can be successfully incorporated into the design of the combined absorption power and cooling systems.

The expander semi-empirical model is a simplified version of the one originally proposed byLemort [11] as shown in Figure 3. The input data for the model are pressure and temperature at the expander inlet, pressure at the outlet and the expander rotational speed. It then calculates mechanical power, exhaust temperature and supply mass flow rate of the expander at accuracy levels of ±9%, ±2 K and ±3%, respectively as highlighted in Figure 4.



(a)



(b)

Figure 3: Semi-empirical model (a) process diagram and (b) block diagram.

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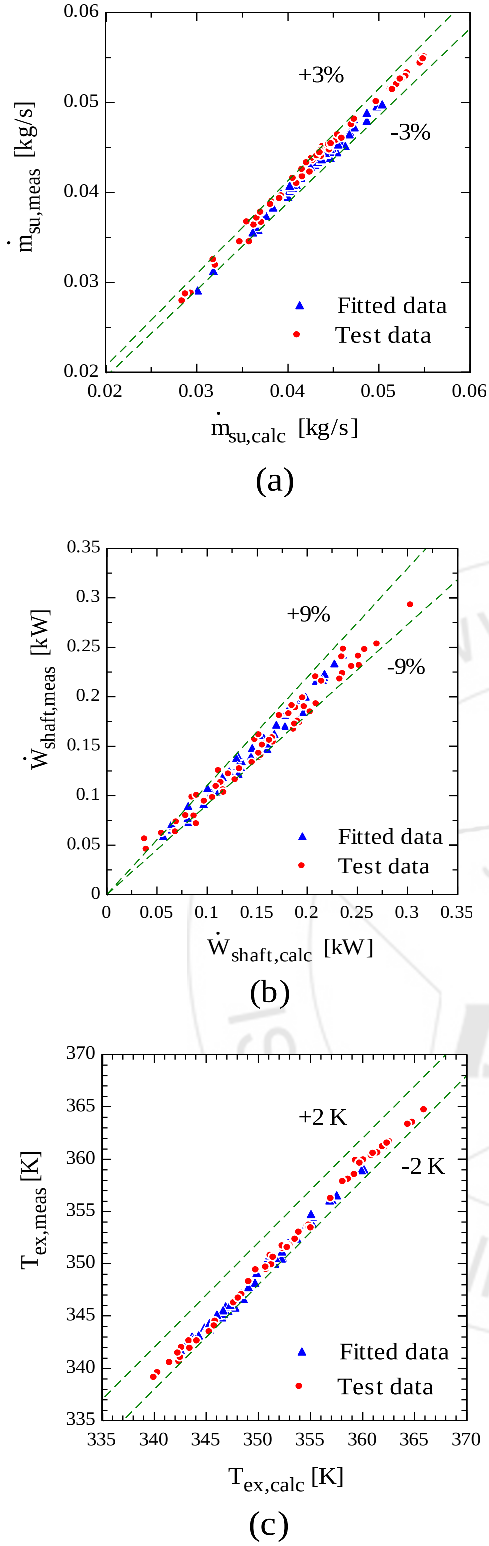

Figure 4: Validation of the model for R134a

5. Integrating the scroll expander to the combined absorption cycle

The semi-empirical model developed in Section 4 is coupled to a larger combined absorption power and cooling cycle simulation model. The working fluid selected is R134a-DEGDME as suggested by Zehioua et al. [12].

5.1. Effects of varying the generator temperature

Figure 5 shows the influence of varying the generator temperature on the performance of the cycle. Generally, increasing the generator increases both the outputs.

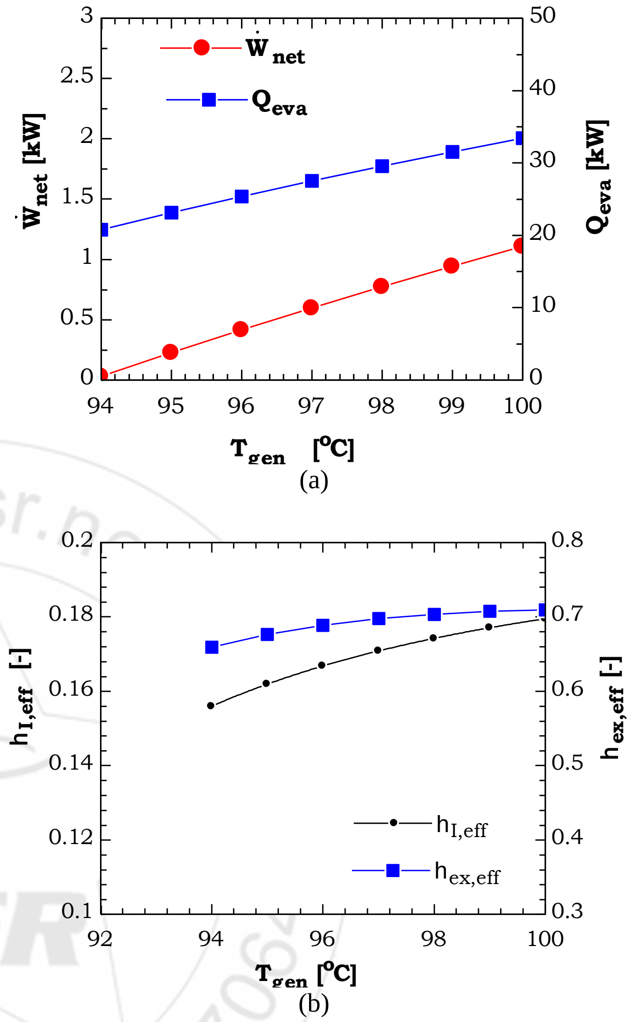

Figure 5: Influence of Generator temperature on: (a) power and cooling outputs, (b) effective efficiencies of the cycle with T_{con} and T_{eva} held at 25^{o}C and 0^{o}C respectively.

The generator could not be operated above the critical temperature of R134a (101^{o}C. At generator temperatures below 94^{o}C, the solution pump power exceeded the expander output. Power production improves from 0.03 to 1.11 kW as the generator temperature is increased (94°C to 100°C) as a result of the increased enthalpy of the working fluid.Cooling output improved by 12 kW. The effective first law efficiency improved at a greater rate than its exergetic counterpart.

5.2. Effects of varying the condenser temperature

Figure 6 depicts the influence of varying the condenser temperature on the performance of the cycle. As the condenser temperature is varied, the generator and evaporator temperatures a fixed at 100^{o}C and 0^{o}C respectively.

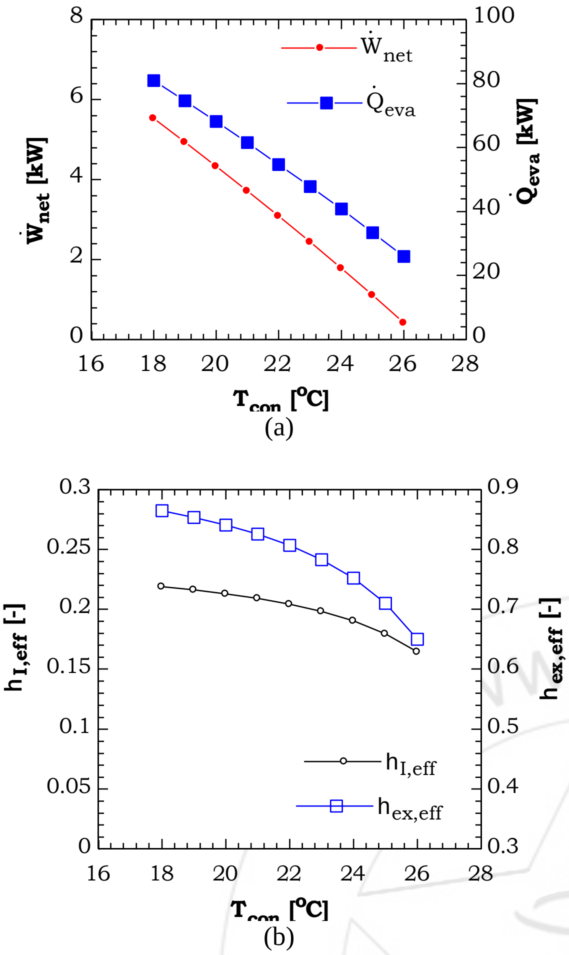

Figure 6: Influence of Condenser temperature on: (a) power and cooling outputs, (b) effective efficiencies of the cycle with T_{gen} and T_{eva} held at 100°C and 0°C respectively.

When the condenser temperature is varied between 18°C and 36°C (at constant generator and evaporator temperatures), cooling capacity drops from 81 to 26 kW while the power output decreases by 5.10 kW. The drop in power output can be attributed to an increase in pump consumption as the condensation pressure is increased. The effective efficiencies also drop with increasing condenser temperature.

5.3. Effects of varying the evaporator temperature

Figure 7 depicts the influence of varying the condenser temperature on the performance of the cycle. As the evaporator temperature is increased, the generator and condenser temperatures a fixed at 100°C and 25°C respectively. At evaporator temperatures below -3°C, the solution pump power exceeded the expander output. Power production improves from 0.29 to 5.66 kW as the evaporator temperature is increased (-2°C to 10°C). The cooling output also increases from 24.07 to 83.67 kW. The exergetic efficiency improves as evaporator temperature is increased and peaks at 2°C. Above 2°C, the exergetic performance drops.

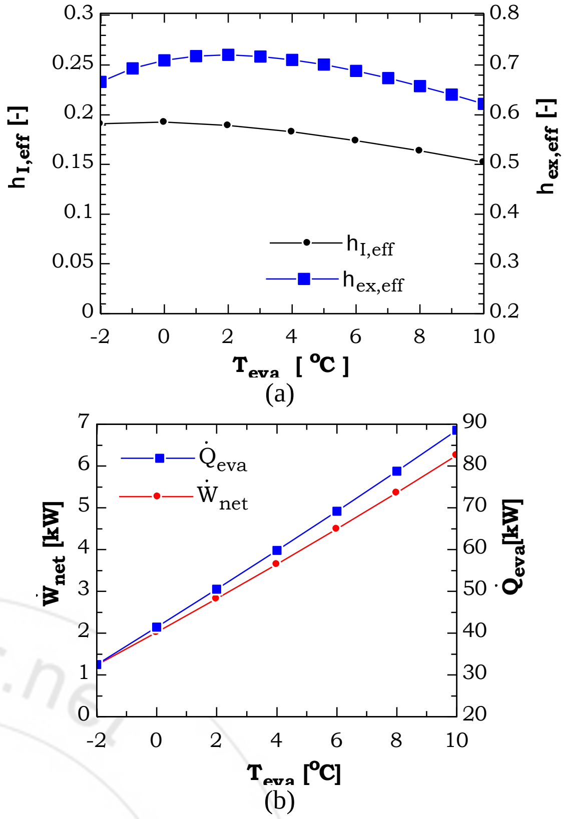

Figure 7: Influence of Evaporator temperature on: (a) power and cooling outputs, (b) effective efficiencies of the cycle with T_{gen} and T_{con} held at 100°C and 25°C respectively.

6. Conclusions

A combined power and refrigeration absorption system adopting a modified scroll expander and R134a-DEGDME working fluid mixture is proposed for the production of mechanical power and cooling. The combined cycle can be driven by low-temperature heat sources such as geothermal, solar or waste thermal energy from industrial processes.
An investigation is carried out to study the influence of the generator, sink and evaporator temperatures on the cycle performance. The key performance indicators highlighted were: net mechanical power output, cooling capacity, effective first law and exergetic efficiencies. A semi-empirical scroll expander model is developed and validated using experimental data. The model can calculate mechanical power, exhaust temperature and supply mass flow rate of the expander at accuracy levels of ±9%, ±2 K and ±3%, respectively

For this particular cycle it is suggested that increasing the generator temperature is beneficial to both power generation and cooling production. However, increasing the condenser temperature is detrimental for both power and cooling. Iincreasing the evaporator temperature is also favorable for both outputs.

This study proves the technical feasibility of employing a modified scroll device as a work producing expander in small capacity combined absorption power and cooling systems. It also proves that the less toxic R134a-DEGDME working mixture could be used in applications where the

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more common but toxic ammonia-water mixture is unsuitable.

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