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# A close-up thermodynamic analysis of a hypothetical sorption heat transformers assembly used to feed an ORC cycle from a low exergy heat source: Part I

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## ABSTRACT

The present study looks at the feasibility of a small scale power plant configuration with nominal electric output of 100 kWe fed from low potential thermal heat source, especially from flat thermal solar collectors operating in the range of 343 to 373K. The power plant uses multiple cascaded sorption heat transformers (AbHT and AdHT) coupled with an Organic Rankine Cycle (ORC). The study revealed for the proposed system, that it would be possible to generate electrical power over the entire range of the proposed temperature within the considered hypotheses. The work was split in two parts. The first part presents a complete state of the art of the thermal cascade assembly of the upgrade of low-potential heat energy technology and small-power generation system. Moreover, the power plant configuration with cascaded sorption heat transformers and small-power generation device is described, followed by the presentation of the developed mathematical model with all equations and fundamental laws specific to each compartment of the proposed power plant. The main results, discussions and conclusions are presented separately in a second part of this work.

## 1 Introduction

In recent years, considerable attention has been given to reduce the use of fossil and thermonuclear energy resources to satisfy the demand for energy and electricity in the world. The exploitation of different alternative thermal energy sources at low potential and low-grade waste heat to generate power is the promising and interesting way to mitigate energy problems and protect the environment. Despite its worldwide abundance, it stays however, without significant interest at such temperature level and not attractive for applications requiring high thermal potential [1]. In fact, the wasted heat still

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contains large amount of energy and exergy that can be reused to deliver heating, cooling or work through suitable heat recovery technologies [2]. In this context, the same authors reported that with rational heat recovery and effective heat transformer technologies, these waste heat could play an important role on energy saving and sustainable development [2].

In the same context, it is reported that a heat transformer technology is attractive because it use waste heat from industrial processes and renewable energy sources such as solar and geothermal [3]. The latter is a device, which can deliver heat at a higher temperature than the temperature of the fluid by which it is fed [3]. Heat transformer technologies are the combination of several physicochemical processes and can be divided into two categories. The first one consists of gas absorption by a liquid solution, while the second category implies the gas adsorption by a solid driven by the van der Waals force among the molecules, and generally happens on the surface of adsorbents with a heat release called isosteric heat of adsorption. Both phenomena occur under certain physical conditions (pressure and temperature) and are reversible.

An absorption heat transformer (AbHT) is a closed cycle thermodynamic system[4], operating in the reverse way of absorption refrigeration, can economically upgrade low grade waste heat into useful heat at a higher temperature with only little electrical energy input for pumping work [3,5]. Up to 50% of the waste heat may typically be recovered [6-9], with a temperature lift of about 50 °C in principle [5]. The integration of AbHTs with different thermodynamic cycles plays an important role in recovering the heat rejected by them or even increasing the energy efficiency of the whole system [3].

Nowadays, adsorption heat transformers (AdHTs) are considered as an alternative to compression and absorption machines [10]. Compared to conventional compression systems, the AdHT can produce cooling/heating effect utilizing waste or renewable heat as driving energy source that can lead to an essential reduction of primary energy consumption [11]. In other hand, the adsorption ones are characterized by the absence of corrosion, crystallization and vibration problems compared to absorption systems. They can operate at a wide range of heat source temperatures with no moving parts [12]. The latter units are already used for converting low temperature heat from renewable sources [13-15]. These new heat sources have significantly lower temperature than that achieved by burning fossil fuels which opens a niche for applying adsorption technologies for heat transformation [16]. It is for these reasons that heat transformers have been termed 'future technology which will be important for energy utilization in the 21st century' by the International Energy Agency [8].

In this context, a new cascade concept consisting of sorption heat transformers fed from low exergy source is proposed and analysed. A very extensive bibliographical research was carried out by Cheikh et al. and can be consulted in the reference [17]. However, the concept of thermal cascade assembly of several heat transformers of the same or different types is reported in [18-19]. According to this, two different terminologies have been reported in the literature. The first term is multi-stage (double stage and triple stage) which indicates the number of times that the heat is introduced at the hot source. The second one is the term multi-effect (double effect, triple effect...) which indicates the number of pressure levels. According to the first terminology, heat transformers with two, three or more stages can be coupled in three different ways. The first is to connect the absorber of the first stage to the evaporator of the second stage and the absorber of the second stage to the evaporator of the third stage and so on. This configuration provides maximum temperature rise over other configurations [21]. The second way is to couple the absorber of the first stage with the generator of the second stage, which will achieve a relatively high COP but relatively lower temperatures. The third way is to deal out the heat delivered by the absorber of the first stage to the generator and the evaporator of the second stage. This combination allows reaching a relatively high maximum temperature but with a lower performance. This last way is interesting solely in case of low heat source potential and where the high temperature requirements are too great[22].

Another version of dual stage absorption heat transformer has recently been proposed by Yang et al. [23]. It provides greater system efficiency at the industrial scale and consists of two absorption heat transformers. The first one operates with the LiBr / H<sub>2</sub>O couple, as for the second with the NH<sub>3</sub> / H<sub>2</sub>O couple. In addition to that, the source of low potential is divided into two heat flows. A high temperature flow intended to feed the LiBr / H<sub>2</sub>O cycle and a low temperature one used to feed the NH<sub>3</sub> / H<sub>2</sub>O cycle. According to the second terminology (multi-effect), several studies have been carried out for the study and the development of heat transformers with double effect [24] and with triple effect [25].

The implementation of thermal cascade of several thermo chemical cycles according to the two terminologies (multi-stage or multi-effects cycles) could raise the temperature level of the driving source and thus improve the energy performance. An identical concept to the thermal cascade proposed in this work has been reported by Ziegler et al. [26] for heat pumps called a multi-effect heat pump. The first feasibility study was carried out in 1982 by the Alefeld team. It has been followed by a development of 7 kW of cooling capacity prototype, which combines a water-zeolite adsorption and

LiBr-water absorption cycles.

A similar concept has been proposed by Stitou et al. [27]. The authors suggest different possible configurations of the trithermal machine by coupling the liquid-gas absorption process with a reversible chemical reaction process to obtain cooling at 5 °C (air conditioning)[17].

The second target is the feasibility of a concept to generate electricity from low temperature heat by using the thermal energy upgraded at the outlet of the cascade heat transformers system and converted into electrical energy. The powers obtained are relatively small compared to conventional conversion cycles. It is reported that a generally speaking, of the concept of small-scale power generation systems can be used to denote small-scale systems with an electric capacity with electrical power less than 100 kW[28]. In fact, the targeted power segment is the one that covers domestic needs or commercial buildings such as electric generation in remote houses, commercial buildings (schools and office building blocks) and domestic buildings of single or multifamily dwelling houses. In the field of small power, the conversion of thermal energy into mechanical (electrical) energy does not seem to have the same development as for large powers.

There has been a great number of promising and interesting technologies emerging to realize the recovery of low temperature waste heat into electricity of small-power capacity, like power generation via Stirling [29-33], Joule [34] and Rankine cycles [35-41]. In terms of small-power production capacity, Stirling engines are a promising solution for installations with nominal electric output between 10 and 150 KWe [42]. Moss et al. [34] has estimated that (1-10KW) of electrical power can be recovered by Joule cycle.

A small thermodynamic solar power plant by Schneider Electric expects autonomous production of electricity by a Stirling engine in isolated sites of developing countries. A typical consumption of 200 kWh of electrical energy per day for 100 households was identified in the project with a power peak of 10 kW. It will have to satisfy well defined specifications in terms of needs to be satisfied and all-round constraints to be respected [43]. However, the Stirling engine is one of the few external heat transfer engines marketed for small power applications.

In literature, small scale Brayton (Joule) cycles are seldom but can be found. In the 1960's, small scale Brayton for automotive applications was proposed and studied by Bjerkle and Warren [44]. A 5 kW Brayton cycle, using a free piston, was studied for automotive applications by Bjerkle [45]. These engines, however, seem difficult to use at heat source temperatures below 300°C and 400°C [46]. Furthermore, no concrete applications of this type of engine were existed [47].

In terms of small scale power production capacity, the knowledge about Rankine cycle a system presenting a small power is still limited nowadays [48]. In this context, it is worth to mention some pilot projects that show the feasibility of small scale Rankine cycles. A small solar power station project has been realized as part of the Solar Power System (SPS) research program. It is a hybrid system based on hermetic volumetric expanders-generators of "scroll" type operating with superposed Rankine cycles. In addition to heat rejections from a cogeneration engine, the system is supplied with heat from solar concentrating collectors with extra-flat reflective slides. Analysis of the project demonstrates clearly the feasibility of the concept, its correct operation over a wide range of power range from a few Watts to a few kWe and finally its adaptation to developing countries [49]. In the same context, a 5 kW Rankine cycle for biomass application was developed by Boudellal [50]. The same nominal power from solar Rankine cycle, using a mono piston expander, was tested by Bouvier[51].

According [52], if it is waste heat that has to be converted into mechanical work that is yielding most benefit, the Rankine cycle is most suitable. At lower temperatures, smaller than 400 °C, not the known Rankine-process with water is the most favourable one that is realized in steam power plants, but a similar process with an organic working medium (refrigerant). This ORC shows a higher efficiency at the same low maximum temperature than the Rankine process with water. This organic fluid evaporates at medium and low temperatures below 300 °C. Many researchers have studied the feasibility of micro scale Organic Rankine cycles. In this context, a 5 kWe Organic Rankine cycle using single screw machine with a measured isentropic efficiency of 59% was presented by Wang et al.[53]. A 20 kWe ORC system for low grade heat source application based on a screw expander is presented in Ref [54]. A nother 20 kWe ORC system was tested and evaluated by Chaiyat [55]. This electricity generation capacity of the integrated system is obtained from low temperature heat (70 to 90°C) upgraded by using an absorption heat transformer (AHT) and supplied to the Organic Rankine Cycle (ORC). In the same context, a 22 kWe Organic Rankine cycle system with single screw expander was built and tested by He et al. [56]. A 50 KWe Organic Rankine cycle using screw expander was tested and compared with the most relevant experimental data available in the literature [57].

A comparison between ORC and other waste heat recovery technologies, such as Stirling engine, Brayton and Kalina Cycle is provided in literature and it highlights that ORC is the best performing technology for heat recovery and power generation using heat sources at temperature ranges between 200 °C to 400°C.[58]. As mentioned in Kongtragool and Wongwiset [59] the key to success of the Stirling engine for low temperature applications are, new materials and good heat transfer to the working fluid. However, the Brayton cycle which is a suitable high temperature waste heat recovery technology [60]. Moreover Cheikh et al. [17] reported a novel cascade concept with electricity generation capacity of 10 kWe, consisting of sorption heat transformers of the up-grade of low-potential natural sources. This upgraded heat will subsequently be converted into mechanical (electrical) energy by conversion devices operating according to Brayton (Joule) cycle and the subcritical and supercritical ORC. The result of this concept clearly shows the feasibility of the second power plant assembly scheme with the organic Rankine conversion machine, which can satisfy the specified electrical power needs of 10 kWe from thermal solar collectors of temperature equal to 358 K for the subcritical ORC and 363 K for the supercritical one. The overall conversion efficiency of the integrated system could be increased to be around 18%.

According to Pippo [61], under identical conditions of ambient temperature and cooling systems, the Kalina cycle will obtain approximately 3 % greater efficiency than the organic Rankine cycle. This performance shows the potential of the Kalina cycle for applications in low-grade heat sources, but the aggressive water-ammonia mixture corroded the turbine, which needed some maintenance after 15 months in operation. Another drawback of the Kalina cycle is the fact that a high vapor fraction is needed in the boiler, resulting in lower overall heat transfer coefficients and a larger heat exchange area [62]. Moreover, it should be mentioned that Pippo concluded that broad claims of 15 – 50 % more power output for the same heat input for Kalina cycles relative to binary ORC are not being achieved for plants in operation so far.

According to Crook [63] and Oluleye et al. [64] electricity generation from low temperature heat flows is possible with several technologies, but the ORC can be considered the most mature technology at the moment. This is supported by the number of manufacturers and running installations [65-66]. Furthermore, their benefits are well proven, including high efficiency and a one stage turbine (as the reason of low enthalpy drops at the turbine), low maintenance costs, favourable operating pressures and autonomous operation [66], less complex [61]. Moreover, it should be mentioned that ORC systems find application not only in industrial waste heat recovery plants, but also for electricity generation from biomass, geothermal or solar sources [67,68]. As mentioned in [69], among all thermodynamic cycles for low-grade heat power conversion, the ORC is so far the commercially most developed one. It was reported by [67], based on an extensive database (collected in spring 2016) containing approximately 95% of ORC systems commissioned and sold worldwide, about 35% of the ORCs operate on excess heat, representing 12% of installed capacity.

For these reasons, this technology is currently recognized as the most promising for exploitation of low/medium temperature heat sources [58, 70].

The first section of this paper presents a complete state of the art of the thermal cascade assembly of the upgrade of low-potential heat energy technology and small-power generation system. At the end of that section, trends are shown in order to highlight the advantages and drawbacks of each system. In the second section, the power plant configuration with cascaded sorption heat transformers and small-power generation device will be described, followed by the presentation of the developed mathematical model with all equations and fundamental laws specific to each compartment of the proposed power plant. The main results, discussions and conclusions are presented separately in a second part of this work.

## 2 Description of the power plant

The proposed configuration schematic diagram of the proposed power plant is shown in figure 1. It consists of two subsystems. The first one is used to upgrade heat coming from low-potential heat source. It is composed of two single-effect heat transformers with different operating cycles. The first is an absorption heat transformer using a refrigerant-absorbent solution rather than pure refrigerant as the working fluid (generally water and LiBr – H<sub>2</sub>O solution respectively) When the refrigerant-absorbent solution passing through the solution pump is referred to as a strong solution, being relatively rich in LiBr. The solution returning from the absorber to the generator contains only a little LiBr compared to the solution being pumped from the generator to the absorber and is therefore referred to as weak solution. The lithium bromide and water (LiBr – H<sub>2</sub>O) working fluid combination is the only working fluid combination in current commercial use [71] and the most commonly used working solutions [2]. This is due to a number of advantages reported by many references works [72-76]. Although water–lithium bromide solution was well suited for use in AbHTs, it still had some disadvantages, namely corrosion, high viscosity, limited solubility and a practical upper temperature limit [3].

In this context, much effort has been focussed upon finding alternatives which have similar (or improved) thermodynamic attributes to LiBr – H<sub>2</sub>O while eliminating some of its negative features. From the bibliographic review related to simple AbHT operating with alternative mixtures (Ammonia–water, TFE refrigerant fluids, Water–Carrol mixture, Additives, Salt and water combinations) reported by [4, 5 and 77], it is clear that LiBr – H<sub>2</sub>O is currently the working fluid standard which is used in almost all analyses and designs and against which all new tested working fluid pairs are compared. Due to the reasons discussed above, this working fluid combination is the only working fluid combination in current commercial use [71].

The operating sequence of the AbHT shown in Fig. 1 is as follows: the working mixture is separated in the generator 1 heated from low potential heat source (Solar flat collector). Strong solution (stream 1) and Refrigerant vapor (stream 7) are obtained in the generator 1. The strong solution (stream 1) is pumped by the solution pump to reach a higher pressure (stream 2). It flows to the absorber (stream 2) through solution heat exchangers (stream 3) where it is preheated. Refrigerant vapor (water) (stream 7) is condensed subsequently in the condenser 1 (stream 8), releasing heat  $Q_{C1}$  to the lower heat sink at  $T_{C1}$ . In the same time, the condensed water is pumped by the first water circulation pump (stream 9) to the evaporator 1 receiving the heat  $Q_{E1}$  from a medium temperature source ( $T_{SC}$ ) where it is completely evaporated before being introduced to the absorber (stream 10). The steam produced (water vapor) is then absorbed by the strong solution (stream 1) in the absorber coming from the generator 1 accompanied by heat release of absorption termed the upgraded heat ( $Q_{U1}$ ) at high temperature  $T_{U1}$ . The weak liquid solution (stream 4) leaving the absorber flows to the generator 1 through the solution heat exchanger (stream 5) and the expansion valve (stream 6). The absorber is then placed in communication with a second heat transformer operating with physical adsorption cycle using the zeolite 13X / water pair. The latter is considered among practically interesting working pairs [78-80]. This, the potential of classical zeolites for heat transformation looks restricted as they have either too high or too low affinity to water [81]. It is reported that, this material could be interesting for double effect AdHT cycles rather than for single effect ones [81]. The choice of zeolite as an adsorbent is subjected to another decisive criterion besides to the hot source temperature influence and zeolite thermo physical characteristics. It is the economic criterion which implies that the price of zeolites is very low on the worldwide market. Lastly, the zeolite shows a good stability with water and other refrigerants such as methanol.

The adsorption heat transformer cycle includes adsorption and desorption phases. During the first phase of cycle, the working pair (zeolite 13X/ water) in the generator 2 receives a medium temperature heat flux supplied by the absorber of the absorption heat transformer (stream 11). Once, the limit desorption temperature is reached, the desorption stops and the steam is conducted to the condenser 2 (stream 12) where it is condensed by rejecting a heat  $Q_{C2}$  (stream 13) at low temperature  $T_{C2}$  which corresponds to a low saturation pressure of the adsorbate  $P_{C2}$ . In the second phase of the cycle, which is carried out at high pressure, the adsorbate (water) is placed in communication with the evaporator 2 (stream 14). The evaporation is carried out by a heat input at average temperature ( $T_{U1}$ ) and at high pressure  $P_{E2}$ . The vapor from the evaporator 2 (stream 15) is admitted into the adsorber, where it is adsorbed by the bed of zeolite 13X (stream 16) accompanied with heat release  $Q_{U2}$  at high temperature  $T_{U2}$ .

This first subsystem is connected to the second one, which is designed for heat conversion and power generation from the heat upgraded by the first subsystem, when the adsorber of the last heat transformer ( $Q_{U2}$ ) feed the boiler of organic Rankine cycle of the converting device, which is a standard (subcritical) cycle without and with heat regenerator. This system performs a closed thermodynamic cycle and consists of four basic components. These are a pump, a boiler, an expansion turbine and a condenser. As a consequence, the load on the condenser increases and installing a regenerator could make even better use of the energy from the expanded vapor, preheating the pump fluid that will enter the evaporator [65, 70 and 82]. As mentioned in [83], the presence of regenerator increases the ORC thermal efficiency, because it allows to reduce the amount of heat needed from the hot source to vaporize the fluid. Moreover, the use of regenerator reduces the heat released to the environment through the condenser. In addition a study of Orlandini [83] reported that if there is no limitation on the hot source temperature cooling, the net power output will not increase by adding a regenerator, and the net work output will approximately be the same. Furthermore, generally the increased pressure drop and the extra cost of the regenerator are negligible [84]. As shown in fig 1, the organic fluid runs successively through the four modules making up the cycle. A circulation pump carries the fluid at high pressure (stream 18). The fluid is then returned to the boiler where it is heated and vaporized (stream 19) by means of heat coming from the adsorber of the adsorption heat transformer ( $T_{U2}$ ). The steam then enters the turbine where it is expanded producing mechanical work (stream 20). The exhaust fluid from the turbine undergoes a total condensation at constant pressure in the condenser (stream 17) where it is pumped back to the steam heat exchanger, which allows the cycle to begin again. In the proposed machine, several fluids can be used such as

pure working fluids and mixture of fluids, as organic and inorganic fluids (Pure Fluids and Mixtures (Azeotrope, Quasi azeotropic and Non-Azeotrope). The ORC working fluids can be categorized under various main classes, according to [70,85]: Hydrocarbons (HCs); Perfluorocarbons (PFCs); Siloxanes (MM/MDM/MD2M); Chloro-Fluoro-Carbons (CFCs); Hydro-Fluoro-Carbons (HFCs); Partially fluoro-substituted straight chain hydrocarbons (HCFCs); Ethers and fluorinated ether Alcohols; Inorganics. The selected working fluid nature has a noticeable effect on machine operation, performance, size and design. According to [86], the organic fluid is chosen according to a compromise taking into account stability criteria, the environmental impact (ODP, GWP) and the fluid curve determined by the saturation curve shape in T-s diagram. Thus, there are three types of fluids: wet, dry and isentropic. Dry or isentropic fluids are recommended for ORC application [38, 87 and 88]. They can be overheated and then expanded isentropically in the turbine, which excludes any risk of erosion because they do not condensate after the fluid goes through the turbine. These fluids are used with moderate operating pressures (subject to the machine design). In addition, they are safe, available and have good thermodynamic performance and low cost. There are other selection criteria directly related to operational conditions that will need to be considered. These are temperature level, hot source type and working fluid critical temperature. In this context, several studies propose to choose the optimal working fluid according to the heat source temperature level [89- 93]. In the work of Tchanche et al. [66], the working fluids were classified in three groups according to the critical temperature: high temperature fluids ( $T_{cr} > 250 \text{ }^\circ\text{C}$ ), medium temperature fluids ( $150 \text{ }^\circ\text{C} < T_{cr} < 250 \text{ }^\circ\text{C}$ ) and low temperature fluids ( $T_{cr} < 150 \text{ }^\circ\text{C}$ ).

Another study carried out in the same context by Fernandez et al. [94] and dedicated to the optimal choice of siloxanes (polymethylsiloxanes or silicone oils) reveals that among many siloxanes tested MM (Hexamethyldisiloxane,  $\text{C}_6\text{H}_{18}\text{OSi}_2$ ) and MDM (Octamethyltrisiloxane,  $\text{C}_8\text{H}_{24}\text{O}_2\text{Si}_3$ ) showed better performances. Several similar works like those of Quoilin et al. [95] are devoted to the study of a new generation of working fluid for ORC such as Solkatherm SES36 (azeotropic binary mixture composed of 65% of a hydrofluorocarbon R365mfc and 35% of a perfluoropolyether), the n-hexane, Novec 649, Novec 774 and Novec 7000 or R1233zd which are considered potential fluids because of their physicochemical properties and low impact on the environment. For the proposed cycle, hexane is selected as working fluids because of their appropriate critical temperature and pressure, which is adopted for hot source temperature level with their positive slope ( $dT/ds$ ) of saturated vapor curve in a T-s diagram, which classified under “dry” fluids category. This characteristic allows dry saturated vapour at the inlet of the turbine and ensures its dry expansion inside the turbine. Therefore, there is no need for superheating the fluid entering the turbine. According to [96], this category ensures the absence of condensation, which reduces the risk of corrosion on the turbine blades and extends its lifetime to 30 years instead of 15 years to 20 years for steam turbines. In addition, hexane is also environment friendly working fluid with a zero ozone depletion potential (ODP) and relatively low global warming potential (GWP) values.



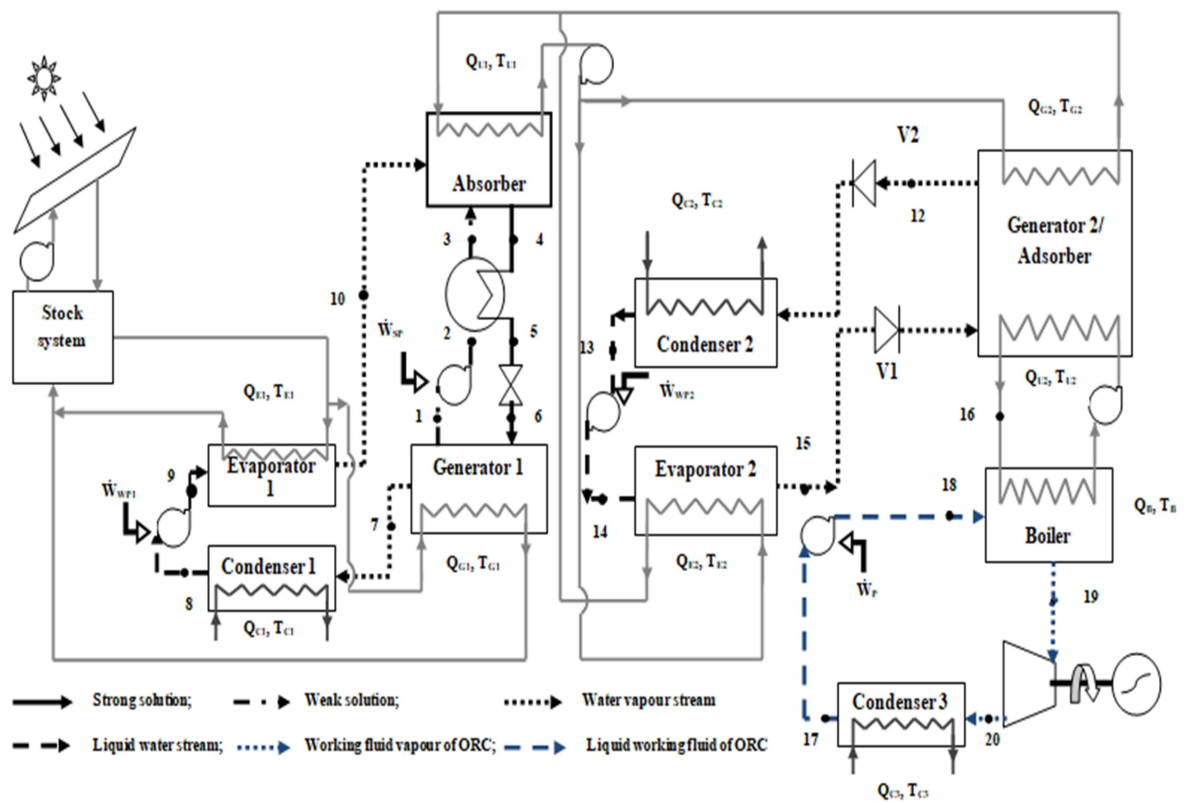


Fig. 1 – Schematic diagram of two cascaded heat transformers coupled to an Organic Rankine Cycle.

### 3 Mathematical modeling

Fig. 2 shows steps for calculating the thermal performance of the proposed power plant shown in Fig. 1. In the simplified model, each component of the system (absorption and adsorption heat transformers and the organic Rankine machine) is treated as a control volume and the principle of mass conservation and the first and second laws of thermodynamics are applied to the component firstly. On the other hand, thermal performance could be predicted under various operating conditions. For simplification purposes, hypotheses on the power plant component operation are adopted in this study. In this context, the following hypotheses are employed for the absorption heat transformer cycle:

- The system is in steady state
- The outlet solution temperature at the generator1  $T_{G1}$  is equal to that of the evaporator1  $T_{E1}$
- The refrigerants, water, at the outlets of condenser1 and evaporator1 were at saturated state
- The weak solution leaving the absorber is a saturated liquid at the temperature  $T_4$  and the absorber concentration ( $X_w$ ) which is a function of the absorber outlet high pressure and temperature  $X_w = f(T_4, P_{E1})$ . Similarly, the strong solution concentration ( $X_s$ ) leaving the generator is a function of the low pressure and the generator temperature  $X_s = f(T_1, P_{C1})$
- The strong solution expanding process in the desorber is isenthalpic
- The recirculation flow ratio ( $FR$ ) is defined as :  $F = \frac{\text{mass flow rate of solution leaving generator}}{\text{mass flow rate of vapour leaving generator}} = \frac{\dot{m}_1}{\dot{m}_7}$
- Solution heat exchanger is assumed to have an efficiency of 80%
- The mass flow rate of refrigerant,  $\dot{m}_R$ , is the essential variable for the full set of calculation on power plant performance, which was assumed to be 0.1 kg/s in this work
- The input parameters are considered temperatures of the generator1  $T_{G1}$ , condenser 1  $T_{C1}$ , evaporator1  $T_{E1}$  and absorber  $T_{U1}$ . However, access to these temperatures is impossible since heat transfer is not considered in this study. Nevertheless, the temperature at each component outlet can be determined according to inlet and temperatures driving force for heat transfer ( $\Delta T_{AbHT}$ ), termed the pinch temperatures at heat exchangers (the generator, the evaporator, the absorber and the condenser) as reported in [97 and 98] by considering:



- both the outlet solution temperature at the generator1  $T_{G1}$  and the evaporator1  $T_{E1}$  are lower than 5K compared to the hot or low potential source temperature ( $T_{SC}$ ),  $T_{G1} = T_{SC} - \Delta T_{AbHT}$
- the absorber outlet temperature  $T_{11}$  is 5K higher than the upgraded heat or high potential temperature  $T_{U1}$ ,  $T_{11} = T_{U1} + \Delta T_{AbHT}$
- the condensation temperature  $T_{C1}$  is 5K higher than the surrounding temperature  $T_0$ ,  $T_{C1} = T_0 + \Delta T_{AbHT}$
- All the plotted properties of LiBr–H<sub>2</sub>O solution in vapour–liquid equilibrium states were calculated by the formulas provided in Patek and Klomfar [99] and Kaita's work [100]

For the adsorption heat transformer cycle, the following assumptions are adopted:

- The adsorption heat transformer operation was considered at stationary mode to calculate the energy balance at each cycle stage and in each converter component.
- Heat is supplied to the system at temperature  $T_{E2}$  for evaporation and at  $T_{G2}$ ,  $T_{G2} = T_{E2}$ , for the desorption process, and extracted at the highest temperature of the cycle,  $T_{U2}$  from the adsorber.
- The upgraded temperature in the adsorber ( $T_{U2}$ ) can be determined by its maximal adsorbate concentration  $x_{max}$  and the high pressure ( $P_{E2}$ ); by the same way, the outlet minimal adsorbate concentration  $x_{min}$  from the generator 2 can be determined by its temperature ( $T_{G2}$ ) and the low pressure ( $P_{C2}$ ).
- The amount of dry adsorbent was assumed to be M kg.
- By the same way of absorption cycle, both the outlet refrigerants temperature at the generator  $T_{G2}$  and the evaporator  $T_{E2}$  are lower than 10K than the supply temperature ( $T_{U1}$ ),  $T_{G2} = T_{U1} - \Delta T_{AdHT}$ .
- The adsorber outlet temperature  $T_{15}$  is 10 K higher than the upgraded heat temperature of high potential  $T_{U2}$ ,  $T_{15} = T_{U2} + \Delta T_{AdHT}$ .
- The condensation temperature  $T_{C2}$  is 10K higher than the surrounding temperature  $T_0$ .
- All the equation properties of zeolite 13X/water pair were calculated by the formulas given by Cacciola and Restuccia [101], Liu and Leong [102] and San and Lin [103].
- The ambient temperature  $T_0$  was assumed to be 298.15 K in this work.

To simplify the analysis of the ORC (subcritical) the following assumptions are employed:

- The system is under steady state condition .
- No pressure drops in connecting pipes and heat exchanger.
- Heat losses in each component are ignored.
- Working fluid entering the expander is saturated vapor.
- The enthalpies and other thermodynamic properties were calculated from the measured pressure and temperature data by using REFPROP ver. 9 developed by NIST [104].
- The condensing temperature of the working fluid in condenser is fixed at  $T_{C3} = 45^\circ\text{C}$ , to transfer heat to the district heating loop for space heating.
- The pinch point temperature differences for both evaporator and condenser are  $\Delta T_{ORC} = 5\text{K}$ .
- The superheating of working fluid at evaporator exit is set at  $\Delta T_{sh} = 2^\circ\text{C}$  for practical consideration [105].
- The turbine and pump isentropic efficiencies are  $\eta_t = 90\%$  and  $\eta_p = 90\%$ , respectively .
- The electrical generator efficiency ( $\eta_g$ ) is assumed as 96% .

Subscripts from 1 to 20 in expressions are corresponding to locations with same numbers in figure 1. Exception is done for the state point 11 and 16', which are not reported in figure 1. They represent the start state of desorption temperature inside the second generator 2 during desorption process and the highest reachable temperature inside the second generator 2 during adsorption process, respectively.

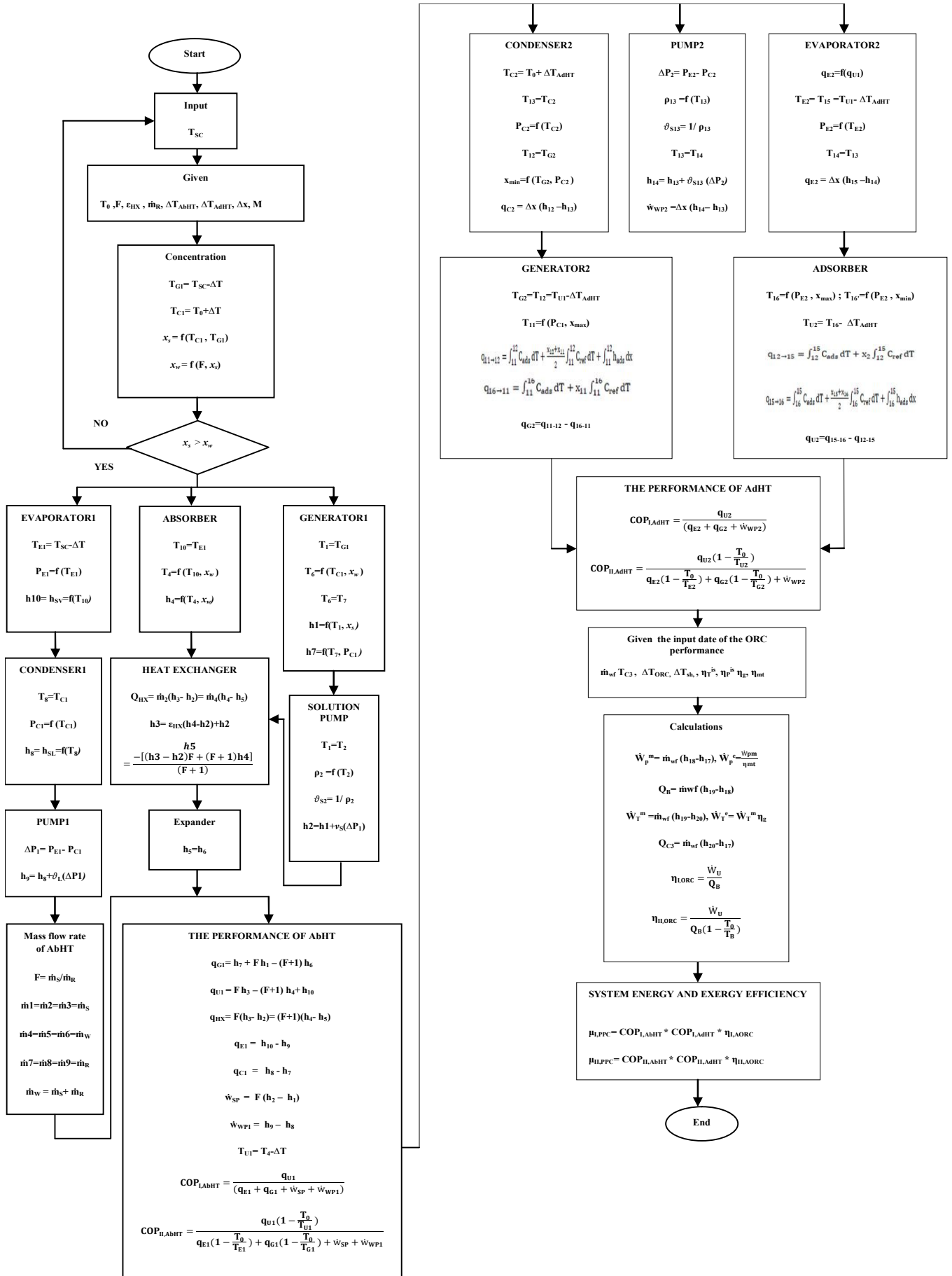


Fig. 2 – The simplified model of the proposed power plant

## 4 Conclusions

The proposed cycle, which is a combination of a new cascade concept consisting of sorption heat transformers fed from low potential natural sources of solar origin and ORC, can beneficially satisfy small scale power electric generation, which are particularly suitable for applications in domestic needs and commercial buildings, such as hospitals, schools, industrial premises. This concept provides an advantageous way of utilizing low energy (natural or industrial) for the increase of the generated electricity with heat recovery and avoids the need for additional fuels, by integration in existing plants. A thermodynamic model was developed by applying the energy analyses for each system component. Furthermore, the model was verified through comparison between results obtained from the current model and those available in the literature for similar operating conditions. It could be noted that based on the abovementioned literature review, there is no sufficient knowledge for the AdHT cycle in literatures.

The second part of this work will be consecrated to the analysis and discussions of the mathematical modeling results. Important recommendations are drawn for such power plants.

**Nomenclature**

COP	coefficient of performance
F	recirculation flow ratio
GWP	Global-Warming Potential
GTL	gross temperature lift [K]
h	enthalpy [kJ kg <sup>-1</sup> ]
m	mass flux rate [kg S <sup>-1</sup> ]
M	adsorbent mass [Kg]
ODP	Ozone Depletion Potential
ORC	Organic Rankine Cycle
P	pressure [Pa]
Q	heat rate [kJ S <sup>-1</sup> ] or [kW]
q	heat rate [kJ]
Q <sub>U1</sub>	the upgraded heat from the for absorption heat transformer
Q <sub>U2</sub>	the upgraded heat from the adsorption heat transformer
T	temperature [K]
T <sub>U1</sub>	upgraded temperature (absorption temperature) for absorption heat transformer
T <sub>U2</sub>	upgraded temperature (adsorption temperature) for adsorption heat transformer
ΔT	temperature driving force for heat transfer [K]
$\dot{W}$	mechanical work rate , [kJ s <sup>-1</sup> ] or [kW]
$\dot{w}$	mechanical work rate , [kJ kg <sup>-1</sup> ]
$\dot{W}_U$	the net power output
X <sub>s</sub>	the strong solution concentration [%]
X <sub>w</sub>	the weak solution concentration [%]
ΔX	concentration difference between strong and weak solution (degassing range) [%]
x	adsorbed mass of adsorbate per mass of adsorbent (adsorbate concentration)[kg kg <sup>-1</sup> ]

$\Delta x$  the net concentration change [kg kg<sup>-1</sup>]adsorption capacity

### Greek symbols

$\varepsilon$  heat exchanger efficiency

$\mu$  efficiency

$\Sigma$  Total

### Subscripts

AbHT absorption heat transformer

AdHT adsorption heat transformer

C1 condenser 1

C2 condenser 2

C3 condenser 3

E1 evaporator1

E2 evaporator2

e electric

G1 generator 1

G2 generator 2

HX heat exchanger

is isosteric

l liquid

m mechanic

max maximal

min minimal

R refrigerant

SC solar collector

SP solution pump

P pump

PPC Power Plant Configuration

v vapour

W	water pump
wf	Working fluid
I	first law (or energy) efficiency
II	second law (or exergy) efficiency
1,2,3,...	state points

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