Comparison of various Modern Heatpump Technologies for unlocking Commercial Value from Ambient Heat

by
Johan Enslin

Heat Recovery Micro Systems

Heidelberg
South Africa

Abstract:

For the evaluation and comparative analysis process, it is required to define heat pump performance in all the categories of use, namely heat generation (50°C warm water), cold (chiller) operation (10°C chilled water), as well as waste heat power generation by coupling the heat pump being evaluated regeneratively to a benchmark organic rankine cycle (ORC).

The warm water generation benchmark is set as the direct electricity (at a benchmark cost) heating of water in a geyser, while the chiller benchmark is produced by the common vapor compression (VC) type heat pump operating between 0°C and 60°C.

The primary input heat used is extracted from a water source at ambient (20°C) temperature, and is therefore assumed available at no cost. Capital investment requirements for the various heat pumps are also omitted in the commercial evaluation.

Comparisons yield several commercially proven, heat of solution (HOS) type absorption heat transformers (AHT) used as heat pumps with highly improved performance ranging from 70 - 90% improvement in heating and cooling performance compared to the benchmarks.

The novel simple HOS Bubble heat pump derived from Hybrid Absorption-Compression heat pump concepts of Jensen [5], is set to very drastically reduce the cost of warm water (as much as 97% reduction) and chilled water (as much as 93% reduction) and seriously challenge coal fired power generation by demonstrating at least a 50% cost reduction!

Introduction:

Using environmental heat and other low grade waste heat sources is not a new concept, as heat pumps have been used for many years to upgrade waste heat for using commercially, as it was recognized to be more cost-effective than using high commercial value electricity to generate useful heat. The heat pump as alternative for heating household geysers and swimming pool water is much cheaper than heating the water with an electrical element, and is therefore used widely.

Different types of heat pumps, however, vary in efficiency of doing this job, and therefore the choice of heat pump technology have far-reaching commercial consequences for the user. It would therefore make sense to seek a comparison of the various technologies available to be able to select the most viable commercial solution.

As the heat pump technologies may be the best choice for either heating-, cooling or power generation application, but not necessarily all three, it would be of value to compare technologies of all three these categories, as a specific technology may be better suited to any one of the categories. We therefore define benchmark solutions to use as measurement base for the comparison to make more sense.

For the comparisons it is assumed that the capital investment related to the heat pump installations may be ignored. Also, the waste heat used is extracted from the environment at ambient temperature and the heat used therefore is free, not impacting on the calculations. For this paper it is assumed ambient water from a pool at 20°C is chilled by a few degrees and the extracted heat used as heat source, and therefore all the heat pumps have a low temperature value of 0°C. The comparison also assume the heat pump delivery temperature is fixed at 60°C so the hot water may be delivered at 50°C via the output heat exchanger. For power generation it is therefore assumed the heat used for driving the power turbine is available from the heat exchanger at 50°C, while heat is rejected in a heat exchanger at 10°C.

For all the models shown in the sketches, comprehensive mass-, species-, and heat balance calculations have been done using the correct thermodynamic properties of ammonia, water and mixtures of NH_3 in aqua for realistic representation of all process variables shown.

The benchmark for Heat delivery:

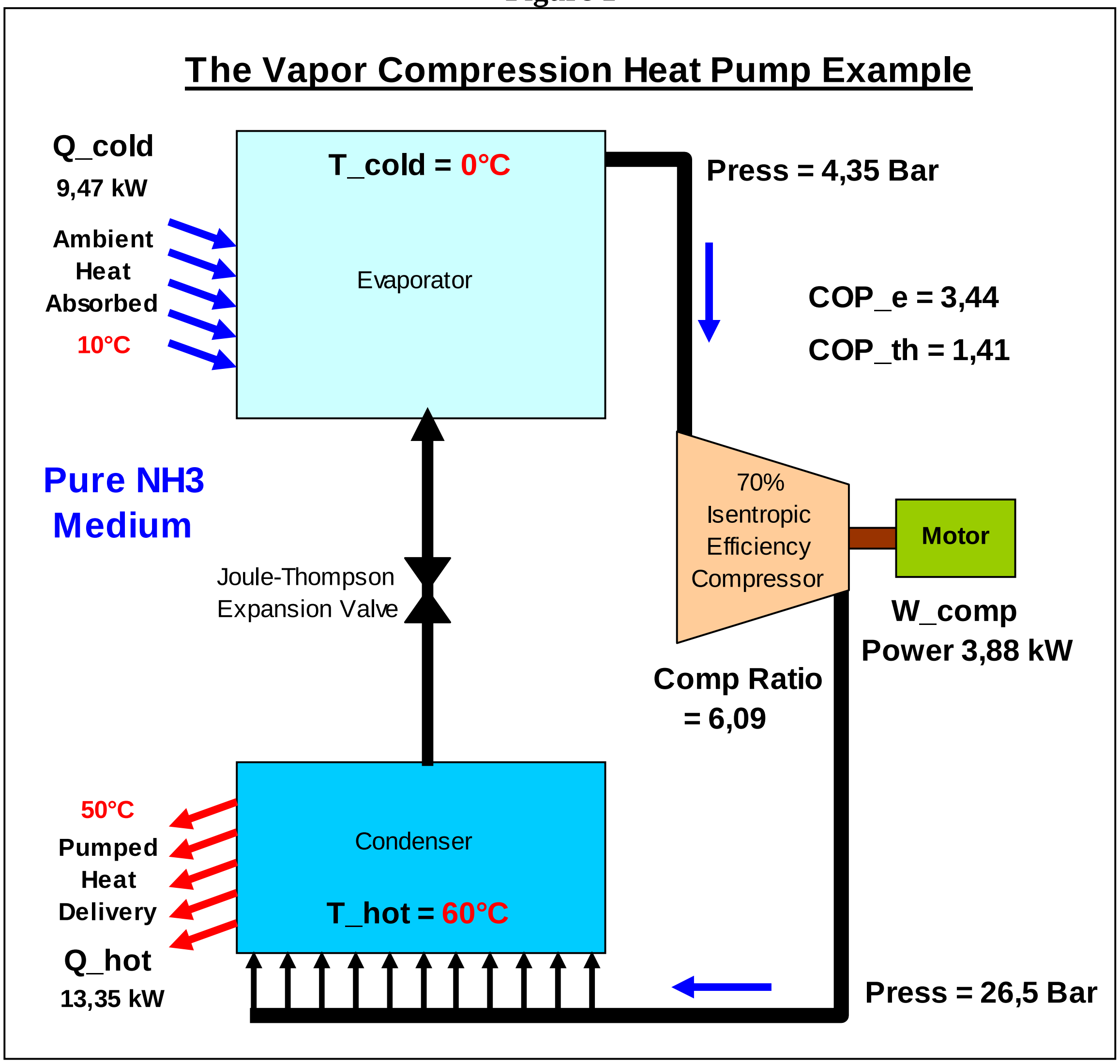
As many households in South Africa still use electricity to heat water in a geyser for household use, a good average cost of this heat is also the cost of domestic electricity from the local utility, Eskom, amounting to about $0-10 / kWh_e. Electrical power used for driving the compressors would therefore be priced at this value. The rate of exchange between USD and ZAR currently is 1 USD = 12-00 ZAR.

Our **Benchmark for Heat Delivery is therefore** the same cost, or **100 $ per MWh_th**.

The benchmark for Chilling (cold) delivery:

A very common vapor compression (VC) heat pump is used as chilling machine benchmark, as sketched in figure 1 below. NH3 vapor at 0°C and 4,35 Bar Abs is compressed with a compressor having a 70% isentropic efficiency, to 26,8 Bar where it condenses at the saturation temperature of 60°C in the condenser. We take note that the compression ratio is 6,09 and the compressor use 3,88 kWe (W_comp) to extract 9,47 kW heat (Q_cold) from the ambient heat source by chilling it to 10°C. High grade heat of 13,35 kW (Q_hot) is delivered at 50°C via a heat exchanger in the condenser.

Figure 1



The electrical coefficient of performance (COP_e) for cooling service is defined as:

$$COP\_e = \frac{Q_{\_cold}}{W_{\_comp}} = 2,44$$

This calculate to the **Benchmark cost for Cold delivery of 41 $ per MWh_th.** The same benchmark VC heat pump may also be used for cost of delivering high temperature heat (at 50°C) where the electrical efficiency is defined as:

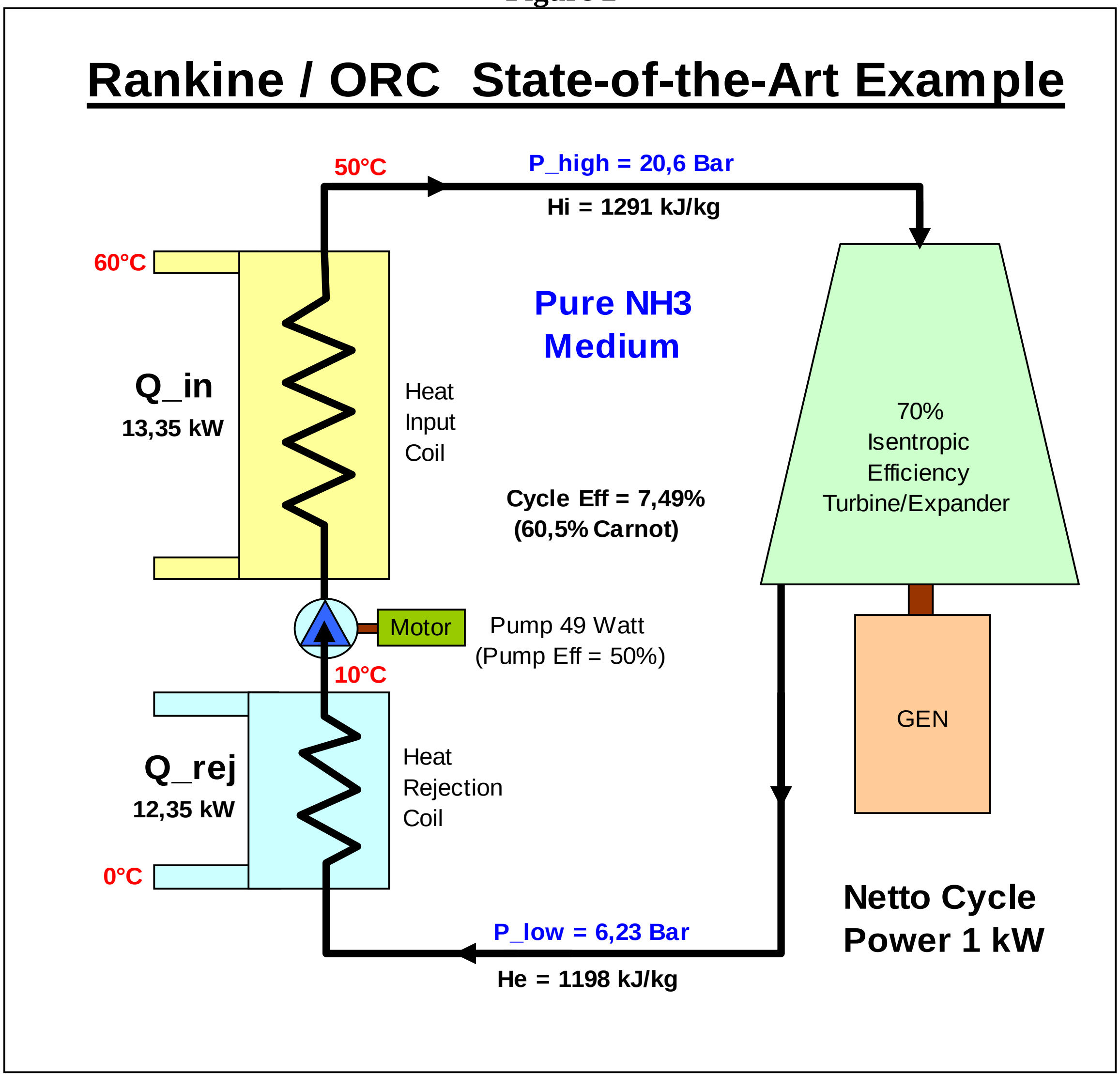
$$COP\_e = \frac{Q_{\_hot}}{W_{\_comp}} = 3{,}44$$

while the thermal efficiency (COP_th) for heating service is defined as:

$$COP\_th = \frac{Q_{\_hot}}{Q_{\_cold}} = 1{,}41$$

This same benchmark VC heat pump therefore deliver high temperature **heat at a cost of 29 $ per MWh_th.**

Figure 2



The benchmark for Power delivery:

Power from waste heat may be generated by employing a simple Organic Rankine Cycle (ORC) as sketched in figure 2 above:

The real process values of a practical NH3 cycle using an expansion power turbine with a 70% isentropic efficiency, to generate power from the high pressure of 20,6 Bar Abs and 50°C saturated, are shown in the sketch. With the thermodynamic cycle efficiency of :

$$\eta_{\_ORC} = \frac{Q_{\_in}}{W_{\_turbine}} = 7{,}49\%$$

The benchmark cycle deliver 1 kWe from the input heat (Q_in) of 13,35 kW at 50°C and reject (Q_rej) 12,35 kW heat at 10°C. The mass vapor flow in the cycle is chosen so that the input heat required (Q_in) of the ORC equals the heat output (Q_hot) delivered by the benchmark heat pump of figure 1 above. The cost of powering the ORC would therefore be the cost of heat provided by the heat pump being evaluated, in the VC type benchmark heat pump this cost would be **29 $ per MWh_th**.

This benchmark ORC is coupled to the heat pumps being evaluated **regeneratively**, with the heat pump output supplying the ORC input heat at 50°C via the heat pump output heat exchanger, which doubles as the heat input exchanger of the ORC power unit. The ORC low temperature heat rejection coil is also placed inside the heat pump evaporator or cold side, where the heat pump absorb and regeneratively re-use the ORC reject heat. This regenerative coupling decrease the amount of heat the heatpump extract from external sources, which is now only the energy difference, being (Q_cold - Q_rej). For this VC heat pump power delivery benchmark, the value of Q_rej is greater than Q_cold, so that the heat required from external sources (Q_cold - Q_rej) is negative, (-2,88 kW_th), requiring 2,88 kWh_th heat to be removed from the cycle by external cooling means for every kWh_e produced. The formed regenerative cycle therefore requires **heat rejection of 2,88 MWh_th at 10°C per MWh_e** produced.

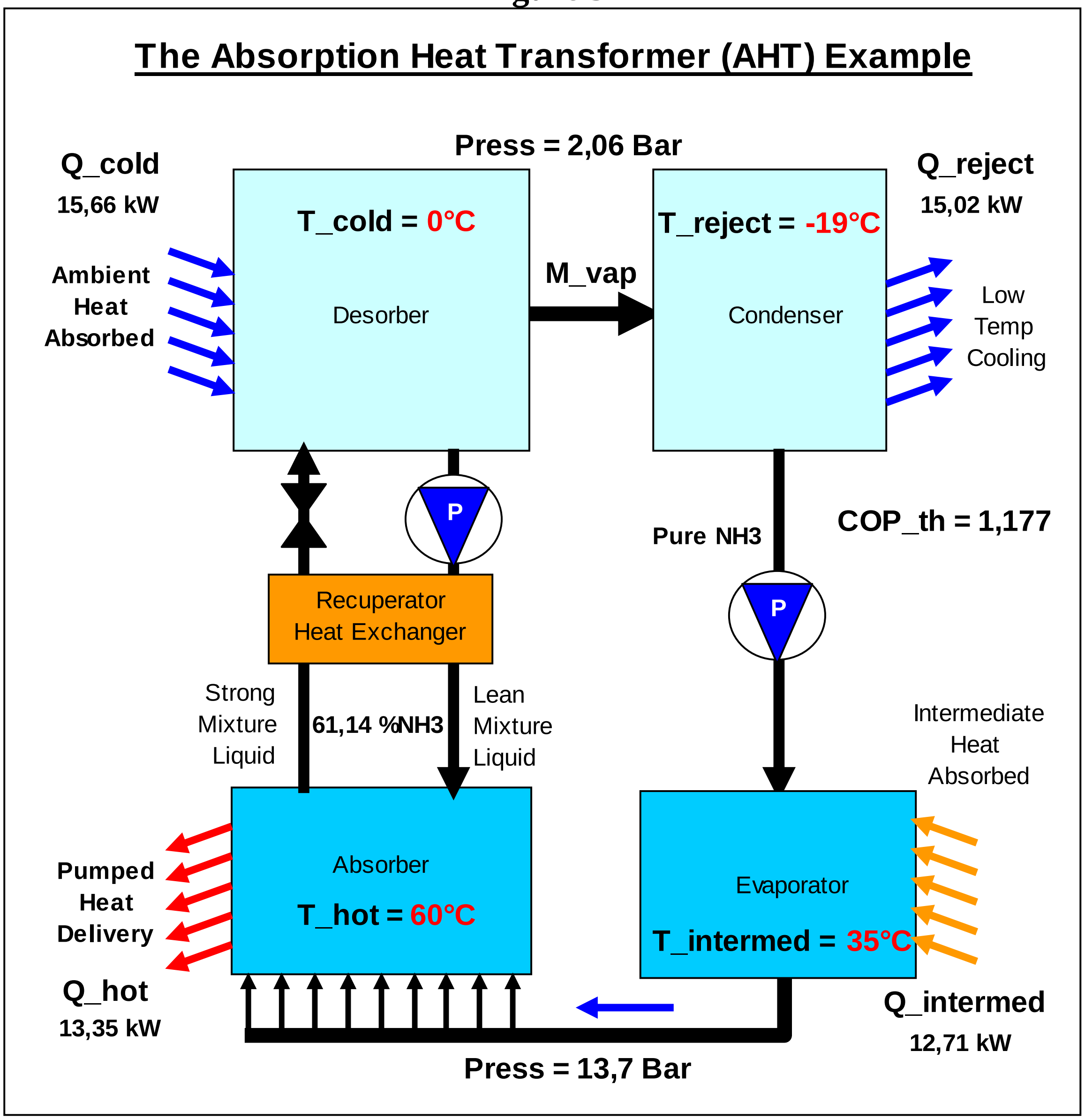
The power delivery benchmark is therefore a cost of 29 $ to deliver the power generated by one MWh_th as heat input to the ORC, which would develop 74,9 kWh_e from this amount of heat. The **Power Delivery Benchmark cost is therefore 387,2 $ per MWh_e.**

The Conventional Absorption Heat Transformer (AHT):

An interesting heat powered heat pump that we are not comparing with the benchmark, but still sketch here (in figure 3 below), have been in use widely for over 50 years, and it teaches us some interesting principles. This information on the standard state-of-the-art AHT is taken from W. Rivera [3] compiled already in 2000. The standard AHT cannot readily be compared to our benchmark, as it is intermediate input heat powered, which we do not have a standard way of costing.

It very clearly demonstrate the principle of upgrading heat to higher temperature using a vapor, releasing the heat of solution (HOS) in addition to the latent heat of condensation of the vapor in an absorber. The great pressure reduction of the vapor being absorbed into a lean binary liquid mixture in the absorber, instead of condensing it at the high saturation vapor pressure of the typical VC heat pump greatly reduce power requirements. In the

Figure 3



example sketched in figure 3, ammonia vapor saturated at 35°C is generated in the evaporator and is absorbed in the absorber (and heat it with the absorption heat) to 60°C, matching the 13,7 Bar pressure of the evaporator. The saturation vapor pressure of pure NH3 at 60°C is 26,5 Bar, double the actual absorber pressure, demonstrating the huge pressure reduction achieved by using an absorber instead of condenser.

The problem in this specific sketched example can clearly be seen in the low temperature heat rejection from the condenser. For this cycle to operate correctly, it requires cooling means at -19°C, which is not practical. To build the AHT for a desorber temperature of 0°C and a pumped heat delivery of 60°C using NH3 in aqua is simply not practical. Either a different temperature range need to be used, or different media eg. LiBr-H2O may be required and temperatures must then be designed so the lowest temperature in the cycle do not go below zero.

With this AHT the vapor required for creating the temperature lift is generated by releasing NH3 vapor at 0°C from the desorber by adding heat Q_cold. This cold vapor is then condensed and the liquid pumped to a higher pressure level from where it is again evaporated in at 35°C in the evaporator by adding additional heat, Q_intermed, at the intermediate higher temperature.

This elaborative way of producing the required high pressure vapor for use in the absorber to raise temperature, may be replaced with a simple vapor compressor, forming the AHT-VC Hybrid heat pump. It is noteworthy that all the further mentioned heat pumps are different members of the same family of HOS heat transformers, delivering heat at higher temperatures than the input heat.

In the VC type heat pumps, all the energy required for pumping the heat from the cold to the hot side comes from the compressor drive, namely electricity energy input. That is why the COP_e value represent the true thermodynamic efficiency of the VC heat pump, as it is calculated by dividing the heat produced by the energy required to produce that heat, namely the compression power used. **In heat transformers (AHT's) however, part of the energy required to pump heat, is sourced from the absorbed heat, while the balance is electrical energy for powering the compressor.** The calculated COP_e values for all the heat transformer type heat pumps only reflect the amount of electricity used, and not the total energy required for pumping the heat. The thermodynamically correct calculation of COP would therefore be:

$$COP_{Thermodynamic} = \frac{Q_{heat\_pumped}}{Q_{used\_heat} + W_{compression}}$$

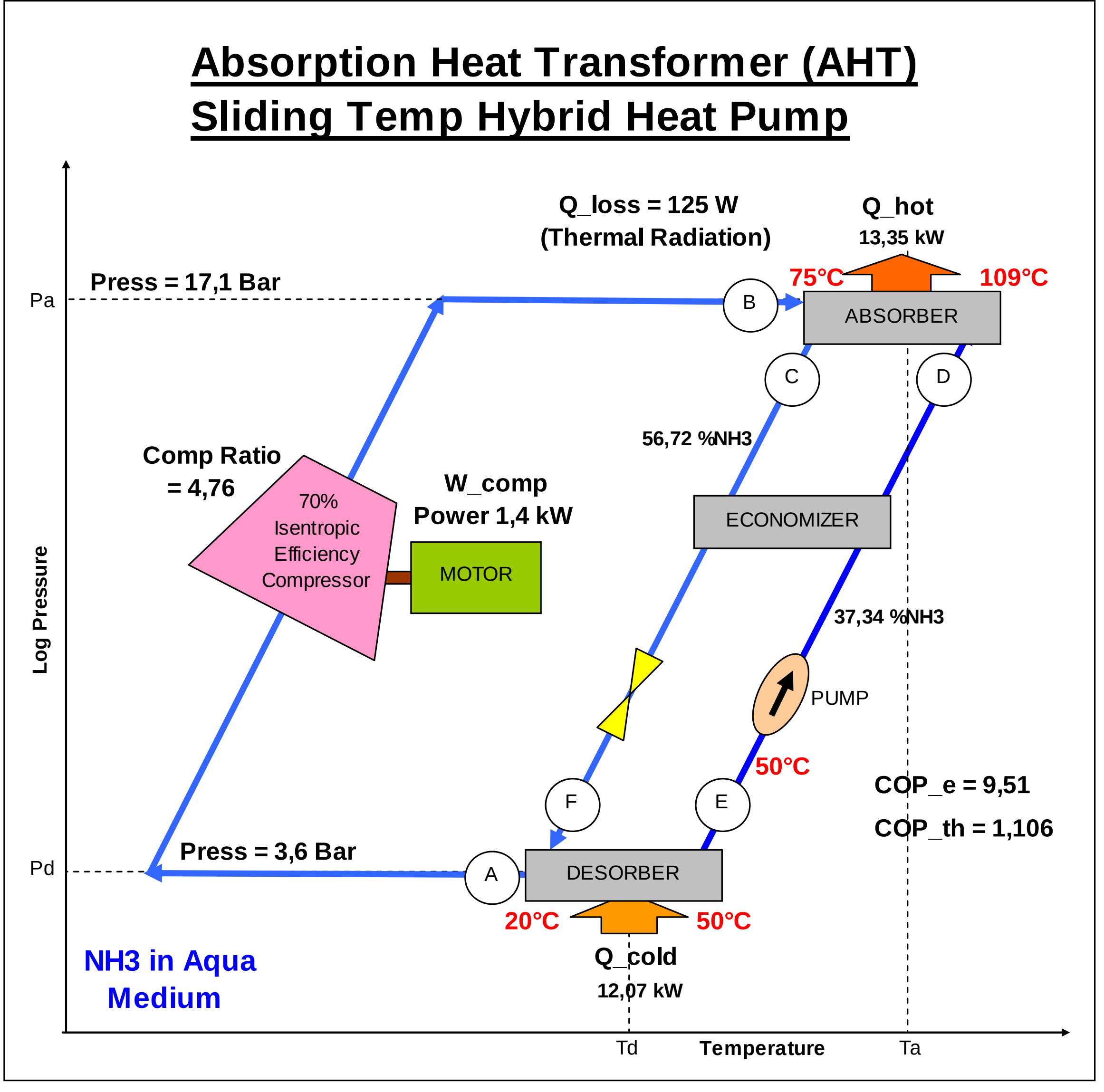
where the pumped heat and the power used for compression are generally known entities, while the heat used to assist in the pumping process, are not always clearly defined. In the AHT heat pump sketched in figure 3, above, however, this used heat is clearly defined as Q_intermed. This real thermodynamically correct COP calculations are also only of academic importance to us for the purpose of this paper, as we are really interested in the cost of operating the heat pump, and therefore only in the COP_e values, as the electricity needed to operate the compressor is the only cost item. The additional heat required by the heat pump is of no real consequence, as it is free.

The AHT-VC Hybrid Heat Pump:

These principles have been proven by Nordtvedt et al [1] compiled into the IEA Handbook [2] detailing a Hybrid Heat Pump delivering 650 kW_th heat for the Norwegian Food Industry built in 2007. Actual process values from this paper are used and sketched in figure 4 below.

In this paper by Nordtvedt et al [1], the principles and use of sliding temperature heat exchangers are made clear. This technology operating on the Osenbrück cycle (condensation and evaporation have been replaced with absorption and desorption processes) is also comprehensively discussed in the Ph.D thesis paper of Jensen [5] compiled in 2015, and confirmed by Borgås [6] in his Masters thesis of June 2014. This,

Figure 4



what Jensen and Borgås call the HACHP, is ideally suited to follow the Lorenz cycle. The high pressure absorber is a zeotropic binary liquid mixture heat exchanger that, during operation, have a sliding temperature gradient across it, spanning 109°C at the hot end and 75°C at the cold end, at the same constant pressure of 17,1 Bar. The binary mixture at the hot end have an ammonia concentration of 37,34% NH3 in aqua, while the cold end have a concentration of 56,72% NH3 in aqua. The vapor absorption heat delivered, and the gradually increasing temperature of the heated water (heat load) introduced into the heat exchanger at 50°C and leaving at 83°C help to maintain these temperature- and species concentration gradients.

Similar temperature- and ammonia concentration gradients exist in the sliding temperature desorber heat exchanger, as heat is extracted from a hot stream gradually cooling down to the exit stream around 30 - 35°C.

The huge advantage in performance visible in the high COP_e value (realizing we ignored the heat used as part of the heat pumping energy), as well as the lower compressor power used of this sliding temperature concepts are very clear if we compare it with the COP_e value and power used by the same heat pump, but using single fixed temperatures in the heat exchangers as sketched in figure 5 below. The main advantages of this Hybrid Absorption-Vapor Compression cycle is reflected in the high COP values attained due to the increased reversibility resulting from the reduction in thermal inefficiencies (exergy destruction) brought about by the sliding temperatures in both the absorber and desorber in coupling with external working streams. The vapor pressure reduction of absorption and desorption pressures as compared to the pure NH3 VC heat pump obviously also drastically reduce power requirements.

This sliding temperature heat exchanger concept is also the differentiator for the well documented Kalina cycle vs. the standard fixed temperature Rankine cycle.

The sliding temperature hybrid cycle COP_e value is 52% higher than the fixed temperature, with corresponding compression power reduced by the same percentage.

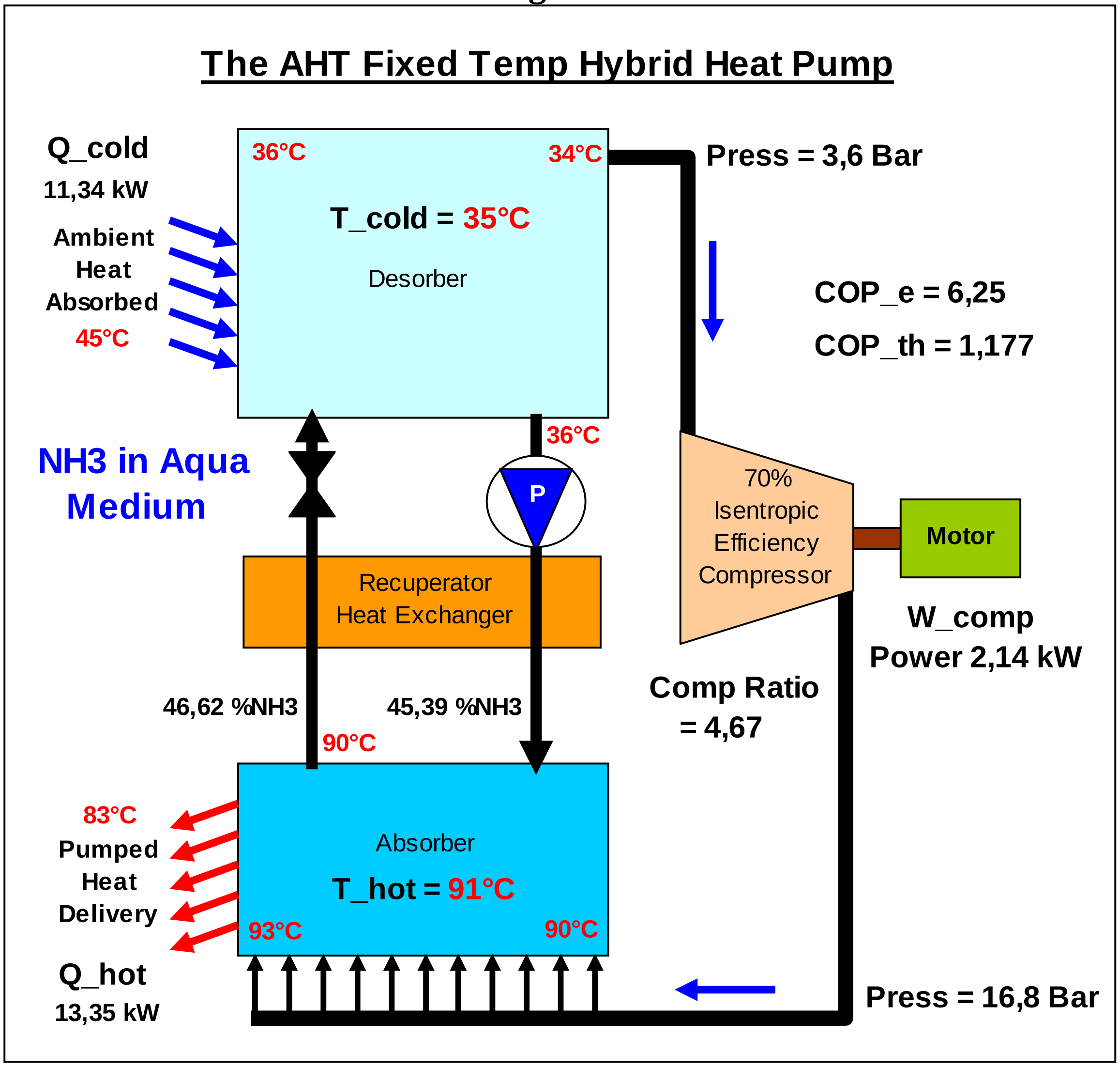
The two sketches, figure 4 and 5, relate to the referenced paper of Nordtvedt, and cannot directly be compared to our defined benchmark heat pumps, as the input and output temperature levels are not the same. A lower temperature replica process is defined as the comparable AHT-VC Hybrid heat pump sketched in figure 6 below. This process is identical to the Nordtvedt and Jensen process (Osenbrück Cycle), except the NH3 concentration levels have been adjusted to result in the correct temperature levels so we are able to compare to our benchmarks.

The same as for our benchmarks, for this AHT-VC Hybrid heat pump, the definition (COP_e) for cooling service is defined as:

$$COP\_e = \frac{Q_{\_cold}}{W_{\_comp}} = 8{,}77$$

This calculate to the cost of **Cold delivery of 11,4 $ per MWh_th.** The same AHT-VC Hybrid heat pump may also be used for delivering high temperature heat (at 50°C) where

Figure 5



the electrical efficiency is defined as:

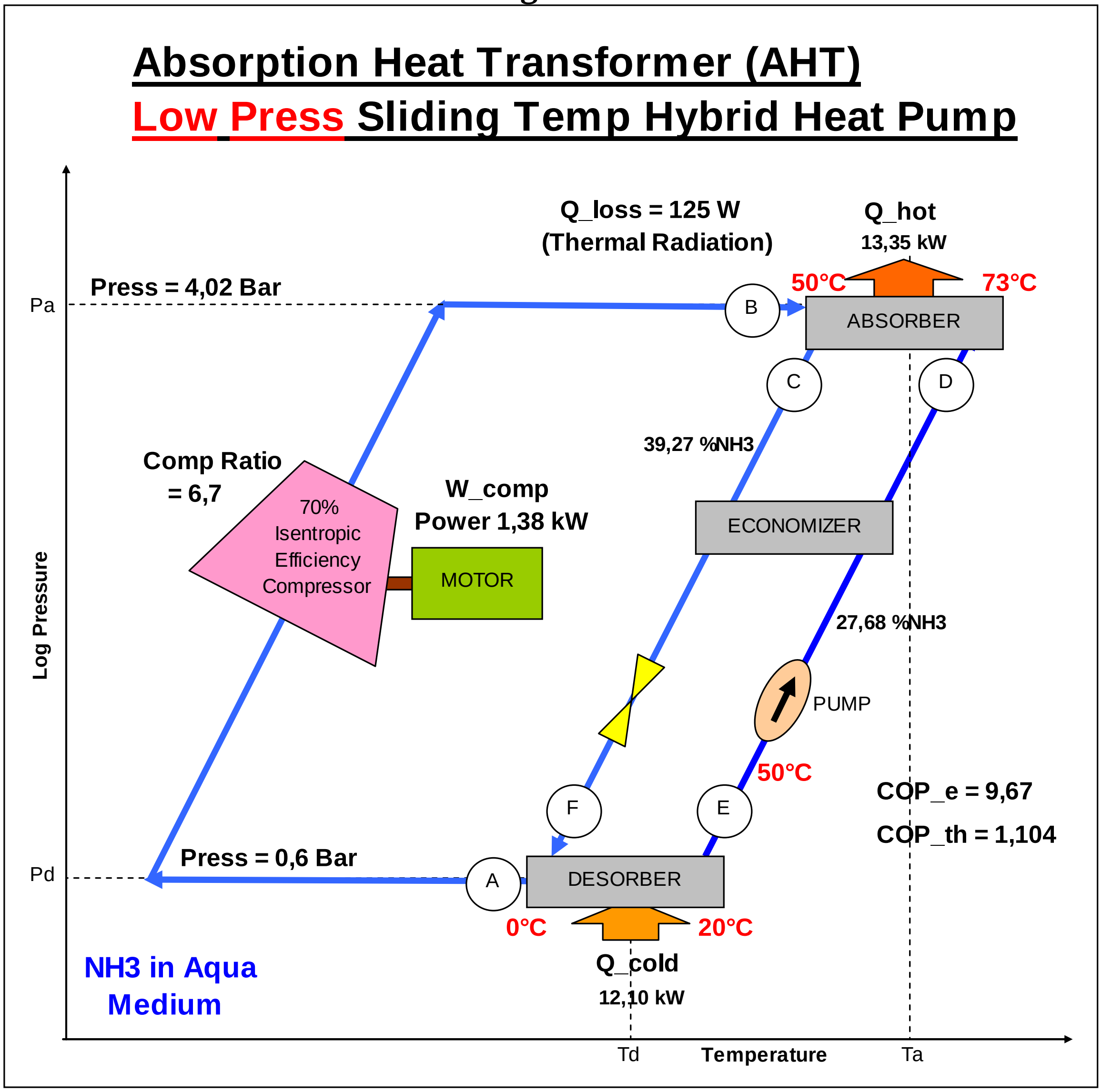
$$COP\_e = \frac{Q_{\_hot}}{W_{\_comp}} = 9{,}67$$

while the thermal efficiency (COP_th) for heating service is defined as:

$$COP\_th = \frac{Q_{\_hot}}{Q_{\_cold}} = 1{,}104$$

This same AHT-VC Hybrid heat pump therefore deliver high temperature **heat at a cost of 10,3 $ per MWh_th.**

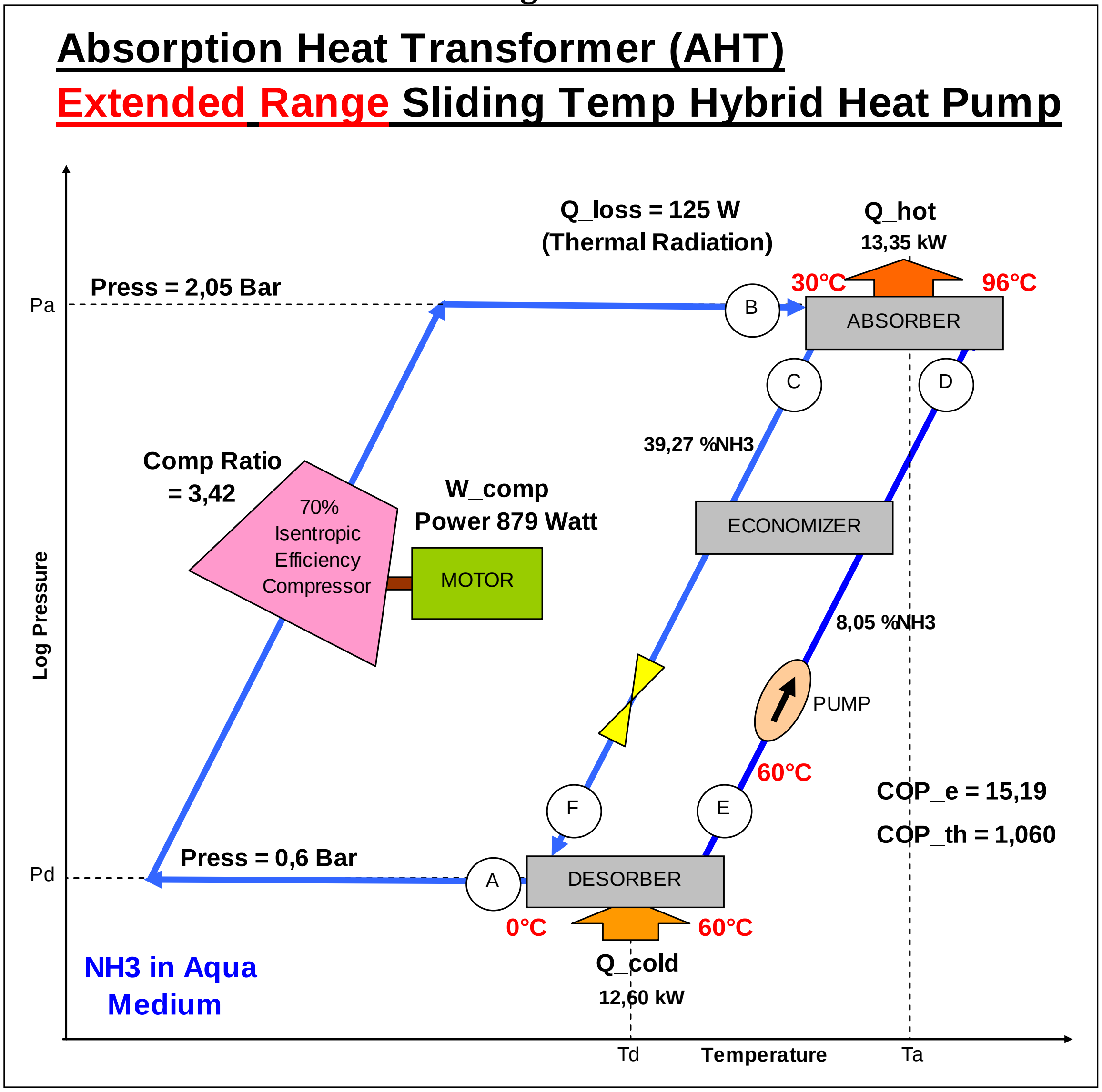
Figure 6



The calculated power delivery value is therefore 10,3 $ to deliver the power generated by one MWh_th as heat input to the ORC, which would develop 74,9 kWh_e with this amount of heat. The **Power Delivery of the cycle formed** (combined AHT-VC Hybrid heat pump and ORC) **would therefore cost 137,5 $ per MWh_e.**

Also, for this regenerative cycle in power delivery, the value of Q_rej is greater than Q_cold, so that the heat required from external sources (Q_cold - Q_rej) is negative, (-0,25 kW_th), requiring 0,25 kWh_th heat to be removed from the cycle by external cooling means for every kWh_e produced. The formed regenerative cycle (combined AHT-VC Hybrid heat pump and ORC) therefore requires **heat rejection of 0,25 MWh_th at 10°C per MWh_e** produced.

Figure 7



The Extended Range Sliding Temperature AHT-VC Hybrid Heat Pump:

Using the same AHT-VC Hybrid model as presented by Nordtvedt, it could easily be recognized that the sliding temperature range of the desorber may be extended to span 0°C to 60°C, even though the input heat exchanger coil may deliver heat (Q_cold) at any temperature between these range extremes. Similarly, the absorber design may be done extending the sliding temperature range to be 30°C to 96°C, even though the heat exchanger coil in the absorber only remove heat (Q_hot) at 50°C. These extended sliding temperature ranges have the effect of also decreasing the high pressure relative to the low pressure, and therefore the required compression ratio, with the corresponding decrease in compression power requirements. This extended sliding temperature range AHT-VC Hybrid heat pump is sketched in figure 7, above.

The same as before, for this Extended Range Sliding Temperature AHT-VC Hybrid heat pump, the definition (COP_e) for cooling service is defined as:

$$COP\_e = \frac{Q_{\_cold}}{W_{\_comp}} = 14{,}33$$

This calculate to the cost of **Cold delivery of 6,98 $ per MWh_th.** The same Extended Range Sliding Temperature AHT-VC Hybrid heat pump may also be used for delivering high temperature heat (at 50°C) where the electrical efficiency is defined as:

$$COP\_e = \frac{Q_{\_hot}}{W_{\_comp}} = 15{,}19$$

while the thermal efficiency (COP_th) for heating service is defined as:

$$COP\_th = \frac{Q_{\_hot}}{Q_{\_cold}} = 1{,}060$$

This same Extended Range Sliding Temperature AHT-VC Hybrid heat pump therefore deliver high temperature **heat at a cost of 6,58 $ per MWh_th.**

The calculated power delivery value is therefore 6,58 $ to deliver the power generated by one MWh_th as heat input to the ORC, which would develop 74,9 kWh_e with this amount of heat. The **Power Delivery of the cycle formed** (combined Extended Range Sliding Temperature AHT-VC Hybrid heat pump and ORC) **would cost 87,9 $ per MWh_e.**

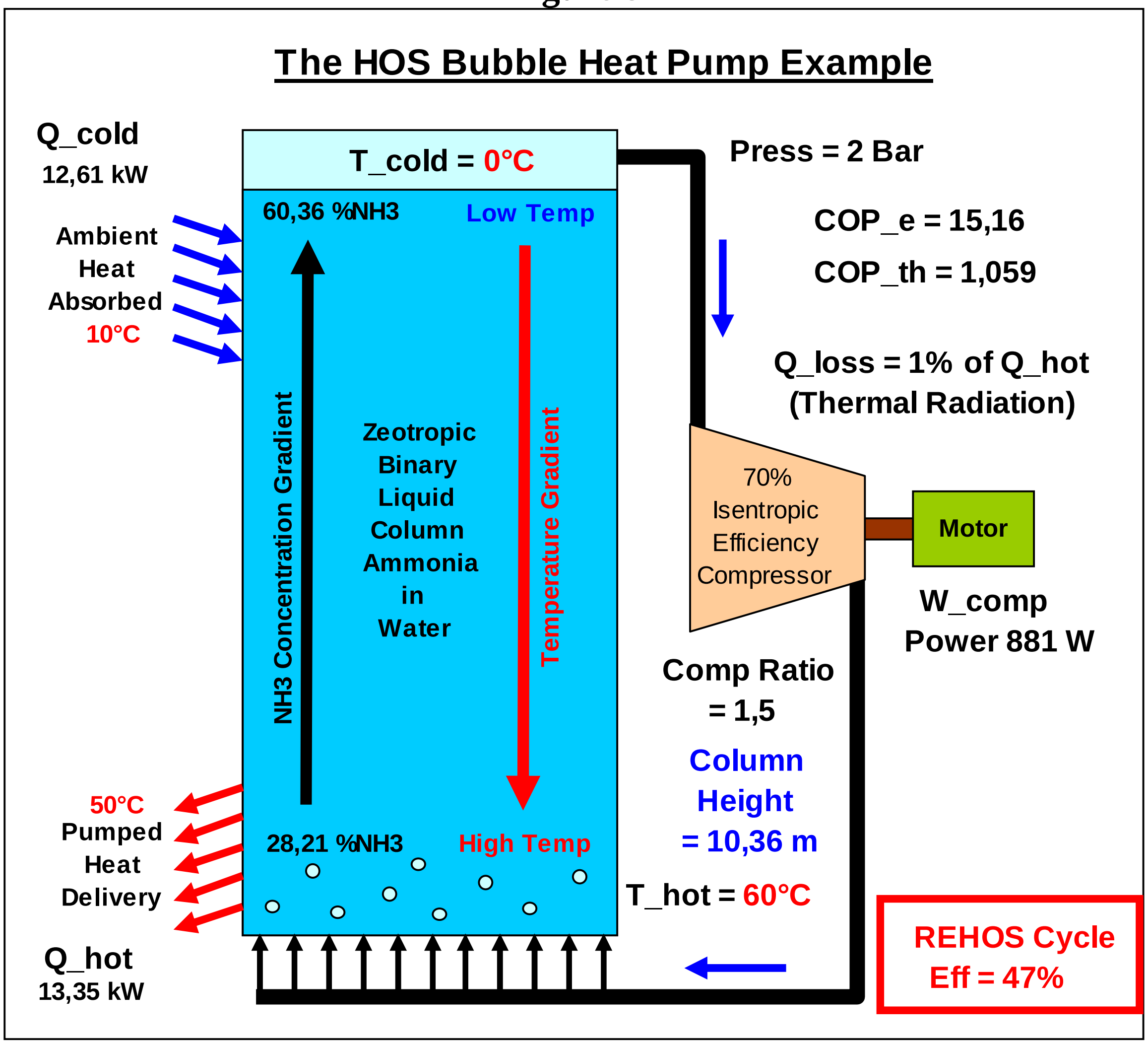
Also, for this regenerative cycle in power delivery, the value of Q_rej is smaller than Q_cold, so that the heat required from external sources (Q_cold - Q_rej) is positive, namely (0,25 kW_th), requiring 0,25 kWh_th heat to be added to the cycle by external ambient heat for every kWh_e produced. The formed regenerative cycle (combined Extended Range Sliding Temperature AHT-VC Hybrid heat pump and ORC) therefore requires **additional heat of 0,25 MWh_th at ambient temperature per MWh_e** produced.

The binary NH3 in aqua mixtures circulating (being pumped) between the desorber and absorber of the Nordtvedt [1], Jensen [50] and Borgås [6], hybrid cycle flow in separate tubes (C - F) and (E - D) and exchange heat in an economizer heat exchanger, but NH3 concentration stay the same during the flow from (C) to (F), and also from (E) to (D). Concentration changes only happen in the sliding temperature absorber and desorber. This does not necessarily have to be like this, however....

The HOS Bubble Heat Pump:

To enhance this hybrid cycle, it would be logical to combine the countercurrent binary mixture flow streams and economizer heat exchanger into a single binary liquid mixture column. This allow direct contact of the two countercurrent liquid streams, resulting in far greater direct contact heat transfer rates between the two streams (enhanced economizer action). The desorption process would then take place in the top section of the column where vapor is withdrawn (and flashed off), cooling this section, while the heat generating, absorption process where the compressed vapor is introduced into the mixture take place in the bottom section of the binary column. This also provide sections

Figure 8



of the column where sliding temperature at a relatively constant pressure take place, very similar to the Lorenz Cycle hybrid models discussed above. The bubbling compressed vapor entering the column at the bottom also perform a vapor lift pumping action, driving the liquid counter flow circulation. During the countercurrent liquid mixture flows, both heat and species exchange would take place, enhancing the single temperature and ammonia concentration gradient formation and maintenance throughout the column, instead of the separate temperature and concentration gradient sections of desorber and absorber. The process sketched in figure 8 represent such an enhanced HOS heat pump.

As can be seen from figure 8, above, the HOS Bubble heat pump with a binary column height of 10,36 meters operating at a cold desorber saturation pressure of 2 Bar Abs, compares quite well in performance (COP values) with the Extended Range Sliding Temperature AHT-VC Hybrid heat pump of figure 7.

This HOS Bubble heat pump sketched in figure 8, combined regeneratively with the benchmark power generating ORC, form the REHOS cycle, and the efficiency may be expressed by the balance of power (Power generated - Compressor power) divided by the heat to be added for balancing the cycle (Q_cold - Q_rej):

$$\eta_{\_REHOS} = \frac{(W_{\_power} - W_{\_comp})}{(Q_{\_cold} - Q_{\_rej})} = \frac{(1000W_{\_e} - 881W_{\_e})}{(12,61kW_{\_th} - 12,35kW_{\_th})} = 47\%$$

The high efficiency is attributable to the differentiation from the Jensen [5] standard HACHP that he had done a comprehensive Advanced Exergy-based analysis on. Jensen concluded the compressor is responsible for the largest exergy destruction in the cycle (26%), closely followed by absorber (24%) and desorber (21%) of exergy destroyed in the cycle. Reducing the compression function drastically by reducing the compression ratio, therefore increase cycle efficiency. Extending the sliding temperature of the desorber to cover the full range (60°C to 0°C) further decrease irreversibility in the desorber function and also increase cycle efficiency, reflected in the higher COP.

The Optimized HOS Bubble Heat Pump:

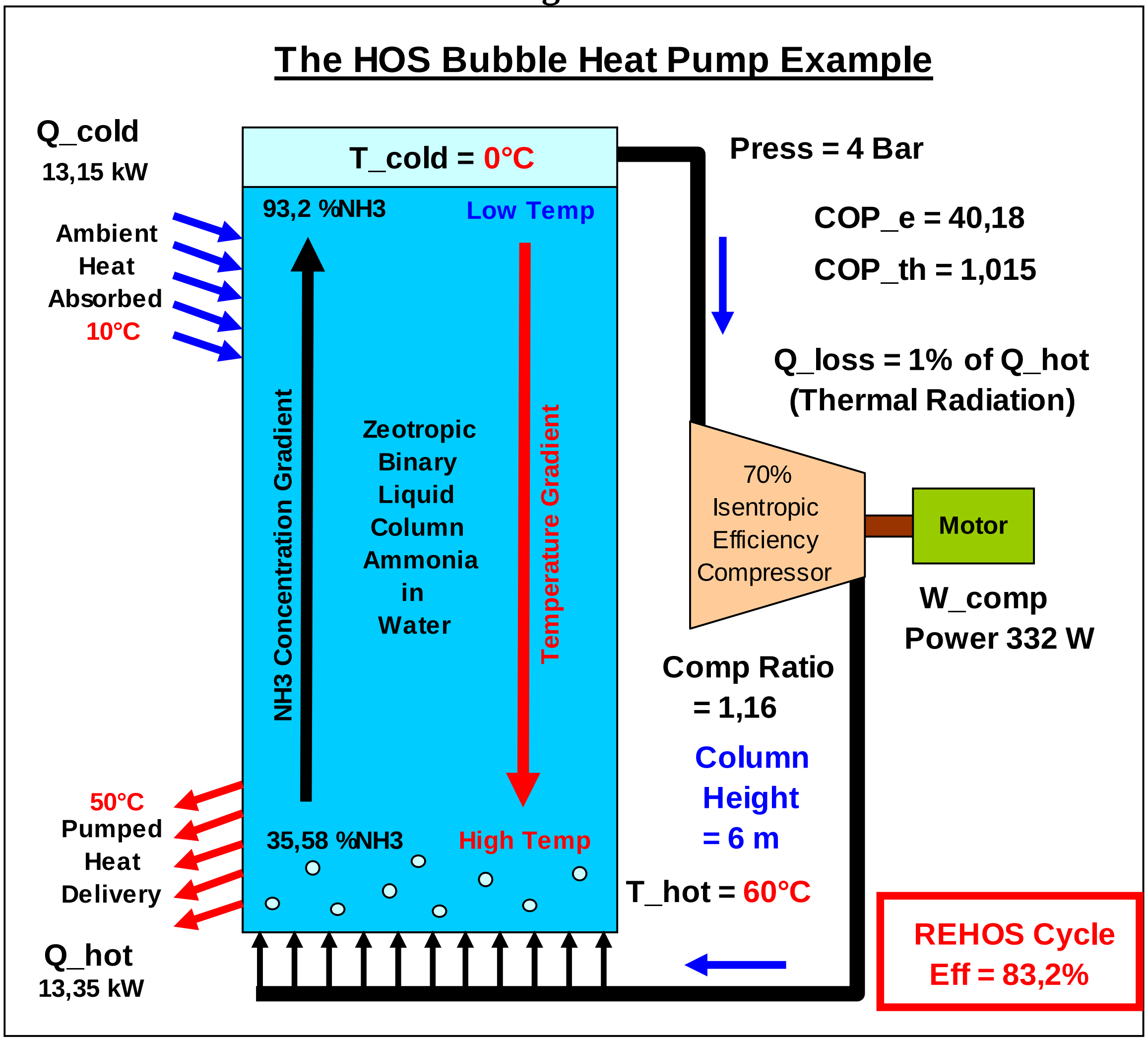
Using this novel HOS Bubble heat pump concept, it is very simple to further increase the performance, by simply increasing the percentage NH3 in the mixture that would increase the operating saturation pressure. The heat pump efficiency would be further increased by decreasing the column height which reduce the compression ratio, and therefore the compressor power consumption. The optimized HOS Bubble heat pump with 6 meter column and operating pressure of 4 Bar Abs is sketched in figure 9 for comparison.

The same as before, for this optimized HOS Bubble heat pump, the definition (COP_e) for cooling service is defined as:

$$COP\_e = \frac{Q_{\_cold}}{W_{\_comp}} = 39,61$$

This calculate to the cost of **Cold delivery of 2,52 $ per MWh_th.** The same optimized HOS Bubble heat pump may also be used for delivering high temperature heat (at 50°C) where the electrical efficiency is defined as:

Figure 9



$$COP\_e = \frac{Q_{\_hot}}{W_{\_comp}} = 40{,}18$$

while the thermal efficiency (COP_th) for heating service is defined as:

$$COP\_th = \frac{Q_{\_hot}}{Q_{\_cold}} = 1{,}015$$

This same optimized HOS Bubble heat pump therefore deliver high temperature **heat at a cost of 2,49 $ per MWh_th.**

The calculated power delivery value is therefore 2,49 $ to deliver the power generated by one MWh_th as heat input to the ORC, which would develop 74,9 kWh_e with this amount of heat. The **Power Delivery of the cycle formed** (combined optimized HOS Bubble heat pump and ORC) **would cost 33,24 $ per MWh_e.**

Also, for this regenerative cycle in power delivery, the value of Q_rej is smaller than Q_cold, so that the heat required from external sources (Q_cold - Q_rej) is positive, namely (0,8 kW_th), requiring 0,8 kWh_th heat to be added to the cycle by external ambient heat for every kWh_e produced. The formed regenerative cycle (combined optimized HOS Bubble heat pump and ORC) therefore requires **additional heat of 0,8 MWh_th at ambient temperature per MWh_e** produced. The thermodynamic cycle so formed (regeneratively combined optimized HOS Bubble heat pump and ORC) we named the **Regenerative Heat of Solution (REHOS) cycle**.

This optimized HOS Bubble heat pump sketched in figure 9, combined regeneratively with the benchmark power generating ORC, forming the REHOS cycle, and the efficiency may be expressed by the balance of power (Power generated - Compressor power) divided by the heat to be added for balancing the cycle (Q_cold - Q_rej) similar to what we had before:

$$\eta_{\_REHOS} = \frac{(W_{\_power} - W_{\_comp})}{(Q_{\_cold} - Q_{\_rej})} = \frac{(1000W_{\_e} - 332W_{\_e})}{(13{,}15kW_{\_th} - 12{,}35kW_{\_th})} = 83{,}2\%$$

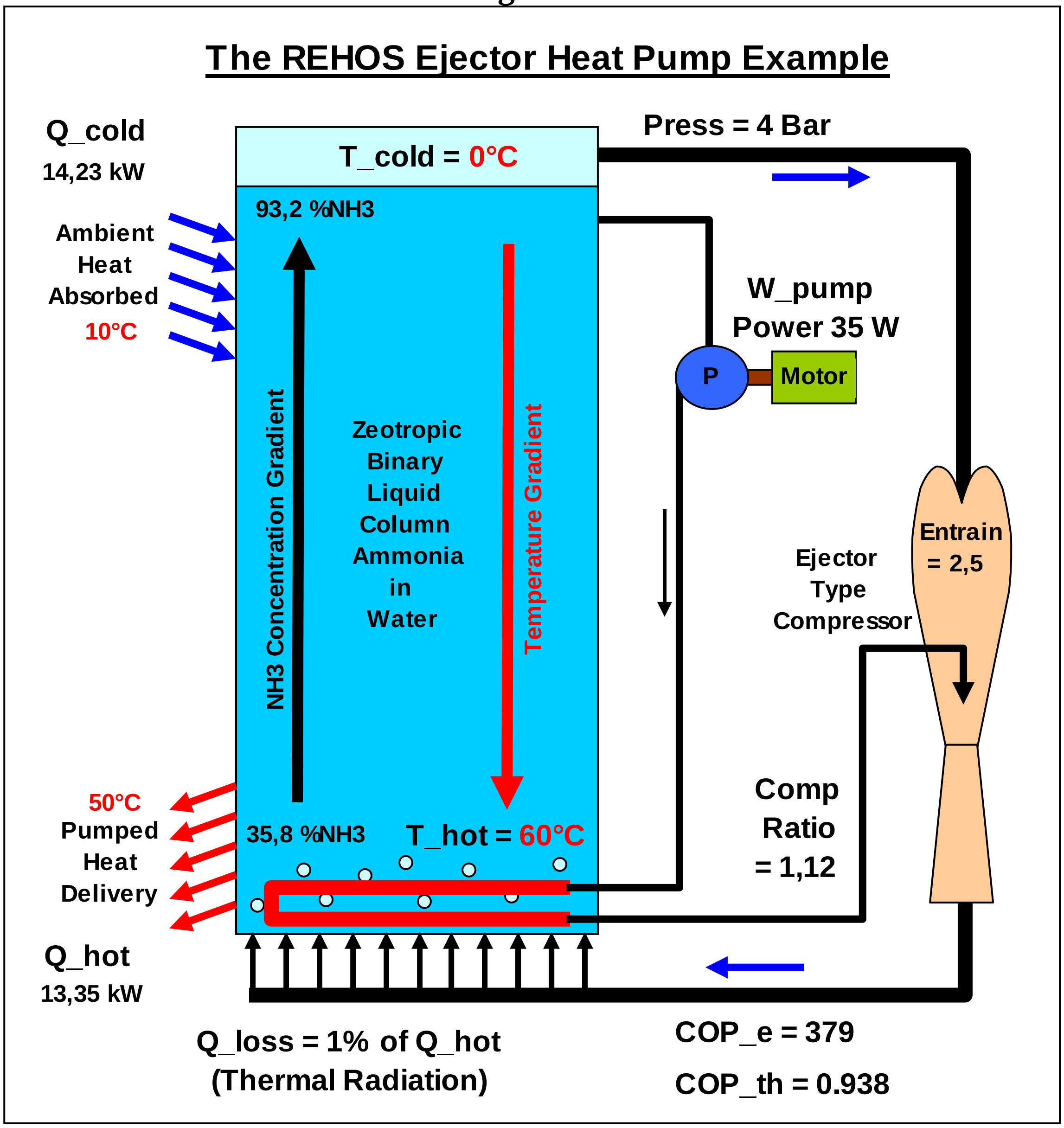
<u>The REHOS Ejector Heat Pump:</u>

This is a derivative of the HOS Bubble heat pump as described with the sketch in figure 9, in that the electrically driven compressor was replaced with an ejector type. This have an advantage resulting from the drastically reduced electric power used (but more heat used), resulting in an extremely high COP_e value.

The REHOS Ejector heat pump utilize some of the pumped heat (18% of Q_hot in our example) to evaporate high pressure NH3 liquid to vapor regeneratively for powering the ejector compressor. This mean that the heat pump use more absorbed heat (from Q_cold input heat), but at least an order of magnitude lower electricity for the compressing function due to the high density of liquid being pumped vs. vapor compression.

Ejector type compressors are only used when the compression ratio's are small, but this fit the developed REHOS heat pump application like a glove. It also have no moving parts, requiring extremely low maintenance, and are very cheap to manufacture. As vapor compressor, the efficiency of the ejector type compressor is fairly low (being only a few percent) compared to the 60 - 80% of mechanical compressors, but the heat generated by the vapor compressor inefficiency is all present in the compressor outlet stream, and is re-used in the bubble reactor regeneratively to evaporate the pumped high pressure NH3 liquid. This regeneration add tremendously to the overall heat pump performance, as can be seen in the high COP values calculated below. It may be very beneficial for heating and cooling applications, although it uses a portion of the heat pumped, and would therefore have lower power generated output for the same heat used when compared to other compressors. The amount of electricity used by this heat pump is very much lower than the previous HOS Bubble heat pump sketched in figure 9, above, resulting from the low power requirements of hydraulic pumping, compared to vapor compression.

Figure 10



The same as before, for this Regenerative HOS heat pump, or REHOS Ejector heat pump, the definition (COP_e) for cooling service is defined as:

$$COP\_e = \frac{Q_{\_cold}}{W_{\_pump}} = 406$$

which is two orders of magnitude larger than previous heat pumps due to the fact that the liquid pump energy use is a few orders of magnitude lower than a vapor compressor. Thiscalculate to the cost of **Cold delivery of $0-25 per MWh_th.** The same REHOS Ejector heat pump may also be used for delivering high temperature heat (at 50°C) where the electrical efficiency is defined as:

$$COP\_e = \frac{Q_{\_hot}}{W_{\_pump}} = 380$$

while the thermal efficiency (COP_th) for heating service is defined as:

$$COP\_th = \frac{Q_{\_hot}}{Q_{\_cold}} = 0{,}938$$

This same REHOS Ejector heat pump is therefore able to deliver high temperature **heat at a cost of $0-26 per MWh_th.**

The calculated power delivery value is therefore $0-26 to deliver the power generated by one MWh_th as heat input to the ORC, which would develop 74,9 kWh_e with this amount of heat. The **Power Delivery of the cycle formed** (REHOS Ejector heat pump and ORC) **would cost 3,47 $ per MWh_e.**

Also, for this regenerative cycle in power delivery, the value of Q_rej is smaller than Q_cold, so that the heat required from external sources (Q_cold - Q_rej) is positive, namely (1,88 kW_th), requiring 1,88 kWh_th heat to be added to the cycle by external ambient heat for every kWh_e produced. The formed regenerative cycle (REHOS Ejector heat pump and ORC) therefore requires **additional heat of 1,88 MWh_th at ambient temperature per MWh_e** produced, relating to a REHOS cycle efficiency of 53,2%.

These ridiculously low costs delivered by the REHOS Ejector heat pump are too revolutionary to include in our comparison summary, making electricity, air conditioning (and water from de-humidifying air) and low temperature heating essentially free! The REHOS Ejector heat pump Proof-of-Concept Model would also be the topic of the next paper.

We therefore ignore it in our further discussions and leave it to your own imagination.....

The Comparison Evaluation Results Summary:

Table 1

Cost of Cold (refrigeration) Delivery:

Technology	COP	Cost $/MWh_th	% of Benchmark
VC Heat Pump	2,44	41,00	100%
AHT-VC Hybrid Heat Pump	8,77	11,40	28%
Extended Range Sliding Temp AHT-VC Hybrid Heat Pump	14,33	6,98	17%
Novel HOS Bubble Heat Pump	39,61	2,52	6,1%

Table 2

Cost of Heat (Geyser) Delivery:

Technology	COP	Cost $/MWh_th	% of Benchmark
Benchmark Electricity Cost		100,00	100%
VC Heat Pump	3,44	29,00	29%
AHT-VC Hybrid Heat Pump	9,67	10,30	10%
Extended Range Sliding Temp AHT-VC Hybrid Heat Pump	15,19	6,58	6,6%
Novel HOS Bubble Heat Pump	40,18	2,49	2,5%

Table 3

Cost of Power Delivery via ORC coupling:

Technology	Cost $/MWh_e	% of Benchmark	Heat Rejection MWh_th/MWh_e
Benchmark Electricity Cost	100,00	100%	
VC Heat Pump	387,20	387%	2,88
AHT-VC Hybrid Heat Pump	137,50	138%	0,25
Extended Range Sliding Temp AHT-VC Hybrid Heat Pump	87,90	88%	-0,25
Novel HOS Bubble Heat Pump	33,24	33%	-0,80

Discussion:

Air conditioning specifically in warm to hot climates and in industries like mining represent a huge percentage of energy consumption globally and the majority of these systems make use of VC heat pump principles. Looking at table 1, it is encouraging to know that more recently proven technologies like AHT-VC Hybrid heat pumps are able to create an electricity cost saving of conservatively calculated to 72% (as real heat pump operational cost is only 28% of the benchmark VC machines), and derivatives of this technology promising as high as 83% cost saving! It definitely makes economic sense to replace the traditional VC heat pumps and, looking at table 1, seriously drive deployment of the novel HOS Bubble heat pump technology able to cut air conditioning and refrigeration power consumption by a whopping 93%!

Globally the extraction of water from air (de-humidifiers) for human consumption, attracts more and more attention, due to water shortages in specific arid countries and drought-stricken cities. Small-scale water extraction machines marketed for potable water supply for building blocks, hotels, hospitals and schools in the 0,5 - 1,0 m3/day capacity range, operate at a power level of about 0,389 - 0,775 MWh_e /m3 water extracted from air with humidity levels around 50% - 70%. On average the power consumption is therefore 0,582 MWh_e /m3 of potable water, at an electricity cost (our benchmark cost) of 100 $ /MWh_e the water would be costing $58-21 /m3 water.

The technology used is largely built around the VC heat pump, and is therefore expensive, but introducing a different, proven heat pump technology of the heat transformer type, like AHT-VC Hybrid heat pumps, could decrease the cost of potable water from these machines by conservatively 72%, bringing the potable water cost down to $16-30 /m3, making a lot of commercial cense. Obviously, introducing the novel, but simple HOS Bubble heat pump technology would see the potable water production cost below $4-07 /m3, creating a real commercial revolution if you compare it to municipal water distributed in cities around the world today. Even desalination concepts would need a commercial viability re-evaluation. Consider that drought-stricken cities like Cape Town in South Africa implemented severe water restrictions already in July 2017, pricing domestic water at $1-48 /m3 for users > 6 m3/month and on a sliding scale $3-64 /m3 for > 20 m3/month usage, while users using > 35 m3/month pay an incredible $9-50 /m3. Schools, Government buildings, industrial and commercial users pay $2-06 /m3. Increased pressure on dwindling water resources have now actually doubled these already high water costs listed here as from February 1, 2018, making the cost of water to Schools, Government buildings, industrial and commercial users now $4-75 /m3! The simple HOS Bubble heat pump and other heat transformer type technology could clearly revolutionize potable water production for the global community in drought-stricken cities across the globe.

Looking at table 2, the reason why heat pumps often replace the more traditional electric geysers for domestic hot water is quite clear, in that the VC heat pump technology create an electricity saving of conservatively 71% on providing washing, showering and bathing household hot water, not to mention larger uses like in-house swimming pool heating and the like. As the figures in table 2 highlight, even more effective money-saving technologies have already been proven in the heat transformer type heat pumps, providing savings around 90% from the electrical heating benchmark. It is also very clear that low temperature (50°C) heat would be practically free using the novel HOS Bubble heat pump technology. Simplicity of this technology process also make it the logical choice for replacing the expensive electrical heating geysers and similar applications.

Looking at table 3, we immediately need to put the power that may be developed regeneratively combining the heat pumps with ORC power machines into another perspective, relating to current energy generation costs using different well-known sources. We therefore provide a short current listing (table 4) of electrical energy produced by some renewable (but intermittent) sources like solar PV and wind, together with some traditional widely used power generators like coal, natural gas, diesel and nuclear power. These numbers were borrowed from the unsubsidized, levelized cost of energy from LAZARD's analysis [4] written November 2017.

Table 4

Range of Cost (minimum & maximum) of Power Delivery numbers from LAZARD:

Generation Technology	Min Cost $/MWh_e	Max Cost $/MWh_e
Crystalline Utility-Scale Solar PV	46	53
Thin-Film Utility Scale PV	43	48
Wind	30	60
Natural Gas -Reciprocating	68	106
Natural Gas -CC	42	78
Diesel Reciprocating	197	281
Coal	60	143
Nuclear	122	183

Remembering that table 3 represent the power generation capabilities of an ORC power cycle coupled regeneratively to a heat pump extracting heat from ambient water (20°C) and using this extracted environmental heat to produce power. It therefore represent baseload power, not plagued by the intermittency of other renewable generators like Solar PV and Wind power. We should therefore discard the first 3 entries in table 4, and only compare the heat pumps listed in table 3 to the baseload generation represented by the other 5 entries in table 4.

Note specifically that the proven AHT-VC Hybrid heatpump model as presented by Nordtvedt, coupled regeneratively as detailed before, deliver power at a cost of $137-50 /MWh_e , which is lower than the minimum cost for diesel generation at $197-00 /MWh_e by a full 30%, actually making diesel generation obsolete when introducing the mentioned coupled heatpump-ORC combinations. Obviously the baseload power cost of $33-24 produced from ambient waste heat by the regeneratively coupled HOS heatpump -ORC combination (REHOS) cycle pull the rug out from all other existing power generation means.

The power generation capabilities as listed in table 3 are not really correct for the last 2 entries in this table. The table 3 listings assume the heat pump is driven electrically, but using the expensive benchmark cost of electricity for generating the heat required by the ORC to generate power. For the last 2 entries in table 3, we note they are able to generate more electrical power than what is used by the heat pumps (obviously keeping in mind the heat pumps not only use electricity, but also a portion of the heat extracted from the environment). Should they be configured for powering their own compressors, instead of using expensive benchmark electricity for this purpose, the surplus electrical power generated by the ORC are actually totally free.............since the ambient waste heat used were said to be without any costs!

Obviously, in practical implementation the compressors used, as well as the ORC expander used would be optimized, and not have the 70% isentropic efficiency we assumed. Heat exchanger approach temperatures would also be optimized and not necessarily be the assumed 10°C we used in these comparisons. Real cycle efficiencies may therefore be substantially improved from those listed here......

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