

Title:
The Simplified REHOS Cycle
(Patent Pending)

Author:
Johan Enslin

Company:
Heat Recovery Micro Systems CC

Country:
South Africa

Economic Viability of Low Temperature Heat Recovery.

Heat extraction from low temperature waste heat sources is not difficult, but it borders on the uneconomical due to the low heat to power conversion efficiency of the current state-of-the-art heat recovery systems. Using the geothermal classification, heat sources $>200^{\circ}\text{C}$ are regarded as high and conversion efficiency may reach $> 25\%$, while heat sources in the $150^{\circ}\text{C} - 200^{\circ}\text{C}$ are classified as moderate and conversion efficiencies range $20 - 25\%$, while a heat source in the range $100^{\circ}\text{C} - 150^{\circ}\text{C}$ is regarded as low and heat to power conversion efficiencies are only $10 - 20\%$. Heat sources below 100°C are regarded as uneconomical to convert to power due to the low heat-to-power conversion efficiency, dropping below 10% , and is used for non-electrical applications only. Conversion efficiency in general are limited by the second law of thermodynamics:

$$\eta = \frac{Q_{in} - Q_{rej}}{Q_{in}} = \frac{Work}{Q_{in}}$$

where Q_{in} = *heatinput* , and Q_{rej} = *heatrejection* (in rankine and organic rankine cycles the latent heat of condensing the low pressure exhaust vapor back to liquid). With a conversion efficiency of only 10% , the latent heat in the exhaust represent 90% of the input heat, and therefore the condenser cooling need huge cooling towers which are very expensive and waste a lot of water (for wet cooling) that is made to evaporate for the cooling effect. Low conversion efficiencies also decrease the power density of the converting machine, and also increasing the required specific capital investment.

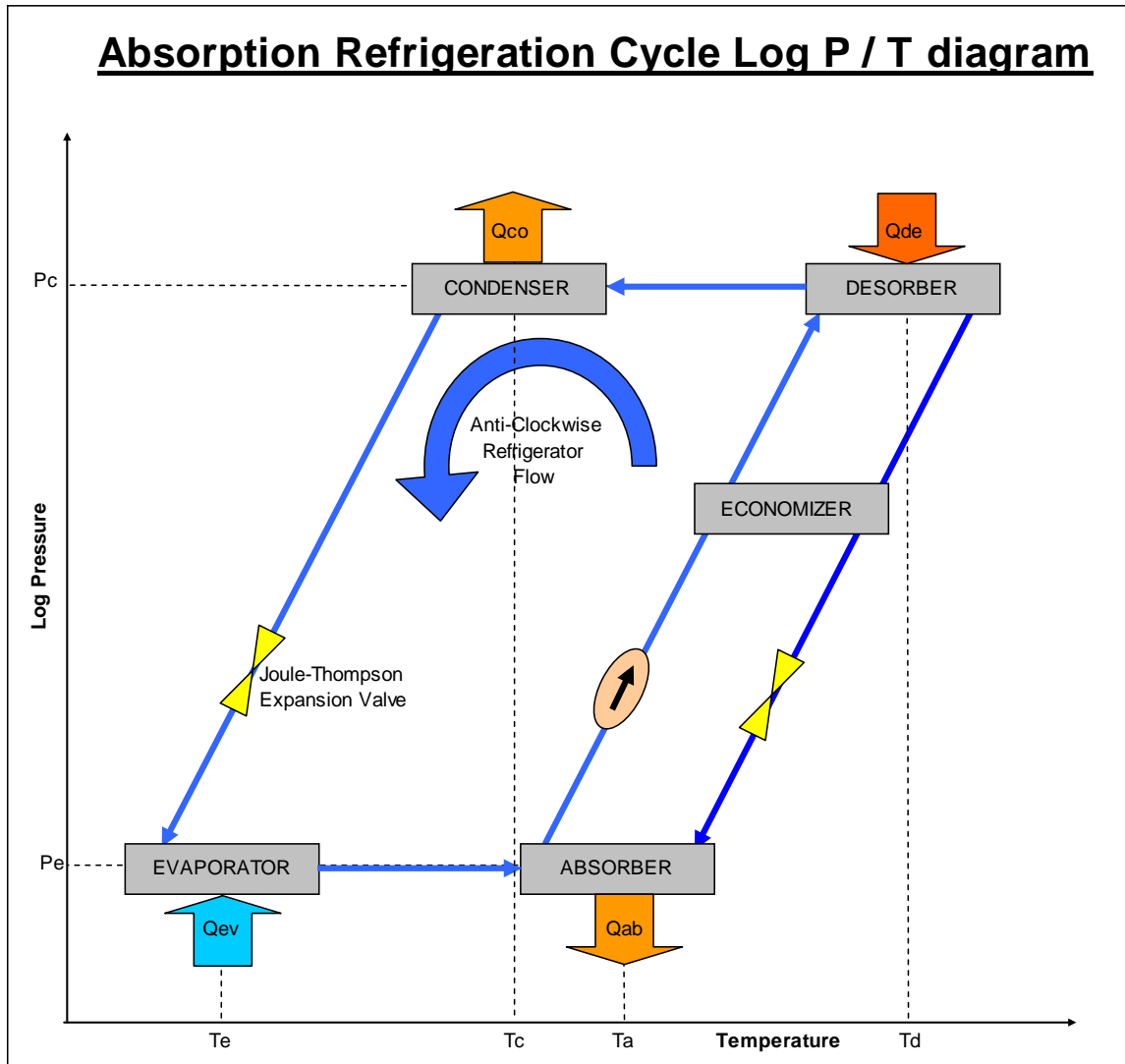
Only the economy of scale may offset some of these low efficiency converters eg. the very large OTEC converters built in the sea which convert the heat in some 20°C surface water using cooling with cold 4°C deep-flowing water streams with conversion to power efficiencies $< 5\%$. These installations are extremely large and capital extensive, as they are designed for the MW range to be economical, but not many exist around the globe, mainly due to the large capital investments and the low return on investments (ROI) required.

Heat Recovery from the Environment using Intermediate Temperature Waste Heat.

Looking at some heat recovery and conversion to power technologies, let us start with the ammonia-water absorption thermodynamic cycle used as a heat-driven refrigerator. The standard binary absorption refrigeration cycle is well known in refrigeration circles already since the fifties. The absorber and evaporator is on the same low pressure, while the desorber and condenser operate on higher pressure. This binary cycle may be made up of ammonia (NH_3) in aqua mixtures (or other mixtures like $\text{LiBr-H}_2\text{O}$ or $\text{CO}_2\text{-H}_2\text{O}$ and many others). In the rest of this document we assume we talk about $\text{NH}_3 - \text{H}_2\text{O}$ mixtures only, to keep the explanation simple.

Looking at the Pressure-temperature layout sketched in Figure 1 below, we notice that strong NH₃ in aqua solution (assume 40% NH₃ in aqua) is pumped from the low pressure (assume $P_e = 3.57$ Bar absolute) absorber via the economizer heat exchanger to the higher pressure (assume $P_c = 10$ Bar abs) desorber, where relatively high temperature (assume $T_d = 90^\circ\text{C}$) heat boil off some nearly pure NH₃ vapor, which flow to the condenser on the same high pressure level ($P_c = 10$ Bar abs), where it is condensed by removing or rejecting some heat to cooling water at the intermediate temperature (assume $T_c = 30^\circ\text{C}$).

Figure 1



The remaining NH₃ in aqua in the desorber after boiling off the NH₃ vapor is a lean mixture (assume 35% NH₃ in aqua) flowing via the economizer heat exchanger back to the absorber at the lower intermediate temperature (assume $T_a = 45^\circ\text{C}$). The pressure is dropped to the lower pressure by a restrictor valve in the lean liquid mixture line. The economizer is just a simple heat exchanger, cooling the returning lean mixture by heating the pumped, upwards flowing rich mixture. High pressure condensed liquid NH₃ from

the condenser expand through the Joule-Thompson expansion valve isenthalpically to the cold ($T_e = -2^\circ\text{C}$) evaporator at low pressure. Evaporating liquid NH_3 in the evaporator absorb heat as refrigeration from the environment as the extraction temperature may be quite low ($T_e = -2^\circ\text{C}$) if the saturation pressure (P_e) is 3.56 Bar absolute. The intermediate heat rejection temperatures are normally kept in the $30^\circ\text{C} - 50^\circ\text{C}$ region, to make sure cooling water slightly above ambient, at say $25^\circ\text{C} - 35^\circ\text{C}$ can absorb the rejected heat from the cycle. The desorber high temperature is determined by available waste heat temp, and may be some $70^\circ\text{C} - 90^\circ\text{C}$ or higher. If we assume the temperature (T_d) to be 90°C the pressure would rise to about 10 Bar absolute, which is the 35% NH_3 in aqua solution saturation pressure.

Although the heat extracted at -2°C from the environment is used to vaporize the liquid NH_3 to vapor (also at -2°C) the absorber temperature is kept higher, in this case 45°C . The process of absorbing NH_3 into a sub cooled lean mixture of NH_3 in aqua is exothermic and the absorption heat would raise the temperature of the absorber if it is not cooled sufficiently, carrying away the rejected heat. The sensible heat required to heat the cold vapor from (-2°C) to the absorber temperature of 45°C amount to 266 kJ/kg, but the heat generated in the absorber by absorbing the heated vapor into the 40% NH_3 in aqua mixture calculate to 1398 kJ/kg. The sensible heating of the vapor therefore amount to only 19% of the absorption heat! Incidentally, if the pure NH_3 vapor was condensed at 45°C instead of absorbed, the heat generated would have been 1154 kJ/kg, which is 83% of the absorption heat. The heat of solution (HOS) is therefore $1398 \text{ kJ/kg} - 1154 \text{ kJ/kg} = 244 \text{ kJ/kg}$ under these pressure, temperature and solution concentration levels.

Pumping Waste Heat to Higher Temperatures to gain Power Cycle conversion Efficiency.

Perhaps a better example of how the heat of solution assist to create higher temperatures can be seen in the Absorption Heat Transformer (AHT). A typical example is sketched as Log Pressure vs. Temperature graph in Figure 2 below.

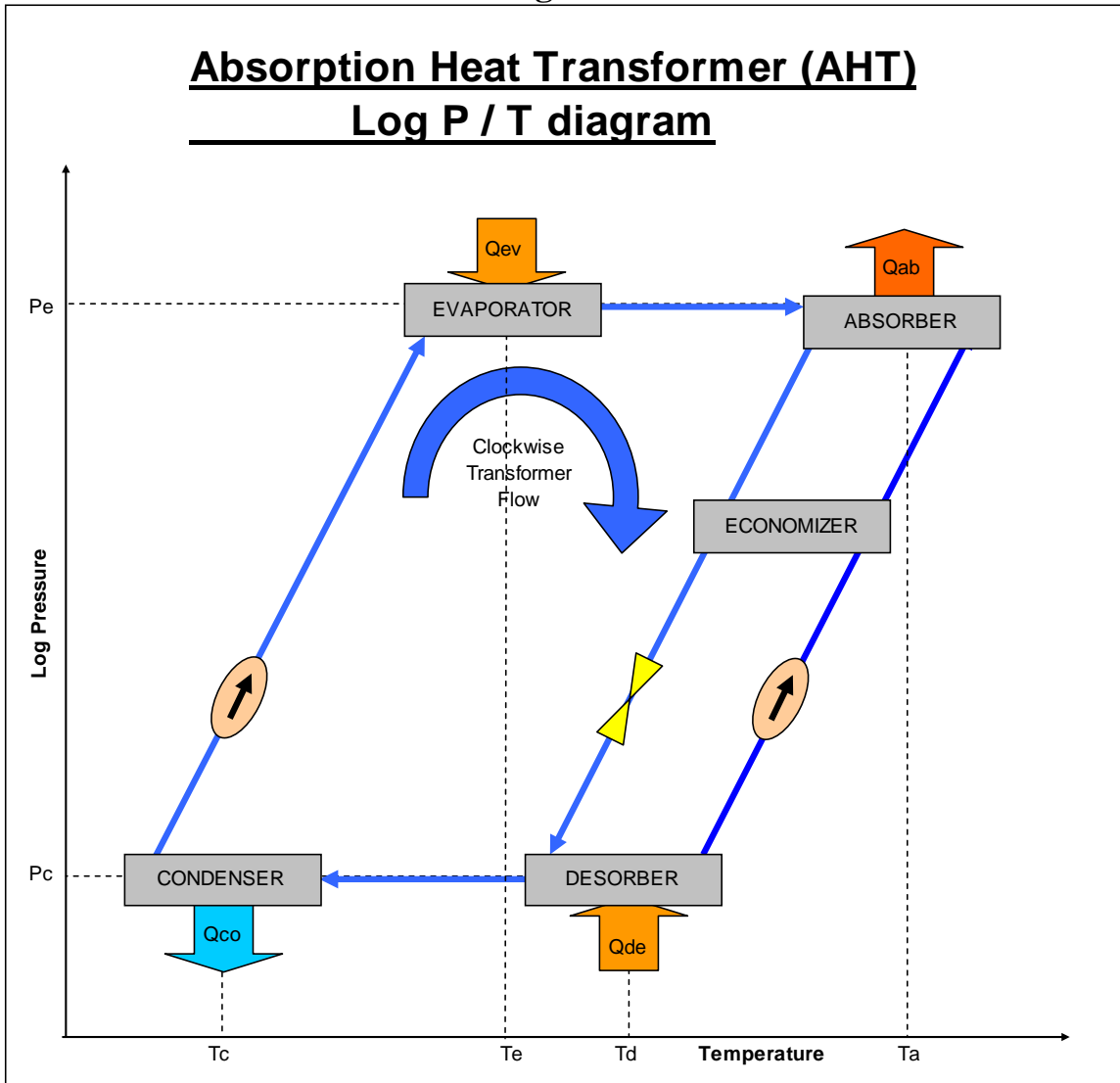
Heat transformers are used to raise temperature levels of large waste heat sources eg. a solar pond, to values more conducive to higher efficiencies of converting the heat to power.

In a solar pond, heat is trapped by a salt gradient which reverse the temperature-density relationship. High temperature heavily salted water has a higher density than the less salty, lower temperature water solution that rise to the top of the solar pond. Heat at temperatures of $60 - 80^\circ\text{C}$ generated by the solar radiation is accumulated in the bottom of the solar pond, trapped by the colder layer of surface water. A heat transformer may be used to pump a very large proportion of the heat at this temperature up to 120°C or more for use in an Organic Rankine Cycle waste heat recovery system operating at 18% efficiency instead of the 9% of the original $70^\circ\text{C} \Rightarrow 30^\circ\text{C}$ system, and this transformer technology has been matured over the past 20 years or so.

Looking at Figure 2, notice that the desorber and condenser is now at the low pressure level (P_c), while the evaporator and absorber is now at the higher pressure level (P_e).

Although the liquid and vapor flows are now flowing clockwise, the operation is basically the absorption refrigeration cycle operated in reverse, and instead of absorbing heat from the environment, it deliver heat at higher temperature. Here we see high

Figure 2

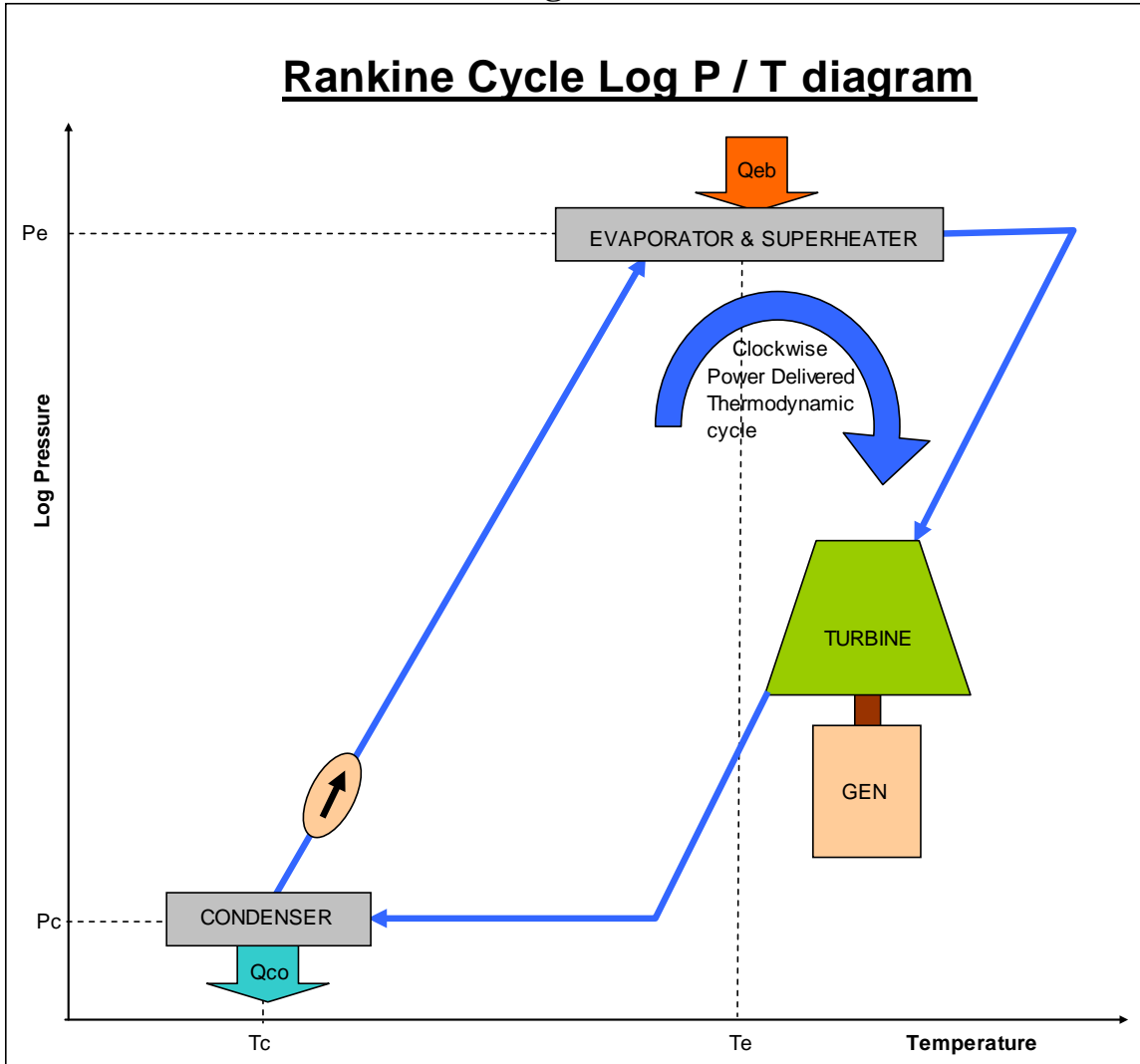


pressure vapor from the evaporator flow to the higher temperature absorber (at the same pressure P_e) and heat the absorber to higher temperature. In practice a very large percentage of the available intermediate temperature (T_e and T_d) heat at 60 - 80°C may become available at the higher absorber outlet temperature (T_a) of 120°C or more, while a smaller amount of water is cooled to condenser temperature (T_c).

Converting Heat to Power using the Rankine Cycle.

For easy comparison, the heat to power conversion cycle we call the rankine cycle have also been plotted on the Log Pressure vs. Temperature graph as Figure 3 below. Condensate is pumped from the low pressure (P_c) condenser at temperature (T_c) to the evaporator at high pressure (P_e), heated and then evaporation take place at temp (T_e) due

Figure 3



to heat input (Q_{eb}), supplemented by some superheat energy. The superheated vapor is then expanded isentropically, delivering power in the turbine and the low pressure (P_c) exhaust vapor is then condensed by rejecting the heat of condensation (Q_{co}) to colder cooling water circulated through the condenser. Superheat is used to avoid the damaging liquid droplets condensing out of the vapor stream while it expand isentropically. If an expander more liquid tolerant than a standard isentropic turbine, (like a liquid piston-, screw- or two-phase expander or even a flooded scroll expander etc.) is used to produce power, superheat is not required.

The rankine cycle is not a binary cycle, but use only one medium in various phases (water / steam). Should an organic refrigerant be used it is called an organic rankine cycle (ORC). This is done specifically when working with very low temperatures where steam density and pressure is too low to be practical.

In these standard power cycles, however, the second law of thermodynamics dictate the conversion efficiency according to the following formula:

$$\eta = \frac{Q_{in} - Q_{rej}}{Q_{in}} = \frac{Work}{Q_{in}}$$

which can also be re-written in temperature terms for the maximum or ideal Carnot efficiency:

$$\eta_{ideal} = \eta_{carnot} = \frac{T_{high} - T_{low}}{T_{high}}$$

with temperatures expressed in Kelvin (K). Using a high temperature of 90°C and low temperature of 45°C (due to availability if cooling water at 30°C from a cooling tower), limit the carnot efficiency $\eta_{carnot} = 12.4\%$. Real practical efficiencies normally would approach 80% of the ideal only, due to the lower isentropic efficiencies of turbines used, calculating to just below 10% efficiency for this example. This low efficiency is a result of the large amount of heat rejection Q_{reject} being the latent heat of getting the low temperature, low pressure turbine exhaust vapor condensed, (or absorbed) back into liquid form. Pumping liquid is a lot less energy intensive to pump to higher pressures than compressing vapor.

Similar to the formula for power delivered, the second law of thermodynamics also dictate that work needs to be done to generate a lower temperature. The efficiency of a refrigerator is called the coefficient of performance (COP):

$$\eta_{refrigerator} = COP = \frac{Q_{absorbed}}{Work}$$

which, again, the ideal or Carnot efficiency for the refrigerator can be re-written in temperature terms as:

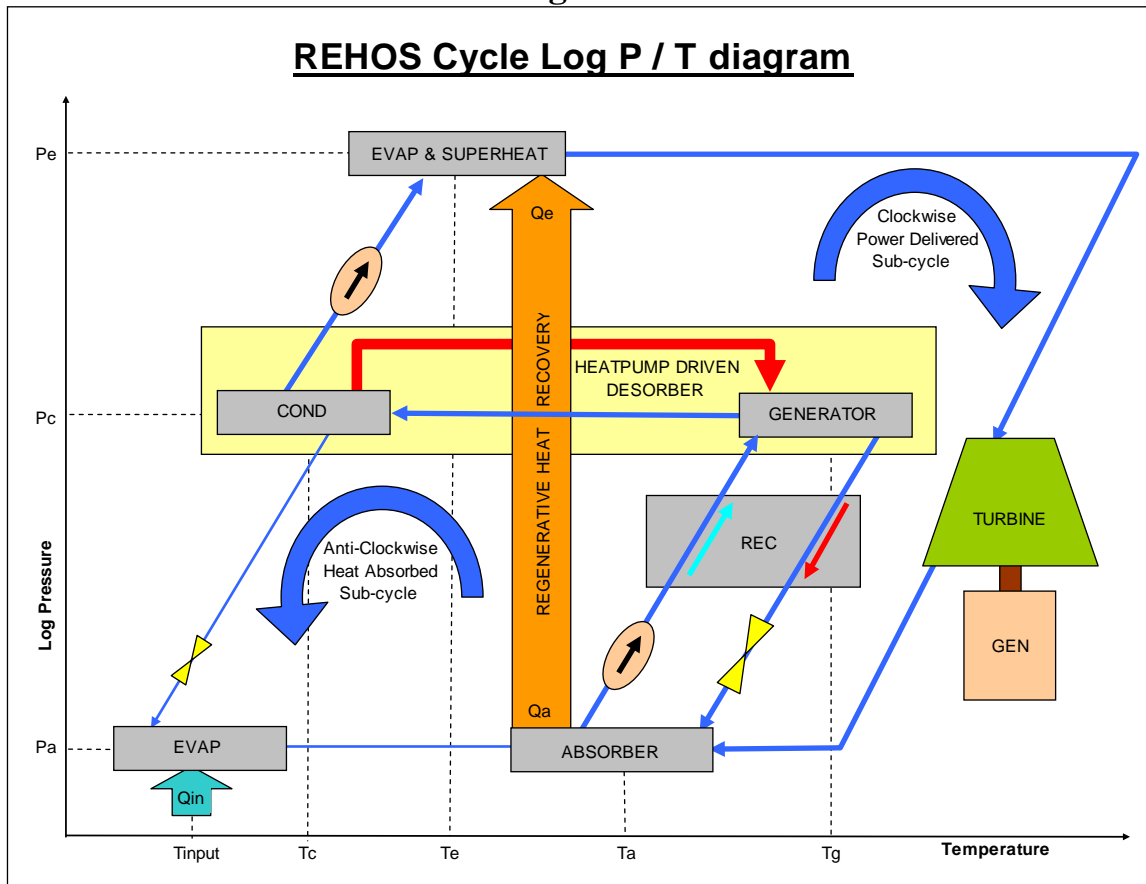
$$COP_{carnot} = \frac{T_{low}}{T_{high} - T_{low}}$$

and again, real refrigeration efficiencies achieved are realistically only ~ 80% of the COP_{carnot} calculated as per the formula above. From this formula it is clear that the smaller the temperature differential between extracting heat and delivering higher temperature heat, required by a heat pump, the larger will be the COP.

Combining Regenerative Heat Recovery with Heat Pump concepts to form the novel REHOS (Regenerative Heat of Solution) Thermodynamic Power Cycle.

Looking at the Log Pressure vs. Temperature graph of the REHOS cycle sketched in Figure 4 below, it is immediately clear that we added another pressure level, (P_e) above

Figure 4



the generator / condenser pressure level (P_c). This allow us to use an evaporator with nearly pure NH_3 at a temperature (T_e), which is lower than the original absorber temperature (T_a), similar to the absorption heat transformer (AHT) sketched in Figure 2 above, so we can **regeneratively use the wasted heat of absorption in the absorber to supply heat to the evaporator** at the elevated pressure (P_e)! This have a result of not having any heat rejection from the absorber, as all the **latent heat is then recovered and regeneratively re-used** as indicated by the orange arrow in Figure 4 above!

Another major innovation comes with the use of a **heat pump to extract the latent heat of condensation from the condenser at the low temperature (T_c) and add it to the generator for boiling off some nearly pure NH_3** . Keep in mind that the evaporation process of boiling off some NH_3 happen at a pressure (P_c) corresponding to the same pressure of condensing the same mass of NH_3 in the condenser. With such a small

differential pressure, the heat pump use very little energy to move the heat back from the condenser to the generator. Due to the very small differential pressure developed by the compressor, the compressor outlet temperature only differ by a few degrees Celsius, with the resulting calculation of COP (as per the formula shown on page 7 of this document), a very high number. The higher temperature generated by the heat pump is actually due to the heat of solution (HOS) of the vapor NH3 into the sub-cooled, lean (low % NH3 in aqua) binary solution. This concept (of having a heat pump COP ~ 100 to 200) is already proven in practice in modern state-of-the-art absorption heat transformers as referenced below. These high COP values correspond to the practical COP values reported eg. from a real experimental heat transformer of 4 MW using only 20 kWe to power the heat pump giving the COP > 200 by Mostofizadeh [9] already in 1998. More on the heat pump is discussed later.

The REHOS cycle sketched in Figure 4 above, actually consist of **2 sub-cycles combined, namely** a standard absorption refrigeration sub-cycle for extracting heat at low temperature from the environment (anti-clockwise flow in the absorption sub-cycle, using a small amount of power to extract heat from the environment, of a small mass of NH3), **combined with** a special regenerative heat recovery power sub-cycle (clock-wise flow of a large mass of NH3 in the power sub-cycle, powering the turbine to deliver a relatively large amount of power). Energy balance around the cycle dictate that the combination of absorbed heat input Q_{in} extracted from the environment, together with any required superheat and heat loss (due to imperfect thermal insulation) at least equal the turbine power output,

$$Q_{in} = Power + Loss$$

with the heat pump compressor as parasitic wasted energy. Not having any heat rejection therefore give a much more favorable cycle efficiency calculation:

$$\text{Cycle efficiency} = \eta = \frac{Power - Compression - Pumps}{Q_{in}}$$

As pumping power may be 4% - 10% depending on temperatures and pressures chosen and compression energy may be very low 6% - 10% due to the high COP >100 the real REHOS cycle efficiency:

$$\eta = 80\% - 90\%$$

Due to the low compression ratio required from the heat pump as well as the high density vapor at the elevated pressure (Pc), the function may easily be fulfilled using a vapor injector type jet compressor with a COP ~ 60 - 150, routing the injector drive vapor from the high pressure (Pe) vapor directly after the evaporator on the 3rd pressure level. This pressure may be some 10 Bar absolute (or higher) in practice. Heat and mass balance calculations show the mass injector drive vapor to be 5% - 10% of the total mass vapor flow and the injector entrainment ratio would be around 16 : 1 (for this specific example).

Heat Pumping with the Bubble Heated, Heat of Solution (HOS) Heat Pump.

Should we fill a 3 meter high vertical tube with a mixture of NH₃ in aqua and kept at a constant pressure of 200 kPa abs, we will find the following: We may call this tube the bubble heated reactor, forming the heart of the REHOS cycle. Due to the liquid column gravity pressure, the column bottom should be about 223.5 kPa abs, as the average density of 800 kg/m³ with a 3m hydraulic height would generate a 23.5 kPa differential pressure. We note that saturated (at different temperatures) solutions of NH₃ in water at a constant vapor pressure of 200 kPa absolute have the following known characteristics:

Table 1

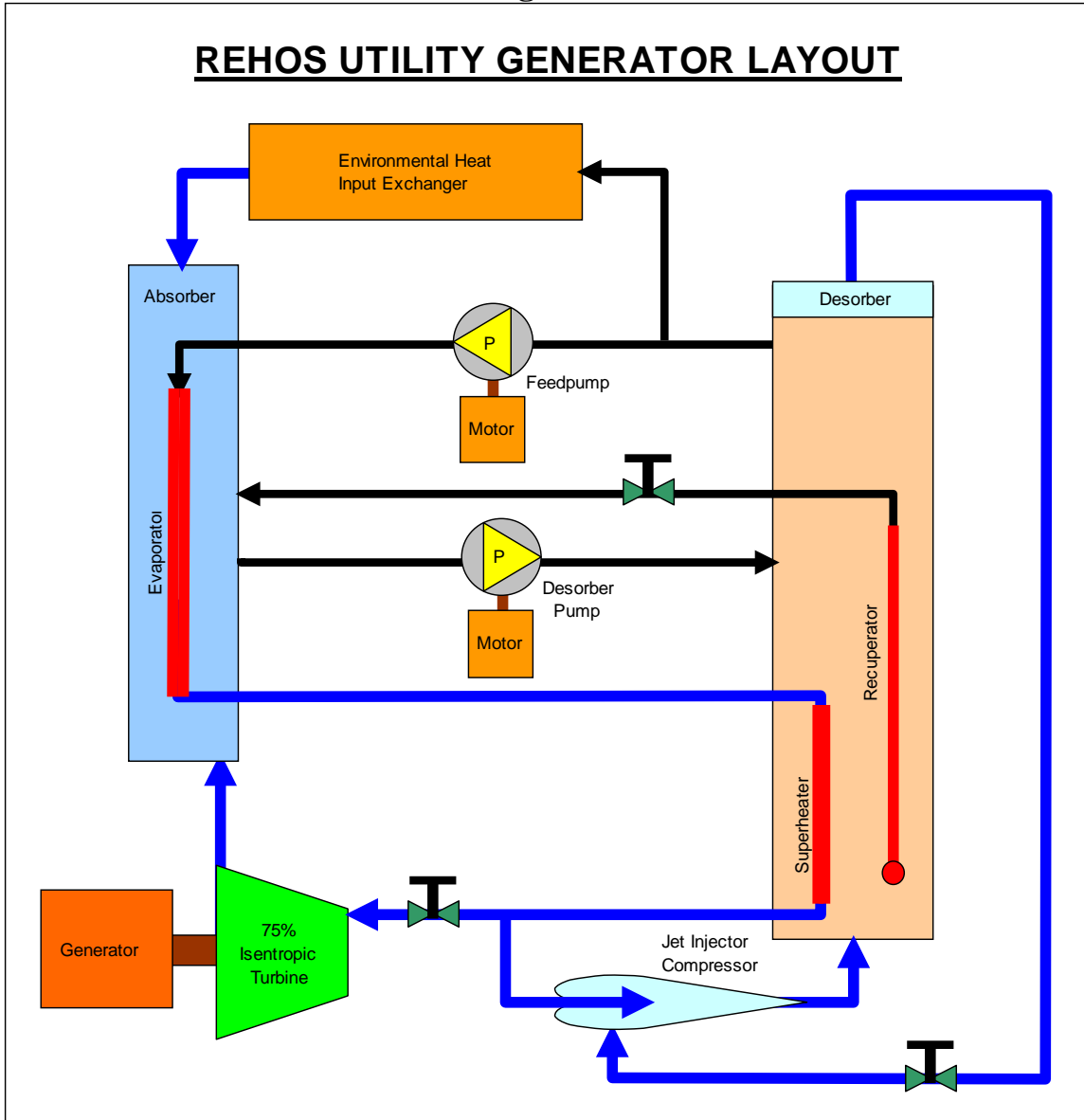
@ 200 kPa	98% NH ₃	-18°C	$\rho = 676 \text{ kg/m}^3$
@ 200 kPa	61% NH ₃	0°C	$\rho = 751 \text{ kg/m}^3$
@ 200 kPa	42% NH ₃	25°C	$\rho = 792 \text{ kg/m}^3$
@ 200 kPa	26% NH ₃	55°C	$\rho = 831 \text{ kg/m}^3$
@ 200 kPa	12% NH ₃	85°C	$\rho = 882 \text{ kg/m}^3$
@ 200 kPa	3% NH ₃	110°C	$\rho = 926 \text{ kg/m}^3$

At the constant pressure of 200 kPa Absolute noted in Table 1 above, the **NH₃ concentration has an inverted proportional relation to both the solution specific density and its specific saturation temperature**. This has the consequence that if you apply heat to any section of the tube filled with a saturated mixture of ammonia in aqua, some vapor is boiled off immediately, rising upwards in the column as bubbles, to be absorbed again higher up in the column, while the heated, now leaner remaining solution have a greater density and higher temperature, and would therefore migrate downwards, displacing liquid with lower density. This has the effect that heat added anywhere along the length of vertical column, would gradually establish both a **temperature gradient with the highest temperature trapped at the bottom of the column**, as well as a **NH₃ concentration gradient with the highest concentration at the top of the column**. These gradients would be maintained with the complete column being saturated at all level sections (but at different temperatures of course) as long as the heat addition continues, but conduction (and radiation) heat leakage from the hot bottom to the cold top of the reactor and NH₃ concentration dispersion resulting from the natural diffusion of the high NH₃ concentration at the top to the low concentration at the bottom, would soon disperse and break up both the concentration and temperature gradients if no heat is continually added to the column.

From the knowledge gained utilizing absorption refrigeration technology, we know that NH₃ vapor would be very strongly absorbed in mixtures of NH₃ in aqua, only **as long as the mixture is sub cooled**. With the mixture saturated, no vapor from a rising vapor bubble would diffuse into the surrounding liquid solution, and therefore no heat would be generated as no absorption will take place. Should a NH₃ vapor bubble come in contact with a sub cooled section of liquid solution, the direct contact between vapor and liquid would facilitate the quick absorption of the vapor into the liquid, collapsing the bubble

and generate absorption heat, consisting of both latent heat of condensation and heat of solution (HOS) as already discussed when we described the absorber operation as part of the proven absorption refrigeration cycle above. Note that we have already proven that the sensible heat of heating cold vapor to the high temperature at the reactor bottom, have

Figure 5



a very small influence compared to the heat generated by absorption. The actual temperature of the vapor in the bubble is therefore of very little importance, as the large direct contact heat transfer is driven by absorption of vapor into the liquid mixture.

With the compressor running, saturated NH₃ vapor is extracted from the top of the column, causing cooling of the solution at the reactor top, by flashing NH₃ from the 90% NH₃ in aqua solution (with heat of vaporization = 1179 kJ/kg) at -2°C. To compress this vapor with the compression ratio of 1.118 using a 70% efficient isentropic compressor

require about 19.2 kJ/kg, making the heat pump performance (COP = Coefficient of Performance):

$$COP = \frac{HeatPumped}{PowerUsed}$$

$$\text{For this example } COP = \frac{1179}{10.2} = 61.4$$

This is also in line with the practical COP values reported from a real experimental heat transformer of 4 MW using only 20 kWe to power the heat pump giving the COP > 200 by Mostofizadeh [9] already in 1998. The compression process use so little energy that the compressed vapor temperature hardly change, at the most 1°C, so vapor is produced at -1°C with enthalpy 1198 kJ/kg. With a heat exchange coil removing high temperature heat from the hot bottom of the reactor, we are correct to assume the bottom of the column is sub cooled (the requirement for heat to be transferred by absorption) by a few degrees to (assume 30°C) below the saturation temperature of 110°C at the stated pressure of 200 kPa.

Introducing the vapor to the bottom of the column into the lean (10% NH₃ in aqua) solution will firstly heat the vapor bubble from -1°C to the liquid temperature of 80°C, using 72 kJ/kg, after which the bubble is absorbed into the sub-cooled hot liquid solution, releasing 1004 kJ/kg of heat to the bottom of the column, raising the temperature back from the sub-cooled 80°C to the saturation 110°C, similar to the absorption heat transformer we described before. In this case the sensible heat (72 kJ/kg) required is only 7.2% of the absorption heat produced.

This continuous bubble heating of the bottom of the column quickly establish and maintain both the temperature and concentration gradient, but if heat is not extracted by the submersed heat exchanger coils the column bottom temperature would be raised to the saturation level of 110°C with continued bubble absorption. At this temperature the column would be totally in saturation if the pressure is kept constant, so the vapor bubbles would pass through the column to the top without any absorption.

The REHOS cycle sketched in Figure 5 above is a typical utility layout of practical power plants, clearly showing the components it consist of.

The interface of this thermodynamic cycle to the waste heat input source is the Environmental heat exchanger. We also take note that the REHOS Thermodynamic Cycle can extract heat from as low as -30°C if the absorber is kept at 100 kPa absolute, although the extracted heat is added to the thermodynamic cycle at the absorber outlet temperature, which is a few degrees higher, and also, no heat is rejected out of the cycle, meaning that no cooling tower would be required! In this layout the desorber would be operated at 500 - 600 kPa abs and the bottom of the desorber would operate at an elevated temperature of 100°C to 120°C. Note also the two pumps required for this layout, namely the feed pump, providing the high pressure required by the turbine inlet

pressure, as well as a desorber mixture pump, providing the circulation of the absorber strong solution liquid to the pressure the desorber operate on.

The flashing vapor extracted from the top of the desorber by the vapor jet injector compressor cool the desorber top down below ambient temperature to some -2°C in this example, establishing the temperature gradient in the desorber of 120°C at the bottom while the top run at -2°C. Heat removed from the desorber bottom by the superheater as well as the recuperator heat exchange tubes provide the necessary sub-cooling to allow heating the desorber bottom by vapor bubble absorption.

The turbine exhaust vapor enter the bottom of the absorber and heat the absorber solution also by vapor bubble absorption. Most of this heat is used regeneratively to heat and evaporate the high pressure feed pump pressurized liquid NH3 contained in the heat exchange tube inside the absorber, while some excess heat is carried in the strong solution (having absorbed the vapor from the turbine exhaust) which is pumped back to the higher pressure desorber.

The Simplified REHOS Cycle Layout

A simpler layout of the REHOS thermodynamic cycle, where the absorption and desorption functions are done in the heated bottom section and the cooled top section respectively, of a single bubble reactor is sketched in Figure 6 below. This layout have the advantage of using less components, like a single high pressure feed pump, a single liquid-liquid heat exchanger for extracting waste heat input into the cycle from the waste heat source, as well as the combined absorber/desorber/superheater/evaporator vapor bubble heated heat exchanger, we call the "bubble reactor" and the normal vapor jet injector compressor as well as an isentropic turbine. The process values added to the layout in Figure 6 were calculated with Mass-, Heat- and Species-balance across each component and reflect a realistic machine with the isentropic turbine efficiency assumed as 75%, while the feed pump efficiency was taken as 33% only. Realistic efficiency assumptions for the design of the jet injector compressor, using the guide provided by Chen et al [14] in 2013 was done. Turbine brutto power calculates to:

$$Power = Mass(H_{inlet} - H_{exhaust}) = (2.118e-2)(1.307e6 - 1.180e6) = 2700 \text{ Watt}$$

while the vapor injector compressor power may be calculated to:

$$Power_{comp} = Mass_{comp}(H_{inlet} - H_{outlet}) = (1.459e-3)(1.307e6 - 1.121e6) = 272 \text{ Watt}$$

$$COP = \frac{Heat_{pumped}}{Power_{comp}} = \frac{29482}{272} = 108$$

This is giving us the realistic heat pump COP = 108. This is also in line with the practical COP values reported from the real commercial absorption type heat transformer of 4 MW

using 20 kWe to power the heat pump, with the resulting COP > 200 as reported by Mostofizadeh [9] already in 1998.

Should we for a moment ignore the regenerative vapor absorption and assume the turbine exhaust vapor is condensed and the heat rejected to cooling water, like the rankine cycle sketched in Figure 3 above, the rejected energy would have been:

$$\begin{aligned} (\text{Heat})_{rej} &= \text{Mass} (H_{\text{exhaust}} - H_{\text{cold}}) \\ &= (2.118e-2)(1.180e6 - 25.5e3) = 24452 \text{ Watt} \end{aligned}$$

while the cycle efficiency would have calculated to:

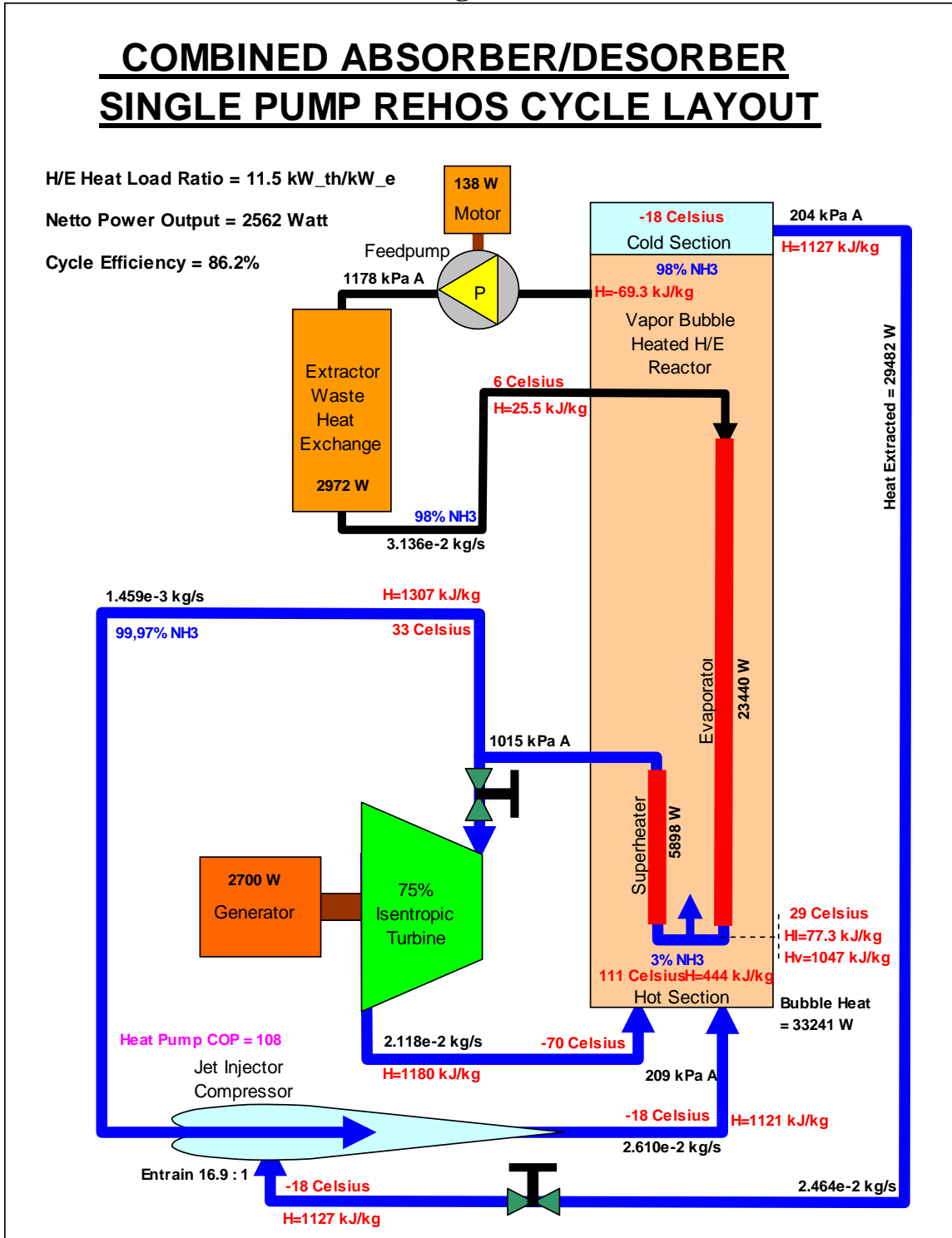
$$\eta = \frac{\text{Power}}{(\text{Power} + \text{Heat}_{rej})} = \frac{2700}{(2700 + 24452)} = 9.9\%$$

which is the standard that we got used to, working with various thermodynamic power cycles like rankine-, kalina-, ericsson-, stirling cycles and many other, all with heat input and heat rejection without the possibility of recovering some of the rejected heat. The maximum efficiency of all these thermodynamic cycles are limited by the second law of thermodynamics, namely, heat inlet and -rejection temperature levels, as already shown in the calculation of carnot efficiency on page 7 of this document.

The REHOS Cycle, however, recover all the exhaust heat and re-use it regeneratively in the form of vapor injected into, and being absorbed in the hot (111°C) bottom area of the reactor where the NH3 concentration is only 3% NH3 in aqua. This is made possible by the heat of solution (HOS) of the vapor being absorbed, driving the solution temperature higher, irrespective of the current solution temperature, demonstrated by heat transformer principles explained already starting on page 4 above, increasing the temperature in this absorption area of the reactor to maintain 111°C. This heat is transferred to the evaporator and superheater tubing where some 29338 Watt is used to produce the superheated vapor at 33°C to power both the power turbine and the injector compressor. Utilizing this heat, sub-cool the absorption area, which is a pre-requisite for vapor absorption to take place, allowing more low pressure vapor bubbles to be absorbed to generate more heat, regeneratively, keeping the temperature gradient and NH3 concentration gradient in tact for the complete length of the bubble reactor. The turbine exhaust vapor and injector compression vapor together produce more bubble absorption heat (total 33241 Watt -some 13% more) than what is used by the evaporator and superheater, to allow for some heat wasted as leakage. To maintain a constant reactor bottom temperature of 111°C, the compressor inlet mass flow is throttled by a valve in the suction line. **This regenerative exhaust heat recovery ensure no heat is rejected out of the cycle.** The heat entering the absorption section of the reactor is continuously recycled via the heat pump, extracting heat from the top of the reactor to add again to the bottom. It therefore does not come from external sources outside the cycle, and therefore any energy removed from the cycle (like electricity produced by the power of the turbine) will decrease the amount of energy in circulation, and would have to be replenished from

outside the cycle to keep all pressures and temperatures constant and balanced in the cycle. Additional energy is therefore extracted from the environment and added to the cycle. The amount of heat required to satisfy heat balance calculations is nearly equal to the power generated. In our example of Figure 6 below, heat required (and extracted

Figure 6



from) the environment to establish energy balance around the cycle is 2972 Watt, absorbed at an average temperature of -6°C, giving the very high REHOS Cycle efficiency of 86.2%, as the electrical power used (taken from the generator) to drive the feed pump, amount to 138 Watt. The injector compressor use 272 Watt, direct from the evaporator/superheater line in parallel to the turbine. The complete cycle netto efficiency therefore calculate to:

$$\eta_{netto} = \frac{Power_{netto}}{Q_{input}} = \frac{Power - Pump}{Q_{input}} = \frac{2700 - 138}{2972} = 86.2\%$$

An important measure is reflected in the ratio of heat exchanged in the reactor vs. netto electrical power produced:

$$\text{H/E Heat Load Ratio} = \frac{Heat_{evap} + Heat_{super}}{Power_{netto}} = \frac{23440 + 5898}{2562} = 11.5 kW_{th} / kW_e$$

The effect of Changing Turbine Isentropic Efficiency.

It is interesting to note that unlike other thermodynamic power cycles without regenerative exhaust heat recovery, the **REHOS Cycle efficiency is nearly independent of turbine isentropic efficiency!** Should the turbine (assumed to be 75% isentropic efficient), be replaced with a turbine having a 65% isentropic efficiency, the mass vapor flow in the REHOS cycle would have to be increased by some 17% to retain the same power output, but the heat exchanged in the reactor increase by 19.5%, keeping the overall cycle efficiency nearly constant. The same goes for a more efficient turbine of 85% isentropic efficiency. The mass vapor flow in the cycle in this case would have to be reduced by 18% to keep the same netto turbine power output, while the heat exchanged in the reactor would decrease by 15%.

For a standard rankine cycle using the same 75% isentropic efficient turbine, we calculated the cycle efficiency before (on page 13) using the following formula:

$$\eta = \frac{Power}{(Power + Heat_{rej})} = \frac{2700}{(2700 + 24452)} = 9.9\%$$

Should we now replace the turbine with another, delivering power with 65% isentropic efficiency in the standard rankine cycle, the power would decrease by 10%, so the vapor flow will have to be increased by ~ 10% to deliver the same power. This however, change the cycle efficiency as follows:

$$\eta = \frac{Power}{(Power + Heat_{rej})} = \frac{2700}{(2700 + 26897)} = 9.1\%$$

Decreasing the turbine isentropic efficiency by 10% therefore decrease the rankine cycle efficiency by ~ 8% from the original 9.9% down to 9.1%, making a huge difference in the economy of waste heat recovery systems operating with low temperature differentials. Higher isentropic efficiency turbines are also much more expensive, again unfavorable to the economy of current state-of-the-art commercial waste heat recovery systems.

The REHOS Cycle efficiency of > 80% and also nearly independent of turbine isentropic efficiency therefore represent a true revolution in waste heat recovery economics!

The effect of extracting Heat Energy from Heat sources at Temperatures at or Below the Ambient.

We note that the pressure pump in the REHOS Cycle layout sketched in Figure 6 above, pump cold liquid solution (nearly pure NH₃) at the reactor cold top section operating on below zero at -18°C through the input heat exchanger, where it absorb heat to emit at 6°C. The average temperature of the liquid solution flowing through the tube-side of the input heat exchanger is therefore -6°C. This heat exchanger would therefore easily extract heat from ambient temperature (let us assume this is 20°C) water entering the exchanger on the shell-side to cool it down by a few degrees. The chilling of the waste heat source is a useful side-effect of operating the REHOS cycle under these conditions, but care should be taken to keep the ambient waste heat source liquid (water) flow speed on the shell-side of the input heat exchanger high enough so it does not freeze in the heat exchanger. It also mean the REHOS cycle is able to produce useful power from ambient water sources, like rivers, lakes, dams and the sea, extracting the ambient temperature heat energy stored in large water masses to deliver as electricity! It could therefore provide the energy for marine propulsion, essentially free, by extracting it from the water in the ocean direct.

A REHOS cycle water pump may also easily be constructed to extract the required pumping energy from the water being pumped, cooling it down by a few degrees centigrade in the process. The high conversion efficiency (> 80%) also have the effect of keeping the input heat exchanger small, only absorbing some 20% more heat than the cycle power output. Typical shell and tube type liquid/liquid heat exchangers using the temperatures mentioned, would have a heat exchanger core of ~ 2.7 MW_{th}/m³, and for a 35 kW irrigation pump powered by a REHOS cycle the required heat exchanger would handle 42 kW_{th} requiring a volume of ~ 0.01556 m³ (or a tubular exchanger 1 m long and 140 mm diameter).

Should we want to extract the same amount of heat energy from ambient air, for powering an electric drive system of a motor vehicle, the heat exchanger absorbing 42 kW_{th} from ambient air would be ~ 100 kW_{th}/m³ (derived from information provided by Raghupatruni et al in the Experimental Study of Heat Recovery from a Diesel Exhaust [13], and automotive radiator supplier info). The required heat exchanger would therefore have a core volume of 0.42 m³ (or 1.3m x 1.3m x 250 mm deep), dropping the inlet air temperature by some 20°C as it pass through the heat exchanger. Extracting energy from air (or gas) require a heat exchanger volume of ~ 27 times the volume of a liquid/liquid heat exchanger, and is therefore also much more expensive. It would, however, be very

practical should the chilled air be used for air conditioning in buildings, shopping centers, in the mining industry etc, where both the chilled air and the electricity produced is of value! It would also be used in electric or hybrid vehicles to supply virtually free energy extracted from the environment.

The energy required by the vapor injector type compressor, is only a function of the amount of heat to be pumped, and not the actual temperature levels, as the small differential pressure across the compressor would generate only a small temperature delta, calculating to be a very large COP as per the formula given on page 7 of this document. Higher temperature of the compressor outlet vapor is generated by the absorption process in the bubble reactor as described before.

The sketched parameters provided in Figure 6, above, suggest the external input waste heat required to power the cycle would be averaged as -6°C , while the reactor pressure is maintained at 204 kPa Abs. Should waste heat at a higher temperature be available, the external input heat exchanger tube-side temperature may be allowed to increase to a higher value eg. 16°C , but this would increase the reactor pressure with 297% to ~ 605 kPa Abs, and assuming we keep all other pressures and temperatures the same, would therefore decrease the power output of the turbine drastically. To maintain the same power output, the vapor mass flow would have to be increased dramatically, creating an increase of 490% to energy used by the compressor, as well as pumping power increase of some 175%, working negatively to decrease the overall cycle efficiency from 86.2% down to 63.3%. This huge effect could be a little less if the pumping pressure is also increased to 20 Bar instead of keeping it on 10 Bar. Then overall cycle efficiency would be 71.7%, but still substantially lower than the original 86.2%!

Should the reactor pressure be further decreased below 204 kPa Abs to 100 kPa Abs, the overall cycle efficiency is effected very little, while the total reactor heat exchanger heat load decrease some 26%. The lower tube-side heat input temperature would decrease substantially, though, entering the heat exchanger at -33°C , increasing the danger of freezing the environmental waste heat source liquid inside the input heat exchanger. The current design of reactor pressure at 204 kPa Abs as sketched in Figure 6 above, is therefore considered to be a very practical compromise, but still low enough input temperature level to extract environmental heat from natural water sources at ambient temperatures.

Costing Aspects of the typical Simplified REHOS Cycles.

All costing values were adjusted to current US dollar at the time of writing, using the US inflation adjuster found on the internet at www.usinflationcalculator.com.

Information published by the two references [11] and [12] were used as a check for quotation information from suppliers to provide realistic budgetary information as listed in Table 2, 3 and 4 below:

Heat and mass balance on the layout as per the simplified layout as per Figure 6 above, has proven the reactor heat exchanger sizing to be about 11.5 kW_{th} / kWe and the environmental heat exchanger ~ 1 kW_{th} / kWe.

Using the info as spelt out above, we calculated the Purchased Equipment Cost (PEC) for a 300 kWe REHOS Power Generator based on a quotation by a supplier (in the USA) of a radial outflow turbine operating on a refrigerant for specific use in direct-drive ORC waste heat recovery systems. Costing values are listed in Table 2 below.

Table 2

300 kWe Radial Outflow OR Turbine	84500 \$	38%
Combined Absorber/Desorber Reactor H/E	69000 \$	32%
Environmental Heat Input Liquid/Liquid H/E	6000 \$	3%
Electrical Generator (50 Hz 1500 RPM)	45500 \$	21%
Ammonia Pressure Pump	13000 \$	6%
Total PEC	218000 \$	

Other elements of the Fixed Capital Investment (FCI) are listed in Table 3 below, with the vapor jet injector compressor included in the piping costs. Table 4 calculate the total capital investment required (Overnight Cost) of the 300 kW REHOS Cycle power generator:

Table 3

Installation	20% of PEC	43600 \$
Piping & Injector Compressor	30% of PEC	65400 \$
Control & Instrumentation	20% of PEC	43600 \$
Electrical	10% of PEC	21800 \$
Total FCI		392400 \$

Table 4

Engineering & Supervision	30% of PEC	65400 \$
Total Legal Cost	2% of FCI	7848 \$
Total Capital Investment (Overnight Cost)		465648 \$

Total Capital Investment = 465648 \$ for 300 kWe

=> Capex = 1552 \$ / kWe

It is normal in power generation circles to include financing and maintenance costs to calculate the total cost over the lifespan of power generating equipment, called the levelized cost of energy (LCOE). The formula used is:

$$LCOE = \left[\frac{Capex \times CRF + O \& M_{Fixed}}{8760 \times CF} \right] + O \& M_{Variable}$$

We take $O \& M_{Fixed}$ as (1.7% of Capex) \$/kW-year (same as Advanced Nuclear Power Generators), and $O \& M_{Variable}$ as (2.3) \$/MWh (same as Advanced Nuclear Power Generators), while the capacity factor, $CF = 90\%$ (for Base Load Generation).

$$CRF = \frac{[i(i+1)^N]}{[(i+1)^N - 1]} = 0.10955$$

where the interest rate $i = 9\%$ while the life expectancy for the machine $N = 20$ years.

$$LCOE = \left[\frac{1552 \times 0.10955 + 26.38}{8760 \times 90\%} \right] + (2.3e - 3) = 0.02721 \text{ \$/kWh for 300kW unit.}$$

This **electricity cost of 0.02721 \$/kWh** is less than half the real utility electricity cost per kWh for the utility (Eskom) in South Africa. Should we repeat the LCOE calculation for a much smaller (60 kWe) turbine from the same supplier costing 45500 \$, the LCOE calculate to 0.05487 \$/kWh, which is still below the South African utility (Eskom) electricity cost. Calculating a larger REHOS Cycle generating 2300 kWe of power using a positive displacement expander in the form of a Lysholm Screw Expander quoted as 190000 \$, give us a electricity cost **LCOE = 0.01566 \$/kWh for the 2.3 MWe unit**. This is only ~ 32% of the current Solar PV and Wind energy cost for utility scale generators, and the REHOS Generator being a base load generator with a capacity factor of 90% compared to the 20% - 25% capacity factors (the sun only shine a few of the 24 hours per day) reported by PV and wind generation make the **REHOS Generator the lowest cost renewable base load power generator ever!**

About 95% of existing power generation systems, coal, oil & gas fired, nuclear, solar thermal etc all have a rankine power block, cooled by water circulating through a cooling tower. This cooling water carry away the > 55% waste heat rejected from the power cycles. The circulated cooling water often leave the condenser at around 35°C - 45°C and is cooled in the cooling tower to 28°C - 32°C either direct by evaporation (wet cooling) or indirect (dry cooling). Cooling water temperatures for dry cooling in general are some 10°C warmer than evaporative (wet) cooling. The REHOS Generator may easily replace the function of the cooling towers, using the remaining waste heat in the cooling water as heat source, and at > 80% heat to power conversion efficiency deliver more power than the original power block, effectively doubling the power station capacity! This will allow the utility to reduce fossil combustion by more than 50% without decreasing the station power output. The low cost LCOE of the REHOS cycle would also ensure that the utility may reduce the cost of generating their power substantially.

The uses of the REHOS cycle machines are extremely widespread, through all industries, and is really only limited by our small human imagination! **We can reverse global warming!**

References

1. Thermophysical Properties of NH₃ + H₂O solutions for the industrial design of absorption refrigeration equipment, by Dr Manuel R. Conde-Petit (M. Conde Engineering, Zurich - Switzerland) 2004.
2. Evaluation of thermodynamic properties of ammonia-water mixture up to 100 bar for power application systems, by N. Shankar Ganesh and T. Srinivas of the Vellore Institute of Technology, Vellore-632014, India. This paper was published in Journal of Mechanical Engineering Research Vol. 3. (1), pp. 25-39, January, 2011.
3. Analysis of Absorber Operations for the 5 kW Ammonia/Water Combined Cycle by Sirisha Devi Govindaraju as Thesis Presented to the Graduate School of the University of Florida in Partial Fulfillment of the requirements for the Degree of Master of Science, 2005.
4. Thermodynamic Properties of Water including Vapor, Liquid and Solid Phases, by J.H. Keenan, F.G. Keeyes, P.G. Hill, J.G. Moore and published by John Wiley & Sons Inc 1969.
5. Thermodynamic Properties of Ammonia by Lester Haar and John S. Gallager, published in J. Phys. Chem. Ref. Data, Vol. 7, No. 3, 1978.
6. Patent Application PCT/ES2010/000208 of 25/11/2010 named Rankine Cycle with Absorption Step Using Hygroscopic Compounds, also publication WO 2010133726 by Rubio Serrano & Francisco Javier of Spain. Also published on the Internet @ Wikipedia under "Hygroscopic Cycle".
7. Experimental Evaluation of a Single-Stage Heat Transformer used to Increase Solar Pond's Temperature, by W. Rivera and published in Solar Energy Vol 69, No. 5, pp 369 - 376, 2000.
8. Quasi-Isothermal Expansion Engines for Liquid Nitrogen Automotive Propulsion, by C. Knowlen, J. Williams, A.T. Mattick, H. Deparis and A. Hertzberg, published October 2011 at the University of Washington.
9. Use of a new type of heat transformer in process industry, by Ch Mostofizadeh & C. Kulick at the Institute for Energy and Process Engineering in Bremerhaven, Germany, published in Applied Thermal Engineering 18 (1998) 857-874.
10. Ammonia-water desorption heat and mass transfer in micro channel devices, by Mathew D. Determan & Srinivas Garimella at the Sustainable Thermal Systems Laboratory, George W. Woodruff School of Mechanical Engineering, Georgia Institute of Technology, Atlanta, USA. It was published by International Journal of Refrigeration 34 (2011) 1197-1208.

11. Cost Estimation of using an Absorption Refrigeration System with Geothermal Energy for Industrial Applications in El Salvador, by Juan Carlos Abrego Castillo, of LaGeo S.A de C.V. 15 ave. Sur, Colonia Utila, Santa Tecla El Salvador and used as Report Number 4, 2007 for Geothermal Training Programme of the United Nations University in Reykjavik, Iceland.
12. Cost Engineering Techniques and their Applicability for Cost Estimation of Organic Rankine Cycle Systems, by Sanne Lemmens, Department Engineering Management, University of Antwerp, Prinsstraat 13, 2000 Antwerp, Belgium, published 23 June 2016 in the journal *Energies* 2016, 9, 485; doi:10.3390/en9070485.
13. Experimental Study of Heat Recovery from Diesel Exhaust at University of Alaska Fairbanks (UAF), by Prasada Rao Raghupatruni, Chuen-Sen Lin, Dennis Witmer, Ed Bargar, Jack Schmid, Thomas Johnson, published February 2012.
14. A 1D model to predict ejector performance at critical and sub-critical operational regimes, by WeiXiong Chen, Ming Liu, DaoTong Chong, JunJie Yan, Adrienne Blair Little, Yann Bartosiewicz from State Key Laboratory of Multiphase Flow in Power Engineering, Xi'an Jiaotong University, China, Université catholique de Louvain (UCL), Institute of Mechanics, Place du Levant 2, Louvain-la-Neuve, Belgium and George W. Woodruff School of Mechanical Engineering, Georgia Institute of Technology, Atlanta, USA. This manuscript was published in *International Journal of Refrigeration* 36 (2013) 1750-1761.
15. Capital Cost Estimates for Utility Scale Electricity Generating Plants, November 2016 published by US Energy Information Administration (EIA), downloaded from www.eia.gov in March, 2017.