

Rankine Cycle Efficiency Increase by the Regenerative Recovery of Historically Rejected Heat_rev2

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Abstract

Steam extraction from Rankine cycle turbines in order to heat the high pressure boiler water in a series of feed heaters, is common practice to increase the thermodynamic efficiency of cycles. Regenerative Cycles are so formed, which form the basis of most modern power generation rankine cycles used by electricity utilities globally.

By utilizing Absorption Heat Transformer (AHT) principles, thermal powered hybrid heat pumps have been developed recently, making it possible to use these heat pumps for reclaiming rejected heat from rankine condensers for providing the required energy for feed water heating, allowing additional power to be generated from the (historically used) extract steam, without requiring more heat to be added in the boiler. Cycle efficiency is therefore increased.

As the AHT-Hybrid heat transformer type heat pumps may generate higher temperatures than the boiler evaporation temperature in some lower temperature rankine cycles, the possibility exist that a much larger portion of the rejection heat in the condenser may be recovered and re-used regeneratively in the cycle. This enhanced regeneration gives rise to the extremely high thermodynamic efficiency achieved by the REHOS Power Cycle.

These Heat Of Solution (HOS) heat transformers, can practically only be used within limited temperature ranges, depending on critical temperature and pressure parameters of the binary media mixtures used e.g. NH₃-in-Aqua may be practical only between -30°C and +140°C making it ideal for ORC systems used e.g. for geothermal power generation.

Keywords: Turbines; High Pressure boiler water; Thermodynamic; Thermal powered hybrid heat pumps

Introduction

The steam engine has been with us for over 100 years now, and utilizes the well-known Rankine cycle, as sketched on the Temperature-Entropy (T-S) Diagram of Figure 1 below. A feed pump is used to increase the pressure of water in moving from (A) to (B) and the water is evaporated at constant temperature and pressure along line (B-C), where all the water is vaporized at (C). The saturated steam is then superheated along line (C-D) before entering the turbine, generating power as the HP steam expand isentropically following line (D-E). Note that the isentropic expansion line (D-E) cross the water-steam saturation line, implying the low pressure and temperature spent steam at point (E) contain a certain percentage liquid droplets in the turbine exhaust (Figure 1).

Heat is added to the cycle in water heating, ($Q_{heating}$) the HP water along line (A-B). More heat is added to evaporate the liquid, (Q_{evap}) along line (B-C), while a third amount of heat is added (Q_{sht}) along line (C-D), to superheat the saturated steam [1]. After the power expansion work (W_{power}) done by the cycle along line (D-E), the remaining latent heat in the low pressure exhaust steam is rejected (Q_{rej}) out of the cycle by water cooled condensing along line (E-A). The power developed by the cycle is represented by the total enclosed (blue) area in Figure 1, and heat rejection may be judged by the length of line (E-A) on the diagram. The thermodynamic cycle efficiency is given by:

$$\eta_{rankine} = \frac{W_{power}}{(Q_{heating} + Q_{evap} + Q_{sht})} \quad (1.1)$$

and because the sum of all energy entering (+) and leaving (-) the cycle add to zero:

$$Q_{input} = (Q_{heating} + Q_{evap} + Q_{sht}) = W_{power} + Q_{rej} \quad (1.2)$$

$$\eta_{rankine} = \frac{W_{power}}{(W_{power} + Q_{rej})} \quad (1.3)$$

While it can be shown that the ideal, or Carnot efficiency of a heat engine operating between the same high temperature reservoir (T_{hot}), and low temperature reservoir (T_{cold}), is limited to a maximum of:

$$\eta_{carnot} = \frac{(T_{hot} - T_{cold})}{T_{hot}} \quad (1.4)$$

Maximum (Carnot) efficiency would be attained then all heat input is done at a single high temperature, unlike the rankine cycle.

Increasing efficiency by regenerative feed heating

Modern power generation use the regenerative (or carnotized) cycle for higher efficiency. Partially expanded steam is extracted from the steam expanding in the turbine cascade at regular intervals and fed to feed water heaters, where it is condensed to use some latent energy from the expanding steam to raise the water temperature stepwise, before it reaches the boiler. This decrease the entropy of the expanding steam stepwise, to form a regenerative cycle as shown in the T-S diagram of Figure 2 below. In the extreme, with numerous very small

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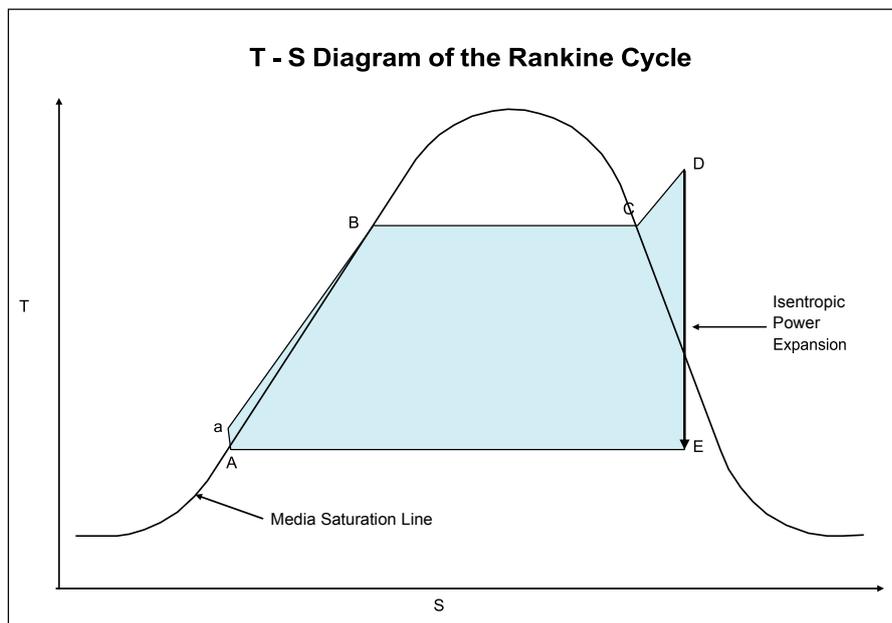


Figure 1: Temperature-Entropy (T-S) diagram.

steps, the regenerative cycle diagram resembles a parallelogram, with the isentropic power expansion line (D-E) very nearly parallel to the water heating line (A-B), as the total heat given off by the expanding steam is added to the pumped feed water in the feed heaters. This result in most heat added as (Q_{evap}) at the single evaporation temperature (T_{hot}), while heat rejection along line (E-A) also occur at the single temperature of the condenser (T_{cold}), with no external heat added to the cycle at different temperatures and therefore the efficiency of the cycle approaching carnot efficiency.

The higher efficiency is graphically visible in the length of the heat rejection line (E-A) of T-S diagram of the regeneration cycle in Figure 2, compared to the length of line (E-A) of the rankine cycle T-S diagram in Figure 1. The decrease in rejected heat (Q_{rej}) in the efficiency formula (1.3) may be used to calculate the efficiency of the regeneration cycle, and should the number of feed heaters in cascade be infinite, the cycle efficiency would calculate to carnot efficiency. In practice the number of feed heaters vary from about 6 to 9, and the number is a trade-off between increased efficiency and the larger capital investment, maintenance and complexity of more feed heaters cascaded in the cycle.

Figure 3 below, show a typical example feed heater (LP-1) from a utility scale (500 MWe) coal-fired power station, being the first LP feed heater closest to the condenser, heated by steam from the last extract point of the LP turbine. Temperatures shown are those of the actual real power station (Figure 2).

Note that the extract steam is at a (saturation) temperature of 68°C and therefore cannot heat the pumped water to a higher temperature than a small approach temperature some 2°C lower than this supply steam temperature. The heater liquid output heated temperature is thus 66°C. Looking at the regenerative cycle diagram of Figure 2, it means that the regenerative heating as used commercially historically cannot quite reach the boiler temperature at point (B), and therefore cannot reach the ideal carnot efficiency.

The only way to change that, is by employing heat pumps to “upgrade” the temperature in each heater. A heat pump is sketched in on the diagram of Figure 4 below, but although it may be used to achieve higher heater output temperatures, the temperatures in this schematic was kept the same as those of Figure 3, specifically so we may evaluate the performance realistically by comparison [2-4].

Heat pump utilization to avoid extract steam

We note that the heat pump installation, pump heat from the lower temperature (condenser) and use that to heat the feed water in the feed heater. No extract steam from the turbine is required for this process, and the idea is that the additional power that may be developed in the turbine as a result of utilizing the formerly used extract steam for generating additional power, should ideally be more than the energy required for powering the heat pump (Figure 3).

We assumed (at first) a standard Vapor Compression (VC) type heat pump using anhydrous NH_3 as medium. For the heat pump to absorb heat from the condenser at 40°C, it would be required for the heat pump evaporator to operate at a temperature lower by a few degrees, so we assumed the evaporator would operate on 35°C. For similar reasons, the heat pump hot condenser (heat output), would have to operate on 73°C, making the saturated NH_3 vapor pressure in the evaporator 13.7 Bar Abs, while the condenser would run at 35.9 Bar Abs. With a compressor isentropic efficiency of 75%, the power needed would be 185.2 kWe/kg NH_3 used, and the heat delivered at 73°C per kg NH_3 used calculate to 919.6 kW_{th}. The efficiency is represented by the Coefficient of Performance (COP) calculated as:

$$COP_{VC \text{ Heatpump}} = \frac{\text{Heat pumped}}{\text{Power used}} = \frac{Q_{heat}}{W_{comp}} = 4.97 \quad (1.5)$$

The ideal heat pump performance (delivering heat) is calculated using temperatures again:

$$COP_{carnot} = \frac{T_{hot}}{(T_{hot} - t_{cold})} = 9.11 \quad (1.6)$$

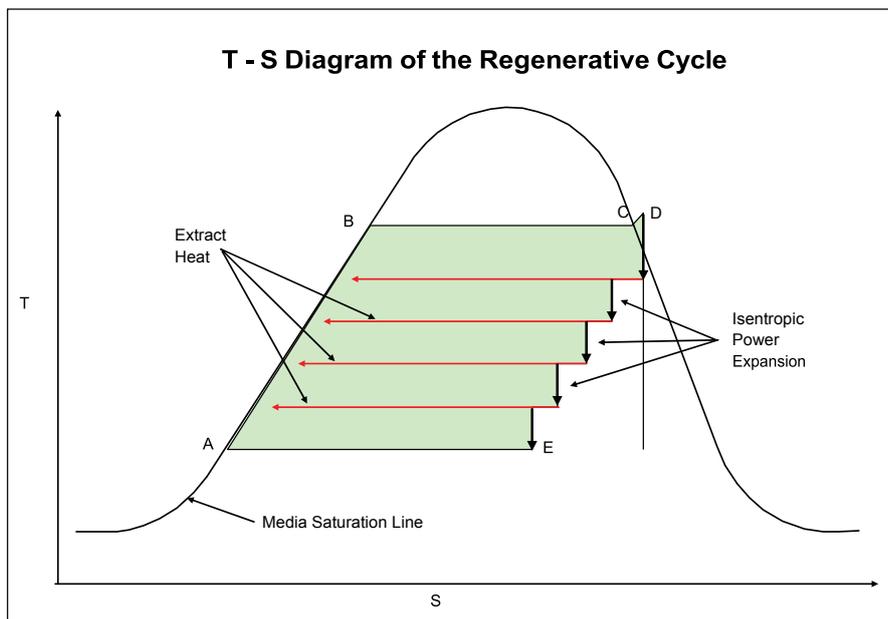


Figure 2: Regeneration cycle.

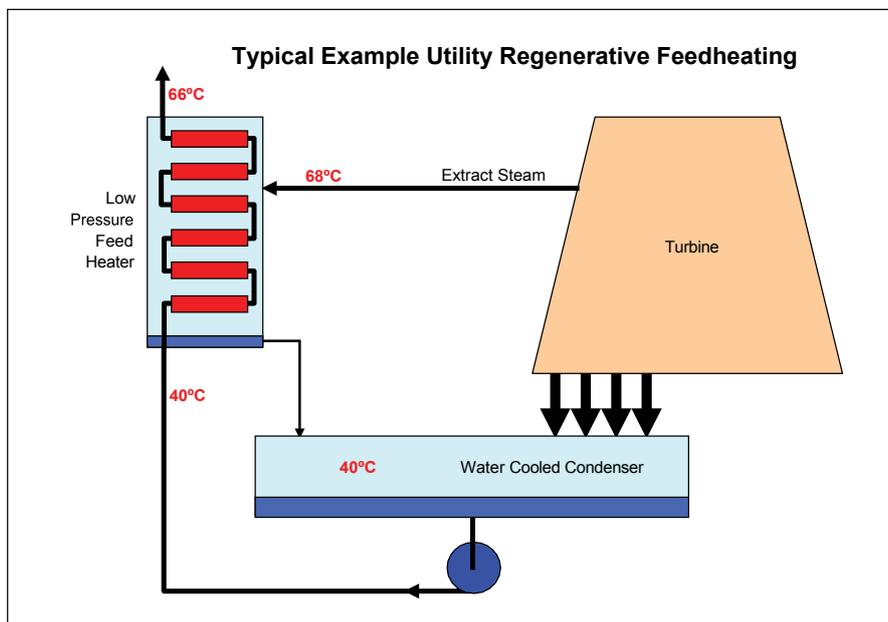


Figure 3: Typical example utility regenerative feed heating.

Interesting to note that the actual VC Heat pump COP is therefore only 54.5% Carnot, as a result of mainly the required increased temperature delta to guarantee heat inflow into the evaporator and outflow from the condenser, as well as the low compressor isentropic efficiency [5-8].

The mass of extract steam from the turbine, that would have been required to deliver the same amount of heat as the VC-Heat pump (using 1 kg/s NH₃ flow), namely 919.6 kW_{th} can be calculated as:

$$M_{extract} = \frac{Heatdelivered}{Enthalpy_of_Condensing} = \frac{919.6kW}{(H_{vapor} - H_{liquid})} = 0.8404kg/s \quad (1.7)$$

and for this mass of steam expanding further down to condenser pressure in the turbine with an assumed 80% isentropic efficiency, the power developed would calculate to 121.37 kWe, which is only 65.5% of the power used by the heat pump compressor (Figure 4).

The standard VC type heat pump can therefore not be used for this purpose, as it uses too much electrical power. We would have to explore heat pumps that use much less electrical power, and combine it with thermal energy for powering the heat pump.

Development of heat transformer type heat pumps

An Absorption Heat Transformer (AHT) work on the principles of using intermediate temperature heat (low temperature) to generate vapor in an evaporator, which is then absorbed into a suitable higher temperature absorber, to release the latent heat in the vapor as well as the Heat of Solution (HOS) of the vapor media into the absorber liquid media to raise the temperature. Temperature “upgrade” or “lift” typically range from 30°C to 80°C in a single stage AHT, depending on the actual binary media combination used. The AHT use mainly thermal energy ($Q_{thermal}$) to generate vapor, and then additionally use a small portion of the total heat pump energy requirement as electrical energy (W_{comp}) to either increase the vapor pressure to the absorber level, or use liquid pumping to increase the pressure to the evaporator prior to evaporation. The AHT used as heat pump would therefore have an efficiency calculated as COP of:

$$COP_{AHT} = \frac{Q_{pumped}}{(Q_{thermal} + W_{comp})} \quad (1.8)$$

The compressor power used is a direct function of the compression ratio, so decreasing this compression ratio drastically, also decrease the electrical power consumed by the AHT (Figure 5).

The Isobaric temperature glide binary reactor H/E

A typical AHT-VC Hybrid heat pump with the same temperature lift to replace the standard VC heat pump mentioned above is sketched in Figure 5 above. Note the very low compression ratio required by this heat pump as a result of utilizing a binary isobaric temperature glide heat exchanger, where the differential pressure is the hydraulic liquid column (2 meter column height was assumed) under gravity pressure of 15 kPa only. The total compressor differential pressure amounts to

132 kPa when the accumulated piping pressure drops are also added in. Although the COP value is actually quite low and calculate to ~1 only, the electrical portion of the heat pump powering energy calculate to only 4.33 kJ/kg NH_3 to power the compressor, as the thermal energy to drive the heat pump account for the balance of the total energy utilized for pumping heat. The low electrical power requirement makes this type of regeneration in feed-heating concept a reality. The temperature-entropy diagram for this example heat pump would typically look like the sketch in Figure 6.

The same Heat of Solution (HOS) type hybrid heat pumps may easily be cascaded for larger overall temperature gains, as a single stage temperature gain would be limited to the 60°C-80°C lift only. A typical low temperature cascade is shown in the sketched diagram of Figure 7 below. Another limitation of this heat pump cascade integration is brought about by the requirement of a zeotropic binary mixture capable of isobar temperature glides, at the proposed operating temperatures. Several different media-pairs are already known among various researchers, but for the NH_3 -in-Aqua mixture, the temperature upper limit would be around 130°C-140°C while the lower limit would be around -20°C to -30°C. This will allow HOS type heat pumps to be utilized replacing the standard feed heaters operating on extraction vapor, but only if the power generation cycle stays within the limitation of the upper and lower temperatures for it's feed heating heat pumps. For lower temperatures (below 140°C e.g. in geothermal heat recovery use) the Organic Rankine Cycle (ORC) would easily form a Regenerative Heat of Solution (REHOS) cycle with very interesting characteristics. This cycle is achieved by integrating the AHT type heat pump sub-cycle regeneratively with an ORC sub-cycle in order to achieve a very high efficiency machine to generate power (Figure 8).

When integrating the AHT type heat pump shown on the T-S Diagram of Figure 6, regeneratively with the rankine (ORC) sub-cycle as shown on the T-S Diagram of Figure 1, above, we can display it graphically in the T-S Diagram sketched in Figure 8, below. Note that the media used in the heat pump may differ from the media used by the ORC, as shown in Figure 8, and in this case the media does not

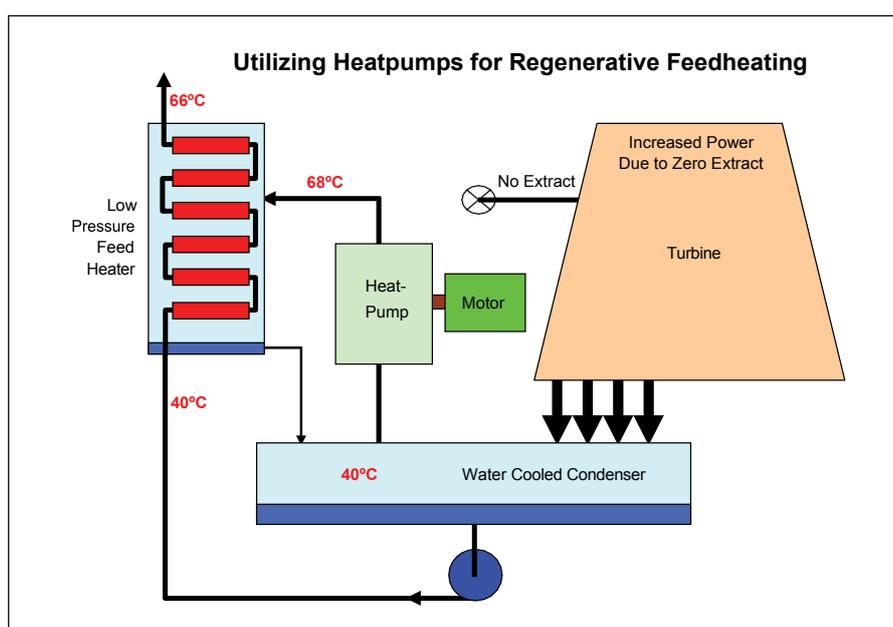


Figure 4: Utilizing heat pumps for regenerative feed heating.

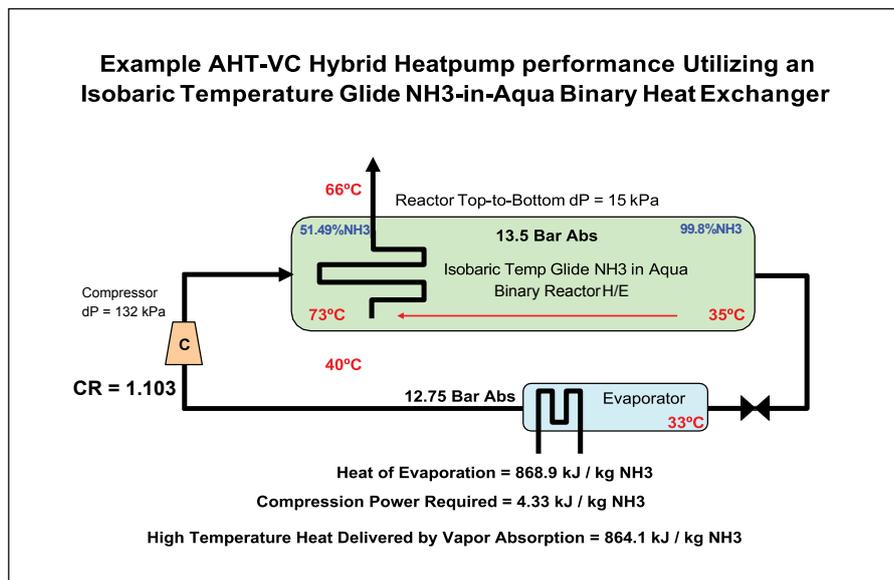


Figure 5: AHT-VC hybrid heat pump.

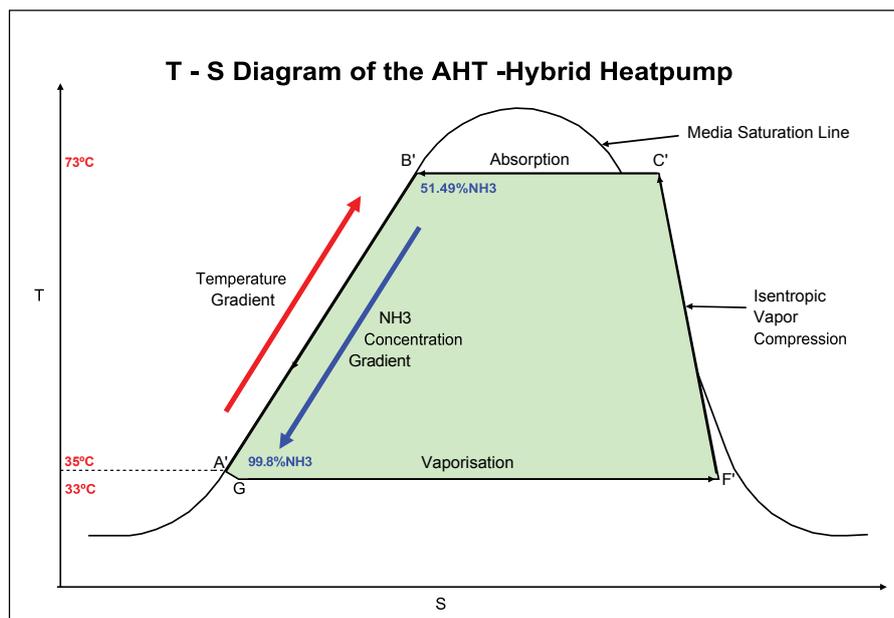


Figure 6: T-S Diagram of the AHT-hybrid heat pump.

mix. The media for the ORC may thus be chosen for the most efficient refrigerant for the operational temperatures and pressures best suited to the turbine, for the specific heat exchangers for heat input and heat reject, available. The heat pump media (unlike the ORC media) need to be a zeotropic binary mixture able to provide isobaric temperature gliding at the intended operating temperatures of the heat pump [9,10].

Complete regeneration is achieved when all the input heat used by the ORC is supplied via the ORC heat exchanger from the heat pump hot absorber, and all ORC Reject heat is absorbed via the ORC Rejection heat exchanger into the heat pump evaporator. This can

only be achieved when the heat pump absorber operate at a higher temperature than the ORC input heat required, as well as the heat pump cold evaporator operate at a lower temperature than the ORC heat rejection temperature to guarantee proper heat inflow and outflow from the ORC, respectively.

Assuming a turbine with 70% isentropic efficiency, using pure NH_3 as operating medium, the pressure at 66°C would be 30.6 Bar Abs, while the low pressure would be 15.8 Bar Abs at a saturation temperature of 40°C. The resulting ORC efficiency calculates to a netto efficiency of 4.4%, which represent a value of about 58% Carnot (Figure 8).

The integrated T-S Diagram sketched in Figure 8, above, try to show the heat pump sub-cycle (in green) interacting with the ORC (in yellow) via the red arrows representing the heat exchangers. All the heat pump heat is sourced from the evaporator (line G-F') and the pumped heat sink deliver this heat to the ORC. The component layout sketch of Figure 9, below, use the same identifier letters and may be easier to explain the fully regenerative interaction of the heat pump sub-cycle with the ORC [11,12]. Critical to the heat pump operation is the pseudo-isobaric temperature gliding bubble reactor containing a strong solution of

the zeotropic binary mixture of NH₃ in aqua. This reactor has a large diameter to facilitate both vapor bubble vertical up flow and lower concentration mixture down flow driven by buoyancy effects against gravity. This density driven counter current flows allow heat-, mass-, and species transfer with the resultant high ammonia concentration and low temperature at the reactor top and low ammonia concentration higher temperature at the bottom of the reactor (Figure 9).

In operation the very rich liquid mixture (in this example 99.8% NH₃ in aqua), flows from the reactor top end through the throttling

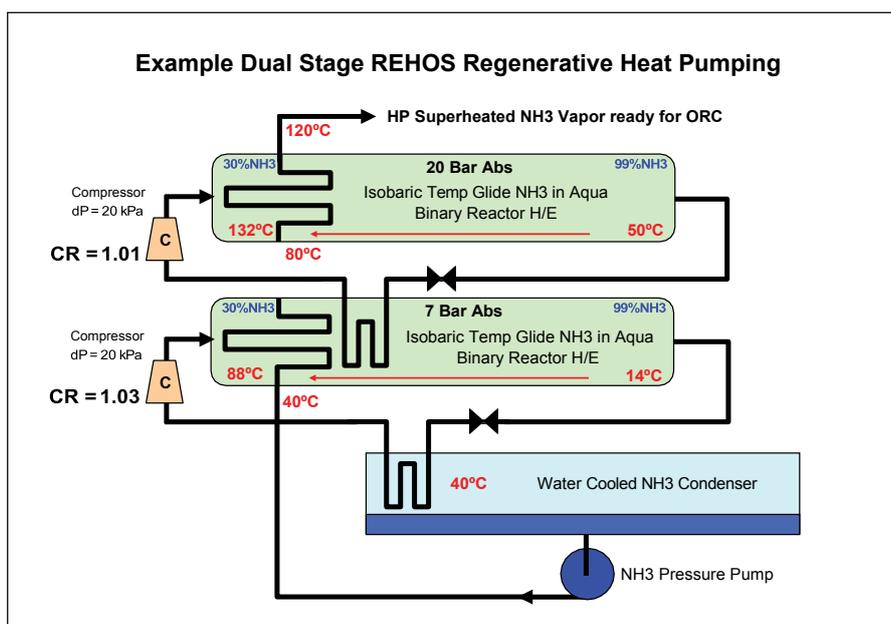


Figure 7: Low temperature cascade.

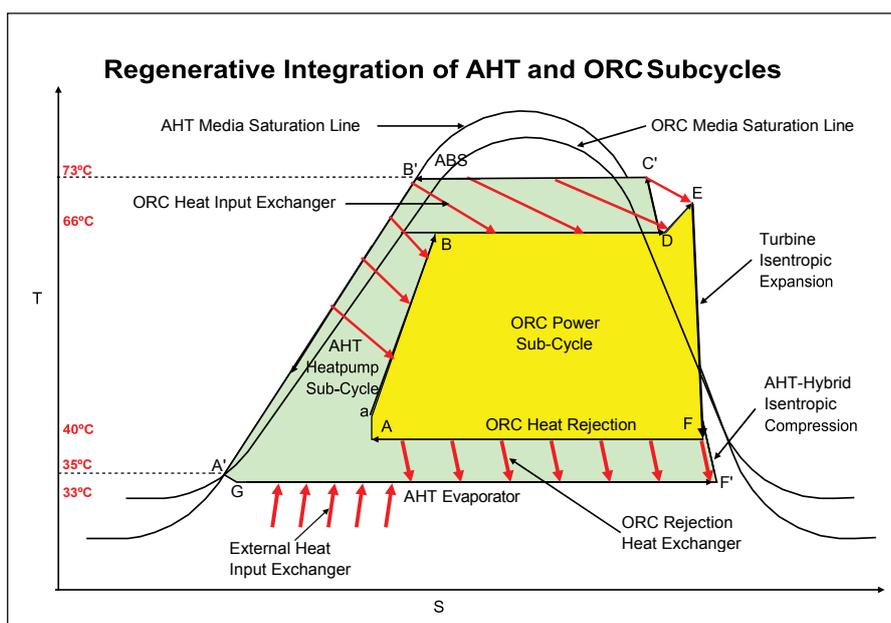


Figure 8: The integrated T-S diagram.

valve into the evaporator with a slightly lower pressure (and 2°C lower temperature) where it is completely flashed into vapor, essentially isothermally. The small % water in this liquid is carried with the vapor to the compressor. Energy required for vaporizing this liquid comes from both the waste heat source H/E coil (water flow H->J) and the ORC heat rejection H/E coil (condensing flow F->A). Should we do an energy balance around the ORC and AHT cycles, we note the following state points of the alphabetically marked positions around the cycles (Table 1).

We calculate the energy clockwise (ORC deliver energy) around the T-S Diagram following the convention that energy added to the cycle is positive and energy removed from the cycle is negative:

Condensate Pumping (A->a)=>

$$W_{pump} = M \cdot (H_a - H_A) = 5.1 \text{ kW} \quad (1.9)$$

Liquid heating (a->B)=>

$$Q_{liquid_heating} = M \cdot (H_B - H_a) = 127.8 \text{ kW}_{th} \quad (1.10)$$

Evaporation Heat (B->D)=>

$$Q_{evap} = M \cdot (H_D - H_B) = 962.6 \text{ kW}_{th} \quad (1.11)$$

Total ORC Heat Input (a->E)=>

$$Q_{heat} = M \cdot (H_E - H_a) = 1090.3 \text{ kW}_{th} \quad (1.12)$$

Turbine Power Delivered (E->F)=>

$$W_{turbine} = M \cdot (H_F - H_E) = -54.7 \text{ kW}_{e} \quad (1.13)$$

The calculated value of $W_{turbine}$ in the formula (1.13) above is negative, as the Table 1 listed value of HF is smaller than the value of HE. This is consistent with the convention mentioned above, that energy removed from the cycle is negative, while energy added to the cycle is positive

ORC Rejection Heat (F->A)=>

$$Q_{rej} = M \cdot (H_A - H_F) = -1040.8 \text{ kW}_{th} \quad (1.14)$$

The calculated value of Q_{rej} in the formula (1.14) above is negative, as the Table 1 listed value of HA is smaller than the value of HF. This is consistent with the convention mentioned above, that energy removed from the cycle is negative, while energy added to the cycle is positive. Heat Balance (clockwise in delivering power) around ORC in the sequence=>

A -> a -> B -> D -> E -> F -> A

$$\sum_{CYCLE}^{ORC} Energy = W_{pump} + Q_{liquid_heating} + Q_{evap} + W_{turbine} + Q_{rej} = 0 \quad (1.15)$$

Similarly, state points are identified around the AHT-VC Hybrid heat pump and listed in Table 2. Note that the mass column have been left blank for the two positions A' and B' because although the actual NH₃ flow should be the same as C', F' and G, the reactor also contain a large mass of water. As vapor is absorbed (and heat produced) close to the bottom end of the reactor, the NH₃-rich vapor bubbles and immediate bubble surrounding NH₃-rich liquid will migrate upwards due to density differences, and the heated, NH₃-lean liquid would migrate downwards. This counter-current internal circulation flow in the reactor facilitate mass-, heat-, and species transfer maintaining the temperature and NH₃ concentration gradients typical of the isobaric temperature glide heat exchanger. Heat continuously being removed

from the hot reactor bottom, as well as the internal liquid circulation described above, make the vapor absorption process nearly isothermal (Table 2).

The evaporator is also providing a nearly isothermal evaporation process as a result of the high NH₃ concentration of the liquid entering the evaporator (99.8% NH₃). The small amount of water is carried out with the NH₃ vapor, to the vapor compressor [13,14].

Note that the heat transformer type heat pumps are powered mainly by thermal energy, as mentioned already. In Table 2 the amount of thermal energy required to power this specific heat pump have been calculated and listed as $Q_{heatpump}$. If we assume the COP may be around 60% Carnot, it calculates to a value of 5.19 as listed in Table 2. The actual amount of thermal energy may therefore be calculated:=>

$$COP_{heatpump} = \frac{Q_{pumped_heat}}{(Q_{heatpump} + W_{comp})} = 5.19 \quad (1.16)$$

calculating to $Q_{heatpump} = 213 \text{ kW}$ with the listed mass flow of

$$M_{AHT} = 1.313 \text{ kg/s.}$$

All input heat enter the AHT-VC Hybrid cycle in the evaporator along line (G-F').

$$Q_{heatpump_input} = M_{AHT} \cdot (H_{F'} - H_G) = 1128.5 \text{ kW}_{th} \quad (1.17)$$

The heat pump compressor energy in line (F'->C') calculate to:=>

$$W_{comp} = M_{AHT} \cdot (H_{C'} - H_{F'}) = 5.65 \text{ kW}_{e} \quad (1.18)$$

Vapor absorption in hot reactor bottom in line (C' -> B')=>

$$Q_{absorption_heat} = M_{AHT} \cdot (H_{B'} - H_{C'}) = -1234.9 \text{ kW}_{th} \quad (1.19)$$

The calculated value of $Q_{absorption_heat}$ in formula (1.19) above is negative, as the Table 2 listed value of HB' is smaller than the value of Hc'. This is consistent with the convention mentioned above, that energy removed from the cycle is negative, while energy added to the cycle is positive.

Absorbed NH₃ moving down the isobaric temperature glide (B'->A')=>

$$Q_{glide} = M_{AHT} \cdot (H_{A'} - H_{B'}) = 100.7 \text{ kW}_{th} \quad (1.20)$$

Leaving the heat exchanger, a small pressure drop across the throttling valve along line (A'->G) would be isenthalpic, flashing a very small amount (~1%) of liquid into vapor as the mixture flow into the evaporator, dropping the temperature only 2°C in this process.

Heat balance around the AHT-VC Hybrid type heat pump sub-cycle (in anti-clock direction as the cycle absorb heat) in the sequence=>

G -> F' -> C' -> B' -> A' -> G

$$\sum_{CYCLE}^{AHT} Energy = Q_{heatpump_input} + W_{comp} + Q_{absorption_heat} + Q_{glide} = 0 \quad (1.21)$$

Integrating the two sub-cycles, it becomes clear the heat rejection energy from the ORC we called Q_{rej} has a large negative value, and would be totally absorbed into the heat pump evaporator, decreasing drastically the amount of external heat required to balance the energy around the heat pump cycle. As heat required by the heat pump flows into the heat pump (and is therefore calculated as positive values) the external heat absorbed into the evaporator may be calculated=>

$$Q_{external} = Q_{heatpump_input} + Q_{rej} = 87.7 \text{ kW}_{th} \quad (1.22)$$

Position	Temp (Celsius)	Press (kPa)	Mass (kg/s)	Enthalpy (kJ/kg)	Entropy (kJ/kg.K)	Quality (x)	Media
A	40	1575	1.0	191.1	0.626	0%	NH ₃
a	40	3058	1.0	196.2	0.652	0%	NH ₃
B	66	3058	1.0	324.0	1.046	0%	NH ₃
D	66	3058	1.0	1286.6	3.884	100%	NH ₃
E	66	3058	1.0	1286.6	3.884	100%	NH ₃
F	40	1575	1.0	1231.9	3.968	94.71%	NH ₃
W_turbine=54.7 kW W_pump=5.1 Kw				η_ORC= %Carnot=		4.4%	
						57.0%	

Table 1: Listed value of HF is smaller than the value of HE.

Position	Temp (Celsius)	Press (kPa)	Mass (kg/s)	Enthalpy (kJ/kg)	Entropy (kJ/kg.K)	%NH ₃ (kg/kg)	Media
A'	35	1352		166.6	0.568	99.80%	NH ₃ -H ₂ O
B'	73	1357		89.9	1.064	51.49%	NH ₃ -H ₂ O
C'	39	1408	1.313	1030.7	4.273	99.80%	NH ₃ -H ₂ O
F'	33	1276	1.313	1026.4	4.230	99.80%	NH ₃ -H ₂ O
G	33	1276	1.313	166.6	0.575	99.80%	NH ₃ -H ₂ O
H	45	200	2.101	188.4	0.639	-	H ₂ O
J	35	195	2.101	146.6	0.505	-	H ₂ O
W_comp = 4.3kJ/kg Q_heatpump = 162.2kJ/kg		Electrical COP_e= Thermal COP_th= Real Thermodynamic COP (~ 60%Carnot)=				200 1.01 5.19	

Table 2: Real thermodynamic COP.

as the evaporator of the heat pump. Remembering that the heat used by the ORC is summed in formula (1.12) as:

$$Q_{heat} = 1090.3kW_{th} \quad (1.23)$$

In this example the heat flow into the integrated two sub-cycles (REHOS Cycle) may be added up providing us with the complete energy balance of the REHOS cycle:

$$\sum_{CYCLE}^{REHOS} Heat = Q_{heatpump_input} + W_{comp} + Q_{absorption_heat} + Q_{glide} + W_{pump} + Q_{heat} + W_{turbine} + Q_{rej} = 0 \quad (1.24)$$

$$\sum_{CYCLE}^{REHOS} Heat = Q_{external} + W_{comp} + Q_{absorption_heat} + Q_{glide} + W_{pump} + Q_{heat} + W_{turbine} = 0 \quad (1.25)$$

or written in real values:

$$\sum_{REHOS}^{CYCLE} Heat = (87.7) + (5.67) + (-1234.9) + (100.7) + (5.12) + (1090.3) + (-54.7) \approx 0 \quad (1.26)$$

The only heat input into the cycle ($Q_{external}=87.74$ kW) comes

from external sources and must be supplied by the Source thermal input.

This allow us to calculate the total REHOS Cycle first order energy efficiency :=>

$$\eta_{REHOS} = \frac{Netto\ Power}{Heat\ Input} = \frac{(-W_{turbine} - W_{comp} - W_{pump})}{Q_{external}} \quad (1.27)$$

$$\eta_{REHOS} = \frac{(54.66 - 5.12 - 5.67)}{87.74} = 50.0\% \quad (1.28)$$

As the REHOS cycle essentially consist of 2 sub-cycles, namely a heatpump for which the ideal, Carnot COP may be calculated in terms of the Source temperature (T_c) and the

Sink temperature (Th) calculated in Kelvin (see also formula (1.6) on page 6):=>

$$COP_{carnot} = \frac{T_h}{T_h - T_c} \quad (1.29)$$

and an ORC for which the ideal, Carnot efficiency may be calculated in temperature terms as:=>

$$\eta_{ORC} = \frac{[(T_h - x) - (T_c + y)]}{(T_h - x)} \quad (1.30)$$

which is the same formula that was provided (1.4) on page 3, but the ORC high temperature is lower than the heat pump Sink temperature (Th), by $x^\circ C$ to allow heat to flow from the heat pump Sink to the ORC hot heat exchanger. Similarly, the ORC heat rejection must be at a temperature higher than the heat pump Source temperature (T_c) by $y^\circ C$ to allow heat to flow from the ORC heat exchanger into the heat pump Source (heat pump evaporator). In the example calculations above, $x=7^\circ C$ and $y=7^\circ C$ to be practical, with ($Th=73^\circ C$) and ($Tc=33^\circ C$). The maximum (Carnot) efficiency of the REHOS cycle can therefore be calculated as:=>

$$\eta_{carnot} = \left[\frac{T_h}{(T_h - T_c)} \right] \cdot \left[\frac{\{(T_h - T_x) - (T_c + y)\}}{(T_h - x)} \right] \quad (1.31)$$

and a few sample calculations was done and is presented in Table 3, below, for illustration purposes (Table 3).

Should the values of x and y be reduced to zero, the Carnot efficiency calculate to 100% for the REHOS cycle maximum efficiency, but every heat exchanger need at least a small temperature differential

to guarantee proper heat flow around the cycle. In the first line of Table 3 the temperatures of this example REHOS cycle is represented, which means that the mass flow of medium in the heat pump sub-cycle set to be ($\dot{M}_{AHT}=1.313 \text{ kg/s}$) for heat balance of the AHT-, ORC-, as well as the REHOS cycle would guarantee system efficiency less than Carnot efficiency! In this case some 4% of the ORC heat input (Q_{heat}) is un-usable and need to be dissipated (rejected) from the hot bubble reactor bottom, while the calculated REHOS cycle efficiency of 50% (see calculation in (1.28) above, correspond to the realistic ~ 79% of the Carnot value listed in Table 3. As expected, Table 3 also show that the higher the temperature span between Source and Sink, the higher the maximum cycle efficiency, and obviously the smaller the heat exchangers approach temperatures (x and y), the higher the maximum cycle efficiency.

Although the example REHOS cycle analyzed above has a thermodynamic efficiency of 50%, optimizing the design for minimum heat exchanger approach temperatures and a more reasonable temperature span between Source and Sink would easily produce cycle efficiencies of 80%-90% in utilization of very low temperature (below 50°C) waste heat to produce power (Figure 10).

The vapor absorption process was already used extensively for years

REHOS Maximum (Carnot) Cycle Efficiency				
T _h (Celsius)	T _c (Celsius)	x (Celsius)	y (Celsius)	Carnot Efficiency
73	33	7	7	66.3%
73	33	5	5	76.1%
73	33	2	2	90.5%
73	33	2	0	95.6%
73	33	0	0	100.0%
120	-30	7	7	92.3%
120	-30	5	5	94.5%
120	-30	2	2	97.8%
120	-30	2	0	99.2%

Table 3: Sample calculations.

in Absorption Heat Transformers (AHT's) for temperature lift, used for the economic temperature "upgrade" of very low temperature heat sources. It is interesting to know that this high REHOS cycle efficiency is attained even though the ORC power sub-cycle have an efficiency of only 4.4%, but may be somewhat larger if we design the ORC sub-cycle to operate on a higher temperature delta. The ORC expander efficiency also have very little effect on the efficiency of the REHOS cycle, and the main (99%) of REHOS cycle inefficiency is due to thermal heat loss as a result of the non-perfect thermal insulation and irreversible heat transfer processes. The secret of this high performance is the large regenerative heat recovery of the ORC heat rejection made possible by the absorption process in the isobaric temperature glide direct contact heat exchanger.

As Figure 9 is quite busy, we have redrawn a simplified version of the complete cycle as Figure 10, above. In this example calculation presented in this paper, the external heat in flow need to come from a waste-heat source with a temperature some degrees above the 33°C heat pump evaporator temperature e.g. an existing utility rankine cycle cooling system with the condenser operating on 45°C-48°C. With the REHOS Cycle, however, the cold end of the isobaric temperature glide heat exchanger may be chosen arbitrarily to suit the waste heat source temperature from which power need to be generated. It is not limited to the availability of low temperature cooling water, as the cycle heat rejection takes place at higher temperatures, so cooling water is not used and the REHOS cycle input heat evaporator may therefore easily be designed to operate at a lower temperature like 0°C to be able to extract ambient heat from the environment at 20°C-30°C. Note that no cooling towers are required, as all the ORC reject heat is recycled by using the heat pump, and heat is rejected out of the cycle by thermal leakage from hot components only. The high cycle efficiency, independent of actual temperature levels makes the REHOS Cycle the perfect power generator for lower temperatures. It has immediate great value in low-grade heat recovery, even recovering heat from existing utility scale rankine cycles to generate a lot of power from the cooling

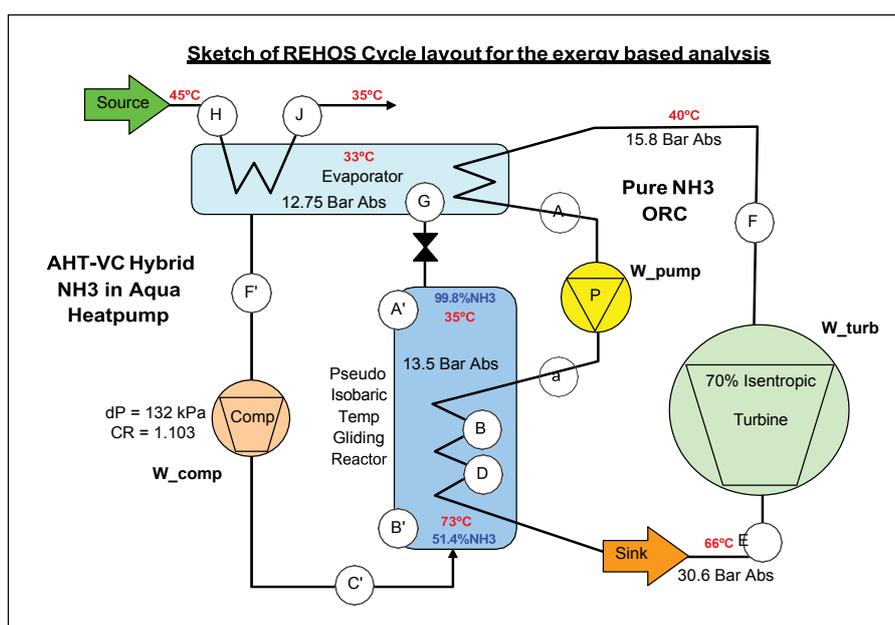


Figure 9: The component layout.

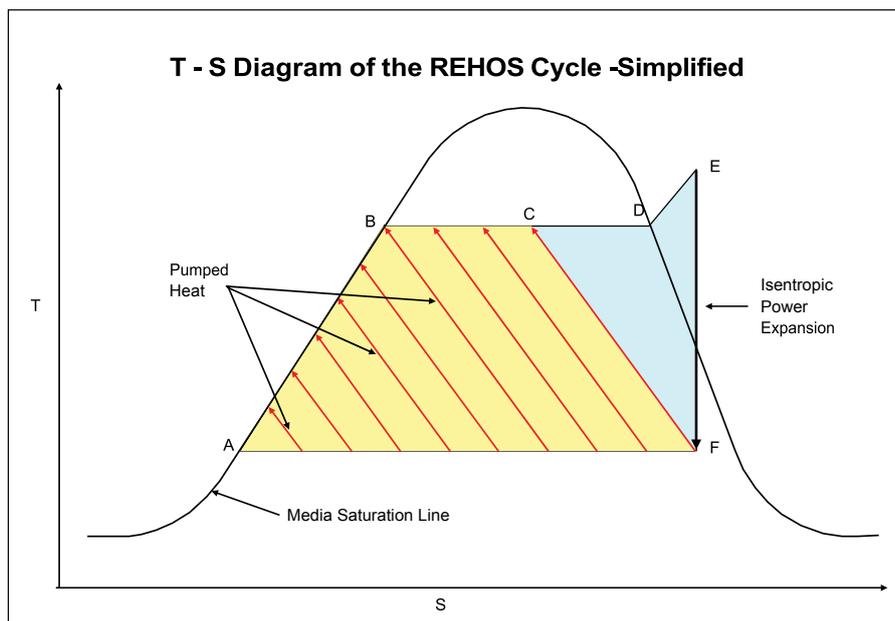


Figure 10: T-S diagram of the REHOS cycle-simplified.

circuits, re-using the reject heat from the condenser. The uses of the REHOS technology are only limited by our small imagination.

In my previous publications (referenced below) I have comprehensively dealt with an AHT Hybrid heat pump design without vapor compressor in [14], while the paper [10] described the same hybrid heat pump, but this time using a vapor jet ejector as compressor. The publication [8] compared the VC type heat pump with various AHT-Hybrids and also introduce the HOS heat pump, as used in this current paper, while [7] clarify some conventions in calculation of COP's in heat powered heat pumps and heat transformers. The referenced paper [13] describe the details of the isobaric temperature glide zeotropic binary reactor heat exchanger, and [3] clarify some of the process parameters used [15,16].

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