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Expander-based Atmospheric Water Harvesting in the Tropics

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Abstract: A novel concept of using an expander to harvest atmospheric water is explored. The main advantages of this concept are its compactness and simplicity. Mathematical models were developed in this project to study the concept. The benchmark system had a crank and piston radii of 5 cm, rod length of 20 cm, operational speed of 60 rev/min, atmospheric temperature of 30°C and relative humidity of 80%. The expander was designed to expand the air to 0.7 bar and 0°C at the end of the expansion process. Because of the expansion, around 11.5 g of water was condensed for every 1 kg of air expanded. Most of the expander power was consumed during expansion to overcome the pressure difference across the two sides of the piston. The average power per cycle was 3.374 W. Therefore, the ratio of energy consumed and condensed water volume produced is 117 kWh/m^3.

The parametric study found that the ratio of energy and water volume was unaffected by the operational speed, increased linearly as the ambient air was hotter, decreased with ambient relative humidity and was unaffected by the expander size.

Key words: Water, vapour, humidity, condensation, expander, tropics.

Introduction

Clean water is a basic necessity for all life. However, due to the growing demand from industry and the increasing human population, it is important to explore alternative sources of clean water. Some of the most popular solutions already in use include desalination (Shrivastava, 2016) and wastewater treatment (Deng and Wheatley, 2016). The challenges facing these methods are the cost and complexity of the system. Moreover, a liquid water source/reservoir is required, hence, limiting the choice of location of implementation.

In the tropics, the air is constantly humid throughout the year. For illustration, in Singapore, which is located in South East Asia, the relative humidity is always above 60% and is often more than 90% (Singapore's National Environment Agency, 2016). The humid condition means that a significant amount of water is available in the air in the form of water vapour. This can be condensed to liquid water to provide an alternative clean water source. This is particularly attractive for places with high humidity but limited access to clean water source, like in many coastal areas and small islands in South East Asia.

Cooling is arguably the most well-known mode of atmospheric water harvesting. However, there are other ways to collect water from the atmosphere. In general, the methods can be classified into three types (Wahlgren, 2001):

1. To cool a surface below the dew-point of the ambient air.

   a. With an artificial refrigeration/heat pump system
   b. With radiative cooling
2. To concentrate water vapour through use of desiccants.
   a. With absorption in liquid desiccant
   b. With adsorption on solid desiccant
3. To induce and control convection in a tower structure.

The strengths and limitations of the three types of methods are tabulated in Table 1 (Harriman, 1990; Khalil, 1993; Meytsar, 1997; Wahlgren, 2001). Type 1a,

Table 1: Strengths and limitations of various methods of atmospheric water harvesting

Strengths	*Limitations*
Type 1: Cooled surface	
Heat pump	
1. Technology is mature	1. Frost may form and degrade the cooling process
2. Efficient when condenser air temperature is low and cooling coil air temperature is high	2. There is mixing of dried and unprocessed air within the processor
3. Easy maintenance	3. High power requirement
	4. Refrigerants may affect the environment negatively
Radiative cooling	
1. No external energy source is needed	1. Existing technology depends on radiation into a clear night sky
2. Simple mechanical requirements	
Type 2: Desiccants	
1. Well-developed technology	1. Energy requirement is fairly high
2. Able to dry air to a low relative humidity	2. Liquid absorbents can concentrate contaminants from the atmosphere
Type 3: Convection induced or controlled in a structure	
1. Adiabatic cooling has the lowest energy requirements of the three design strategies	1. Large structure is required
2. Engineering experience in removal of water from industrial compressed air systems is well-developed	2. No existing prototype is known yet

which is the artificial refrigeration system, is the most common method for artificial condensation. However, this system involves a lot of components, because even in its most basic form, a typical refrigeration system consists of at least four main components (a compressor, an evaporator, an expansion device and a condenser). A much simpler solution is to achieve cold temperature by directly expanding the air in an expander. This method involves only one component, i.e. an expander. In an expander, air is expanded and its temperature decreases. This low temperature of the expanded air will induce condensation.

The oldest, most common and most reliable machine design is the reciprocating mechanism. The schematic is shown in Figure 1. It consists of a stationary cylinder, a piston that moves in a reciprocating motion in the cylinder chamber, an intake port with its valve and an exhaust port with its valve. The piston is linked to a crank mechanism that causes it to move up and down in the cylinder. Reciprocating machine is known to be very reliable and has been earlier used for expanders (Baek et al., 2002).

The working cycle starts with the piston at the bottom dead centre (BDC), which is the furthest point of the piston from the cylinder's top wall. The fluid inside the chamber is currently low in pressure and temperature. The intake valve is shut while the exhaust valve is opened. The piston then begins to move up, towards the cylinder's top wall, pushing the low pressure fluid to flow out of the chamber through the exhaust port. This process continues until the piston reaches the top dead centre (TDC). This is the closest the piston can reach with respect to the cylinder's top wall. A narrow gap between the piston and the top wall remains for

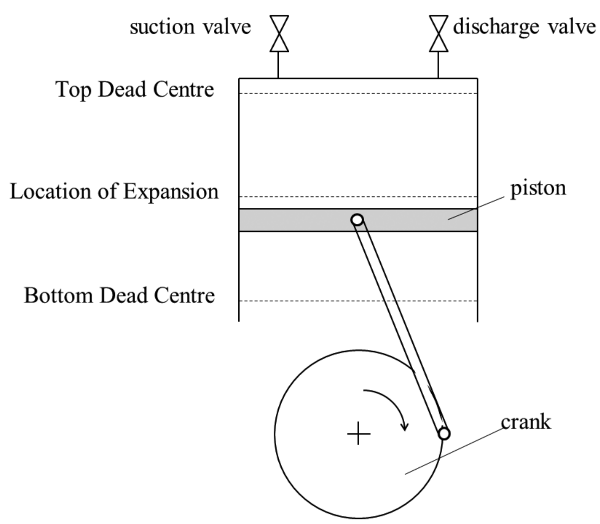

Figure 1: Schematics of a reciprocating expander.

practical reasons. However, this gap should be made as narrow as possible for better performance. At this moment, the exhaust valve is shut while the intake valve is opened. The piston then moves away from the top wall, increasing the volume of the working chamber. This induces high pressure fluid to flow into the chamber through the intake port. This process continues until the piston reaches the designed location of expansion (LOE) at which the intake valve is immediately shut, stopping the intake flow. Meanwhile, the piston continues to move down, increasing the chamber volume further. This causes the fluid pressure in the chamber to decrease. This expansion process continues until the piston reaches the BDC location again. At this moment, the fluid pressure is at its lowest and the cycle is completed.

Currently, expanders are mostly used to generate useful power, like in Rankine cycles (Badr et al., 1985a, 1985b). Expanders have also been recently proposed for increasing energy efficiency of refrigeration and heat pump systems (Tamura et al., 1997; Henderson et al., 2000; Robinson et al., 1998; Lorentzen, 1994). In this study, a reciprocating expander was used to harvest atmospheric water by directly expanding the air to reduce its temperature to induce condensation. The focus is on the feasibility of the concept. The issue of energy requirement and effects of various parameters to the performance were also investigated.

Mathematical Models

Expander

The mathematical model of the expander was derived based on the crank mechanism schematic and the free body diagram shown in Figure 2. It was assumed that the upper side of the piston was exposed to the working chamber while the lower side was exposed to atmosphere.

The definitions and force balance equations of the piston mechanism are expressed in Equations (1-4). The masses of the crank and connecting rod were ignored in this model.

$$F_p = (p_f - p_{atm})A_p \quad (1)$$

$$F_{fric} = \eta F_n \quad (2)$$

$$F_{PN} \sin \varphi = F_n \quad (3)$$

$$-F_{PN} \cos \varphi - F_p + F_{fric} = m_{piston}\, \alpha_p \quad (4)$$

where F_p is the pressure force (N), p_f is pressure of fluid in the working chamber (Pa), p_{atm} is atmospheric pressure (Pa), A_p is piston's cross sectional area (m^2), F_n is the piston side contact force (N), F_{fric} is the piston side friction force (N), η is the friction coefficient, F_{PN} is the force provided by the link PN to push the piston (N), φ is the connecting rod angle (rad) and m_{piston} is the piston mass (kg).

The torque acting at the crank centre, T_O, is calculated according to Equation (5) and the power required at the crank centre is calculated according to Equation (6).

$$T_O = (x + \eta r_p)\left(\frac{\tan\varphi}{\eta\tan\varphi - 1}\right)(m_{piston}\, \alpha_p + (p_f - p_{atm})A_p) \quad (5)$$

$$P_O = T_O \omega \quad (6)$$

where ω is angular speed of the crank (rad/s) and r_p is the piston radius (m).

Thermo-physics of the Fluid

The working fluid in this project was humid air. It was assumed that throughout the process, the fluid was in thermodynamic equilibrium and can be treated as a homogenous mixture. It was also assumed to behave

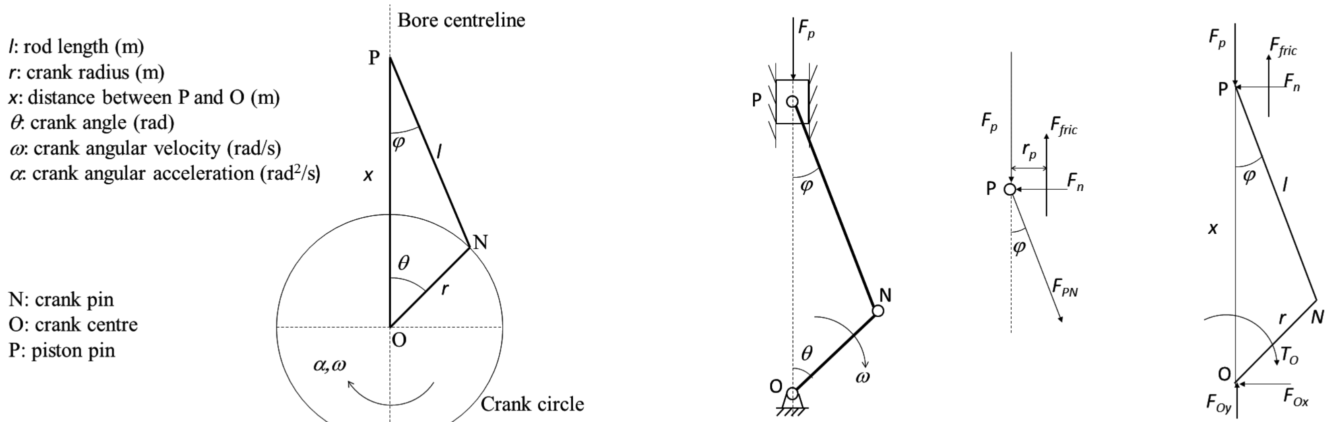

Figure 2: Crank mechanism schematic and free body diagrams of a reciprocating machine.

as an ideal gas. Therefore, the properties of the fluid are according to Equation (7).

$$p_f V_f = m_f R T_f \tag{7}$$

where p_f is fluid pressure (Pa), V_f is fluid volume (m^3), m_f is fluid mass (kg), R is specific gas constant (287 J/kg·K for air) and T_f is fluid temperature (K).

The location of expansion (LOE) needs to be defined beforehand. This was done by first defining the final expansion temperature. In this project, this was set to be 0°C to avoid frosting. During expansion, the process was modelled according to the polytropic model as expressed in Equation (8). The working fluid's pressure is reduced together with its temperature.

$$p_1 V_1^{\gamma} = p_2 V_2^{\gamma} \tag{8}$$

where γ is the polytropic index, which is around 1.4 for air (Moran et al., 2000).

The enthalpy change because of the expansion is calculated according to Equation (9). This is the available thermal energy to be used for condensation process. Specific heat of air can be estimated according to Equation (10) (Moran et al., 2000).

$$h_2 - h_1 = \int_{T_1}^{T_2} c_p dT \tag{9}$$

$$c_{p,air} = 287(3.653 - 1.337 - 10^{-3}T + 3.294 \times 10^{-6}T^2 - 1.913 \times 10^{-9}T^3 + 0.2763 \times 10^{-12}T^4) \tag{10}$$

where h is the specific enthalpy (J/kg) and c_p is specific heat at constant pressure (J/kg·K).

Enthalpy of humid air can be computed following the step-by-step procedure developed by Devres (1994), resulting in Equation (11).

$$h_{air} = (T_{\mathrm{air}} - 273.15) + W(2501 + 1.805 (T_{air} - 273.15)) \tag{11}$$

where T is dry-bulb temperature (K) and W is humidity ratio of humid air (kg/kg).

Finally, mass of condensed water produced is calculated with Equation (12).

$$Q_{evap} = m_{cond} h_{evap}$$
$$\rightarrow m_{air} \Delta h_{exp} = m_{cond} h_{evap} \tag{12}$$

Results and Discussion

The results gathered from the models are presented and discussed in this section. A benchmark system is first discussed. Later, a parametric study is presented.

Throughout the study, the following assumptions were adopted:

- Working fluid was air
- Air was ideal gas
- Adiabatic expansion
- Air was expanded to 0°C
- Valves opened and closed instantaneously
- The port sizes were large enough such that the fluid flows did not affect the thermodynamics of the fluid in the chamber.
- Suction and discharge reservoirs were always at the ambient pressure and temperature
- Perfect sealing (no leakage)
- Heat caused by frictions of expander components was negligible
- Constant crank rotational speed
- Weights of the crank and connecting rod were ignored

Benchmark System

The dimensions and operating conditions of the benchmark system are tabulated in Table 2.

Table 2: Dimensions and operating conditions of the benchmark system

Item	*Value*
Crank radius	5 cm
Rod length	20 cm
Crank angular velocity	60 rev/min
Dead volume	39.3 cm^3
Maximum volume	824.7 cm^3
Location of expansion	115°
Piston radius	5 cm
Piston thickness	0.2 cm
Piston density	2700 kg/m^3
Piston mass	42.4 g
Friction coefficient	0.1
Atmospheric pressure	101325 Pa
Atmospheric temperature	30°C
Atmospheric relative humidity	0.8

The movement of the piston and the valve dynamics varied the fluid mass in the working chamber as shown in Figure 3. At the beginning of the cycle, air was induced into the chamber through the open suction valve, increasing the fluid mass in the expander. The discharge valve was shut during the process. When the piston reached the location of expansion (LOE),

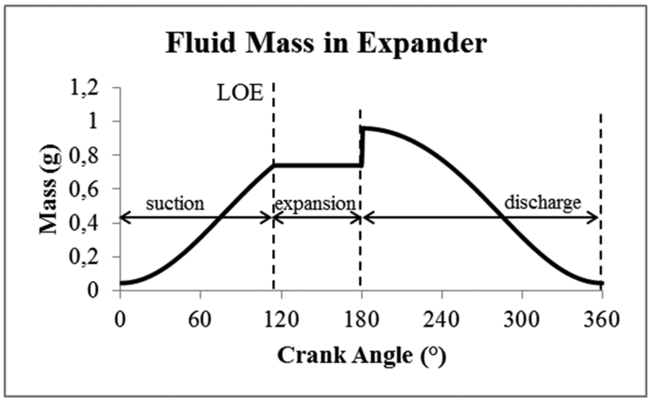

Figure 3: Variation of fluid mass with crank angle.

the suction valve was shut, stopping the flow of air into the expander. Hence, the fluid mass was constant during this expansion process. This continued until the piston reached its bottom dead centre at a crank angle of 180°. At this moment, the discharge valve was opened while the suction valve was kept shut. Air from the discharge reservoir gushed into the working chamber because the pressure inside the chamber was lower than the reservoir. Therefore, the fluid mass in the chamber increased rapidly. As the piston moved up, the fluid in the chamber was gradually discharged out through the discharge valve. This process continued until the end of the cycle. The amount of air mass expanded in the expander per cycle was around 0.7 g. Since the angular speed was 60 rev/min, the expanded air mass flow rate was equal to 0.7 g/s.

The variation of fluid pressure and temperature in the chamber with crank angle is shown in Figure 4. At the beginning of the cycle, as the volume was increasing, ambient air was flowing into the chamber. The pressure and temperature inside the chamber were constant at 101325 Pa and 30°C, respectively. When the piston reached the LOE, the pressure and temperature gradually dropped because no more air was added into the chamber while the volume was still increasing. This decrease in pressure and temperature continued until the volume reached its maximum at the bottom dead centre position. The minimum pressure and temperature were 70359 Pa and 0°C, respectively. The pressure was then increasing rapidly at the beginning of the discharge because of the sudden flow of air into the chamber when the discharge valve was opened. After the pressures in the chamber and the ambient were in equilibrium, it stayed constant throughout the discharge process until the whole cycle was completed.

The low temperature caused by expansion was the cause of the condensation of the water vapour in the expander and, indirectly, at the outer wall of the expander. Using the psychrometric equations by Devres (1994), the dew point temperature of air at 30°C and 80% relative humidity is 26.2°C and the absolute humidity is around 21 g of water per every kg of air. The enthalpy difference provided by the expansion process of air from the dew-point temperature of 26.2°C to 0°C was 26.4 kJ/kg of air. Considering that 0.7 g of air was expanded every cycle, there was 18.4 J of enthalpy to be used for water condensation. The latent heat of vaporization of water at 0.7 bar is 2283.3 kJ/kg of water while the specific heat capacity is around 1.86 kJ/kg·K at room temperature. Therefore, assuming that the available enthalpy was used to first cool the water vapour to the dew point temperature and then to condense it, 0.008 g of water was condensed per cycle. This was equal to 29 g of water per hour because the rotational speed was 60 rev/min here. This also meant that around 11.5 g of water was condensed for every 1 kg of air expanded.

The variation of expander power requirement with crank angle is plotted in Figure 5. It can be seen that most of the power was consumed during the expansion.

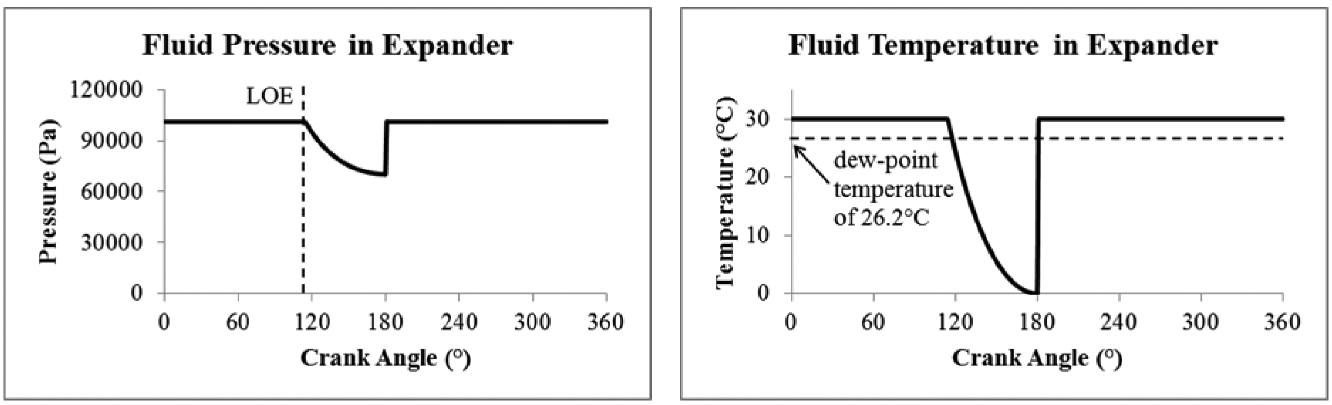

Figure 4: Variation of fluid pressure and temperature with crank angle.

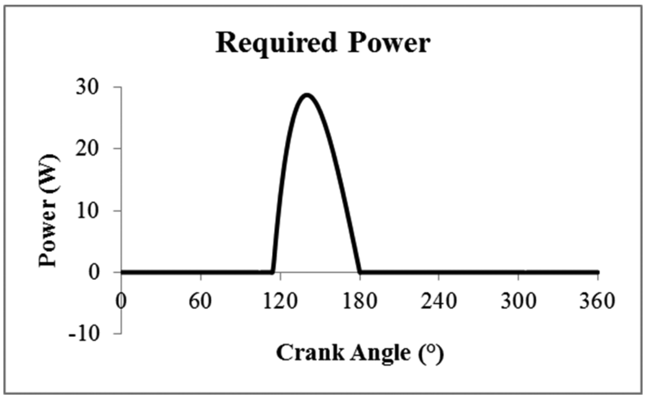

Figure 5: Variation of expander power requirement with crank angle.

This power was mostly used to overcome the pressure difference across the two sides of the piston. The average power per cycle was 3.374 W. It is interesting to observe that the curve is continuous, unlike those of mass, pressure and temperature. Moreover, the maximum power was not at crank angle of 180°. This is because power was affected by not only pressure difference across the piston, but also the angle of the connecting rod and the acceleration of the piston. As the piston was approaching the bottom dead centre, the connecting rod angle was decreasing and, hence, the required power was also decreasing.

It had been calculated above that 0.008 g of water was condensed every cycle. Therefore, the ratio between energy usage and amount of water produced was found to be 0.117 kWh/kg of water. Assuming that the water density was 1000 kg/m^3, the ratio between the energy usage and volume of water was 117 kWh/m^3. This was around 17% of the standard ratio of 681 kWh/m^3 (Wahlgren, 2001).

The gathered benchmark data are summarised in Table 3.

Table 3: Summary of gathered benchmark data

Item	*Value*
Air mass/cycle	0.7 g of air
Air mass flow rate	7 g/s of air
Available enthalpy difference	26.4 kJ/kg of air
Available enthalpy diff./cycle	18.4 J
Condensed water mass/cycle	0.008 g of liquid water
Condensed water mass/hour	29 g of liquid water
Ratio of energy and condensed water mass	0.117 kWh/kg of liquid water
Ratio of energy and condensed water volume	117 kWh/m^3 of liquid water

Parametric Study

After discussing the benchmark data, parametric study is presented in this section. The focus of the analysis is on the ratio of energy and condensed water volume. The varied parameters include the following and the results are shown in Figure 6.

- Rotational speed
- Atmospheric temperature
- Atmospheric relative humidity
- Expander size

It was found that the ratio was practically unaffected by the operational speed. The simulation results had a standard deviation of 1% over the range of speeds tested, i.e. from 30 to 3840 rpm. This deviation was within the expected numerical error. It is useful, however, that the effect of frictional heat was ignored. In practice, this heat will increase the temperature of the air in the expander and, in turn, increase the ratio as the speed increases.

The ratio increased significantly as the ambient air was hotter. From the figure, it can be seen that the increase is linear with a gradient of around 4 kWh/m^3/°C. The reason was because less air can be expanded in every cycle if the ambient air was hotter because the final expansion temperature has been fixed at 0°C. Moreover, the fluid had to be expanded more to achieve the desired expanded temperature.

The ratio was found to decrease with a higher relative humidity. Air flow rate and required power were unchanged here. However, more humid air had a higher dew point temperature and, therefore, was easier to condense the moisture.

Lastly, the ratio was unaffected by the expander size. This was because with a bigger expander, more air was processed per cycle. However, at the same time, more power was required, cancelling out the increase in air mass.

Conclusions

In this study, an expander is used to induce condensation to harvest atmospheric water. The main advantages of this concept are its compactness and simplicity.

Mathematical models were developed and employed to study a benchmark expander system. The system had a crank and piston radii of 5 cm, rod length of 20 cm, operational speed of 60 rev/min, atmospheric pressure

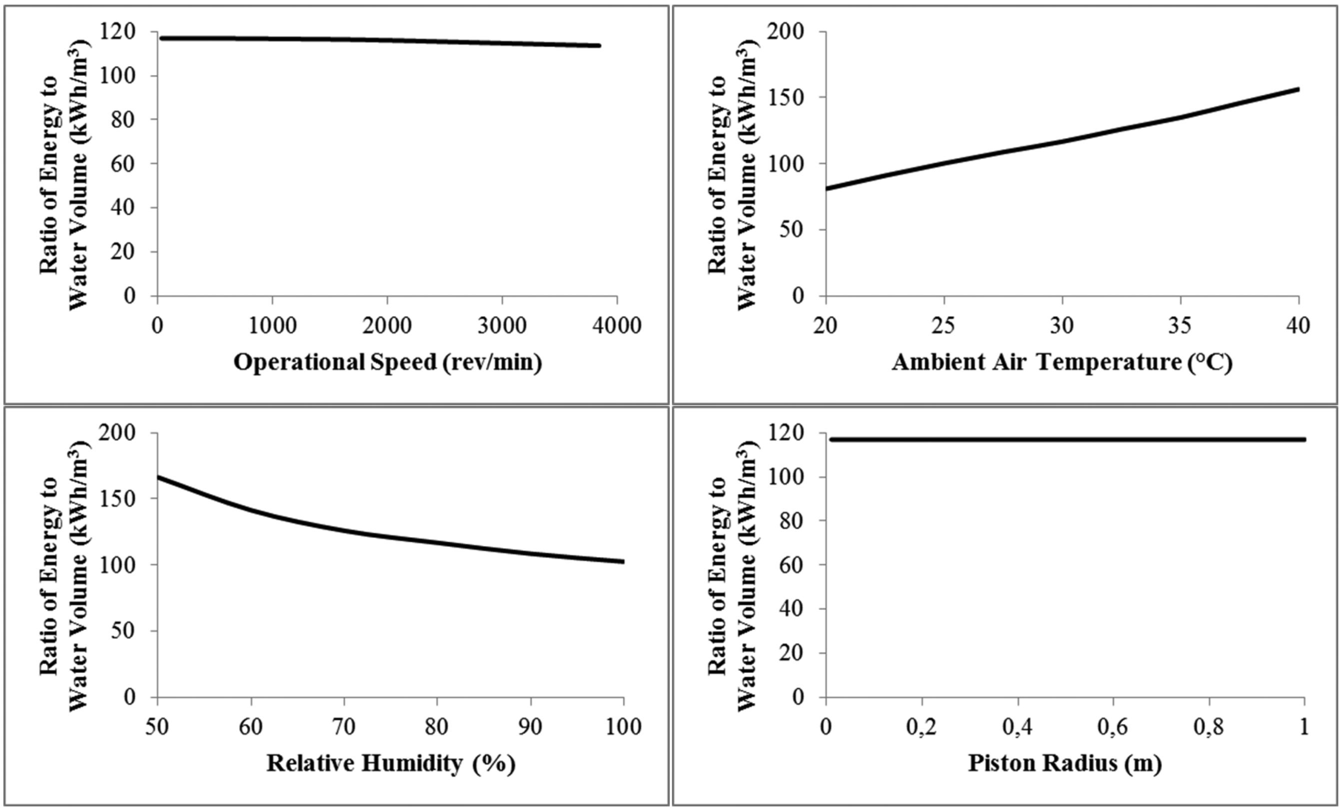

Figure 6: Variation of ratio of energy and water volume with operational speed, ambient air temperature, ambient relative humidity and piston radius (expander size).

of 101325 Pa, atmospheric temperature of 30°C and relative humidity of 80%. The following were found:

- The amount of air mass expanded in the expander per cycle was around 0.7 g. Since the angular speed was 60 rev/min, the expanded air mass flow rate was equal to 0.7 g/s.
- The fluid temperature was constant during suction and discharge. During expansion, it decreased to 0°C. From the process, around 11.5 g of water was condensed for every 1 kg of air expanded.
- Most of the power was consumed during expansion to overcome the pressure difference across the two sides of the piston. The average power per cycle was 3.374 W.
- The ratio of energy consumed and condensed water volume produced was 117 kWh/m^3. This was around 17% of the standard ratio of 681 kWh/m^3 in the literature.

A parametric study was conducted to complete the analysis. The focus of the analysis was on the ratio of energy and condensed water volume. From the parametric study, the following were found:

- The ratio was unaffected by the operational speed.
- The ratio increased linearly with the ambient air temperature.
- The ratio decreased with a higher relative humidity.
- The ratio was unaffected by the expander size.

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