Economic Aspects of Utilizing Heat Transformer Technology

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Abstract:

Heat Transformer technology (HT-technology), although commercially available, is relatively unknown. The fact that HT-technology use only ~ 1% of the electricity of the conventional vapor compression (VC)-technology for the same heat load, drive rapid revolutionary new heat recovery possibilities, however.

Some advances in heat transformer development open new doors for lowering the cost of air conditioning (A/C), water pumping and extraction (de-humidification) from the air, as well as power generation by combining with Organic Rankine Cycles (ORC), from utility scale down to micro-scale of a few kW_e for single household use. This paper try to realistically present cost calculations based on cost correlations for process components often used in the literature for estimating overall system costs.

A/C making use of HT-technology allow cost savings of > 99% over the traditional VC-types, while de-humidifiers built on HT-technology can decrease the cost of water extracted from the air to values of < $R5/m^3$ water produced, or in USD terms $0.36\,\$/m^3$.

Modern HT-principles make the recovery of heat even from ambient temperature water practical for utilization of small ORC coupling to pump water using the thermal energy in the water being pumped as power source! REHOS Autarkic Water (RAW)-Pump costs are calculated to ~ 5 x the standard electrical pump cost, but savings on not having to use electricity, repay the difference in as little as 3.5 years for the larger pumps.

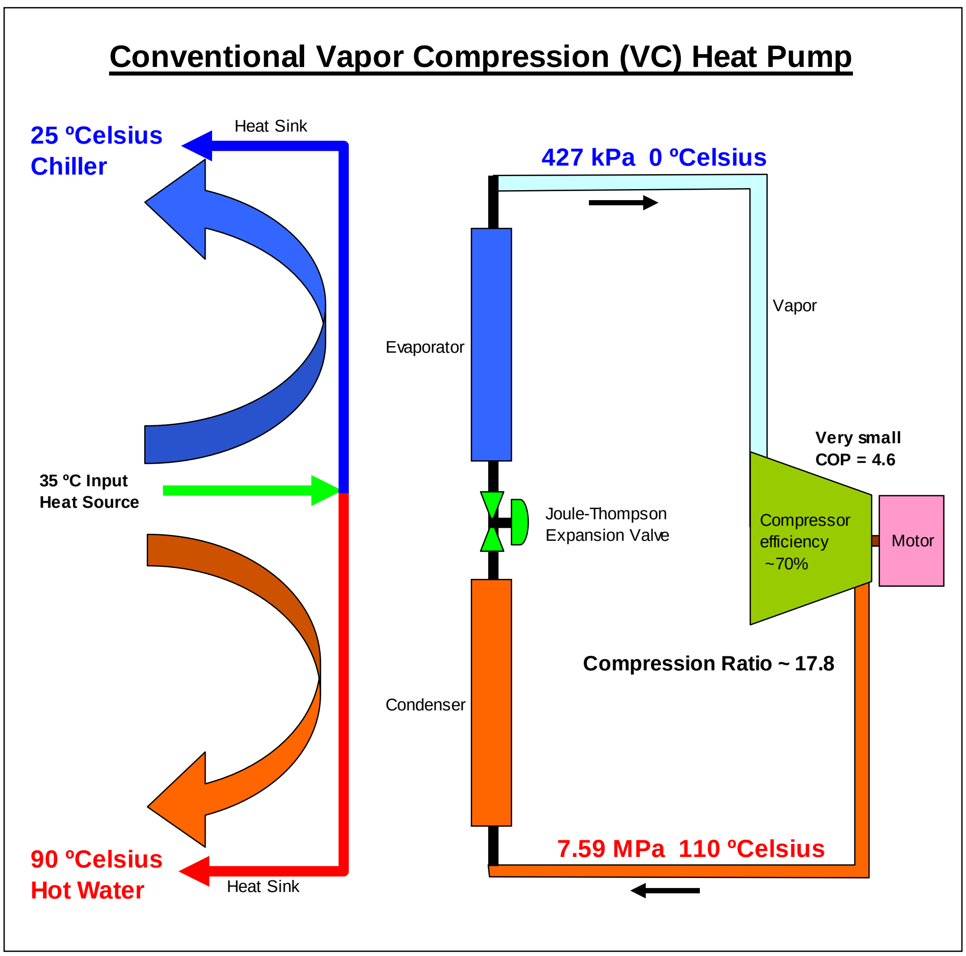
Utilizing HT-technology with ORC integration allow utility cooling water (CW) heat recovery for power generation as low cost as $22.1\,\$/MWh_e$ allowing huge Carbon Dioxide (CO_2) emission-, and water savings, while phasing in very practical, affordable stepwise de-carbonization of the fossil combustion Power Station (P/S). The same Regenerative Heat of Solution (REHOS) cycle may, on micro-scale, generate power using a swimming pool (solar pond) as heat source delivering electricity at extremely low rates (~ 50% of grid parity) eg. Levelized Cost of Electricity (LCOE) calculated for a $20\,kW_e$ for a REHOS-Pond be: $LCOE_{20kW_e} = 35.95\ \$/MWh_e$ making a very strong business case for home power generation, even though capital investments are still high.

Heat Transformer technology even allow heat recovery from ambient air for mobile generation with costing as low as $1569\,\$/kW_e$ for $30\,kW_e$ Power Packs. This could very practically be utilized in electric vehicles, road- and rail transport, as well as aero-applications making practical electrical planes possible by eliminating some of the weight of batteries.

1.) Introduction

Heat normally come at a cost, but the actual cost strongly depend on the temperature level. Thermal energy below ~ 50°C is very often regarded as "free", but the cost of low temperature or "waste" heat is a function of the equipment used to extract the heat. Heat extraction equipment have a capital investment cost component, as well as an energy component for powering the equipment. We may use a heatpump to absorb heat at a low temperature (eg. 45°C) and deliver it as "upgraded" heat at a higher temperature (eg. 80°C). This heatpump would then consist of 2 heat exchangers (H/E), namely a condenser and evaporator, a Joule-Thompson (JT-expansion) valve and a vapor compressor. While the cost of this equipment (with some piping in between) would represent the capital cost, the electrical energy required to power the compressor represent the operating cost (or energy cost). This is named the conventional vapor compression (VC) Heatpump, as sketched in figure 1, below.

Figure 1



In the VC-heatpump, a suitable refrigerant vapor (eg. ammonia -NH3) is compressed to a high pressure by the compressor and is condensed at the high saturation temperature and pressure, delivering the latent heat of condensation to a heat sink at high temperature.

Condensate pressure is dropped via the JT-valve and the liquid flashed to vapor in the low pressure evaporator, absorbing the latent heat of evaporation from the evaporator heat source. The difference between the temperature values of the condenser and evaporator is known as the temperature lift.

The VC-heatpump have a compression ratio fixed by the temperature lift, and the electrical energy required to operate the VC-heatpump is strongly tied to the refrigerant mass flow and the compression ratio. The efficiency or coefficient of performance (COP) for any heatpump in heating service is defined as the amount of heat pumped (Q_{cond}), divided by the compressor work done ($W_{compress}$) to pump the heat, while for cooling service it is defined as the heat removed (Q_{evap}) divided by the compressor work done ($W_{compress}$):

$$COP_{heating} = \frac{Heat_{delivered}}{Energy} = \frac{Q_{cond}}{W_{compress}} \tag{0.1}$$

A typical domestic A/C heatpump used by millions of people is making use of VC technology eg. the Dunham-Bush split unit A/C of 18000 BTU pumping heat with an efficiency expressed as COP = 2.78, making the electricity used ~ 3.6 kW_e for every 10 kW_{th} heat pumped.

$$COP_{cooling} = \frac{Heat_{removed}}{Energy} = \frac{Q_{evap}}{W_{compress}} \tag{0.2}$$

Heat transformers are different in the energy used for powering the temperature lift. In contrast to the VC-heatpump using electrical energy to power temperature lift, the thermal energy used in heat transformers for this purpose decrease the electricity use substantially for creating the same temperature lift.

In the typical conventional thermally powered absorption heat transformer (AHT) sketched in figure 2, low to moderate temperature level (Te and Td) heat, (Q_{ev} and Q_{de}) is used to generate high pressure (Pe) vapor at the intermediate temperatures (Te and Td). This vapor is then routed to the absorber, and the latent heat of condensation, added to the heat of solution (HOS), elevate the temperature of the absorber to the high output temperature of (Ta). This type of heat transformer involve some heat to be rejected from the condenser (Q_{cond}), making the efficiency lower.

$$COP_{heating} = \frac{Heat_{delivered}}{Energy} = \frac{Q_{ab}}{(Q_{ev} + Q_{de} + W_{pump})} \tag{0.3}$$

The biggest advantage of using an AHT is that it utilize waste heat for heat pumping instead of expensive electricity. The waste heat is normally abundantly available at low or no cost, and the liquid pumps used have a power consumption at least two orders of magnitude smaller than the heat flow. Liquid pumping energy requirements (W_{pump}) are therefore sometimes ignored in efficiency calculations.

Real COP values for this type of heat transformer is relatively low, eg. around 0.35, compared to the VC heatpump example mentioned above with COP = 2.78, but realizing the amount of electricity used by the heat transformer is extremely low (eg. W_{pump} ~ 100 Watt for a heat transformer where the heat pumped, $Q_{delivered}$ = 10 kW), it makes sense to define two different COP values to represent the electrical efficiency, COP_e and the thermal efficiency COP_{th} separately:

$$COP_e = \frac{Q_{delivered}}{W_{pump}} \sim 50+ \tag{0.4}$$

$$COP_{th} = \frac{Q_{delivered}}{(Q_{ev} + Q_{de})} \sim 0.3 \text{ to } 0.5 \tag{0.5}$$

We recognize that the **heat transformer is actually defined by four criteria,** namely:

1. the fact that it is a **thermodynamic cycle, which is (at least partially) heat powered** to upgrade (or lift) the temperature of heat from low- to moderate levels to higher temperature commercial heat;

2. **temperature lift is generated by a vapor absorption process,** releasing the heat of solution (HOS) combined with the latent heat of condensation of the vapor into a hot absorber;

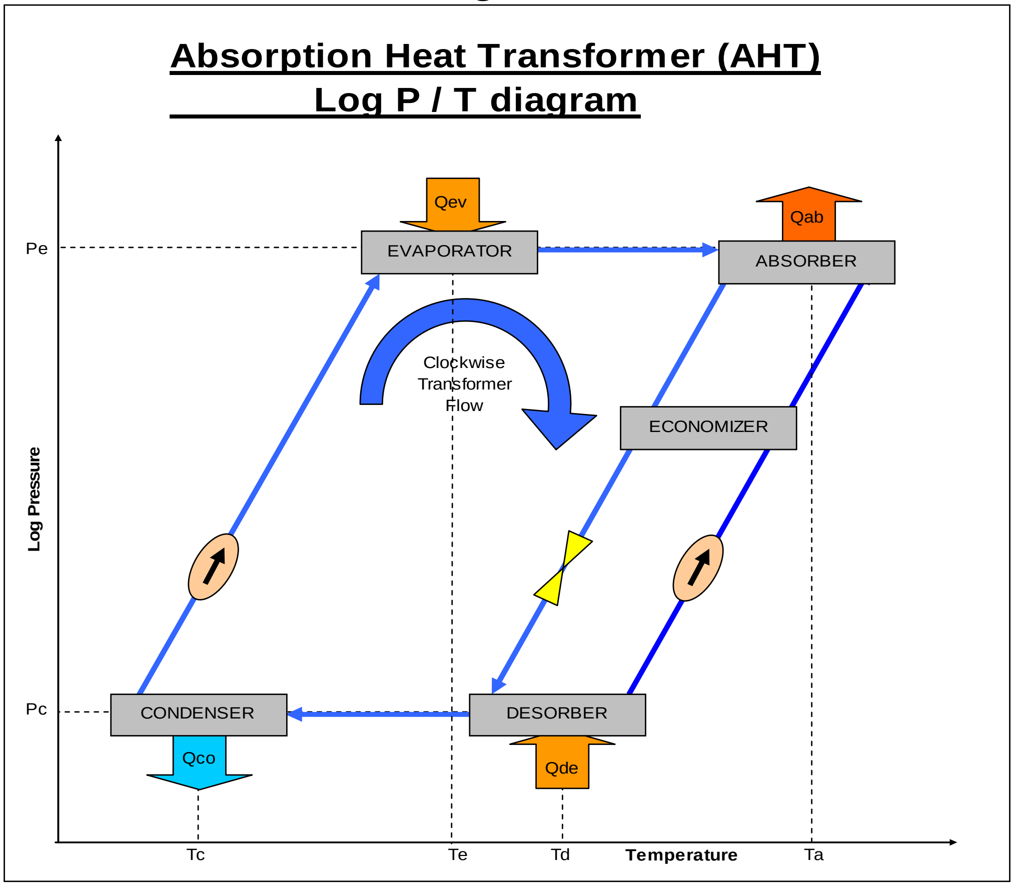
3. the cycle also has a **means of producing the vapor at the absorber pressure**, although it may be much lower in temperature; and,

4. **means is provided in the absorber to sub-cool the liquid present**, eg. heat removal, allowing the vapor absorption to take place. (As we know, vapor will not be absorbed into a saturated liquid! It has to be subcooled before vapor will be absorbed.).

The unique aspects of the heat transformer is represented in items 1, 2 and 3 of the definition above, as item 4 is also present in the VC heatpump. Item 4 is achieved by heat removal from the hot end of the heatpump or -transformer.

In the conventional AHT sketched in figure 2, all four these criteria are met. The 3rd criteria is done by generating vapor at a low temperature (and pressure) by utilizing moderate temperature heat in a low pressure vapor generator (desorber) and condensing it in the condenser, rejecting the latent heat (Q_{cond}), after which a liquid pump increase the pressure to the evaporator/absorber pressure, where vapor is produced by vaporizing the liquid at higher pressure in the evaporator with the addition of more heat at moderate

Figure 2



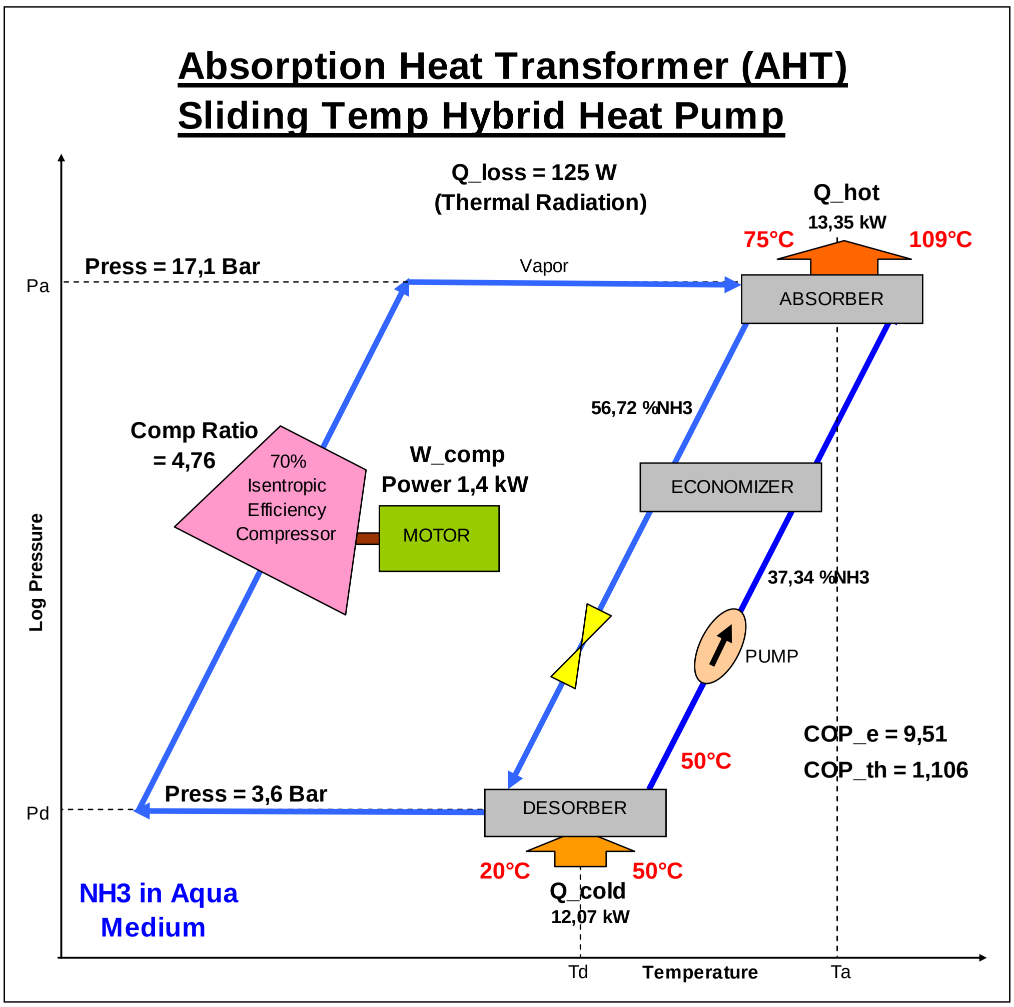
temperature level. Heat transformers have been explored extensively for more than 30 years to "upgrade" low temperature heat to higher temperatures, like the evaluation done by Rivera [5] in 2000.

Over the years, attempts have been made by researchers to increase the efficiency by using heat recovery of the rejected heat in the heat transformer condenser. This give COP values above 0.6 and is really commercially quite valuable.

Others used a complete VC heat pump to recover all the latent heat in the vapor flowing from the generator to the condenser, and pumped this heat to the evaporator, making the condenser heat rejection completely redundant. With this modification, additional heat exchangers were used, making the non-zero H/E temperature differential still a non-ideal machine. With this modification they were able to measure the COP to have increased from 0.5 to 0.8 which makes huge commercial sense. 80% of the low to moderate temperature heat may now be upgraded to temperatures in excess of 100°C, making use of a heat powered machine!

Different ways of generating the required high pressure vapor led to the development of hybrid-type heat transformers. Figure 3 represent such a hybrid heat transformer where the high pressure vapor is generated by an isentropic vapor compressor instead of the conventional way. Although this technology (of replacing the condenser and evaporator in the VC heatpump with an absorber and desorber) was already reported as the Osenbrück cycle over a century ago, described in handbooks eg. "Thermally driven heat pumps for heating and cooling" also known as the IEA Handbook [4] 2013. This concept was only recently dusted off and studied again in 2005 by Nordtvedt [3] used for waste heat recovery in the Norwegian Food Industry. He used the concept of isobaric temperature glides of binary mixtures in both the heat Sink and Source. Glides of some 40°C in the absorber and some 32°C in the desorber were used, more closely following the temperature changes in the heat Sink-, and Source water flows, forming the more efficient Lorenz cycle. The higher efficiency of this cycle follow from the decreasing irreversibility's in heat exchange in both the Sink and Source heat exchangers.

Figure 3



Nordtvedt also paved the way for the comprehensive development and testing of the compression/absorption heat transformer (CAHT) reported in 2014 by Anders Borgås in his thesis [7]. He developed the CAHT using temperature glides of 50°C in the absorber and 40°C in the desorber. His experiments and simulations of heating water from 110°C to 160°C in the absorber H/E (Heat Sink) and cooling water of 45°C to 5°C in the desorber H/E (Heat Source) resulted in a COP values of 1.6 - 1.8 when the compressor pressurize vapor with 95% - 99% NH3 by mass, mixed with a small % water vapor. As can be noted by the COP values of the hybrid machines, the values fall between the characteristic heat transformer (COP ~ 1) and the VC heatpump (COP ~ 2.78). It is therefore not surprising that the electrical portion COP_e = 9.51 as shown in figure 3 below.

In 2015 Jensen [6] dedicated his PhD thesis to the hybrid absorption-compression heat pump, providing a temperature lift of some 30°C with the absorber high temperature ranged 120°C - 150°C although he used smaller temperature glides of roughly 10°C, achieving similar COP's like Borgås a year earlier. Jensen did very elaborate energy-, and exergy-, as well as advanced exergy analysis to prove his findings.

The higher electrical efficiency (COP_e ~ 10 to 50) of these hybrid machines can be attributed to the much lower compression ratio's eg. 4.7 in figure 3, vs. 17.8 of the VC machine in figure 1. The temperature gliding effect shown in figure 3 highlight that the one end of the absorber may be at 109°C while, at the same pressure, the other end may be at 75°C as the internal saturated binary liquid concentration differ, forming a concentration gradient in the opposite direction than the temperature gradient. This type of hybrid heat transformer with much reduced compression ratio is utilized and fully described in my paper [2] where it is used as heatpump in the place of a VC heatpump as a result of the much lower electricity consumption combined with the abundance of low temperature waste heat.

Isobaric temperature gliding in binary zeotropic mixtures is actually a well known concept exploited extensively in the development of the Kalina cycle since the 80's and have recently also involved other binary mixtures of hydrocarbons apart from the frequently-used NH3-H2O and LiBr-H2O. In one example in a Kalina cycle boiler, isobaric (at 34.5 Bar Abs) temperature gliding from the bubble-point temperature of 93°C to the dew-point temperature of 184°C (a gliding span of 91°C) for a 70%NH3 in H2O binary mixture was reported in 1996 by Kiesela et al [8] and since then, a large gliding span has become common practice for zeotropic binary mixture H/E design.

With a further reduction of compression ratio as well as the increased use of the isobaric temperature gliding effect in binary liquids, the pseudo-isobaric temperature gliding heat transformer as sketched in figure 4 is made possible. These concepts are more comprehensively described in my publication [1] where I highlight the key principles the regenerative heat of solution (REHOS) cycle is built on. The heat transformer form the basis for this novel cycle.

The Pseudo-Isobaric Temperature Gliding Binary heat transformer as (P-T Diagram sketched in figure 4), or simply named the Bubble Reactor Heat Transformer (BRHT) sketched in figure 6 below, generate the required high pressure vapor (item 3 of the criteria defining it as a heat transformer) by making use of the temperature gliding effect and utilizing a liquid pump to increase the high concentration low temperature binary liquid pressure to a value higher than the absorber high temperature reactor pressure, and therefore completely avoid the use of a vapor compressor.

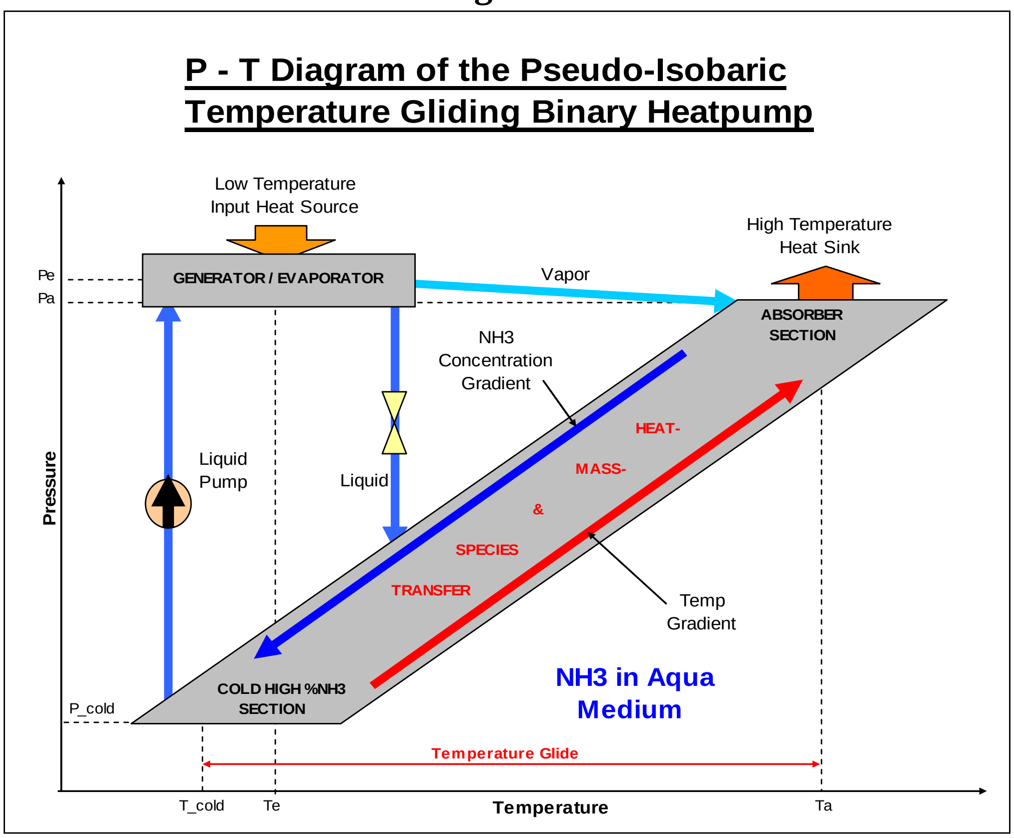
As no vapor compressor is used, the electrical energy required by the BRHT is even lower, as liquid pumping using such low differential pressures is very small compared to the latent heat flow. In the BRHT the electrical pumping energy is therefore about 3 orders of magnitude smaller than the heat flow, making the COP calculations:

$$COP_{e\_BRHT} = \frac{Q_{delivered}}{W_{pump}} \sim 500+ \tag{0.6}$$

$$COP_{th\_BRHT} = \frac{Q_{delivered}}{Q_{evap}} \sim 1 \tag{0.7}$$

This low electricity use for temperature lift (heatpump action) obviously have a tremendous economic impact on A/C and refrigeration, heat pumping and waste heat utilization by converting it to power using an ORC.

Figure 4



2.) Component Costs and Correlations

To be able to make a reasonable ballpark comparison of capital investment as well as operational costs of the heat transformer and some applications of its use, it is most practical to make use of cost correlations presented by authors who have made comprehensive studies to propose realistic cost values, all converted to USD and compensated for inflation and adjusted to represent real cost in 2018. All cost information in this paper is either ZAR (written as R) or $\$_{2018}$ written as $ and the rate of exchange between the two currencies (1$ = R14.00).

2.1) Shell & Tube H/E Costs

The cost correlation used by Nusiaputra et al [12] published in 2014 was checked with local shell & tube H/E real cost and found to be very close:

$$C_{H/E} = 14498 + 658.(A)^{0.85} \tag{1.1}$$

where the parameter A represent the heat exchange area in m^2 and the correlation calculate for a shell & tube H/E assuming a Carbon-steel shell and Stainless-steel tubing.

For heat extracted from a liquid media like water, the H/E area density is calculated at averaging ~ 90 m^2 / m^3 while typical average heat exchange rate calculates to values of about ~ 4 $kW / m^2.K$. This make the capital cost of the H/E that need to recover **heat from ambient temperature (LMTD = 20°C) water**

$$C_{H/E\_water\_20C} = 6.20\$ / kW_{th} \tag{1.2}$$

while the H/E if needed to recover **heat from a water source** of higher temperature (LMTD = 45°C) as

$$C_{H/E\_water\_45C} = 2.76\$ / kW_{th} \tag{1.3}$$

Using the shell & tube H/E for recovering heat from air or a gas would have the H/E area density much lower at ~ 12 m^2 / m^3, while the typical heat exchange rate would be as low as ~ 400 $W / m^2.K$. Capital cost of the H/E to recover **heat from air** (LMTD = 20°C) can be estimated as

$$C_{H/E\_air\_20C} = 207.68\$ / kW_{th} \tag{1.4}$$

while the H/E if needed to recover **heat from air** (or a gas) at the higher temperature (LMTD = 45°C) calculate to

$$C_{H/E\_air\_45C} = 92.30\$ / kW_{th} \tag{1.5}$$

These examples highlight the huge difference in capital costs of recovering heat from a low density medium like air vs a high density medium like water. The difference is about a factor of 30. These calculations also provide an indication of cost implications when the log-mean-temperature-difference (LMTD) between the heated and the heating streams are increased from 20°C to 45°C.

2.2) Bubble Reactor Costs

The bubble reactor is actually a vertically positioned column heat exchanger, or it may be seen as a distillation column as it has internally different binary mixture streams flowing both vertically upwards as countercurrent flowing downwards. Vapor enter from the bottom and is partially absorbed into the lean liquid present in the reactor bottom creating a lot of heat. The balance of the vapor not yet absorbed, creates a vapor-lift action and drive the internal circulation flow. The vapor-rich upflow stream absorb vapor and generate heat as it flows upwards, while the leaner, denser downflow stream is also heated (in direct contact heat exchange) by the upflow stream, and in the heating process boil off more vapor to become even leaner, hotter and more dense.

Heat exchange (together with mass and species exchange) within the reactor is therefore complex. One type of exchange is vapor-liquid exchange, where vapor is absorbed and generate heat in the up flowing liquid stream. Another type of heat transfer is liquid-liquid direct contact exchange of the two countercurrent flowing binary liquid mixture streams of different concentrations, while a third type involve the highly turbulent two-phase mixture transferring vapor absorption heat to an internal H/E tube used for the high temperature heat output coil near the reactor hot bottom. This heat removal create the required sub-cooling to allow vapor absorption.

During the theoretical evaluation of reactor performance, the length of the reactor was divided into a number of circular discs forming flow segments, and each segment was balanced individually for heat-, mass-, and species balance at a constant pressure, partially determined by the liquid column hydraulic pressure of the column above the segment. For this balancing the thermo-physical properties of the NH3-H2O were used as look-up tables derived from the formulations from the literature nicely grouped together by Ganesh and Srinivas [13] and published as recently as in 2017. In these balancing calculations done at the process parameters shown in figure 6 it was found that the Nameplate H/E of 3 kW was only 1.15% of the real total heat exchanged in the reactor. The bubble reactor real total average H/E = 87 times the Nameplate heat load, as a result of also the NH3 concentration increase as the mixture flow upwards! The Nameplate heat load is defined as the amount of heat removed by the heat output H/E tube coil.

Process intensification research has shown the vapor-liquid H/E contact area due to the vapor hold-up can be averaged at ~ 100 m^2 / m^3 while the average overall H/E rate (vapor-liquid as well as liquid-liquid direct contact and tubing contact boiling transfer) was measured as ~ 4.8 MW_{th} / m^3.

In evaluating the reactor cost, the work done by Altinbalik et al [11] and published in 2016 proved very valuable. This benchmark design of a pressurized liquid storage tank was done for a 1.5 m^3 of ~ 1m diameter with various differently shaped end-pieces, manufactured from SA-240 304L Stainless Steel and rated at 10 Bar using all the SME safety factors for pressure vessels. The cost correlation was adjusted to include a real SS cost of $7\,\$/kg$ as the local price for SS in large diameter pipe was found to be 5 $ for 304 grade and 6.5 $ for 316 grade.

$$C_{reactor} = 1852\ \$/m^3 \tag{1.6}$$

giving us the cost of the reactor related to the internal holding volume. With the mentioned H/E real transfer rate divided by 87 to deliver the Nameplate H/E rate calculate to the bubble reactor costing rate per Nameplate H/E of:

$$C_{reactor} = 33.57\ \$/kW_{th\_nameplate} \tag{1.7}$$

This bubble reactor cost correlation is actually conservative and may be optimized considerably by using process intensification principles like adding a swirl to add a centrifugal component to the binary liquid in the column, enhancing heat and mass transfer. The column may also be of a smaller diameter compared to the length to save material mass even with the high pressure specification. The column may also be manufactured from fiber-reinforced synthetic material to save a lot on mass etc.....

2.3) Combined Pump & Motor Costs

The correlation provided by Nusiaputra et al [12] published in 2014 adjusted to reflect 2018 $ values is perceived as reasonable

$$C_{pump+motor} = 5197.(\frac{kW_e}{30})^{0.7} \tag{1.8}$$

and normally working with pump-motor combinations, especially the smaller sizes of a few kW power, it became clear that the pump cost is around 30% of the combination cost, with the electrical motor the more expensive part. This is, however, a real thumb suck estimate.

2.4) Pump Only (without Motor) Costs

As mentioned under 2.3 above, the estimation of the pump only costs is really a part of the pump-motor combination, and a thumb-suck guess would be around 30%. It would be acceptable to use in further calculations of this paper, however, because the relative percentage of the pump cost to the complete machine discussed further is low. The correlation therefore:

$$C_{pump} = 1559.(\frac{kW_e}{30})^{0.7} \qquad (1.9)$$

2.5) Generator Only Costs

The correlation presented by Toffolo et al [9] published in Appl. Energy 2014 was found to be the most accurate, and after adjustments to bring the cost to 2018 $, the correlation is

$$C_{generator} = 2161910.(\frac{kW_e}{11800})^{0.94} \qquad (1.10)$$

This is a very popular correlation used by many researchers in the ORC range of power generators.

2.6) Power Expander Costs

Correlations for power expanders also differ widely, as does the ORC applications and ranges served by power expanders, but the one chosen has a very realistic approach in costing specifically Screw expanders available on the market. It is the correlation presented by Astolfi [10] at the International Conference on Concentrating Solar Power and Chemical Energy Systems 2014. Astolfi use the low pressure exhaust refrigerant volume flow in m^3/s as parameter. This eliminate the effect of varying higher or lower inlet temperature and pressure conditions, as mainly the outlet volume has the biggest impact on the physical dimensions of power expanders used as ORC prime movers.

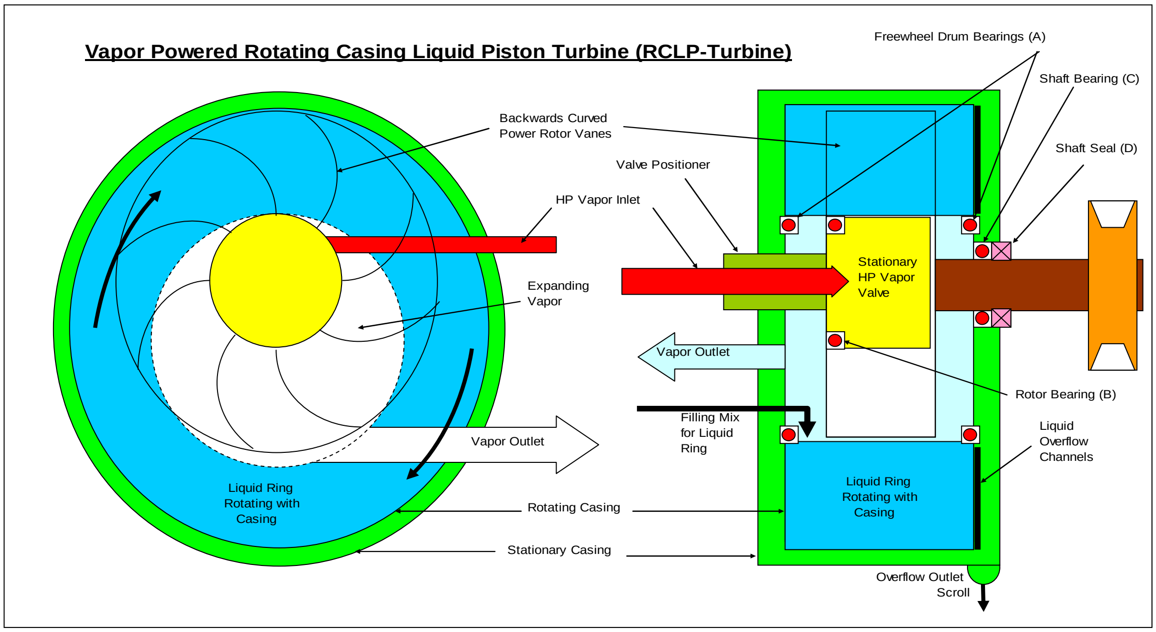
$$C_{screw} = 217423.Vol + 9596.4 \qquad (1.11)$$

The practical range of screw expanders on the market are used from ~ 10 - 200 kW_e but they really start to be readily available only from ~ 30 kW_e. Larger power outputs normally use turbines, priced completely different than this attempt presented in this paper. In the lower power output category very few suppliers can be found, and specifically below ~ 10 kW_e virtually all available positive displacement power expanders are custom-designs, with the exception of some scroll devices used for automotive A/C. The small-scale scrolls, however, are designed for compression service and not for expanders. The porting does not quite suite expander service and the isentropic efficiency is low.

I therefore tried to cost my own very simple custom design rotating casing liquid piston (RCLP) turbine that may be suitable as expander for low cost applications in the smaller (micro) ranges. The sketch in figure 5 highlight the simplicity of this design.

The internally off-centre double shrouded rotor coupled to the power shaft rotate inside a liquid ring enclosed in a free-wheeling rotating casing. The rotating casing avoid high liquid friction on the inside of the stationary casing, and allow higher rotation speeds (and therefore smaller turbines) with reasonable expander efficiency.

Figure 5



Costing of this expander would start with the cost of a simple centrifugal pump of similar power rating, doubled to account for the additional rotating casing, and further multiplied by 4 to account for other complication factors like additional bearings and balancing as well as the vapor channels and valves etc. Cost of the complete RCLP -Turbine could therefore be estimated as power equivalent pump cost x 8, namely

$$C_{RCLP-Turbine} = 12472.(\frac{kW_e}{30})^{0.7} \tag{1.12}$$

This type of liquid piston type expander designed for 1500 (1800) or 3000 (3600) RPM should be very practical on the smaller sizes of around 1 - 30 kW_e, coupled directly to a generator, avoiding expensive gearboxes.

2.7) Summary of Component Cost Correlations

Table 1

Component	Correlation	Equation #
Shell & Tube H/E	$C_{H/E} = 14498 + 658.(A)^{0.85}$	(1.1)
Shell & Tube Water H/E (LMTD = 20°C)	$C_{H/E\_water\_20C} = 6.20\$ / kW_{th}$	(1.2)

Shell & Tube Water H/E (LMTD = 45°C)	$C_{H/E\_water\_45C} = 2.76\$/kW_{th}$	(1.3)
Shell & Tube Air H/E (LMTD = 20°C)	$C_{H/E\_air\_20C} = 207.68\$/kW_{th}$	(1.4)
Shell & Tube Air H/E (LMTD = 45°C)	$C_{H/E\_air\_45C} = 92.30\$/kW_{th}$	(1.5)
Bubble Reactor	$C_{reactor} = 1852\ \$/m^3$	(1.6)
Bubble Reactor	$C_{reactor} = 33.57\ \$/kW_{th\_nameplate}$	(1.7)
Combined Pump+Motor	$C_{pump+motor} = 5197.(\frac{kW_e}{30})^{0.7}$	(1.8)
Pump Only	$C_{pump} = 1559.(\frac{kW_e}{30})^{0.7}$	(1.9)
Generator Only	$C_{generator} = 2161910.(\frac{kW_e}{11800})^{0.94}$	(1.10)
Screw Expander	$C_{screw} = 217423.Vol + 9596.4$	(1.11)
RCLP-Turbine Expander	$C_{RCLP-Turbine} = 12472.(\frac{kW_e}{30})^{0.7}$	(1.12)

3.) Economy of Heat Transformer Systems

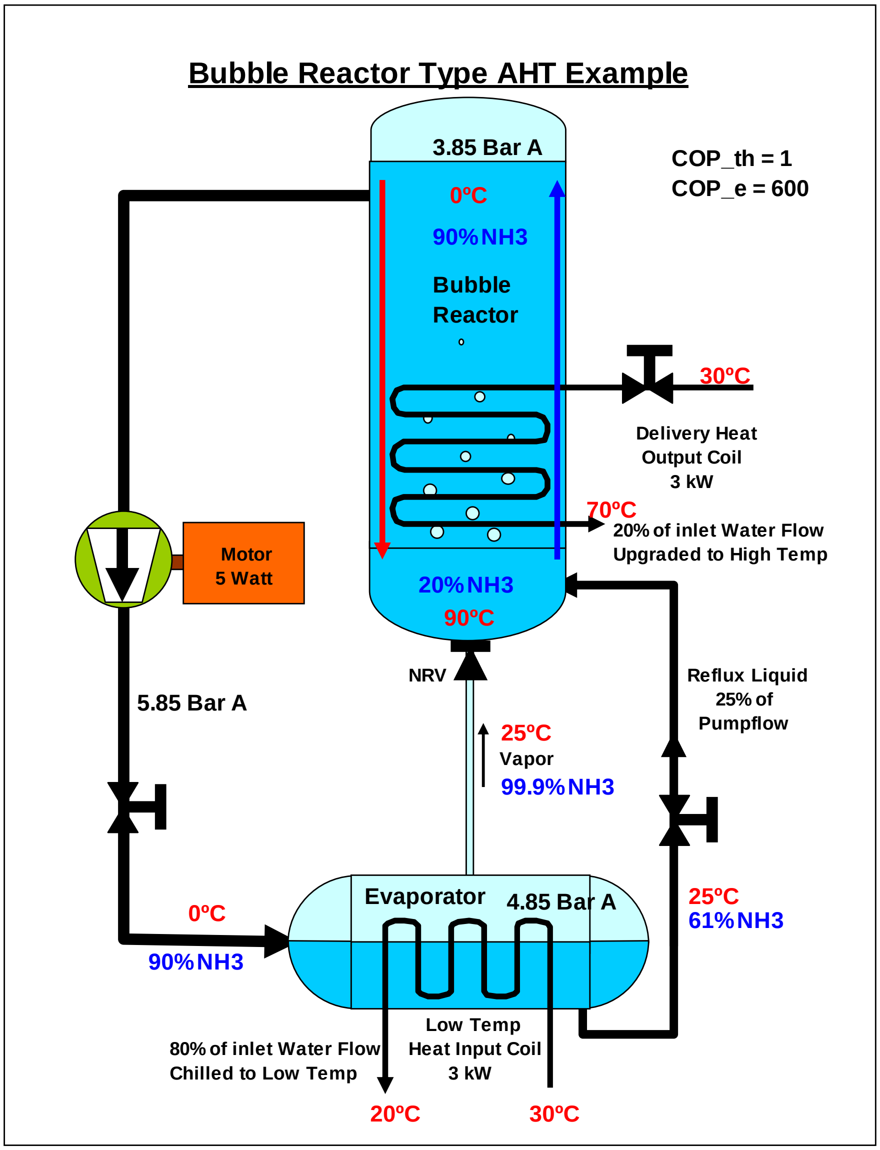
Evaluating the capital investment required to produce a system utilizing several components, is not only the cost of the components, but also the construction, engineering, commissioning and other costs, for the purpose of this paper altogether estimated as 20% of the summed total component costs. This correlate well with the recommendations of many authors on the same topic.

3.1) Economy of the BRHT used in A/C and Refrigeration

Among all the mentioned heat transformers and hybrid heat pumps used today, the BRHT represent the lowest electricity cost and simplest (therefore cheapest) design boasting an efficiency of $COP_{e\_BRHT} >$ 500 as per equation (0.6), and in the specific example design as sketched in figure 6, above, the value is 600. The electrical energy used by the liquid pump is calculated to only 5 Watt for a heat load of the machine of 3 kW!

Noteworthy as comparison the residential electricity price in Heidelberg, South Africa currently is ~ R1-50 / kWh_e supplied from the local municipality, so to use my A/C (the 18 000 BTU unit mentioned earlier) using the standard VC type heat pump ($COP_e = 2.78$) every day for 6 hours, calculate to 342 kWh_e /month costing me R513-00 /month. If the same size A/C was designed using the BRHT technology ($COP_e = 600$), the unit would have used only 1.58 kWh_e /month at a cost of R2-37 /month. This represent a saving of 99.5% for replacing the VC technology with Heat Transformer technology!

Figure 6



Taking into account that A/C globally use ~ 17% of all electricity generated globally, savings in electricity usage and cost of living would really make a huge impact on efforts to mitigate global warming. Heat transformers is not a new technology, but it is not really so well known, even among academics....

3.2) Economy of the BRHT used as De-humidifier for Water Production from Air

Small de-humidifier units designed for delivering water from the air available commercially use VC technology, and for a unit of 500 - 1000 Liters/day water extracted from air at a humidity ~ 50% use ~ 389 - 775 kWh_e / m^3 of potent water supply. This represent an average ~ 582 kWh_e / m^3 and with the assumed VC-heatpump COP = 2.78 the thermal energy extracted from the air calculate to Heat = 1670 kWh_{th}. Using heat transformer technology where COP_e = 600, the electrical energy used calculate to a mere 2.78 kWh_e / m^3 !

The VC technology therefore run ~ R573-00 $/m^3$ of fresh water, while the BRHT technology would run ~ R4-18 $/m^3$, using electricity supplied from the local municipality. Should the low power requirement be provided for by using a Solar PV panel or a small wind turbine, the water produced would be free....apart from the capital investment!

Realizing that many municipal water purification plants operate at a cost of ~ R5-00 $/m^3$, and the municipality selling water to their citizens at ~ R15-00 to R25-00 $/m^3$, the utilization of BRHT-technology would revolutionize potable water production, even in draught-stricken area's and cities of the world. This would also put a complete new perspective on the cost of coastal desalination plants....

4.) Heat Transformers as Source for ORC Power Generation

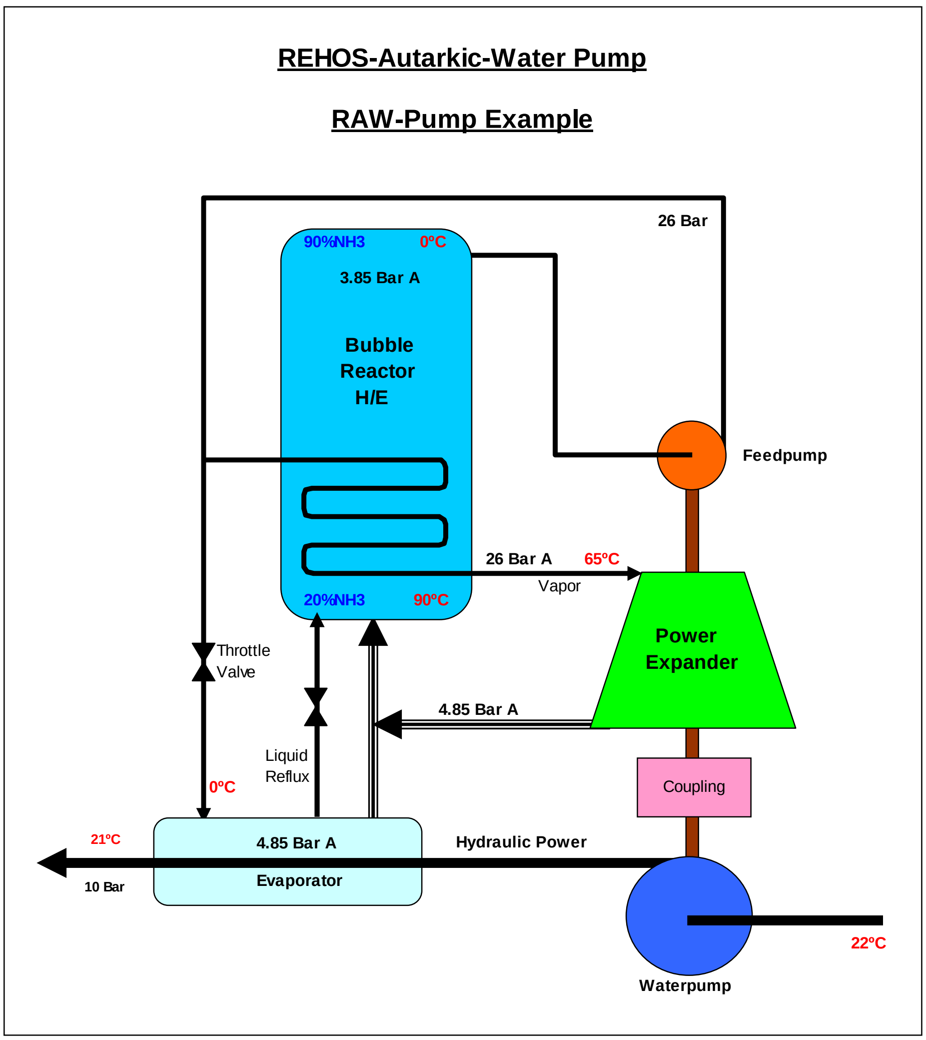
Apart from the utilization of heat transformer technology for producing cold in A/C and refrigeration service, it would also be very beneficial to use it to replace VC heatpumps as a very cheap alternative in providing domestic hot water and heat swimming pools as well as provide space heating in the colder climate countries around the globe. The most revolutionary application of heat transformer technology, however, would be to provide an ideal configuration for power generation, making use of already existing ORC technology. The heat transformer is able to recover heat, even from sources at lower temperatures than ambient, and upgrade the temperature to a higher value, suitable to run an ORC for power production, at an extremely low heat pumping electrical energy requirement.

Looking at the BRHT sketched in figure 6 we recognize that the bubble reactor deliver the pumped, higher temperature heat to a H/E coil integrated into the reactor hot bottom. This coil would form the ideal evaporator for a simple ORC. Also, the bubble reactor is heated by the vapor generated in the heat transformer evaporator. The bubble reactor therefore form the **ideal vapor recycler**, provided the expander exhaust vapor to be recycled is at the same (or slightly above) the system evaporator pressure, in our example design of figure 6 being 4.85 Bar Abs. Should vapor be supplied from an ORC expander exhaust at this pressure, all the latent heat in the ORC exhaust vapor would be recycled completely regeneratively, displacing most of the required vapor generated from the transformer evaporator! Figure 7 sketch an example of this Regenerative Heat of Solution

(REHOS) cycle so formed as a fully autarkic water pump (RAW)-Pump. This pump is powered via an ORC, by the thermal energy in the water being pumped!

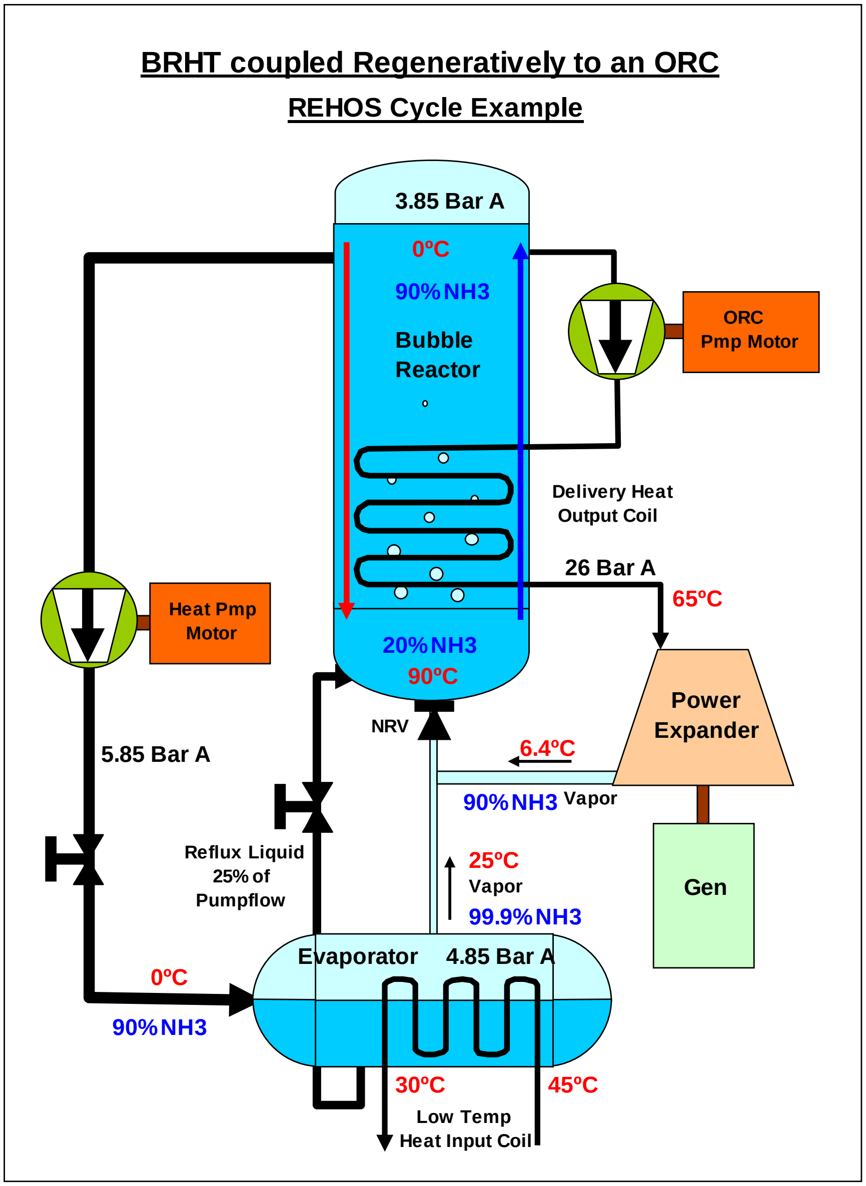
The ORC liquid pressure pump provide liquid binary mixture (90%NH3 in aqua in this example) at 26 Bar Abs to the H/E coil inside the bubble reactor, that absorb the heat to evaporate the liquid NH3 at 65°C. The ORC power expander deliver ~ 10% of this absorbed heat as power, while the 90% balance is available as latent heat in the exhaust vapor, recycled regeneratively by exhausting it into the reactor bottom. This way the heat transformer evaporator only have to supplement vapor to make up for the 10% the expander delivered as power (+ a few % other losses) to achieve overall energy balance.

Figure 7



Obviously some radiant heat losses occur due to the hot components radiant heat loss, and the expander netto delivered power is also decreased as it needs to power not only the ORC liquid pump, but the heat transformer liquid pump as well. Nevertheless the REHOS cycle example designed as sketched in figure 8 operate at ~ 80% heat to power conversion efficiency. More detail of this cycle is provided in my paper [1] where the key principles of the cycle is explained in more detail.

Figure 8



The power developed by the power expander which forms part of the ORC cycle, may of course be used in any mechanical power as needed. One way would be to couple a generator to the expander and generate electricity (like figure 8) from the heat recovered in the system evaporator. A different application would use the mechanical energy from the expander to power a water pump (like figure 7) for agricultural irrigation pumping requirements. A third option would concentrate the design on the chilling of the pumped water, delivering also some electricity and hydraulic pressure in an underground mine chilling application. Obviously a combination of the above may see both hydraulic water power, water chilling and electricity being produced in any combination or % split required.

Let us evaluate the simple REHOS water pump as sketched in figure 7, above, from an economic point of view first:

4.1) Economy of the RAW-Pump for Agricultural Irrigation

The BRHT is coupled fully regeneratively to the ORC, but the power shaft of the ORC drive a water pump directly. The REHOS cycle so formed therefore operate completely autarkic, as the energy required for the water pumping is recovered from the actual water being pumped, chilling the water in the process by a degree or two, depending on the mass water flow. Thermal energy is extracted from the water being pumped, and converted to hydraulic power for pumping.

In this design of the RAW-Pump operating temperatures and pressures were designed as shown on the sketch in figure 7. The ORC power expander deliver a netto 10% of the heat flowing through it as power and the feedpump power requirement is assumed to be 10% of the expander power produced. All the expander power is used to drive the water pump, with an assumed isentropic efficiency of 65%, and the RAW-Pump hydraulic output pressure, although shown in figure 7 as being 10 Bar, may vary according to the required pumped water volume flow. Obviously the lower the water flow rate, the higher the pressure would be for a specific power delivered, and also the larger the chilling effect cooling the pumped water.

In our evaluation of the relevant total RAW-Pump costs, we compare it with a traditional electrically powered water pump using electricity at R1-50 $/kWh_e$. Also assume the pumps run for 6 hours per day, 365 days of the year and the pump life is 10 years. In table 2 below, the correlation used for calculation of the evaporator costs is equation (1.2) as heat is recovered from ambient temperature water.

Table 2 (Small RAW-Pumps)

Component	**1 kW**	**4 kW**	**16 kW**	**32 kW**
RCLP-Turbine Expander	1153.32 $	3043.64 $	8032.21 $	13048.40 $
Water Pump Only	144.17 $	380.46 $	1004.03 $	1631.05 $
ORC Pump Only	14.42 $	38.05 $	100.40 $	163.11 $
Bubble Reactor	335.7 $	1342.8 $	5371.2 $	10742.4 $
Evaporator (Water H/E)	7.75 $	31.00 $	124.00 $	248.00 $
Other 20%	331.07 $	967.19 $	2926.37 $	5166.59 $
Irrigation Cost (10 years)	**1986 $**	**5803 $**	**17558 $**	**30999 $**

Table 3 (Small Elec-Pumps)

Component	**1 kW**	**4 kW**	**16 kW**	**32 kW**
Combined Pump+Motor	480.58 $	1268.26 $	3346.97 $	5437.17 $
Cabling&Switchgear 20%	96.12 $	253.65 $	669.39 $	1087.43 $
Electricity	21900 $	87600 $	350400 $	700800 $
Irrigation Cost (10 years)	**22477 $**	**89122 $**	**354416 $**	**707325 $**

Although the initial capital investment of the RAW-Pump is ~ 3.5 times to 5 times the price of the standard electrical pump, the fact that the RAW-Pump need no expensive electricity, cabling and switchgear to deliver water, make a huge difference to the irrigation costs, even at this small scale!

As seen by comparing the RAW-Pump cost of irrigation with the normal electrical pump, the cost of the RAW-Pump at 1 kW sizing is only 8.8% of the electrical equivalent pump over the pump life, while it decrease even further as pump size increase. For 32 kW RAW-Pump irrigation the cost is only 4.4% of the electrical equivalent, or put differently, calculated as if repaying the difference in cost of the RAW-Pump vs the Electrical pump as:

Table 4 (Small RAW-Pumps)

RAW-Pump Size	**1 kW**	**4 kW**	**16 kW**	**32 kW**
Cost Diff Repaid from Electricity Savings	**6.0 years**	**4.6 years**	**3.6 years**	**3.3 years**

4.2) Economy of the RAW-Pump used as Mining Chiller

The RAW-Pump cost calculated above is 100% the same for a small unit used as chiller for the mining industry. The water pump used is just designed with a smaller water volume flow (and therefore a higher pressure) delivered by the pump, so that the RAW-Pump outlet temperature is lower than the inlet temperature by several degrees Celsius.

The lower pumped volume flow, essentially dictate smaller diameter pumped water lines, increasing the water flow friction to dissipate more pressure per unit length. This way it would be practical to install several cascaded RAW-Pumps followed by a high pressure drop radiator (mining heat absorber) in sequence to repeatedly increase the pumped water pressure and chill it again, ready for the next Pump.

Deep mining spend a lot of electricity on chillers and considering that electricity cost represent a very large percentage of mining operation costs (some mines as high as 25-30% of total mining costs), cost savings as demonstrated in irrigation water pumping of table 2 and 3 would increase mining profitability hugely.....

4.3) Economy of the REHOS-Generator Recovering CW heat from a Utility P/S

As we know, large utility sized power stations (P/S) make use of a Rankine cycle to generate power from heat produced by combustion processes. Gas, coal or nuclear energy is partially (about ~ 40%), converted to electricity, and the balance of heat rejected to cooling water (CW) that dissipate the other 60% of the primary heat normally in a cooling tower. Dry-cooling is more expensive, but sometimes used in dry countries and use radiation H/E to dissipate the heat to ambient temperature air flowing through cooling towers, while the normal wet-cooling P/S dissipate the heat by flashing off (vaporizing) a portion of the CW, cooling the water some 15°C with the latent heat of evaporation of a mass of water lost in the air.

A typical utility wet-cooled P/S of 500 MW_e therefore reject ~ 750 MW_{th} heat (if the cycle efficiency was 40%) by flashing some 317 kg/s water to vapor, or 2.28 kg / kWh_e of power generated!

Should we use a heat transformer to recover some of this low temperature CW heat and lift the temperature from the 45°C to a higher temperature eg. 90°C (like our example BRHT design of figure 6) it allow us to generate power from it using an ORC. The BRHT, regeneratively coupled to the ORC is sketched in figure 8.
Recovering the CW heat using a REHOS Generator in this way have several simultaneous advantages, like:

- As less reject heat need to be dissipated into the air, less water need to be evaporated, providing a **water saving**;
- power generated by the REHOS Generator without using any fuel, **decrease the P/S fuel bill**, making the complete station more economical to run;
- as the REHOS Generator cost structure (LCOE) is much lower than the existing P/S cost structure, the difference between the cost structures represent **additional profit for the P/S**, or, alternatively, decreasing the LCOE of the combined Rankine-REHOS combination;
- less fuel combusted by the P/S also produce **less** CO_2 released into the air;
- as the power from the Rankine cycle is decreased and replaced by power from the REHOS cycle, the **Rankine cycle de-rating** also decrease the high temperature and pressure levels in the superheat stages, decreasing metal fatigue and elongate the station life.
- the REHOS add-on being modular and not interfering in the existing Rankine cycle (using CW interface only), it facilitate the gradual, stepwise according to the P/S budget, phasing out of fossil fuel combustion (**phased de-carbonization**) without being a financial burden to the P/S, utility, or the country.

The cost calculations only involve the use of positive-displacement Screw Expanders very much dedicated to the power range of 10 - 200 kW_e, but the same principles will be applicable in the larger ranges of a few MW_e units, only replacing the positive displacement expander with a much cheaper (at higher power levels) ORC turbine.

For the temperatures, pressures and NH_3 concentration levels as indicated in figure 8, the expander exhaust volume vapor flow was calculated at ~ 1.77198e-3 m^3/s per kW_e power produced. The evaporator cost calculation uses equation (1.3) for 45°C water.

Table 5 (Screw Expander REHOS Generators)

Component	**20 kW**	**50 kW**	**100 kW**	**200 kW**
Screw Power Expander	17302 $	28860 $	48123 $	86650 $
Combined Pump+Motor	781 $	1483 $	2409 $	3913 $
Bubble Reactor	6714 $	16785 $	33570 $	67140 $
Evaporator (Water H/E)	69 $	173 $	345 $	690 $
Generator Only	5373 $	12715 $	24393 $	46799 $
Other 20%	6048 $	12003 $	21754 $	41038 $
Total Capital Investment	**36287 $**	**72019 $**	**130524 $**	**246230 $**
	1814 $\$/kW_e$	**1440** $\$/kW_e$	**1305** $\$/kW_e$	**1231** $\$/kW_e$

Knowing that the Capital Recovery Factor (CRF) value is calculated and we assume an annual interest rate applicable as i = 9% pa while the REHOS machines are built for life expectancy of 20 years:

$$CRF = \frac{i.(1+i)^n}{(1+i)^n - 1} = 0.10955 \tag{2.1}$$

Assuming the operation maintenance fixed cost similar to large PV installations, we have

$$Fixed\_O\&M = 20\$/kW - year \tag{2.2}$$

while we assume variable O&M to be very low:

$$Var\_O\&M = (1.3)\$/MWh \tag{2.3}$$

and we assume the capacity factor to be the same as utility baseload P/S like the new Eskom coal-fired P/S's currently being built, Medupi and Kusile, namely

$$CF = 85\% \tag{2.4}$$

the Levelized Cost of Electricity (LCOE) from this complete power plant may therefore be calculated as:

$$LCOE = [\frac{(Capex.CRF + Fixed\_O\&M)}{(8760.CF)}] + Var\_O\&M \qquad (2.5)$$

Table 6 (Screw Expander REHOS Generators)

REHOS Generator	**20 kW**	**50 kW**	**100 kW**	**200 kW**
Capex per kW	**1814** $\$/kW_e$	**1440** $\$/kW_e$	**1305** $\$/kW_e$	**1231** $\$/kW_e$
LCOE in USD	**30.67** $\$/MWh_e$	**25.17** $\$/MWh_e$	**23.19** $\$/MWh_e$	**22.10** $\$/MWh_e$
LCOE in ZAR	**R0.43** $/kWh_e$	**R0.35** $/kWh_e$	**R0.32** $/kWh_e$	**R0.31** $/kWh_e$

Although the new Eskom P/S's Medupi and Kusile have similarly calculated LCOE values of R1.05 $/kWh_e$ and R1.19 $/kWh_e$ respectively, the national grid average currently **(Grid Parity) is said to be R0.78** $/kWh_e$. The fuel cost for Medupi and Kusile is claimed to be ~ R0.26 $/kWh_e$**.**

For South Africa the CO_2 **emission factor is said to be 0.94 Tonne** CO_2/MWh_e power generated.

With the results as mentioned above, we may summarize the total effect of **adding a 200 kW REHOS Generator** to Medupi P/S to generate power from the recovered heat in the CW in table 7 below:

Table 7 (Summary 200kW REHOS-Generator)

Power Generated	1489 $MWh_e/annum$	
P/S Fuel Saving	R287192.00 $/annum$	20514 $\$/annum$
P/S CO_2 emission Decrease	1400 $Tonne/annum$	
CW Saving (if it was Wet-Cooled)	3395 $m^3/annum$	Medupi is dry-cooled
LCOE Gain Profit	R1102008 $/annum$	78715 $\$/annum$
Total Additional Profit	**R1389200** $/annum$	**99229** $\$/annum$

Obviously, this additional profit may be used to decrease the P/S specific LCOE, or it may be used to finance an acceleration of de-carbonization by adding more REHOS Generators! Also, larger REHOS Generator units of a few MW_e using turbine expanders would be even more economical!

When the REHOS Generation approach 50% of the Rankine P/S capacity, heat for additional REHOS installations may be made available from solar thermal sources, constructed on the P/S premises, therefore allowing the complete phase-out of the fossil combustion generation with time.

The use of the REHOS Generators in this way is very practical, as the electrical and control infrastructure for electricity delivery to the national grid, as well as the operation and maintenance personnel and infrastructure is already in place on the P/S premises.

4.4) Economy of a Micro-scale REHOS-Generator extracting ambient heat from a REHOS-Pond

As the heat transformer primary sub-cycle of the REHOS Generator sketched in figure 8 may also recover heat from the environment at a temperature ~ 20°C (for our example calculation) by utilizing a H/E as per equation (1.2) for recovering heat from 20°C water, it would be practical to evaluate the economics of a "Solar Pond" storing solar irradiation energy as thermal heat at ambient temperature (20°C) for electricity generation on micro-scale. Heat storage at ambient temperature also guarantee zero thermal losses for storage! With solar irradiation of ~ 2200 $kWh_{th} / m^2.annum$ (in the largest part of South Africa) and the extremely high thermal to electrical conversion efficiency of the REHOS Generator (~ 80%) ~ 1760 $kWh_e / m^2.annum$ may be generated, in sharp contrast to solar PV installations where only ~ 88 $kWh_e / m^2.annum$ is generated!

Solar pond surface area for the REHOS-Pond delivering ~ 1000 $kWh_e / month$ would therefore have to be ~ 7 m^2, so even a small swimming pool would be large enough. Because home-owners do not necessarily qualify for the low utility-scale interest rate for financing this type of equipment, the interest rate for this application is assumed at 12% and the equipment life is adjusted to 10 years instead of the 20 years used with utility installations. This change the CRF:

$$CRF = \frac{i.(1+i)^n}{(1+i)^n - 1} = 0.17698 \qquad (2.6)$$

Table 8 (RCLP-Turbine Expander REHOS Pond)

Component	3 kW	6 kW	10 kW	20 kW
RCLP-Turbine Expander	2488.49 $	4042.57 $	5780.31 $	9390.14 $
Combined Pump+Motor	206.90 $	336.10 $	480.58 $	780.71 $
Bubble Reactor	1007.10 $	2014.20 $	3357.00 $	6714.00 $
Evaporator (Water H/E)	23.25 $	46.50 $	77.50 $	155.00 $
Generator Only	903.16 $	1732.73 $	2800.71 $	5373.25 $
Other 20%	925.78 $	1634.42 $	2499.22 $	4482.62 $
Total Capital Investment	**5555 $**	**9807 $**	**14995 $**	**26896 $**
Capex per kW	**1852** $\$ / kW_e$	**1634** $\$ / kW_e$	**1500** $\$ / kW_e$	**1345** $\$ / kW_e$

LCOE in USD Equations (2.2 - 2.6)	48.01 $\$ / MWh_e$	42.82 $\$ / MWh_e$	39.64 $\$ / MWh_e$	35.95 $\$ / MWh_e$
LCOE in ZAR	**R0.67** $/kWh_e$	**R0.60** $/kWh_e$	**R0.55** $/kWh_e$	**R0.50** $/kWh_e$

Immediately obvious is the comparison of even micro-scale REHOS Pond electricity produced to the residential electricity price in Heidelberg, South Africa currently of about ~ R1-50 $/ kWh_e$ supplied from the local municipality. The REHOS Pond supply even at 3 kW_e scale is < 50% of the local municipal cost!

To my mind the days of electricity utilities are numbered -they should very urgently re-think their business model!

4.5) Economy of the REHOS-Generator extracting ambient heat from the Air for Micro-scale Mobile Applications

For mobile applications the REHOS Generator would be identical to the REHOS Pond application shown above, apart from the evaporator, that need to be priced using equation (1.4) as heat is recovered from environmental temperature air, and not from the more economical water.....

Table 9 (RCLP-Turbine Expander REHOS Mobile Power-Pack)

Component	10 kW	20 kW	30 kW	40 kW
RCLP-Turbine Expander	5780.31 $	9390.14 $	12472.00 $	15254.30 $
Combined Pump+Motor	480.58 $	780.71 $	1036.94 $	1268.26 $
Bubble Reactor	3357.00 $	6714.00 $	10071.00 $	13428.00 $
Evaporator (Air H/E)	2596.00 $	5192.00 $	7788.00 $	10384.00 $
Generator Only	2800.71 $	5373.25 $	7866.16 $	10308.70 $
Other 20%	3002.92 $	5490.02 $	7846.82 $	10128.7 $
Total Capital Investment	**18018 $**	**32940 $**	**47081 $**	**60772 $**
Capex per kW	**1802** $\$ / kW_e$	**1647** $\$ / kW_e$	**1569** $\$ / kW_e$	**1519** $\$ / kW_e$

For mobile applications, weight may be reduced by manufacturing the pressure vessels from fiber-reinforced synthetic materials and the tubing from a suitable non-metallic material eg. PTFE. Even the power expander may have several synthetic material components, and only the most critical, like the power expansion rotor being Stainless Steel.

Looking at the cost of these Power Packs in table 9, the cost may seem high, but remember they produce electricity on demand, from thermal energy in the air, and do not require any costly fuel! Power Packs like these may be utilized to provide power for an electric airplane, keeping it in the air indefinitely, as the propulsion energy is sucked from the air!

5.) Conclusions

The utilization of **Heat Transformers** for the economical recovery of heat from both waste sources and ambient or solar thermal supplemented heat is vastly superior to the conventional VC type heatpumps, making "temperature upgraded" heat available for the very economical use in applications like the following:

- Extremely low electricity consumption A/C systems and Refrigeration to replace the traditional VC technology;
- Extremely low electricity consumption De-humidifiers and water-from-air pumps to provide water in draught-stricken cities;
- Regenerative combinations with ORC to make RAW-Pumps possible, pumping water fully autarkic, without any electricity, to be used in all water pumping applications like eg. agricultural irrigation water pumping, mine chiller applications and marine propulsion to name just a few;
- Regenerative combinations with ORC to make REHOS-Generators possible with heat-to-power conversion efficiencies > 80%, for use as bottoming cycles to facilitate electricity utility phased de-carbonization, micro-power supplies for buildings, shopping centre's, large buildings and even individual households etc.;
- REHOS Power Packs to make mobile electricity generation from heat extracted from ambient air a reality....

It may be argued that the introduction of the REHOS-Pond micro-scale power generator would render utility grid-electricity obsolete, and it is probably correct, but the high capital investment required even for the micro-scale would decrease the speed of adoption by many communities to a slow trickle, focused on areas where grid-electricity is difficult and expensive to implement, leaving ample opportunity for utility generation for the next 10 - 20 years, provided the utility make use of the phasing de-carbonization proposed in section 4.3 of this document to be able to **decrease the electricity selling price to consumers.**

Further R&D around the bubble reactor is recommended for the purpose of not only increasing heat-, mass-, and species exchange rates in order to be able to use physically smaller (cheaper) equipment, but also to decrease weight with the view of producing Power Packs suitable for the Aero-industry and electric mobility with higher power-to-weight ratio's.

Selected Previous Publications by the same author:

1. The document "Key Principles of the REHOS Cycle" was written by Johan Enslin in November 2018 and published in the Open Access Bioenergetics Journal at https://www.omicsonline.org/open-access/key-principles-of-the-rehos-cycle-2167-7662-19-154.pdf in January 2019, as well as on my own website http://www.heatrecovery.co.za/.cm4all/iproc.php/Key Principles of the REHOS Cycle.pdf

2. The document titled "Rankine Cycle efficiency increase by the Regenerative Recovery of Historically Rejected Heat" was written by Johan Enslin in October 2018 and published in the Open Access Bioenergetics Journal at https://www.omicsonline.org/open-access/rankine-cycle-efficiency-increase-by-the-regenerative-recovery-ofhistorically-rejected-heatrev2-2167-7662-1000155.pdf in January 2019 as well as on my own website http://www.heatrecovery.co.za/.cm4all/iproc.php/Rankine Cycle efficiency increase by the Regenerative Recovery of Historically Rejected Heat_rev1.pdf

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3. Hybrid Heat Pump for Waste Heat Recovery in Norwegian Food Industry, Stein Rune Nordtvedt (2005), Institute for Energy Technology, Instituttveien 18, N-2027 Kjeller Bjarne R. Horntvedt, Hybrid Energy AS, Ole Deviks vei 4, N-0666 Oslo, Jan Eikefjord, John Johansen, Nortura AS, Rudshøgda, Norway, Stein.Nordtvedt@ife.no. *(This paper was published in the proceedings of the 10th International Heat Pump Conference 2011.)*. This paper is also published as part of the IEA Handbook [2], page 57 - 61.

4. Thermally driven heat pumps for heating and cooling, compiled and edited by Annette Kühn (Ed.) as the Universitätsverlag der TU Berlin 2013, as the IEA Handbook available as handbook ISBN (online) 978-3-7983-2596-8 at email publikationen@ub.tu-berlin.de

5. Experimental Evaluation of a single-stage Heat Transformer used to increase Solar Pond's Temperature, by W. Rivera of Centro de Investigación en Energia-UNAM, P.O. Box 34, 62580, Temixco, Mor. , Mexico and published in Solar Energy Vol 69, No. 5, pp. 369 - 376, 2000.

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Energy and Process Engineering, Norwegian University of Science and Technology.

8. An Introduction to the Kalina Cycle, reprinted by Henry A. Mlcak, PE , first published in PWR- Vol.30, Proceedings of the International Joint Power Generation Conference, with editors: L. Kielasa and G.E. Weed, Book No. H01077-1996.

9. A Multi-criteria approach for the Optimal Selection of working fluid and Design Parameters in Organic Rankine Cycle systems, by Toffolo, A; Lazaretto, A; Manente, G and Paci, M published in Appl. Energy 2014, 121, 219-232.

10. Techno-economic Optimization of low temperature CSP systems based on ORC with Screw Expanders, by Astolfi, M presented at the International Conference on Concentrating Solar Power and Chemical Energy Systems, Solar PACES 2014 and available online from Science Direct Energy Procedia 69 (2015) 1100-1112.

11. Comparative Design and Cost Analysis of Cylindrical Storage Tanks with Different Head Types by using COMPRESS, by Altinbalik, M, T and Isencik, S presented as part of Proceedings of the 2nd World Congress on Mechanical, Chemical and Material Engineering (MCM'16), Budapest, Hungary in August 22-23, 2016.

12. Thermal-Economic Modularization of small Organic Rankine Cycle Power Plants for Mid-Enthalpy Geothermal Fields, by Yodha Y Nusiaputra, Hans-Joachim Wiemer and Dietmar Kuhn, published in Energies 2014, 7, 4221-4240 Open Access Journal 2 July 2014.

13. Development of Thermo-physical Properties of Aqua-Ammonia for Kalina Cycle systems, by N. Shankar Ganesh and T. Srinivas published in the Int Journal Material and Product Technology Vol 55, Nos. 1/2/3, 2017.