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Key Principles of the REHOS Cycle

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Abstract

The REHOS (acronym for "Regenerative Heat of Solution") cycle consist basically of an Absorption Heat Transformer (AHT)-Hybrid heat pump, coupled fully regeneratively to an ORC to produce power. Three key principles govern the high efficiency.

The 3 key principles of the extremely high efficiency of the REHOS cycle are explained as:

1. The utilization of an absorption heat transformer type of heat pump as primary sub-cycle to guarantee a very large thermal energy utilization with exceptionally small electrical component for heat pumping.
2. Maximizing the isobaric temperature gliding in the zeotropic medium used in the heat exchangers of the heat pump, allowing it a calculated COP=1.0 with a very high % of the energy required for heat pumping coming from thermal input, with the electrical component at least 2 orders of magnitude smaller that the heat flow.
3. Combining the thermally powered heat pump and power generating ORC fully regeneratively, allowing all ORC-reject heat to be used to offset the heat requirement for the heat pump from an external heat source.

This paper also show the recent developments in thermally powered heat pumps (AHT's) achieving COP's gradually increasing from the turn of the century to 2018 with values like 0.5 optimized to 0.8 during 2014-17 with major increases with the development of high temperature Compressor/Absorption Heat Transformers (CAHT) by authors like Nordtvedt, Borgas and Jensen.

Keywords Turbines; High pressure boiler water; Thermodynamic; Thermal powered; Hybrid heatpumps

Introduction

The REHOS (acronym for "Regenerative Heat of Solution") cycle consist basically of an Absorption Heat Transformer (AHT)-Hybrid heat pump, coupled fully regeneratively to an ORC to produce power. The heat pump is therefore of critical importance to the REHOS cycle. Different commercial aspects of various types of heat pumps have been compared in my paper [1,2] written April 2018, also to highlight the great commercial value of a simple machine like an efficient (mainly) thermally powered heat pump.

The heat pump used as primary sub-cycle of the REHOS cycle, is a thermally powered heat transformer, that use very little mechanical (or electrical) power, as more than 95% of the energy required to drive the heat pump is thermal energy.

Principle 1: Utilizing an absorption heat transformer

Figure 1 represent a typical Vapor Compression (VC) type heat pump where all the energy required to pump heat from the source (at low temperature) to the sink (at higher temperature) is required to power the vapor compressor (Wc). In the VC type heat pump, saturated compressed vapor is condensed at the high pressure (P1) delivering the latent heat of condensation isothermally at the saturation pressure of the medium [2,3]. Condensate is passed through the Joule-Thompson isenthalpic expansion valve to the lower pressure evaporator, where the saturated liquid at the low pressure (P0) flash to vapor, extracting heat from the source for the isothermal evaporation process.

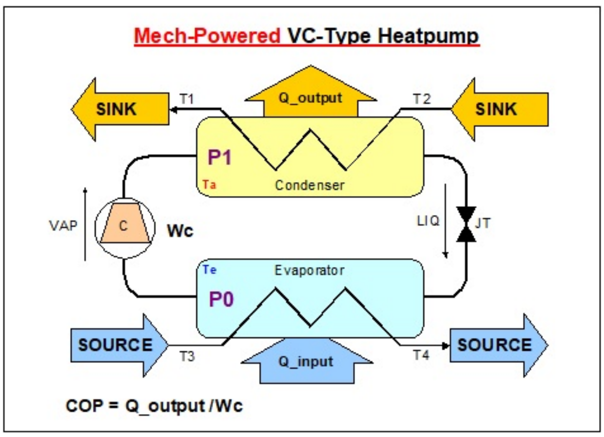

Figure 1: Mech-Powered VC-Type heat pump.

The efficiency of the heat pump, or the Coefficient Of Performance (COP) is calculated as the heat pumped (Q_out put), divided by the power required by the heat pump, in the case of the VC-type heat pump, the power requirement equals the compressor power used (Wc), which may be mechanical or electrical to drive the compressor [4,5].

In the typical thermally powered heat transformer (AHT) sketched in Figure 2, low to moderate temperature level (Te and Td) heat, (Qev and Qde) is used to generate high pressure (Pe) vapor at the intermediate temperatures (Te and Td). This vapor is then absorbed in the absorber, and the latent heat added to the heat of solution elevates the temperature of the absorber to the high output temperature of (Ta). The heat pumping (temperature upgrading) process is not very efficient, and some heat is also rejected from the condenser (Qco). The biggest advantage of using an AHT is that it utilize waste heat, which is abundantly available, and the liquid pumps used have a power consumption at least two orders of magnitude smaller than the heat flow. Liquid pumping energy requirements are therefore normally ignored in efficiency calculations.

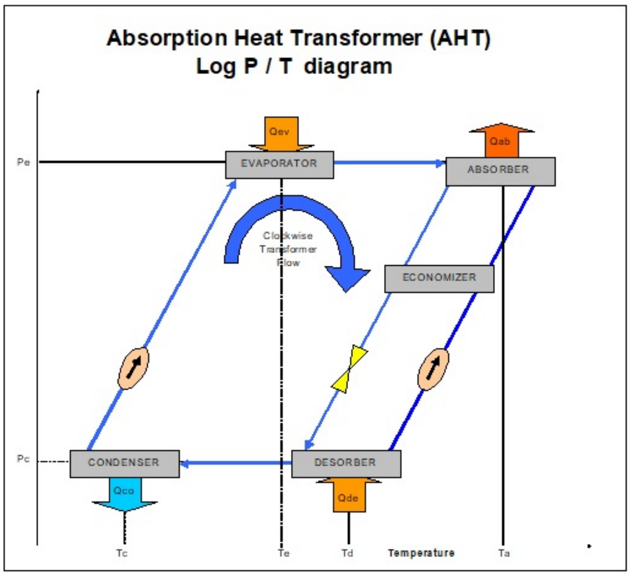

Figure 2: Absorption Heat Transformer (AHT) Log P/T diagram.

We take specific note that the absorption heat transformer (AHT) as sketched in Figure 2 above, has liquid/vapor media flowing in a clockwise direction, with the evaporator and absorber at high pressure. In the Absorption Heat Pump (AHP), however, used for refrigeration purposes, have anti-clockwise media flow, and the evaporator and absorber is at the low pressure side of the cycle. This paper only deals with heat transformers [6,7].

A typical example commercial use of this type of AHT was reported on by Rivera [6] in 2000 already. A Solar Pond operated between 30°C and 85°C, and an AHT was used to upgrade the temperature high enough to generate industrial steam at 125°C. This AHT generated a temperature lift of some 50°C, at a COP of 0.35, meaning that 35% of the intermediate Solar Pond heat flowing through the desorbed and evaporator heat exchangers was delivered as high temperature heat (125°C) in the absorber used for commercial applications. Even with this low efficiency, the installation was deemed commercially viable, as the real cost of the Solar Pond low temperature heat was very low.

More recently, in 2014 it was reported by Parham et al [7], in a comprehensive technology review paper, across several different configuration types of AHT's using several different binary operating media, proved the average COP values obtained was 0.40 to 0.48, with a few exceptions reaching values >0.52. Even 52% of waste heat upgraded to high temperature is fairly poor, so it is no surprise that several research facilities investigated ways to increase this efficiency.

We recognize that the heat transformer is defined by four criteria:

1. The fact that it is a thermodynamic cycle, which is (at least partially) heat powered to upgrade 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 combined with the latent heat of condensation of the vapor into a hot absorber
3. The cycle also have 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, e.g. 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 sub cooled before vapor will be absorbed.)

In the standard AHT, all four criteria are met. The 3rd criteria is done by generating vapor at a low temperature (and pressure) by moderate temperature heat in a vapor generator and condensing it in the condenser, rejecting the latent heat, after which a liquid pump increase the pressure to the evaporator/absorber pressure, where vapor is produced by vaporizing it in the evaporator with the addition of more heat at moderate temperature level. The 4th criteria is met by pumping a lean binary mixture from the generator to the absorber, with heat exchange in an economizer, producing the required sub-cooling to facilitate vapor absorption [8,9].

Optimizing the AHT:-Heat recovery by Heat pipe: Some Mexican researchers, Heredia et al [10], reported September 2017 that major strides have been made in efficiency improvements of the AHT's. Using an injector for the vapor just before it enters the absorber, would apparently increase the COP by between 14% and 30%. In this paper they also proposed condenser heat recovery by using a series of heat pipes between the evaporator and condenser. This concept is reported as increasing the COP by 20%, which is substantial. This give COP values above 0.6 and is really commercially quite valuable.

Optimizing the AHT:-Osenbruck cycle and CAHT: Figure 3 represents a hybrid heat pump where the high pressure vapor is generated by an isentropic vapor compressor instead of the generator, condenser, condensate pump and evaporator. Although this technology (of replacing the condenser and evaporator in the VC heat pump with an absorber and desorber) was already reported as the Osenbrück cycle over a century ago, it was only recently dusted off and studied again by Nordtvedt [4] used for waste heat recovery in the Norwegian Food Industry. He used the concept of isobaric temperature glides 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.

Nordtvedt also paved the way for the comprehensive development and testing of the Compression/Absorption Heat Transformer (CAHT) reported by Anders Borgas in his thesis [8] done at the Norwegian University of Science and Technology. He developed the CAHT using

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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% NH_3 by mass, mixed with a small % water vapor [10].

Jensen 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 Borgas a year earlier. Jensen did very elaborate energy-, and energy-, as well as advanced energy analysis to prove his findings [11].

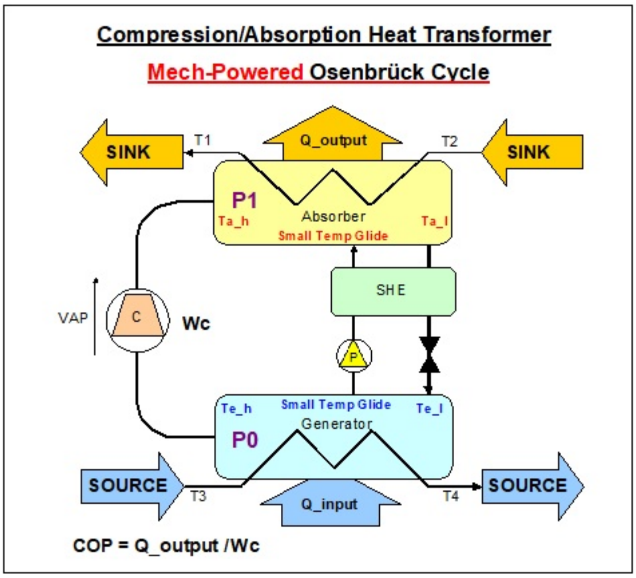

Figure 3: Compression/Absorption Heat Transformer Mech-Powered osenbrück cycle.

It is very interesting to note that the CAHT cycle as sketched in Figure 3 above, and researched by Nordtvedt, Borgås and Jensen, actually do not comply to all 4 criteria for being called a heat transformer, as the large pressure differential between the absorber and desorber require a relatively large energy component to be provided by the vapor compressor, instead of using thermal energy. The COP values of 1.6-1.8 also indicate operation very close to VC-type heat pumps, and the COP calculation ignore the relatively small portion of heat used with Wc to power the heat pump. This configuration like Figure 3, only change to become a true heat transformer when the compression ratio decrease substantially to decrease the compressor power consumption (Wc) drastically relative to the heat flow. In this special case the COP then becomes 1.0. In one of the previous publications [1], the different calculation methods used by various authors for the calculation of COP to better understand the term (COPe) representing the efficiency of the electrical portion of the energy required to power the AHT (in addition to thermal energy requirements) were highlighted.

Principle 2: Maximizing isobaric temperature gliding of the zeotropic mixture

Isobaric temperature gliding in binary zeotropic mixtures is 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 NH_3-H_2O and LiBr-H_2O. 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% NH_3 in H_2O binary mixture was reported in 1996 by Kiesela et al [9] and since then, a large gliding span has become common practice for zeoptropic mixture H/E design.

In an NH_3/H_2O binary heat exchanger with a specific mass of ammonia dissolved in the balance of water would give a specific saturation pressure with the NH_3 completely dispersed and equally distributed throughout the H/E, but will, during active operation exchanging a specific amount of heat would decrease in pressure, and when kept at 2 Bar Abs (by controlling the heat flow) distribute the NH_3 concentration internally giving a temperature glide of 65°C across the H/E so:

- one end of the H/E would be 47°C with a NH_3 concentration of 30% NH_3, while
- the other end of the H/E would be -18°C with concentration of 99% NH_3
- Should the same H/E be adjusted to 7 Bar Abs (by adjusting the heat flow), internal distribution would give a temperature glide of 74°C:
- one end of the H/E would be 88°C with a NH_3 concentration of 30% NH_3, while
- the other end of the H/E would be 14°C with concentration of 99% NH_3
- Also with the same H/E but adjusted to 20 Bar Abs (by adjusting the heat flow), internal distribution would give a temperature glide of 82°C:
- one end of the H/E would be 132°C with a NH_3 concentration of 30% NH_3, while
- the other end of the H/E would be 50°C with concentration of 99% NH_3

Clearly the internal pressure of the zeotropic binary H/E, being a function of the volatile component concentration, but also a function of the residual internal heat in the H/E. When heat inflow into the H/E exceed the outflow, the pressure will rise (also raising the temperatures) and when more heat is extracted from the H/E than heat inflow, the pressure (and temperatures) would drop, making pressure control by heat load adjustments logical. The complete internal volume of the isobaric temperature gliding H/E is in saturation, and when once-through operation takes place, the completely liquid high concentration inlet to the H/E is heated along the length of the H/E, and as the heat boil off more volatile component, the liquid flowing with the vapor has a lower concentration. The pressure being constant, force the remaining liquid saturation temperature higher as more and more liquid evaporate, until the dew point is reached where all the liquid have been evaporated. When the H/E diameter is suitably enlarged, decreasing the flow speed of the saturated mixture to small values, also counter flow of a liquid stream can be expected.

With the H/E diameter enlarged suitably to allow liquid counter flow, and physically positioned vertically to allow gravity liquid density separations, the isobaric temperature gliding would still take place exactly like explained above, however, lean heated liquid where the volatile component has been boiled off have a higher density that the richer, colder vertically up-flowing liquid-vapor mixture, migrating downwards due to gravity. This gravity-separation create a low temperature, high concentration volatile mixture at the H/E top, while the hot, lean mixture with very low volatile concentration segregate to the H/E bottom. Whether the H/E top contains high percentage vapor, or more liquid, depend on the heat added/removed from the H/E along the complete length. Heat balance calculations done on small length-wise virtual "discs" prove the temperature gliding at constant saturation pressure from a hot bottom to the cold top of the H/E.

Optimizing the AHT: Radical dP reduction and integrating components into a Bubble Reactor H/E: From the examples of a H/E at different pressures provided above, it should be very clear that the complete H/E would be in saturation at all points in the H/E, and that temperature distribution would be the inverse of the concentration distribution. This happen spontaneously accompanied by liquid counter flow in the H/E tube and is driven by heat flows as well as gravitational effects (higher concentration NH_3 mixtures are less dense than leaner mixtures). The "temperature glide" would therefore make high concentration NH_3 liquid (rich mixture at low temperature) available at one end (top) of the H/E, while the other end (bottom) would have low concentration NH_3 (lean mixture at high temperature). This zeotropic binary H/E may therefore be used as a complete NH_3 liquid producing Reactor, and coupling the cold NH_3-rich liquid from the reactor top through a Joule-Thompson isenthalpic expansion valve would deliver slightly lower temperature liquid to an external evaporator at a slightly lower pressure than the reactor. There the liquid could be flashed to vapor, absorbing ambient heat for the latent vaporization heat with a very small isobaric temperature glide (nearly isothermally). A vapor compressor may be added to increase the pressure high enough to force vapor flow into the bottom side of the reactor H/E, bubbling the vapor through the reactor, -hence the name Bubble Reactor H/E. The pressure drop from the reactor to the evaporator need only to be about 0.5-1 Bar, allowing enough pressure in the compressed vapor to facilitate the bubbling flow through the lean hot mixture of the reactor, delivering heat by vapor bubble absorption to the reactor.

Analysis show this Bubble Reactor with very low dP compressor and evaporator to conform to all the criteria set for AHT's as set out before in this document, but the compressor power used (Wc) is only a small fraction of the energy requirement for heat pumping. The balance (thermal energy) come from external heat inflow and extracted in the evaporator.

Optimizing the AHT: Abs-Evap pressure inversion and maximizing Isobaric Temperature Glide: Although the Bubble Reactor-Compression/Absorption Heat Transformer (BR-CAHT) described above would have a COP ~ 0.9-1.0 and would therefore be extremely commercially valuable, the practical implementation would still require a costly vapor compressor, though it may be very small. This could easily be eliminated by using a liquid pump to increase the cold, NH_3-rich liquid pressure slightly (by about 0.5-1.0 Bar).

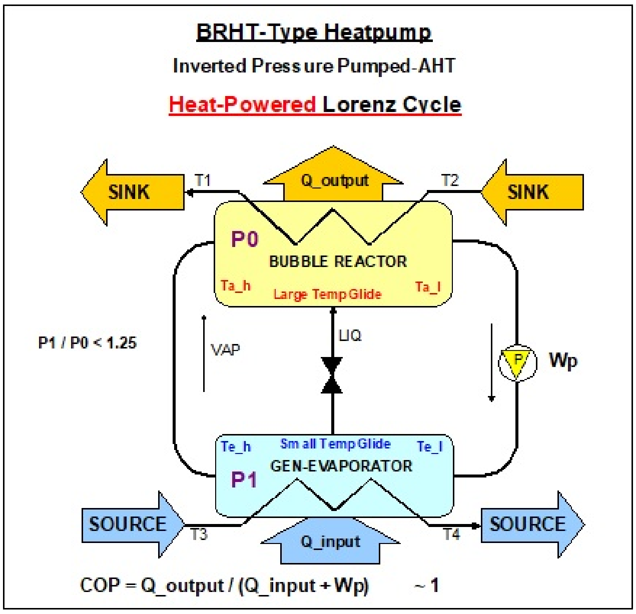

Figure 4: Heat-Powered lorenz cycle.

Above the reactor pressure, pumping it into the evaporator, where external heat (from a waste heat source) would vaporize it.

Figure 4 above, show a schematic of a heat transformer like that. The vapor formed in the evaporator, is already at the higher pressure, allowing spontaneous flow into the hot Bubble Reactor bottom, without the need of a vapor compressor.

This pressure inversion where the isobaric temperature glide taking place in the Bubble Reactor is at a lower pressure (only very slightly) that the evaporator, avoid the need of a vapor compressor, as well as decrease the pressurization energy requirements, as liquid pumping requires far less energy than compression, giving a BRHT-Type heat pump with COP ~ 1, meaning that nearly all low-to-moderate heat of the Source may become available as higher temperature heat (with a very small % lost as result of heat leakage) in Sink, useful commercially e.g. in power generation.

Full details of this heat pump, designed as a Syphon-pump type to emphasize the low differential pressure requirements is disclosed in paper [3]. As the reactor absolute pressure, and therefore the operating temperatures are adjustable by changing the heat input (with constant heat output) from the external waste heat source (or ambient temperature heat source), controlling the reactor low and high temperatures becomes a simple liquid inflow throttling control of the liquid stream of the external waste heat Source.

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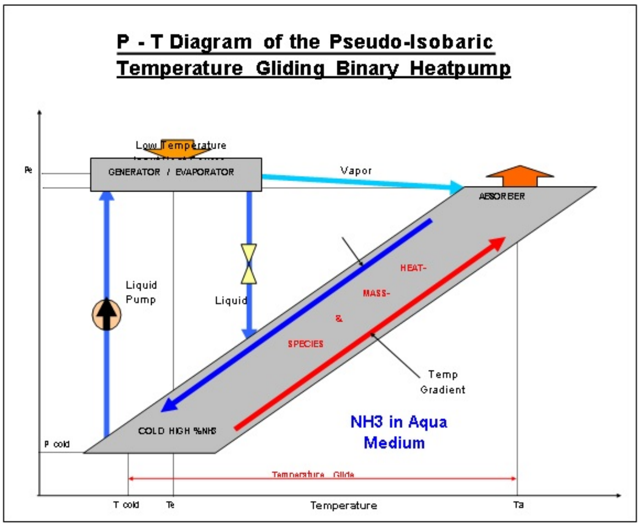

Figure 5: P-T graph.

The BRHT-Type heat pump shown in Figure 4 above, form the maximum efficiency, thermally powered heat pump with extremely low electricity consumption (Wp being 2 orders of magnitude smaller than the heat flow) required to form a key part of the REHOS cycle. Figure 5 provide a P-T graph of this BRHT-Type heat pump to be able to compare with the basic AHT P-T diagram represented by Figure 2. Clearly the absorber, economizer and desorber with circulation pump of Figure 2 have now been combined into the single Pseudo-Isobaric Temperature Gliding H/E with internal gravity-driven density segregation flows [12].

To be able to provide real process values in an example BRHT-Type heat pump, another sketch where all the process points have been alphabetically identified is provided in Figure 6. This allow us to list all the relevant state information of the process variables in table format as Table 1.

Position	Temp (Celsius)	Press (kPa)	Mass (kg/s)	Enthalpy (kJ/kg)	Entropy (kJ/kg.K)	$\%NH_3$ (kg/kg)	Media
A	0	413	0.1	-16.3	0	96.00%	NH_3-H_2O
B	0	511	0.1	-16.1	0.23	96.00%	NH_3-H_2O
C	10	511	0.075	1090.1	4.62	99.99%	NH_3-H_2O
D	20	-	0.388	83.9	0.297	0.00%	H_2O
E	70	-	0.388	293	0.955	0.00%	H_2O
F	20	-	1.937	83.9	0.297	0.00%	H_2O
G	10	-	1.937	42	0.151	0.00%	H_2O
H	10	511	0.025	-67.3	0.149	84.30%	NH_3-H_2O
J	80	413	-	181.5	1.121	24.70%	NH_3-H_2O
K	0	413	-	-16.3	0	96.00%	NH_3-H_2O

Pump dP 98 kPa | Pumpn Isentropic Eff=65%
Electrical COP_e=3530 | Wp=23.0 Watt
Q_input=81176 Watt | Thermodynamic COP=1.00
Q_output=81199 Watt | Reactor Temp Glide=80 Celsius

Table 1: List of all the relevant state information of the process variables.

The output, temperature upgraded liquid stream (E) carry the ~ 82 kW heat at the high 70°C value away from the heat pump where it was heated from the Sink incomer (D) at 20°C. This is upgraded temperature is high enough for use as domestic hot water system, and just need a thermally isolated reservoir to store it in.

The ambient temperature water inflow as Source (F) for the heat pump operation, is chilled from the ambient 20°C Source input to the chilled output at 10°C available at (G), providing an additional 81 kW_th chiller action for use in the domestic air-conditioning (A/C) system. All this by utilizing electricity of only 23 Watt to drive the liquid pump, revolutionary.

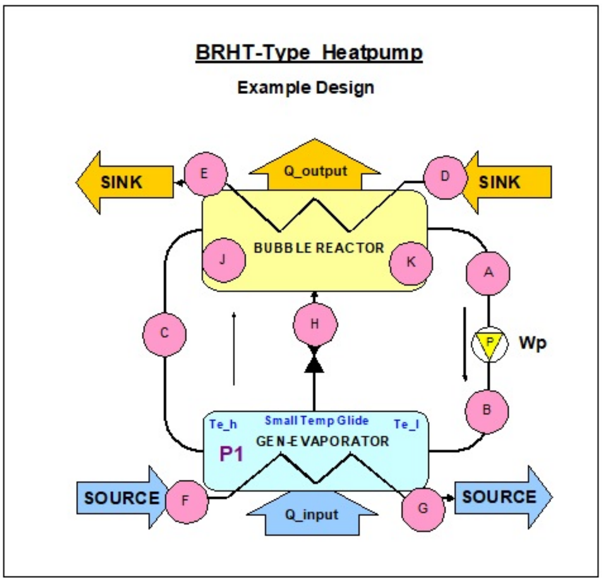

Figure 6: BRHT-Type heat pump.

Principle 3: Combining the thermally powered heat pump and power generating ORC fully regeneratively

When you have a heat transformer with an extremely high efficiency (COP=1) like the BRHT-Type heat pump sketched and described in Figures 4-6 above, delivering higher temperature upgraded heat to the H/E coil of an ORC, power may be generated, with an efficiency depending on the actual temperature levels of ORC heat input and heat rejection, as well as the isentropic expansion efficiency of the turbine. A typical ORC operating between 70°C high temperature and 10°C low temperature may deliver power at realistically 60% of Carnot, (e.g. 10.5% of the heat flowing into the turbine, rejecting 89.5% of this energy as heat. This rejection heat from the ORC is used by the heat pump to offset some of the heat required as Q_input. Overall energy balance tell us that the heat input into the overall REHOS cycle (Q_input) from the external Source would be the leakage heat (Q_leak) lost to the external Sink, as well as the turbine power delivered (Wt). This fully regenerative coupling of the ORC with the thermally driven heat pump is shown in the sketch of Figure 7 below.

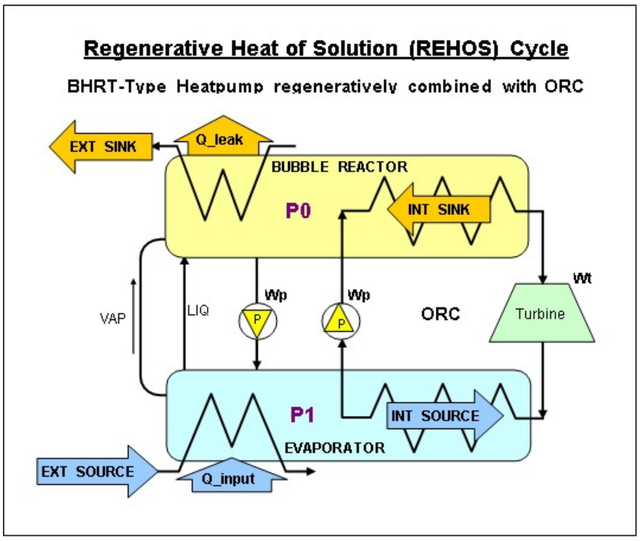

Figure 7: Regenerative coupling of the ORC.

For the REHOS cycle as a complete system, the power output would be:

$W_{nett}=W_{turb}-W_{orc\_pump}-W_{BRHT\_pump}$

and using the values presented in Table 1, we calculate and W_{nett} may be as high as 10.5% (-about 1.0 kW ORC pumping energy) of the heat pump delivered heat of 82 kW_th, calculating to ~ 7.35 kWe with ORC reject heat of ~ 71.2 kW_th. Total heat pump input heat requirement of 81.2 kW_th is partially satisfied with the ORC reject heat, leaving the heat input of Figure 6, (Q_input=81.2-71.2=10 kW_th) with the leakage heat loss would be (Q_leak=2.5 kW_th). The overall cycle efficiency would then be:

$n_{REHOS} = \frac{W_{nett}}{Q_{input}} - \frac{7.35}{10} = 73.5\%$ZThis means that ~ 74% of the thermal energy in the 20°C external heat Source may be converted to electricity, while the balance is heat loss at the higher (70°C) H/E due to imperfect thermal insulation. This may obviously still be optimized to deliver higher REHOS cycle efficiency.

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