

COABSORBENT HEAT PUMPS

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Abstract: Heat pumps based on the coabsorbent cycles, recently introduced by the author, may be a feasible challenge for the classic mechanical vapour compression heat pumps, because of at least two main characteristics of coabsorbent technology: a) it has the capacity to process depleted heat/sink sources having a small temperature gap between them (e.g. 15-40), due to a particular technique, that of truncation, and b) given that classic condensing absorption technology is a particular case of the coabsorbent one, it is versatile, offering a broad technical solutions portfolio to a given application, which range from highest efficiency + highest complexity to lowest efficiency + most reduced complexity of a heat pump. First, a short introduction of the coabsorbent cycles is outlined, with emphasis of the truncated coabsorbent cycles and of the heat pumps based on the heating fractals, operated in an absorption and hybrid mode. Out of all coabsorbent heat pump cycles analysed, three applications have been promoted for this work: a) nontruncated hybrid, b) simple truncated hybrid and c) quadruple truncated, respectively, in connection with heat/sink sources coming of naturally (underground water/ambient air), or industrially (Steam Rankine Cycle (SRC) and Organic Rankine Cycle (ORC) related). Generally, the heat pumps COP and capacity depend on truncation columns complexity in such a way that the higher the complexity, the higher the COP, but the lower the capacity. Their COP depend on temperature lift and gap and generally increase with truncation column high increase from several times to one and two magnitude orders as compared to mechanical vapour compression existing heat pumps COP. Heat (depleted) source use effectiveness, E , is smaller than that of a mechanical vapour compression existing heat pump, ranging from about 24 to 47 percent for c) to a) applications, in absorption working mode, and from about 30 to about 70 percent in the hybrid working mode.

Key words: *heat pumps, coabsorbent technology, efficiency, absorption, hybrid, heating and cooling nontruncated coabsorbent cycles, heating and cooling fractals, truncated cycles.*

1 INTRODUCTION

Heat pumps are based today on mechanical vapour compression (e.g. Radcenco 1985). This method is easy to apply as long as low grade heat sources and electrical power are available. However, from second law point of view, it suffers from a known major mismatch between supplying source (electricity=high quality energy) and the task to be accomplished (thermal energy producing, of low quality, usually at 60-90 C, for domestic hot water and house heating), with poor primary energy use and negative environmental impact consequences. The absorption technology can be utilized in order to decrease this source-task mismatch. Heat transformers (e.g. Alefeld 1982, or Gutkowski 1987) are to be mentioned as absorption most simple units. They are supplied by heat sources which are upgraded with a thermal COP (useful heat output/heat source heat input) usually of ~ 0.45 - 0.48 . For cycles with higher temperature lifts, condensing temperature might become critical, with too low values, which limits application. Absorption heat pumps COP and temperature lift increase was a permanent research concern.

Here, the multi-effect cycle works must be cited first (e.g. Alefeld 1982a, Ziegler and Alefeld 1987). Unfortunately, their extensive application is hindered in many cases by natural drawbacks which working fluid-absorbent combinations are confronted with (e.g. limited solubility field, absorbent volatility). Absorbent own pressure existence, may require vapour rectification use in refrigeration applications, which significantly reduces COP, or may cause absorbent inventory problems, which hamper plant continuous working. Besides pure absorption, hybrid operation (synergy between absorption and mechanical vapour compression) was also thoroughly investigated, as offering a broader heat pumping design solutions. Works of Altenkirch and Osenbrück (Niebergall 1959) paved this way in the early 20th century, and relative recent theoretical works in this direction are due for example to (Alefeld 1982b).

Coabsorbent technology was introduced recently on the heat pumping scene (Staicovici 2006a-2007h), shortly presented above, aiming at solving some of mentioned technical problems, and bring new opportunities in efficient absorption design and operation. In this work, we shall present theoretical aspects concerning coabsorbent heat pumps, with emphasis on the second law meet in heat production, through a new feasible design and low grade heat sources capitalization proposals.

2 THE NONTRUNCATED HEATING (HEAT TRANSFORMER) COABSORBENT CYCLE

The basic cooling and heating nontruncated cycles are introduced elsewhere (Staicovici 2006a, g). Here, the heating one is represented in the $\log p - 1/T$ diagram in Fig. 1. Formally, it can be obtained from the cooling nontruncated cycle through an 180 degree rotation in the $\log p - 1/T$ diagram plane, and vice versa. Neglecting the internal heat recovery, its operation is as

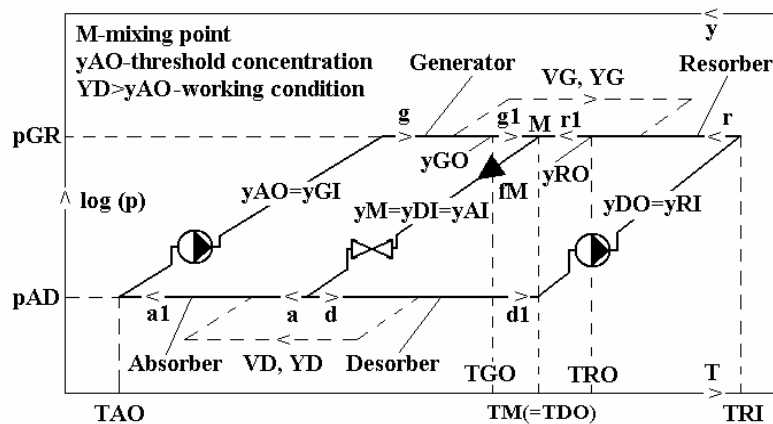


Figure 1: The nontruncated heating (heat transformer) coabsorbent cycle.

follows (see arrows): the absorbents exiting the high pressure resorber and generator are being mixed up in the mixer M, the resulting cumulated flow f_M lives the mixer with concentration y_M and then is expanded and distributed in pre-established complementary quantities to the low pressure absorber and desorber, and finally, the absorbents exiting the desorber and the absorber are pumped to the high pressure resorber and generator, in order to close the cycle. Useful heating effect takes place in resorber. The nontruncated heating coabsorbent cycle has as input data, the desorber outlet temperature (heat source related) and concentration, T_{DO} and y_{DO} , respectively, the resorber inlet temperature (related to useful heating effect maximum temperature), T_{RI} , and the absorber outlet temperature (sink source related), T_{AO} . From these

four parameters it results the rest of the useful parameters: p_{AD} , p_{GR} , $T_M (= T_{DO})$, y_M , and y_{AO} . As compared to the classic condensing heat transformer, which has $y_{AO} = y_{GI} \cong 1.0$, the coabsorbent cycle operates with a smaller temperature lift, equal to $(T_{RI} - T_{DO})$, and a smaller absorber end temperature T_{AO} , for same generation end temperature $T_M (= T_{DO})$. Similarly to the condensing cycle, temperature lift increase is requiring absorber cooling till temperatures T_{AO} which sink sources not always can supply. In order to solve this problem, usage of a special technique, named truncation (Staicovici 2006d, g, 2007h), is particularly interesting in our context. Truncation is possible only with coabsorbent technology. Next, we shall give its short description and of the cycles making use of it.

3 COABSORBENT TRUNCATED CYCLES

Truncation meets second law, enabling nontruncated coabsorbent cycles to work either with smaller temperature heat sources for generation and desorption processes and same sink source temperature, in case of the cooling operation, $T_{GO,t} \leq T_{GO,nt}$, Fig. 2a, or with higher temperature sink sources for absorption and resorption processes and same heat source temperature, $T_{AO,t} \geq T_{AO,nt}$, in case of the heating (heat transformer) operation, Fig. 2b. The

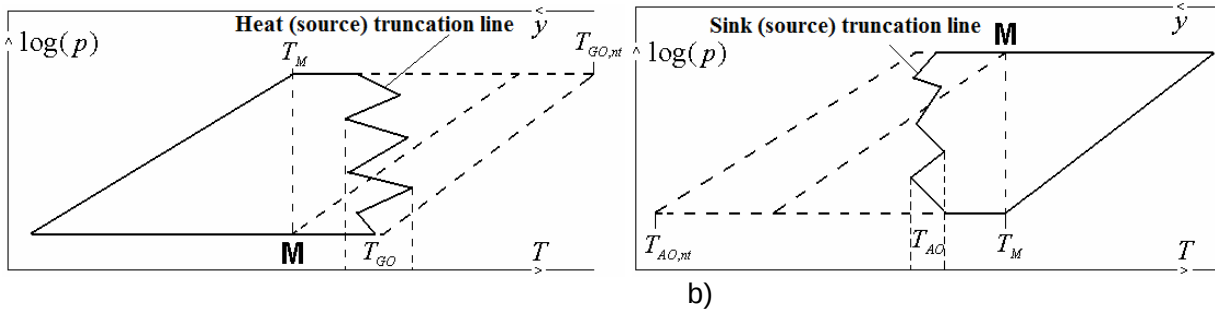


Figure 2: Truncation of the coabsorbent cycles: a) – cooling cycle; b) – heating cycle.

truncation is done upon a certain *heat (source) truncation line*, which marks the end temperature of the generation and desorption processes in case of the cooling cycles, Fig. 2a, and upon a *sink (source) truncation line*, marking the end temperature of the absorption and resorption processes in case of the heating cycles, Fig. 2b. Truncation lines can be designed in a free or in a pre-established way, according to sources characteristics, Fig. 2. Both cooling and heating nontruncated coabsorbent cycles have same time desorption-generation and absorption-resorption processes, therefore can benefit of a double truncation. The two truncation lines of a same nontruncated cycle define a *truncation column*, intrinsically related to a truncated cycle and referred to later in this paper. The importance of truncation is emphasized shortly next. Many applications, not all, have quasi-constant temperature heat and sink sources, equal within the accuracy limits of the necessary heat exchange temperature pinch to mixing points temperatures, $T_{GO} = T_{M_{heating}} = \text{const.}$, e.g. (40-70) °C, and $T_{RO} = T_{M_{cooling}} = \text{const.}$, e.g. (10-40) °C, respectively. Heat and sink lines are separated by a minimum temperature difference (gap) $\Delta T_M = T_{M_{heating}} - T_{M_{cooling}} \geq (8-10)^\circ\text{C}$, which is required in order that cycle operation be possible. The higher this temperature difference, the better for cycle COP increase and complexity decrease. Unlike other known absorption cycles, most important feature of a truncated cycle is that in theory it is capable to process low temperature supplying sources

forming heat/sink pairs of small temperature differences, e.g. $\Delta T_M = (10 - 50)^\circ\text{C}$, with extremely high efficiency ($\text{COP}=15$ to 200), both in cooling and heating working mode. This is why, despite a higher complexity, which is chosen by the designer upon heat/sink sources availability, truncated cycles can have very important applications in clean heat pumping and power production. Nature and human industrial activity create such pairs of heat/sink sources, the problem is to identify and use them appropriately. One of the most attractive might be that offered by a thermal power plant (e.g. Steam Rankine Cycle (STR), Organic Rankine Cycle (ORC)) where condenser cooling water and the cooling tower cooled water could be the truncated cycle heat/sink sources, respectively. Recent published works (Staicovici, 2006e, f, 2007a-h) emphasize these aspects describing for instance the author proposal of using nontruncated and truncated cycles (hybrid or not) with ammonia/water in coupling with the thermal power stations for most efficient district power, heat and cooling (trigeneration) in cities (power plant global efficiency higher than 0.9), Fig. 3.

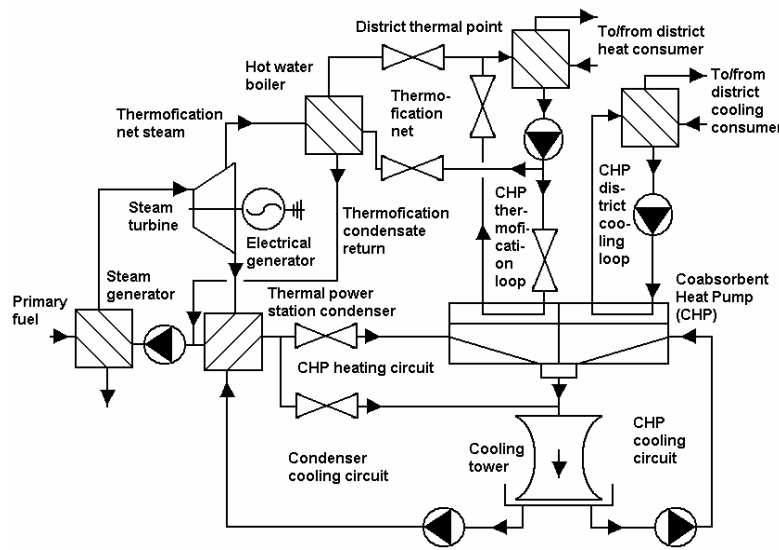


Figure 3: District heating and/or cooling flow chart using coabsorbent heat pumping plant-STR (ORC) coupling.

The special technique required by truncation will be described next (Staicovici, 2006d, g, 2007h). A truncated heating cycle is plotted by solid line in Fig. 4 (see also block flow chart). It comes of the nontruncated heating cycle, represented in the same very diagram by the external solid line contour on the right side and by the dashed line on the left side. It is including a high pressure resorption process (1-2), p_h , e.g. (10-50) bar, where useful heating effect occurs, e.g. $T_{RI,n} = (353,15 - 453,15)\text{K}$ and a truncation column having pressures between p_h and a low pressure value p_l , e.g. (0.1-2) bar. The truncation column is made up by a high pressure generation process (3-4) coupled on vapour side with the high pressure resorption process (1-2), a high pressure mixing process of absorbents coming of the high pressure generation and resorption processes in order to generate the one mean y_M concentration and temperature $T_M(p_h, y_M) \leq T_{M\text{heating}}$ absorbent, point M, a stage of low pressure absorption ($5_{AI} \equiv 5_{DI} - 5_{AO}$) and desorption processes ($5_{AI} \equiv 5_{DI} - 5_{DO}$) and a series of i stages, $i=1, \dots, n$, $n \in \mathbb{N}$, of isobar opposite intermediary generation ($5_i - 5_{GO,i}$) and resorption processes ($5_i - 5_{RO,i}$), with stages pressures decreasing from p_h to p_l , $p_l \leq p_{i+1} < p_i < p_h$ and temperatures between

$T_{M_{cooling}}$ and $T_{M_{heating}}$ and being coupled on vapour side. The vapour have essentially decreasing mean concentrations $Y_{G,m,i+k-1} \geq Y_{G,m,i+k} \geq Y_{D,m}$ and fulfil the concentration threshold condition $y_{RO,i+k} \leq Y_{G,m,i+k}$, $k \leq n-i$, $i, k, n \in N$, $y_{AO} \leq Y_{D,m}$. The stages are supplied in pre-established quantities by uniform absorbents coming, for the first stage of the mixture of the one mean concentration absorbent with that coming of the second stage of resorption, point 5_1 , for each of the following $i=2, \dots, n-1$ next stages of the mixture of the absorbent coming of the first superior resorption stage $i+1$ with that coming of the first inferior generation stage $i-1$, points 5_i , $i=2, \dots, n-1$ (see arrows), for the $i=n$ stage by the mixture of the absorbent coming of

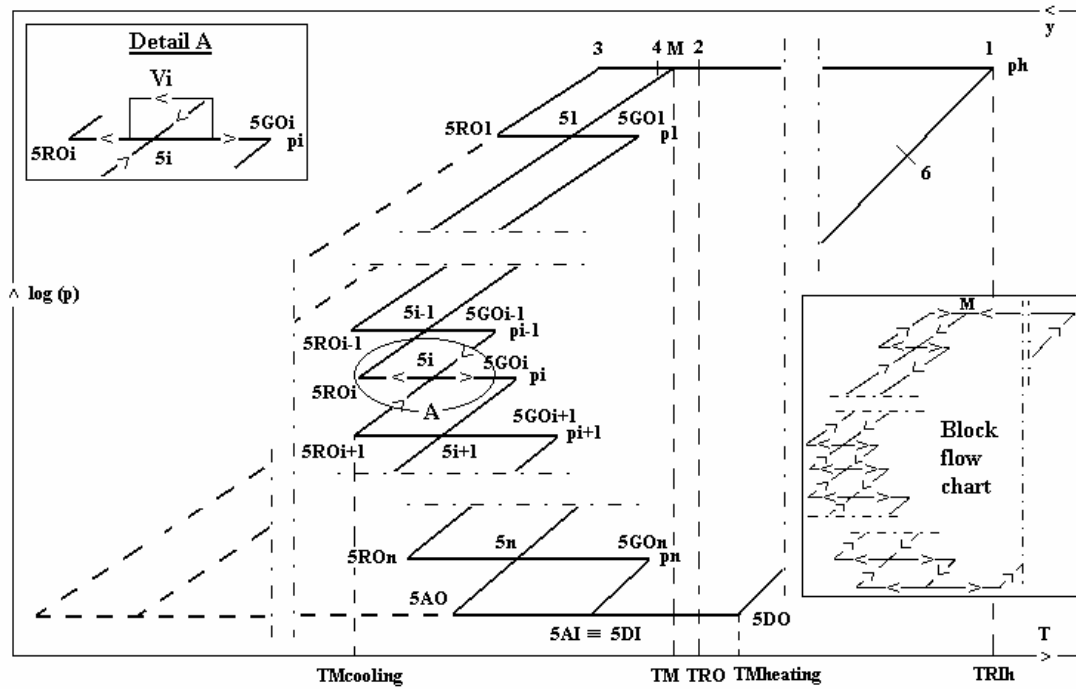


Figure 4: The truncated heating (heat transformer) coabsorbent cycle.

the $i=n-1$ generation process with that of the low pressure absorption process ($5_{AI} \equiv 5_{DI} - 5_{AO}$), and for the low pressure processes stage of the absorbent coming of the n generation process ($5_n - 5_{GO,n}$), point $5_{AI} \equiv 5_{DI}$. The truncated column also has processes between its successive stages, of reducing the pressure of the one mean concentration absorbent from p_h to p_1 , of the absorbent coming of the generation processes from p_{i-1} to p_i , $i=2, \dots, n$ and from p_n to p_1 , of increasing the absorbent pressure coming of the absorption process from p_1 to p_n , and of the resorption processes from p_i to p_{i-1} , $i=2, \dots, n$ and from p_1 to p_h , and of heat recovery between opposite processes of absorption-generation and of resorption-generation, of gas and absorbent to absorbent type. Functionally, the poor absorbent leaves the desorption process ($5_{AI} \equiv 5_{DI} - 5_{DO}$) at p_1 , it is pumped till p_h , it is preheated in a recovering way in the resorption process (1-2) at p_h from $T_{DO} \leq T_{M_{heating}}$ till a temperature close to T_{RI} , portion $5_{DO} - 6$, it is suffering the resorption process where it is rejecting the resorption heat to the medium which must be warmed up and that of preheating, reaching state parameters $T_{RO} \geq T_{M_{heating}}$ and $y_{RO} \leq y_M$, and the absorbent coming of the resorption process ($5_1 - 5_{RO,1}$)

of the truncation column first stage it is pumped from p_l till p_h , it is preheated in a recovering way, it is suffering the generation process (3-4) of the refrigerant vapour at p_h , it is mixed up at the high pressure with the absorbent coming of the high pressure resorption process and generates the one mean concentration absorbent which is covering the truncation column downstairs until the last stage and finally the stage at p_l , in the way described above, in order to close the cycle.

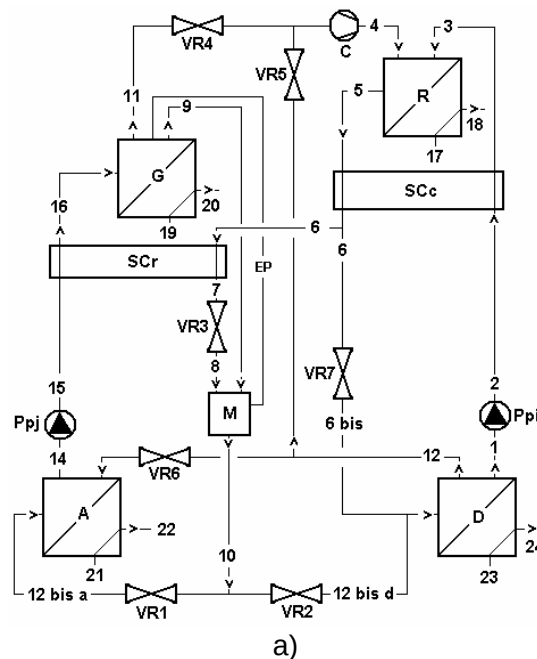
The truncated cooling cycle can be obtained formally, rotating by 180 degrees in the plane of the $\log p - 1/T$ diagram the nontruncated heating cycle plotted in Fig. 4, according to lemma 1 in (Staicovici 2006g), and has similar working as that described above. Truncated cycles can be obtained through overlapping nontruncated cycles in the way shown in (Staicovici 2006g) by lemma 2. Also, absorbent flow of a truncated cycle has certain global similarities to that of a nontruncated one, which is coming of from, according to lemma 3 (Staicovici 2006g). In other words, every fragmented geometric shape, that is the truncated cycle, of a nontruncated cooling and heating (heat transformer) coabsorbent cycle, considered as a whole (origin), can be subdivided in parts, each of which, namely the nontruncated cycle, being a reduced size of the whole from the geometric point of view, according to the lemma 2, and behaves as a whole from the absorbent flow point of view, according to the lemma 3. For these reasons, shortly outlined here, nontruncated cycles are cooling and heating fractals, as stated by the theorem given in (Staicovici 2006g) as well. This theory serves not only to understand truncated cycles construction and working, but lays down to the basis of their thermal computation as well.

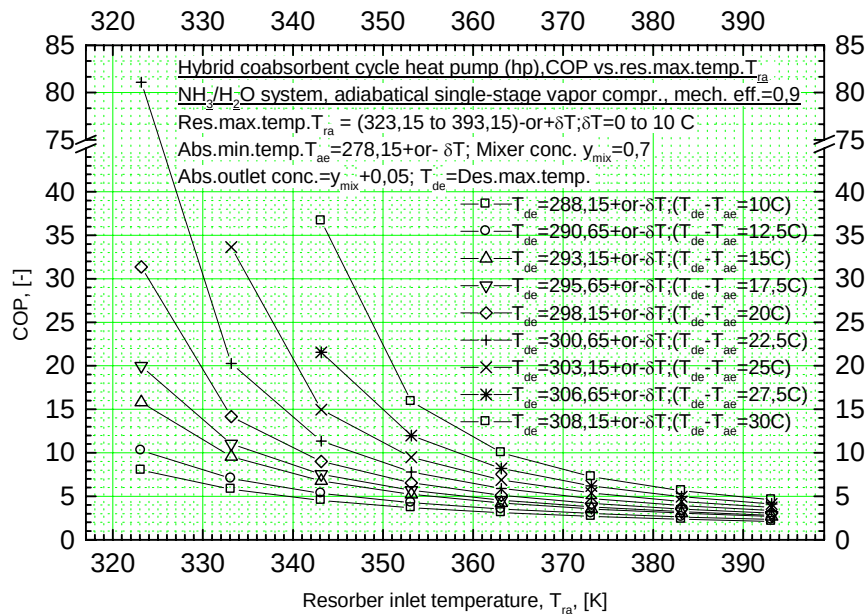
4 COABSORBENT HEAT PUMPS APPLICATIONS

During preparation of this work, different coabsorbent cooling and heat pumps cycles, operating with $\text{NH}_3/\text{H}_2\text{O}$ and $\text{H}_2\text{O}/\text{LiBr}$ working combinations, were thoroughly considered. Concerning cooling cycles, which, as a general characteristic, work more often than not in cogeneration of cooling and heating with high global COP, they were not included in this work, for the sake of clarity and homogeneousness. Further on, concerning just heat pumps, the heating nontruncated cycles bring not advantage over the classic heat transformers, and were not promoted in this study. Also, the cycles with small truncation columns ($n \leq 2$) have smaller temperature lifts, and were not interesting for this study, as well. Besides this, multi-effect coabsorbent cycles with double, triple and quadruple pressure stages working in the heat pump mode (Staicovici 2006g, 2007a, b, d, f) haven't brought satisfaction as well, because they have either small temperature lifts, or at higher temperatures lifts become too complex. Finally, hybrid absorption truncation and absorption high truncation column ($n \geq 3$) cycles, operating with $\text{NH}_3/\text{H}_2\text{O}$, were considered best options for our analysis. To this extent, applications presented here include heat producing model results with three coabsorbent heat pumps types, nontruncated hybrid, simple truncated hybrid and quadruple truncated, respectively. They will be analysed separately, next. Applications will emphasize two characteristic coabsorbent technology features. The first one has been already mentioned, referring to its capacity to process depleted heat/sink sources having a small temperature gap between them. The second one, given that condensing absorption technology is a particular case of the coabsorbent one, it is referring to its versatility, offering a broad technical solutions portfolio to a given application (known by its working parameters, sources availability, operation risk constrains, required environment protection degree etc.), ranging from highest efficiency + highest complexity to lowest efficiency + most reduced complexity of a heat pump.

Hybrid Nontruncated Heat Pump

This heat pump type, Fig. 5a, has been presented elsewhere (Staicovici 2006a, b, f, 2007a, b, d, e, f, h). We repeat it here for the sake of completeness, with additional comments. Constructively, it is the simplest, and it comes of a nontruncated coabsorbent cycle, which has non-isobar generation and resorption processes. Model heating results, obtained with the $\text{NH}_3/\text{H}_2\text{O}$ working combination, are given in Fig. 5b, where COP, defined as usually, by (useful effect power/mechanical power of compressor and pumps input), is plotted against resorber inlet temperature, T_{RI} , and takes into account compressor and pumps mechanical losses, $\eta_{mech,c} = 0.9$, and $\eta_{mech,p} = 0.6$, respectively. For COP assessment we considered single stage compression. In practice, if required, two or three stage compression with intermediary cooling must be applied, and known COP benefit. The curves in Fig. 5b have as parameter the temperature gap between desorber outlet and absorber outlet temperatures, respectively, $\Delta T = T_{DO} - T_{AO}$. The scheme in Fig. 5a has been chosen in order to enable heat pump operation as a hybrid coabsorbent unit (with four main component devices, A, G, R and D + compressor), when temperature gap is enough high, of 12-15 C, or more, and as an Osenbrück unit, with just the desorber D and resorber R + compressor, when this temperature gap is below 12-15 C. As compared to the Osenbrück unit, coabsorbent unit heating COP is 30-45 percent better, for $\Delta T = (15 - 20)^\circ\text{C}$, but increases to 200-300 percent and more for higher ΔT values and all T_{RI} temperatures. The heat source use effectiveness, defined by $E = (\text{useful effect power} / \text{depleted source heat input})$, is in favour of the Osenbrück unit, because coabsorbent unit has the generator G as an additional device which needs heat input. Usually, E values of the coabsorbent unit are about 0.45-0.47. Referring to what we have commented in the beginning of this paragraph, the Osenbrück unit has lowest efficiency, but most reduced complexity amongst absorption heat pumps. The results in Fig. 5b show COP increase with temperature gap parameter values for same resorber inlet temperature. When $\Delta T \geq 15^\circ\text{C}$, the coabsorbent heat pump COP exceeds by far the COP values of known mechanical vapour compression heat pumps. For this reason, this type of heat pump is recommended in heat production in the range of 50-100 C when appropriate heat/sink sources are available, as it has the potential of high primary energy savings, and it is still quite simple constructively.





b)

Figure 5: a)-coabsorbent hybrid heat pump: Resorber, Generator, Absorber, Desorber, SC=HE, Mixer, Pump, Compressor, VR-valve; b)-NH₃/H₂O coabsorbent hybrid heat pump COP (adiabatic single-stage vapour compression; mechanical efficiency = 0.9; resorber maximum temperature: $T_{ra} = (323.15 \text{ to } 393.15) - \text{or} + \Delta T$, where $\Delta T = 0 \text{ to } 10 \text{ }^{\circ}\text{C}$; absorber minimum temperature: $T_{ae} = 278.15 + \text{or} - \Delta T$; mixer concentration: $y_{mix} = 0.7$; absorber outlet concentration: $y_{mix} + 0.05$).

Simple Truncated Hybrid Heat Pump

Generally, and particularly in our case, for a given heat pump cycle temperature lift, the hybrid working mode is considered for design when high unit complexity is intentionally avoided, and/or when heat and sink sources availability is limited. The most simple truncated coabsorbent heat pump includes a single stage of isobar opposite intermediary generation and resorption processes between high and low pressure compartments, only (see Fig. 4). Here, the simple truncation is applied to a hybrid nontruncated cycle, having non-isobar generation and resorption processes, as previously mentioned, and a compressor must be added for operation. Schematically, the unit is given in Fig. 6. Functionally, absorbents coming of the high pressure resorber R and intermediary pressure generator G are mixing up in the mixer M, generating the one mean concentration absorbent, which subsequently is mixed up in the mixer M1 with the absorbent coming of the absorber A. The resulting absorbent is supplying further in pre-established quantities the isobar low pressure generator G1 and resorber R1. Absorbent exiting G1 is divided in pre-established amounts in order to supply the low pressure absorber A and desorber D. Finally, absorbers exiting D and R1 supply resorber R and generator G, respectively, in order to close the cycle. Vapour VG generated in G is introduced in R by means of the compressor C. Vapours VG1 and VD generated in G1 and D, respectively, are absorbed in R1 and A, respectively. Heat exchangers HE1, HE2 and HE3 hold for internal heat recovery devices. Absorbent flow is made possible by means of pumps P1, P2 and P3, and valves shown in Fig. 6, but not noted. Unit gap and lift temperatures are given by $\Delta T = TM_{hs} - TM_{ss}$ and $\Delta T_{lift} = TRI_{heating} - TM_{hs}$, respectively.

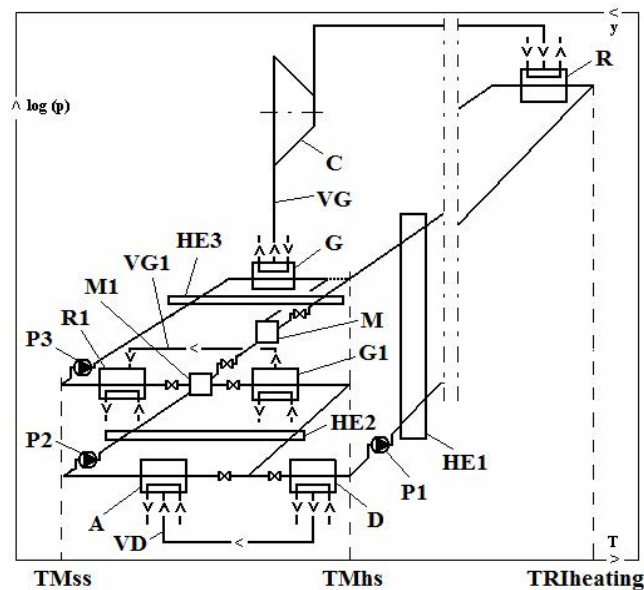


Figure 6: Simple truncated hybrid coabsorbent heat pump.

Bearing in mind that normally heat pumps upgrade natural low grade heat sources, or industrial ones, resulted ultimately as a rejected (depleted) heat of a high quality energy producing process, we have analysed here three cases with practical application respecting this remark. Model results are given in Fig. 7 for heat pump COP and heat (depleted) source use effectiveness, E , both defined to paragraph 4.1. In computation we selected the mixer concentration value equal to 0.3, in order to avoid operation at pressures exceeding 60 bar. For COP assessment we considered single stage compression. Again, in practice, if required, two or three stage compression with intermediary cooling must be applied. The first case considered the Underground Water-Ambient Air (UW-AA) heat/sink sources, with a temperature gap of $\Delta T = 12 - (-10) = 22$ C. Cold geographic regions may benefit of this efficient heat producing way for house domestic water or industrial steam preparation, when underground water keeps a quasiconstant temperature during whole year, of cca 14-15 C, and ambient air has preferably negative values. This case COP and E values are plotted in Fig. 7 with dotted line. The second case considered the Steam Rankine Cycle (SRC) based power plants and heat/sink sources coming of the condenser cooling at 40 C and of the cooling tower or another cooling source (e.g. cold winter ocean water or ambient air), with a temperature gap of $\Delta T = 35 - 10 = 25$ C. COP and E values are plotted in Fig. 7 with dashed line. This heat producing case may be applied in parallel to district heating by thermofication, improving power plant global efficiency, or especially for industrial steam production. Completely replacing thermofication by coabsorbent heat pumps heating is technically less feasible, given thermofication high capacity potential, and the benefit, expressed by the small electrical power savings (Staicovici 207b, d). The third case considered the Organic Rankine Cycle (ORC) based power plants and heat/sink sources coming of the condenser cooling at 90 C and the cooling tower or another cooling source (e.g. ambient air), with a temperature gap of $\Delta T = 85 - 45 = 40$ C. COP and E values are plotted in Fig. 7 with solid line. Efficient industrial high temperature applications can be expected in this last case. Concluding, concerning COP, normal mechanical vapour compression heat pumps cannot compete with it. Indeed, its COP values are 4-5 to more than one order of magnitude higher than those of mechanical vapour compression based heat pumps. For this reason it is recommended for high and very high temperature lifts achievement, as it results from Fig. 7 as well, where mechanical vapour compression heat pumps are not feasible, because their global efficiency,

including electrical power plant efficiency effect, is less than that of a fuel burning boiler heating efficiency. However, the price paid for this is its increased complexity, with 6 main devices, one compressor and three heat exchangers + three pumps, and the smaller heat source use effectiveness, which is 2-4 times less than that of an existing heat pump (see Fig. 7). This last drawback can be diminished, if mixer concentration increases. In this case, taking $y_M = 0.7$ for example, E increases by 30 percent, COP by 10 percent, but cycle operating pressure goes up to 90-95 bar for highest cycle temperature lifts.

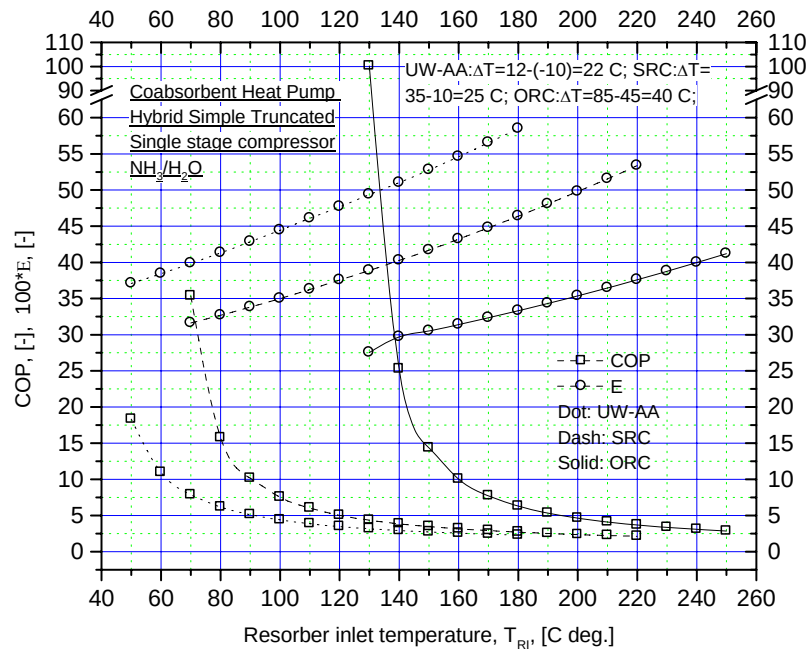


Figure 7: Model results of the simple truncated hybrid coabsorbent heat pump COP and heat (depleted) source use effectiveness E, plotted against resorber inlet temperature, for Underground Water- Ambien Air (UW-AA), SRC and ORC heat/sink sources.

Quadruple Truncated Fractal Heat Pump

Last application is considering a coabsorbent heat pump which is using just absorption, without mechanical vapour compression. The only electrical power is consumed by unit solution pumps. Here we don't give a special flow diagram, heat pump operation is described by cycles plotted in Fig. 4, where truncated column number of stages is considered $i=n=4$, that is four isobar opposite intermediary generation and resorption processes are interposed between the low and high pressures compartments. We modelled both cooling and heating fractals having this same truncation column. Results are given in Fig. 8 for cooling and heating (heat pump) COP, against desorber and resorber inlet temperatures, respectively, for same temperature gap $\Delta T_M = T_{M_{heating}} - T_{M_{cooling}} = 40^\circ\text{C}$ but differently placed in the low grade source region:

A: $\Delta T = 50 - 10$; B: $\Delta T = 55 - 15$; C: $\Delta T = 60 - 20$; D: $\Delta T = 65 - 25$; E: $\Delta T = 70 - 30$; F: $\Delta T = 75 - 35$; G: $\Delta T = 80 - 40$; H: $\Delta T = 85 - 45$. Heat (depleted) source use effectiveness, E, is 0.09951-0.09684 and 0.2535-0.2336 for cooling and heating units, respectively and all A-H points shown in Fig. 8. This means that about only 10 percent and 24.5 percent of the depleted sources become useful cooling and heating, respectively, at temperatures indicated in the abscissa, for each case A to

H. However, because these sources are usually free, the very low electrical power consumption, expressed by COP values makes this type of cooling and heating (heat pump) units still very interesting energetically and economically, despite their high complexity (12 main components, 6 heat exchangers, 5(6) solution pumps).

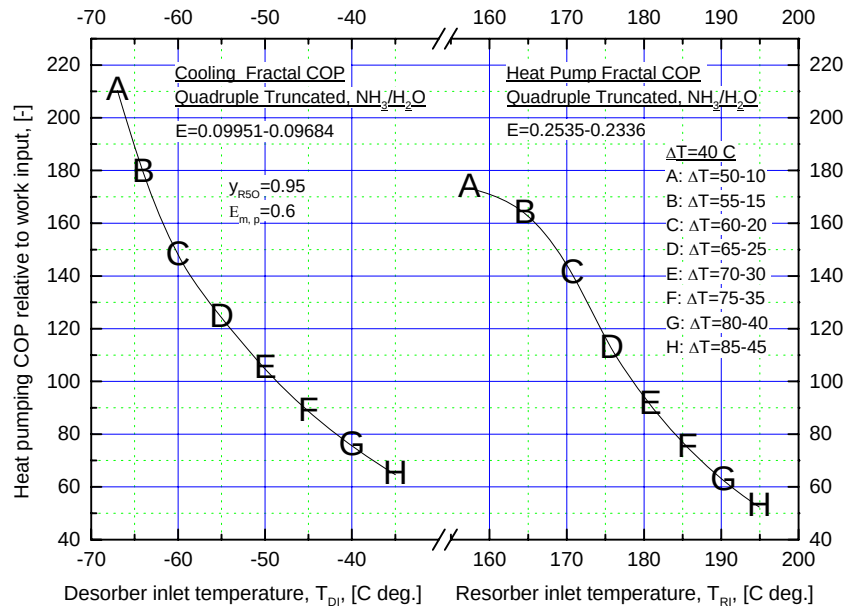


Figure 8: Cooling and heating quadruple truncated fractal model results, giving cooling and heating (heat pump) COP, against desorber and resorber inlet temperatures, respectively, for same temperature gap of 40 C but differently placed: A: $\Delta T = 50 - 10$; B: $\Delta T = 55 - 15$; C: $\Delta T = 60 - 20$; D: $\Delta T = 65 - 25$; E: $\Delta T = 70 - 30$; F: $\Delta T = 75 - 35$; G: $\Delta T = 80 - 40$; H: $\Delta T = 85 - 45$.

5 CONCLUSIONS

Coabsorbent heat pumps may be a feasible challenge for the classic mechanical vapour compression heat pumps, because of at least two coabsorbent technology main characteristics: a) it has the capacity to process depleted heat/sink sources having a small temperature gap between them, due to a particular technique, that of truncation, and b) given that condensing absorption technology is a particular case of the coabsorbent one, it is versatile, offering a broad technical solutions portfolio to a given application, which range from highest efficiency + highest complexity to lowest efficiency + most reduced complexity of a heat pump. Out of all coabsorbent heat pump cycles analysed, three applications have been promoted for this work: a) nontruncated hybrid, b) simple truncated hybrid and c) quadruple truncated, respectively. Their COP depend on temperature lift and gap and generally increase with truncation column high increase from several times to one and two magnitude orders as compared to mechanical vapour compression existing heat pumps COP. Units complexity increases in the same way as COP. Heat (depleted) source use effectiveness, E, is smaller than that of a mechanical vapour compression existing heat pump, ranging from about 24 to 47 percent for c) to a) applications, in absorption working mode, and from about 30 to about 70 percent in the hybrid working mode.

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