

Calculations

for an

[Implementation of the Absorption Cooled Energy Tower](#)

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[Tucson Arizona, USA](#)

Desert Climate, Low Humidity, 6 months of sub-zero C temperatures

Document Version 1.0

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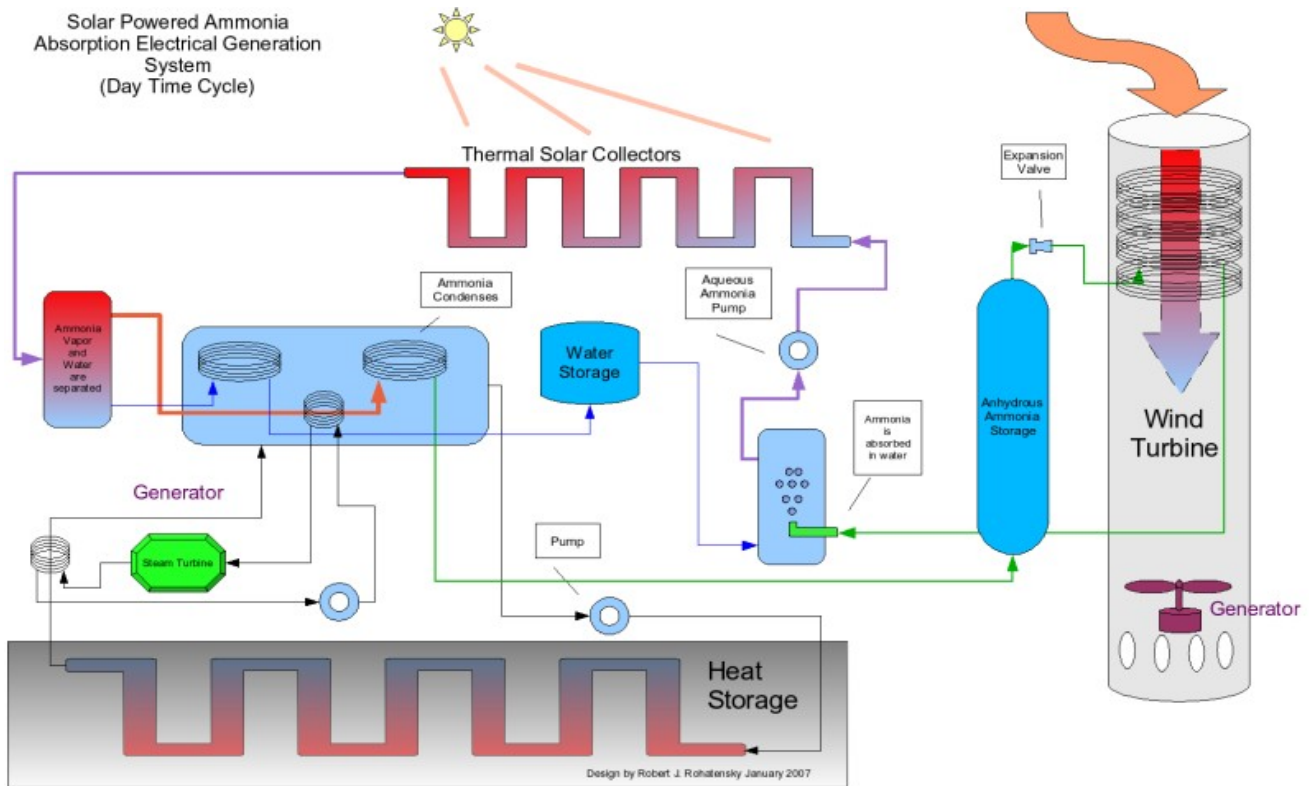
Overview

The climate, annual solar radiation and economic need for this location would best be suited by an bi-directional tower with a smaller scale daily heat storage. This location has low humidity, high daytime temperature and low nighttime temperature.

The tower structure would be hourglass shaped to increase air velocity through the wind turbine and include a moderate sized thermal storage to hold and extract heat on a daily cycle. The intake and outlet could rotate to prevailing winds. The system would include large trough solar collectors and a bore-hole thermal storage system operated throughout the year.

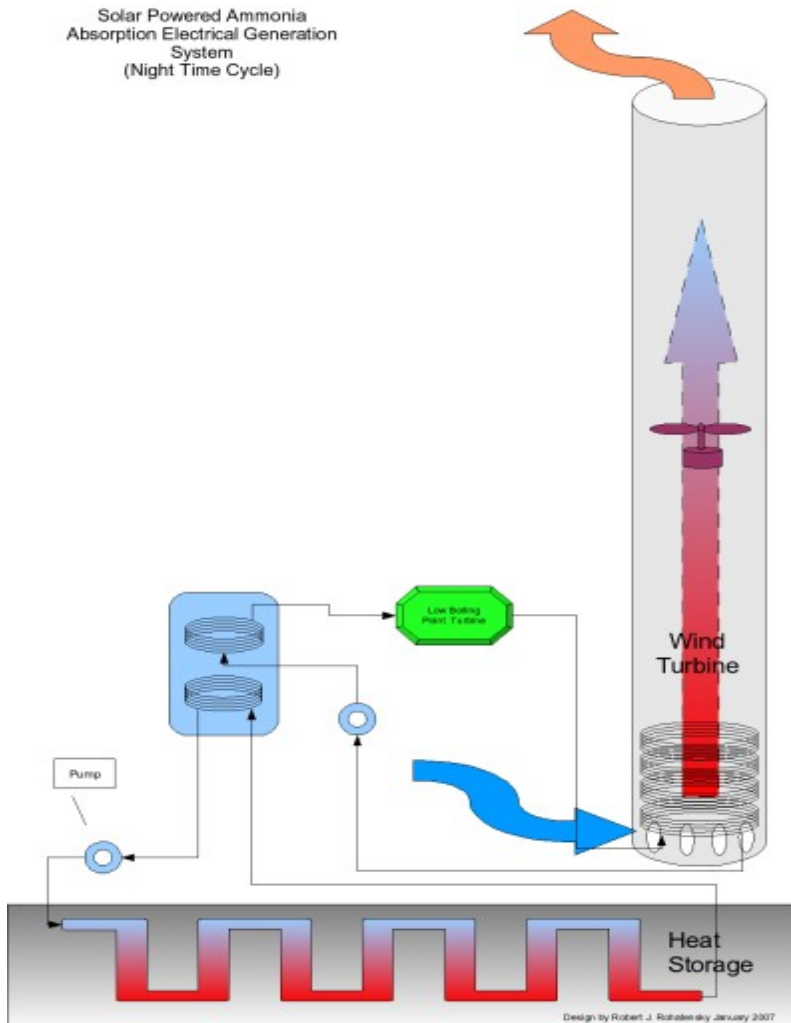
The day time system functions as a large scale solar powered ammonia absorption chiller. The air at the top of the tower is cooled with expanded ammonia causing the water vapor to condense and the air density to increase due to the lower temperature and humidity. The condensed water may be captured and utilized domestically, for irrigation or for hydrogen electrolysis. The ammonia is absorbed in water and then fractionally distilled with solar heat and re-pressurized. All of the heat from the air and distillation is stored in a large underground thermal storage system. At night the system is reversed and the heat is extracted from the thermal storage and converted to electricity in a system very similar to geothermal generation with the turbine air cooled causing an updraft in the tower.

Daytime system



Night System

At night the system would extract heat from the thermal storage and generate power while cooling off the thermal storage which increases efficiency of the day cycle.



The night operation is very similar to existing low gradient geothermal utilizing a low boiling point fluid steam turbine. There are 3 major efficiency improvements over existing systems like at [Chena Hot Springs](#).

1. The heat source is very close and has low pumping cost over deep geothermal.
2. The turbine is air cooled and the heat from the cooling causes an updraft in the tower. The low nightly air temperature gives a large thermal gradient.
3. Some additional energy is recovered in the wind turbine. The steam turbine is cooled by the night air and some of the energy in the convection is captured increasing the efficiency of the system as a whole.

Local Climate and Solar Data

NASA Surface meteorology and Solar Energy
At Latitude 32 and Longitude -111

Monthly Averaged Insolation Incident On A Horizontal Surface (kWh/m²/day)

Lat 32 Lon -111	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
10-year Average	3.1	3.82	5.05	6.47	7.18	7.26	6.29	5.76	5.26	4.2	3.4	2.74

Monthly Averaged Air Temperature At 10 m Above The Surface Of The Earth (° C)

Lat 32 Lon -111	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Annual Average
10-year Average	5.43	7.42	10	14.7	20.2	25.9	28.4	27.7	23.8	17.6	9.25	5.45	16.3

Average Daily Temperature Range (° C)

Lat 32 Lon -111	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
10-year Average	15.3	15.8	16.9	18.7	19	18.8	0	13.4	16.1	17.7	17.2	14.7

Average Minimum, Maximum and Amplitude Of The Daily Mean Earth Temperature (° C)

Lat 32 Lon -111	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Annual Amplitude
Minimum	-1.23	0.54	3.35	9.16	15.1	21.8	27.2	25.6	19.4	12.6	1.14	-2.89	
Maximum	13.4	15	17.2	23.1	30	36.5	37.8	36.9	34.3	25.7	17.9	12.7	
Amplitude	7.32	7.26	6.92	7.01	7.45	7.35	5.3	5.65	7.45	6.52	8.4	7.84	20.3

Monthly Averaged Atmospheric Pressure (kPa)

Lat 32 Lon -111	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Annual Average
10-year Average	88.3	88.1	88	87.9	87.9	87.9	88.1	88.2	88.1	88.2	88.1	88.2	88.1

Monthly Averaged Relative Humidity (%)

Lat 32 Lon -111	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
10-year Average	57.4	53.3	47.4	38.4	28.5	23.5	32.9	37.5	35.7	38.4	48.2	58.6

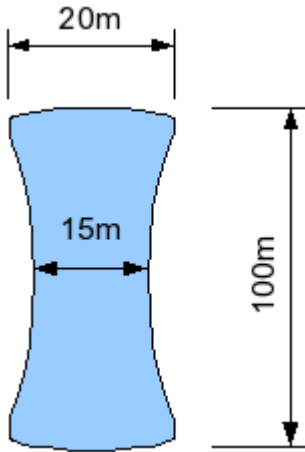
Yearly average of the daily maximum temperature: 25.04°C

Yearly average of the daily minimum temperature: 10.98°C

The power output of the wind turbine in the tower is based on the air velocity and the diameter of the tower. The air velocity is based on the buoyancy of the air external to the tower, the drag loss across the heat exchanger and tower walls and the exit loss. The buoyancy difference is based on the air temperature and absolute humidity difference. Because there is no energy expended in expanding the pressurized ammonia in the cooling coils, the power output of the system is based on the height and diameter of the tower, the surface area of the heat exchanger and the ambient air temperature and humidity.

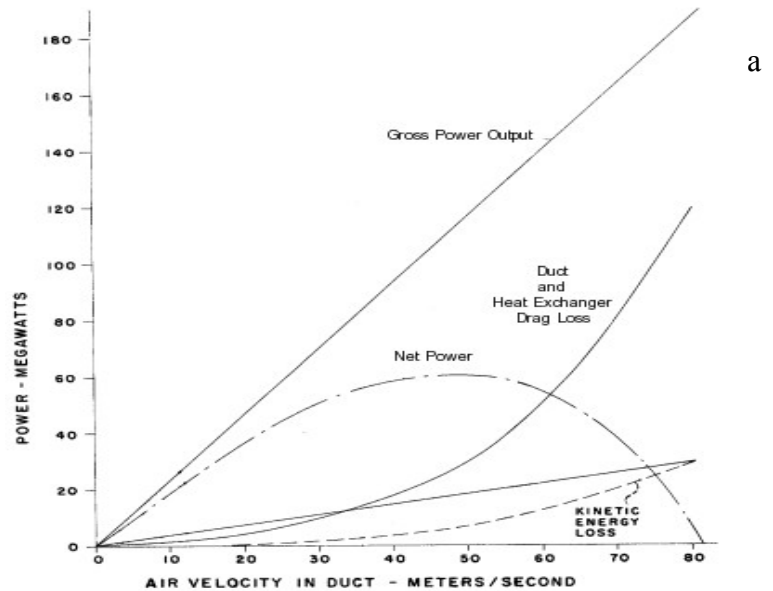
Assuming a medium scale system with a tower height of 100m and a diameter of 20m narrowing to 10m in the center. The tower will function bi-directionally seasonally and the wind turbine would be located at the center of the tower with a diameter of 15m.

Model System Sizing and Calculations



Wind Turbine Power Calculation (daytime)

This chart from patent US003894393 for the water spray tower has a peak net output with a wind velocity of 50 m/s. The ambient air temperature and tower height are too low to achieve this air velocity, but a sustained 20 m/s (72 km/h) is reasonable target air velocity. The system will be designed to control the ammonia flow and the resulting air cooling to keep the air velocity in the tower at a constant rate. A constant air flow will allow the system to have a reliable turbine rotation speed and to utilize AC alternators at a fixed frequency rather than DC systems and inverters for AC output. This lowers the cost of the generating system substantially over traditional wind turbines with varying RPM and DC generators.



Daytime Air Density and Flow Calculations

Target Air Velocity: **20m/s**

The temperature used for the daytime flow calculation is 80% of the annual average of the daily max temperature.

$$T = \text{annual average of daily maximum temperature}$$

$$T = \frac{25.04 \times 80}{100} = 20^\circ C$$

$$\text{Area of turbine: } A = \pi r^2$$

$$A = 7.5^2 \times 3.14159$$

$$176.71 m^2 - \text{hub area} = 150 m^2$$

$$\text{Air volume through turbine: } V_a = U \times A$$

$$20 m/s \times 150 m^2 = 3000 m^3/s$$

To achieve the desired 20 m/s air velocity 3000 cubic meters of air per second need to be moving through the turbine. The top and bottom of the tower are wider so although the same volume of air is moving into the top of the tower the air velocity is lower through the larger area. The exit loss where the output air meets the external air is reduced by having a larger area and lower velocity.

$$\text{Area of top inlet: } A = \pi r^2$$

$$A = 3.14159 \times 10^2$$

$$314.59 m^2 - \text{heat exchanger area} = 250 m^2$$

$$V_a = U \times A$$

$$3000 m^3/s \div 250 m^2 = 12 m/s$$

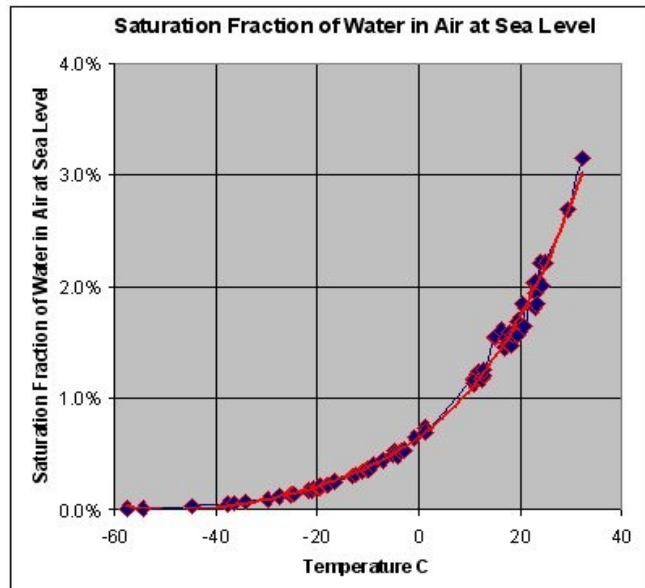
This assumes a heat exchanger horizontal surface area of ~ 64 m². Constructing the tower in an hourglass shape allows for the surface area of the inlet not to impede the airflow and cause a vacuum restriction through the heat exchangers.

The goal is to remove the required amount of heat from the air at the top of the tower to cause the negative buoyancy to achieve the required air velocity across the tower height. In this model the condensed air is allowed to drip off the cooling coils and is ignored in the calculation.

The formula for air density is:

$$\rho = \frac{p}{R \times T}$$

The specific gas constant "R" for dry air is:



$$R_{dry,air} = 287.05 \frac{J}{kg \times K}$$

The target is to lower the 20°C average ambient air to 0°C. For dry air the density at 88.1 kPa at 20°C:

$$\rho = \frac{88100}{(287.05 \times 297.13)} \rho = 1.033 \text{ kg/m}^3$$

Water vapor pressure:

$$E_s = 611 \times 10.0^{(7.5 \times 16.3 / (237.7 + 20))}$$

$$E_s = 1821 \text{ Pa}$$

Actual vapor pressure:

$$E = (RH \times E_s) / 100$$

$$E = (40 \times 1821) / 100$$

$$E = 728 \text{ Pa}$$

Absolute Humidity:

$$D = P / (T \times R_w)$$

$$R_w = \text{gas constant for water vapor} = 461.5 \text{ (J/kg} \cdot \text{K)}$$

$$D = 728 / (287.45 \times 461)$$

$$D = 0.0055 \text{ kg/m}^3$$

For 20°C air at 40% relative humidity, the air is approximately 0.6% water vapor with a density of 0.804 kg/m³. This offsets the density by 0.804*0.006:

$$1.033 \times 0.994$$

$$+ 0.804 \times 0.006$$

$$= 1.031 \text{ kg/m}^3$$

For dry air the density at 88.1 k Pa at 0°C:

$$\frac{88100}{(287.05 \times 273.15)} = 1.124 \text{ kg/m}^3$$

Although the relative humidity of the air in the tower after cooling is 100% (below outside air dew point) the 0°C temperature means that the absolute humidity is near the same as the 40% relative humidity input air at around 0.06% and the water vapor density offset is near the same as the warm input air.

$$1.124 \times 0.994 + 0.804 \times 0.006 = 1.122 \text{ kg/m}^3$$

	External	Internal
External Air Temperature	20°C (293.15°K)	0°C (273.15°K)
Relative Humidity	40%	100%
Air Pressure (586m + 100m above sea level)	81.100 k Pa	81.100 k Pa
Air Density ρ	1.031 kg/m ³	1.122 kg/m ³

The volume of the air inside the tower is 2 times the volume of a truncated cone:

$$V = \Pi (R^2 + rR + r^2) h / 3 \times 2$$

$$V = \Pi \times (10^2 + 10 \times 7.5 + 7.5^2) \times 50 \times 2$$

$$V = 72649.33 \text{ m}^3$$

The available potential energy is the negative buoyancy of the denser cold air inside the tower relative to the air outside the tower.

$$F = V \times (\rho_i - \rho_o) \times 9.8$$

$$F = 72649.33 \times 1.189 - 72649.33 \times 1.121 \times 9.8$$

$$F = 4940.15 \times 9.8$$

$$F = 48413.5 \text{ N}$$

The average velocity of the air moving down the tower without drag:

$$v_a = \frac{\sqrt{2gd}}{2}$$

$$V_a = \frac{\sqrt{2 \times 9.8 \text{ m/s}^2 \times 100 \text{ m}}}{2}$$

$$V_a = 22.14 \text{ m/s}$$

The peak velocity of the air after it falls 100m (excluding exit loss):

$$v_a = \sqrt{2gd}$$

$$V_a = \sqrt{2 \times 9.8 \text{ m/s}^2 \times 100 \text{ m}}$$

$$V_a = 44.27 \text{ m/s}$$

The assumption is that although the wind turbine is located in the center of the tower vertically, and the air velocity at the center would be 1/2 of the velocity at the bottom of the tower (if there was no exit loss). The exit loss is substantial and is caused by the downdraft air having to push the static air at the bottom. With exit loss, the real velocity of the air will be much less than the peak velocity. This is complicated to understand where the ideal location for the turbine would be, but the bi-directional nature of the design lends to having the turbine in the vertical center of the tower so the average velocity of the air is used in the calculations and it is assumed that this is a reasonable estimate.

The gross power is the flow rate in kg/s (of buoyancy) times 9.8 times height.

$$W = \text{flow rate} \cdot g \cdot h$$

$$W = (1.122 - 1.031) \times 3000 \times 9.8 \times 100$$

$$W = 267540 = 270 \text{ kW gross power}$$

Or using a wind turbine formula based on the 22.14 m/s wind speed with a 0.33 turbine power efficiency the actual power is:

Power delivered = C_p x area of wind turbine x $\frac{1}{2} \rho v^3$

$$P = 0.33 \times 150 \times \frac{1}{2} \times 1.122 \times 22.14^3$$

$$P = 301 \text{ kW}$$

In other words, the power available is the difference in density (total mass) in the cold air inside the tower relative to the air outside the tower and how fast it is falling. The efficiency of the wind turbine is much better than a conventional natural wind turbine due to much lower tip loss because the turbine is in a duct. The assumption is that with a well designed turbine much more of the energy could be converted to electricity. The air in the tower can also be routed into a vortex to increase the angle of attack and efficiency.

The electrical power output of the wind turbine in the tower is ~ 200kW consistently 24 hours per day.

Alternate Flow Calculations Using the Stack Effect Formula

Another method of calculating the air flow through the tower is by using the Stack Effect formula:

$$Q = CA \sqrt{2gh \frac{T_h - T_c}{T_h}}$$

Q = stack effect flow rate, m³/s

A = flow area, m²

C = discharge coefficient (usually taken to be from 0.65 to 0.70)

g = gravitational acceleration, 9.8 m/s²

h = height, m

T_h = warm temperature, K

T_c = cold temperature, K

This will give the flow rate in m³/s, but doesn't take into account differences in absolute humidity.

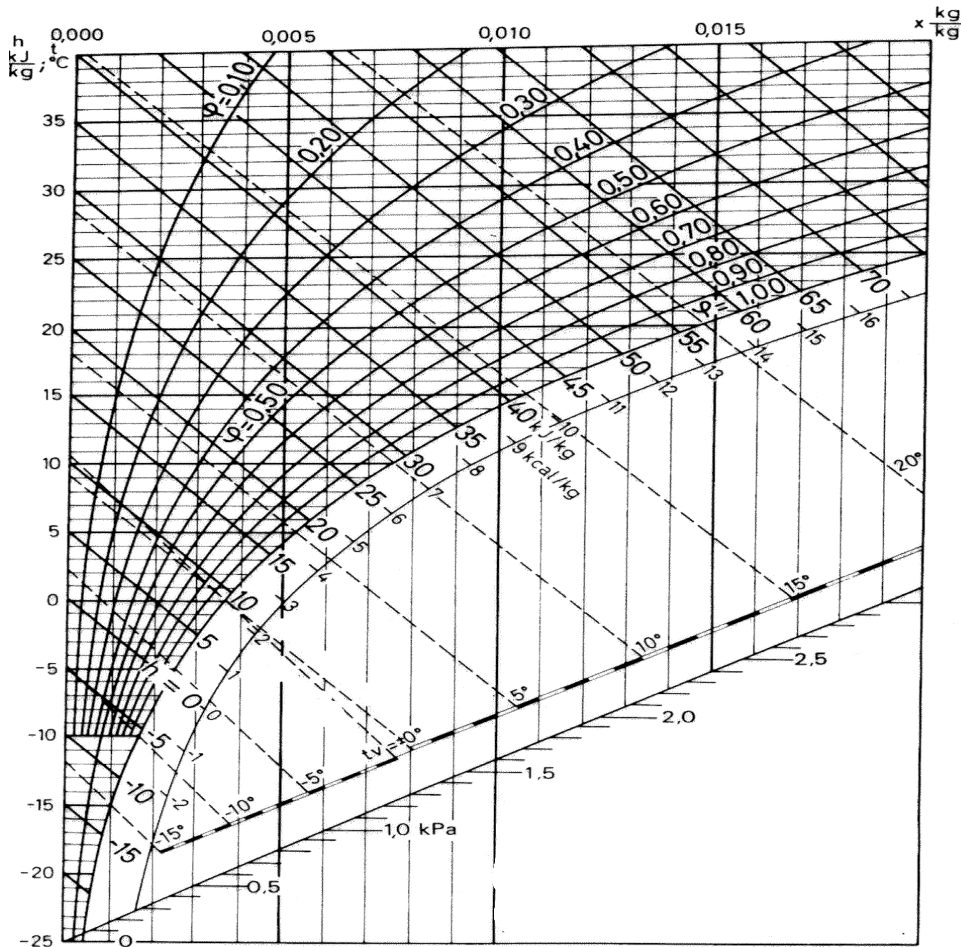
$$0.65 * 314 * \sqrt{2 * 9.8 * 100 * \frac{(293.15 - 273.15)}{293.15}} = 2360 \text{ m}^3/\text{s}$$

The exact calculation of the power output of the wind turbine is complicated and there are a lot of fluid dynamics involved. The Second Law of Thermodynamics limits the amount of power out of the wind turbine to the amount of power put into the heat pump minus efficiency loss. An estimate of < 50% of the solar energy input of the heat pump should be reasonable for the electrical power output of the wind turbine.

Estimated daytime electrical power output of wind turbine
12h X 250kW = 3000 kWh/day
(annual average)

Air Cooling Calculation (daytime)

Mollier Diagram



Based with an ambient air temperature of 16.3°C and relative humidity of 40% across the tower height of 100m to lower the temperature of 3000m³/s of air requires removal of 16.3° X 3000m³ X 1.058 kg/m³ with air having a specific heat of 1.012 J per g per K or 1012 J per Kg per K

1 m³/s of air at 16.3°C and relative humidity 40% (A) is cooled down to 0°C (B). The surface temperature of the cooling coil is -20°C (C). The density of air at 16.3°C is 1.058 kg/m³ 81.1 kPa.

Using the [Mollier diagram](#) the state of the cooled air (B) is in the intersection between the straight line between (A) and (C) and the 15°C temperature line.

From the Mollier diagram the enthalpy in (A) is 20 kJ/kg, in (B) 5 kJ/kg and in (C) -15 kJ/kg. The Contact Factor can be calculated as:

$$\beta = \frac{(20 \text{ kJ/kg}) - (5 \text{ kJ/kg})}{(20 \text{ kJ/kg}) - (-15 \text{ kJ/kg})} = 0.429$$

The total heat flow can be calculated as:

$$q = (1 \text{ m}^3/\text{s})(1.058 \text{ kg}/\text{m}^3)((20 \text{ kJ}/\text{kg}) - (5 \text{ kJ}/\text{kg})) \\ = 15.87 (\text{kJ}/\text{s}, \text{kW})$$

The sensible heat flow can be calculated as:

$$q_s = (1 \text{ m}^3/\text{s})(1.058 \text{ kg}/\text{m}^3)(1.01 \text{ kJ}/\text{kg}/^\circ\text{C})(16.3^\circ\text{C} - 0^\circ\text{C}) \\ = 17.42 (\text{kW})$$

According to the Mollier diagram the specific humidity in (A) is 0.096 kg/kg and in (B) 0.001 kg/kg and the latent heat flow can be calculated as:

$$q_s = (1 \text{ m}^3/\text{s})(1.058 \text{ kg}/\text{m}^3)(2,502 \text{ kJ}/\text{kg})((0.004 \text{ kg}/\text{kg}) - (0.003 \text{ kg}/\text{kg})) \\ = 2.1 (\text{kW})$$

At 3000m³/s the total heat removed from the air is:

$$15.87 \times 3000 \text{ m}^3/\text{s} \\ = 47,610,000 \text{ j}/\text{s} \\ = 47,610 \text{ kW} \\ = 47 \text{ MW}$$

To drop the 3000m³/s of air by 16.3°C requires approximately 16 kW/m³
=
47 MW of heat removed from the ambient air moving at 3000 m³/s

Pressurized Anhydrous Ammonia Requirements

The heat capacity of ammonia vapor is 35.06 J/mol K with molar mass of 17.0304.

$$\frac{35.06}{17.0304} = 2.059 \text{ J/g K}$$

The latent heat of ammonia is 1369 J/g K with a boiling point at atmospheric pressure of -33°C.

To remove 47 MW of heat with liquid ammonia changing state and being warmed to -13°C, the latent heat is approximately 600x the specific heat.

To raise ammonia vapor 20°K it takes $597.085 \text{ J} \times 20 / \text{g} = 11940 \text{ Joules per gram}$.

$$\begin{aligned} g/s \times \text{specific heat} \times K + g/s \times \text{latent heat} \\ 47,610,000 &= Q \times 20 \times 2.059 + Q \times 1369 \\ \frac{47,610,000}{Q} &= 41.2 + 1369 \\ \frac{47,610,000}{Q} &= 1410 \\ Q &= 33,761 \text{ g/s} \end{aligned}$$

To remove 47 MW of heat from the air:

33,761 g/s of liquid ammonia must be evaporated and heated 20°K (to -13°C)

Liquid ammonia density is 681.91 g/L

The pressurized liquid ammonia input is 4068g/s at 681.91 g/L

$$33761 / 681.91 = 49.5 \text{ L/s}$$

49.5 litres/s of liquid ammonia to transfer 47MW of heat from 3000m²/s of air

Operating 12 hours per day is a total consumption of $49.5 \times 12 \times 60 \times 60 = 2 \text{ million litres per day}$ of liquid ammonia.

Ammonia Recovery System

The solar collectors would need to boil enough aqueous ammonia to meet the 5.24 L/s constant intake during sunlight hours.

Ammonia dissolves in water at 89.9 g/100 ml at 0C

To absorb the ammonia vapor at the 33 kg/s rate is $33761 / 899 = 37.5 \text{ L/s}$ of cold water.

The absorption of ammonia in water is exothermic (gives off heat), as the ammonia vapor is dissolved the aqueous ammonia solution increases in temperature and absorbs all of the heat energy in the ammonia vapor. The temperature of the water increases and the aqueous ammonia remains at its vapor point. If there is no increase in pressure, any increase in aqueous ammonia temperature will cause the ammonia to boil.

The aqueous ammonia is pumped to approximately 200 psi pressure prior to entering the heating stage. The pump uses energy but substantially less than a compressor due to the smaller volume. The amount of heat to extract the ammonia is the latent heat of the output ammonia which is 2 million litres or 1.4 million kg per day plus the heat required to raise the water the same temperature. The aqueous ammonia solution is at approximately 50% and should be at the vapor point at all times.

The heat to boil off the ammonia is:

$$\begin{aligned} &= 33,761 \text{ g/s} \times 1369 \text{ J/g K} \times 12\text{h} \\ &= 46.2 \text{ MJ/s, MW (12h)} \\ &= 554 \text{ MWh per day} \end{aligned}$$

Local Solar radiation average 6kWh/m².

$$554,000 \text{ kWh} / 6 = \sim 92,333 \text{ m}^2 \text{ of solar panel}$$

$$\begin{aligned} &\text{Solar panel daily input} \\ &92,333 \text{ m}^2 \times 6 \text{ kWh/m}^2 \\ &= \\ &\mathbf{554 \text{ MWh / day}} \text{ solar thermal input} \end{aligned}$$

Water Output

In a low humidity environment the cooling coils don't produce enough water to use it to produce electricity, but the source of clean water is very valuable in an arid climate. If the heat exchanger is built from stainless steel or glass coated, the water is clean and distilled quality and can be used domestically, for irrigation or for hydrogen electrolysis. The water is also produced high in the tower and may be delivered at a reasonable pressure without pumping energy.

At airflow of 3000m³/s of 16.3°C 40% relative humidity air, the tower will extract almost all of the available water vapor.

Saturated Vapor Pressure:

$$E_s = 611 * 10.0^{(7.5 * 16.3 / (237.7 + 16.3))}$$
$$E_s = 1850.7 \text{ Pa}$$

Actual Vapor Pressure:

$$E = (RH * E_s) / 100$$
$$E = (40 * 1854) / 100$$
$$E = 740 \text{ Pa}$$

Absolute Humidity:

$$D = P / (T * R_w)$$

R_w = gas constant for water vapor = 461.5 (J/kg·K)

$$D = 740 / (297.15 * 461)$$
$$D = 0.005 \text{ kg/m}^3$$

Absolute humidity of 16.3°C 40% RH=0.005kg/m³

$$3000 \text{ m}^3 / \text{s} * 0.005 = 45 \text{ kg/s}$$

15L/s of distilled water

The system will produce **15 L/s of distilled water** or almost 640,000 litres per day (daytime only) at a height of 100m.

If the water drives a water turbine at the base of the 100m tower, the gross power is:

$$W = \text{flow rate} * g * h$$
$$W = 15 * 9.8 * 100 \text{ m}$$
$$W = 14.7 \text{ kW gross power}$$

With the efficiency of a water turbine at 50% the net power from the condensed water:

$$W = 44.1 \text{ kW} * 0.5$$
$$W = 7.5 \text{ kW Net Electrical}$$

The condensed water capture:

15L/s (4 million L/day) at 100m
which can generate:
7.5kW (90kWh/day) Net

A water turbine probably wouldn't be feasible in this environment, but the clean water is very valuable.

Night System Power Calculation

The night system is very similar to a conventional geothermal with the addition of air cooling. In an arid location the night air cools off on average 12°C from the daytime temperature.

The temperature used for the nighttime power calculation is 120% the annual average of the daily minimum temperature.

$T = \text{annual average of daily minimum temperature}$

$$T = \frac{10.89 \times 120}{100} = 13^\circ \text{C}$$

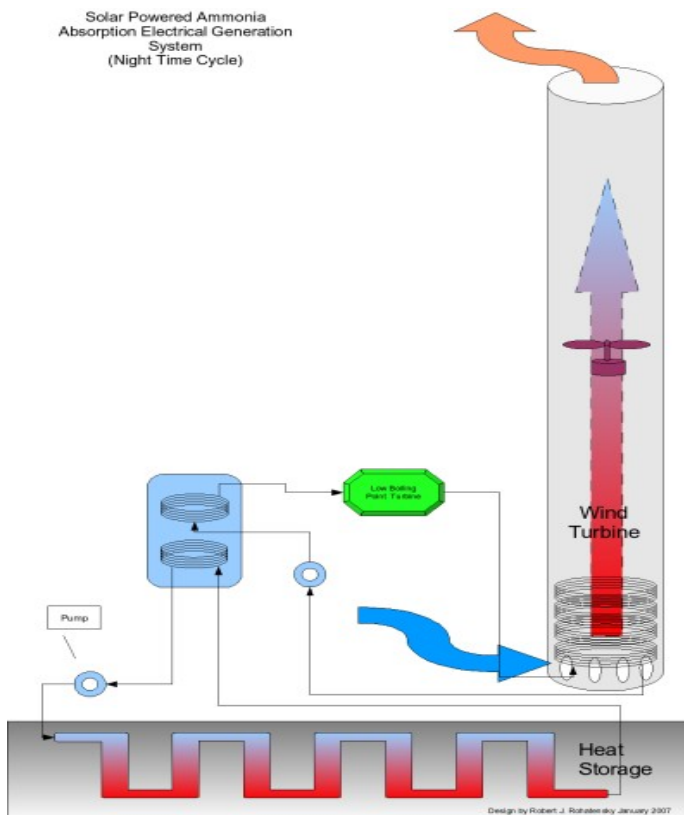
In this system during the daytime cycle 47MWh of heat is removed from the air and 60MWh of heat is collected in the solar collectors. All of this heat is stored in the underground thermal storage over the 12 hours of daylight.

$H = \text{daily thermal storage}$

$$H = (47\text{MW}_{(t)} + 60\text{MW}_{(t)}) \times 12$$

$$H = 1116 \text{ MWh}_{(t)} / \text{day}$$

Solar Powered Ammonia
Absorption Electrical Generation
System
(Night Time Cycle)



Depending on the design of the thermal storage there will be losses with heat dissipating into the earth, as an estimate 50% of the stored heat can be used at night. The intent of the night system is to generate some power while “cooling off” the thermal storage.

The average temperature of the thermal storage is difficult to calculate without knowing the material and transfer system. The input heat from the solar collectors could approach 100°C, but the input heat from condensing the ammonia will be lower.

The system functions like existing low-gradient geothermal and the geothermal turbine and wind tower calculations will be estimated at ½ the daytime output with the intent of cooling off the thermal storage as much as possible with the 13°C night air temperature.

If the thermal storage is 90°C and the ambient air is 13°C, the maximum Carnot efficiency of the whole system (both the geothermal and wind turbine) is:

Maximum Carnot Efficiency for Complete Night System

$$\eta = 1 - \frac{T_c}{T_H}$$

$$\eta = 1 - \frac{(13 + 273.15)}{(90 + 273.15)}$$

$$\eta = 0.23$$

There is 1.1GWh_(t) per day of thermal storage from the daytime system with an assumption that the night system can convert 20% of the maximum Carnot Efficiency (4% of thermal transfer) into electricity and accounting for a 20% loss of heat from the thermal storage.

$$P = (47MW_{(t)} + 46 MW_{(t)}) \cdot 12h \cdot 80\% \cdot 4\%$$

$$P = 1116 MWh_{(t)} \cdot 80\% \cdot 4\%$$

$$P = 35 MWh_{(e)} \text{ per day}$$

The 4% thermal to electrical energy conversion is a *very* low estimate and should be much higher with an efficient system.

Night Complete System Electrical Power Output

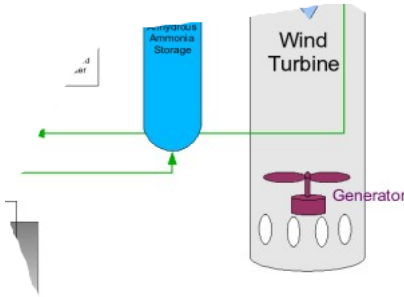
2.98MW x 12h

35 MWh per day

System Losses

There direct losses in thermal conversion are attempted to be factored into the calculations. There are additional losses in the system in pumping the aqueous ammonia, cycling the working fluid in the thermal storage, thermal collection and electrical generation. It would be possible to actively cool the electrical components to increase efficiency. The electrical power output is constant and much higher than any direct solar system with an output of > 100% of the solar input.

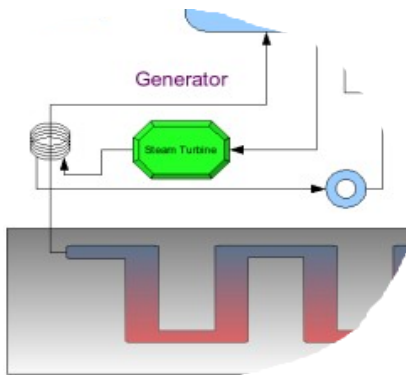
Electrical and Thermal Output



Daytime wind turbine output = $200 \text{ kW} \times 24\text{h} =$
4.8 MWh(e) per day gross output

Daytime gross thermal storage = $(47 \text{ MW}_{(t)} + 46\text{MW}_{(t)}) * 12\text{h}$
1.1GWh per day of thermal storage

The $1.1\text{GWh}_{(t)}$ is extracted nightly, the total thermal transfer = $1.1\text{GWh}_{(t)}$
 $\times 2 =$ **2.2GWh_(t) / day total thermal transfer**



During the day, the second heat recovery turbine should convert 4% of the the heat going into thermal storage into electricity during daylight hours.

$93\text{MW} \times 2\% \times 12 = 44 \text{ MWh(e) per day gross}$
 (during daylight hours, the night system is calculated separately)

At night the whole system produces: **35 MWh per day**

Total System Statistics

- Daytime wind turbine electrical output: **4.8 MWh_(e) / day**
- Daytime heat recovery turbine output: **44 MWh_(e) / day**
- Night system total output (includes both turbines): **40MWh_(e)**
- **Total electrical output: 40 MWh_(e) / day**
- Thermal storage: **1.1GWh of heat / day** (moved twice)
- 300 m tower
- 10,000 m² trough concentrated solar collectors
- medium thermal storage (vertical bore-hole 100m deep 200m x 200m= 4,000,000 m³)
- 20,000 litres pressurized anhydrous ammonia storage
- 20,000 litres water storage
- 20,000 litres aqueous ammonia storage