Compressor-assisted heat transformer for waste-heat powered district heating applications

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Abstract:

The recovery of low-temperature waste heat can be promoted by upgrading its temperature. Heat transformers are capable to increase the temperature of a heat stream with a minor contribution of electrical energy. However, the lifting capability of heat transformers are limited by the operating conditions and often insufficient to meet the temperature required by many applications. To overcome this limitation, in this paper a compressor assisted heat transformer is numerically investigated, targeting the application in medium and high temperature district heating networks. Using waste heat at 50 and 60 °C and heat rejection temperature at 0 °C and -10 °C, it is possible to upgrade heat up to 120 °C with an electrical COP always above 4, a thermal COP of about 0.5 and an upper pressure below 40 bar.

1 Introduction

District heating is often considered a key option in the process of the decarbonization of the heating sector, thanks to its capability to both establish energy communities characterized by high renewable penetration and to match availability of waste heat with heating demand, usually not geographically close. However, while high and medium temperature waste heat can be easily recovered in district heating, it also finds other applications either directly in the industry or for electricity generation with ORC. On the contrary, low grade waste heat is harder to be recovered, so finding a way to reuse it in a district heating network would represent an interesting option both from the economic and the environmental point of view. This would also benefit by the fact that often low-temperature waste heat is generated relatively close to the urban areas, as in the case of water-cooled datacenter [1]. One option to couple low-grade heat with district heating is the use of low-temperature networks, which could allow direct recovery. However, many existing networks operate at higher temperature and would benefit by an increase of the waste heat temperature. The option normally considered is the use of a vapor compression heat pump to upgrade the heat to the required temperature. However, the potential of this option is limited by the efficiency reduction at high thermal lifts, which leads to high electricity consumption. A second opportunity is the use of a heat transformer, which faces limitations in terms of working range, i.e. its thermal lift cannot go beyond a certain level.

A compressor assisted heat transformer (CHT) merges in a single cycle the features of the two solution, exploiting the capability of a heat transformer to upgrade low temperature heat and the flexibility provided by the compressor to extend the working range. This solution has been investigated by [2] with the purpose of recovering very low heat ($T \approx 15$ °C), rejecting heat toward a low temperature environment ($T < -10$ °C) and providing heat for low temperature heating ($T \approx 45$ °C). The compression assisted heat transformer proved to require lower electrical input than a water source heat pump operating at the same conditions. However, the explored working condition represent a niche in the heating sector, due to the very low ambient and water temperature, which are favorable for the cycle performance and allow the use of very low temperature waste heat. Thus, in this work, more challenging conditions are investigated, exploring the capability of a CHT to provide efficient heat in hard-to-decarbonize networks, i.e., high (up to 120 °C) or medium temperature (80 °C), starting from a driving temperature of about 50-60 °C. This temperature range also covers a fraction of the industrial heat demand, extending the applicability of the explored solution.

2 Cycle layout and modelling approach

2.1 Cycle layout

Two options are available when adding a compressor to a single stage heat transformer. In the former, the compressor is installed between the generator (GEN) and the condenser (COND) allowing the condensation pressure to be higher than the generation pressure. Alternatively, as depicted in Figure 1, the compressor can be

installed between the evaporator (EVA) and the absorber (ABS), allowing the generation pressure to be lower than the condensation pressure. In both cases the use of the compressor extends the working range of the heat transformer, enabling the use of heat sources (for GEN and EVA) at lower temperatures or heat sinks (for COND and ABS) at higher temperatures.
For the present work, the second option has been selected because it allows to use a smaller compressor, given that the specific volume of the vapor is smaller due to the higher pressure. This is beneficial for the dimensions of the compressor, especially when a volumetric machine (e.g., a membrane compressor) is used.
Two internal heat exchangers are used for heat recovery purposes: the refrigerant heat exchanger (RHE) and the solution heat exchanger (SHX).

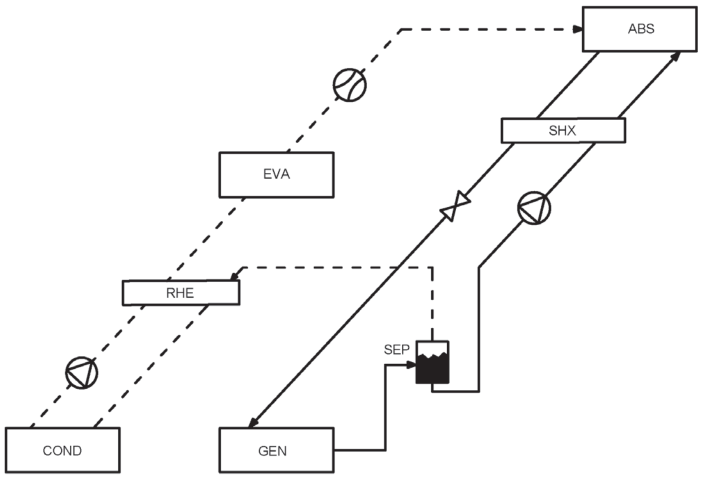

Figure 1 – Layout of the cycle

2.2 Modelling approach

It was assumed that counter-current heat exchangers are used for the heat transformer. For what concerns the generator, a phase separator is added at the solution outlet and a tray column is used to assure a high ammonia mass fraction in the vapor. In the absorber, vapor and poor solution are adiabatically mixed at the inlet of the heat exchanger.
The cycle was numerically modelled using STACY, a tool described in [3], which provides a mathematical framework for steady-state calculation of absorption cycles. The tool is based on a modular approach and support the calculation for each component of the energy, mass and species balances. Additionally, the system of equations is completed with pressure and heat transfer conditions. In particular, the following were assumed within this study:

- Pressure losses are negligible in the pipes and in all heat exchangers but the absorbers where a fixed value of 20 kPa is assumed [4, 5].
- Heat losses are negligible.
- Throttling is isenthalpic.
- The efficiency of the internal heat exchangers (SHX and RHE) is 0.85

The conditions at the absorber, condenser and evaporator were set based on the following assumptions:

- in the generator, a minimum pinch of 5 °C is set between heat source fluid and solution outlet [4].
- at the absorber, a fixed subcooling of 1 °C at the solution outlet and minimum temperature difference of 5 °C is imposed along the heat exchanger [6];
- at the condenser, a fixed subcooling of 1 °C is set at the refrigerant outlet and the minimum temperature difference of 3 °C is imposed between refrigerant and cooling water [6];
- in the evaporator, the minimum temperature difference between the heat source fluid and the refrigerant is 3°C and the refrigerant leaves the heat exchanger with a vapor quality of 0.95 [7].

The mass flow rate of the external circuits was adjusted to maintain fixed temperature differences:

- generator and evaporator are connected in series and the overall temperature difference between generator inlet and evaporator outlet is 10 °C;
- a temperature difference of 30 °C is set between inlet and outlet of the absorber;
- a temperature difference of 5 °C is set between inlet and outlet of the condenser.

The cycle performances are evaluated based on the electric COP, defined as in eq. 1, which provides the ratio between the heating capacity released at high temperature and the electric power needed to run the compressor and the refrigerant and solution pumps. The electric power is calculated assuming an efficiency of the electric motors of 0.9 and isentropic efficiencies for pumps and compressor of 0.8 and 0.7, respectively.
Moreover, the thermal COP, defined in Eq. 2, expresses the ratio between the heating capacity at high temperature and the heating power at intermediate temperature provided to the cycle. It is worth mentioning that the COP_{th} is not strictly the share of the driving heat, which is upgraded to a higher temperature, as in the balance it must be taken into account the contribution of the compressor.

$$COP_{el} = \frac{Q_{ABS}}{W_{COMP} + W_{P\,REF} + W_{P\,SOL}} \tag{1}$$

$$COP_{th} = \frac{Q_{ABS}}{Q_{GEN} + Q_{EVAP}} \tag{2}$$

3 Results and discussion

In this section the cycle performances, calculated numerically for various operating conditions, are presented. The operating conditions are identified with the inlet temperature of the intermediate ($T_{M\ in}$) and the low ($T_{C\ in}$) temperature circuits, which represent the temperatures of the available waste heat and of the available sink for the heat rejection, respectively. For what concerns the absorber, the outlet temperature ($T_{H\ out}$) is used as reference, since it is the temperature of the upgraded heat.

3.1 Influence of the high pressure

The outlet pressure of the compressor (P_{out}) can be freely fixed, provided that it is above a minimum value which allows the cycle operation. Thus, the first analysis which has been carried out is on the impact of this quantity on the cycle performances. Figure 2 reports the variation of the COP_{th} and the COP_{el} with the compressor outlet pressure, for various $T_{H\ out}$. The calculations have been done using -10 °C as the $T_{C\ in}$ and 50 °C as $T_{M\ in}$. The results show that as P_{out} increases, the COP_{el} experiences a maximum before decreasing. In fact, at low pressure, the cycle is close to cut-off conditions and the heat duty at the absorber is very low, as can be deduced looking at the trends of COP_{th}. Then, as the value of COP_{th} becomes practically constant with P_{out}, the additional power required to run the compressor does not translate into an increase of the heat duty at the absorber.
Worth to be noticed is that P_{out} which maximizes the COP_{el} is always below 40 bars, even when $T_{H\ out}$ is 120 °C.

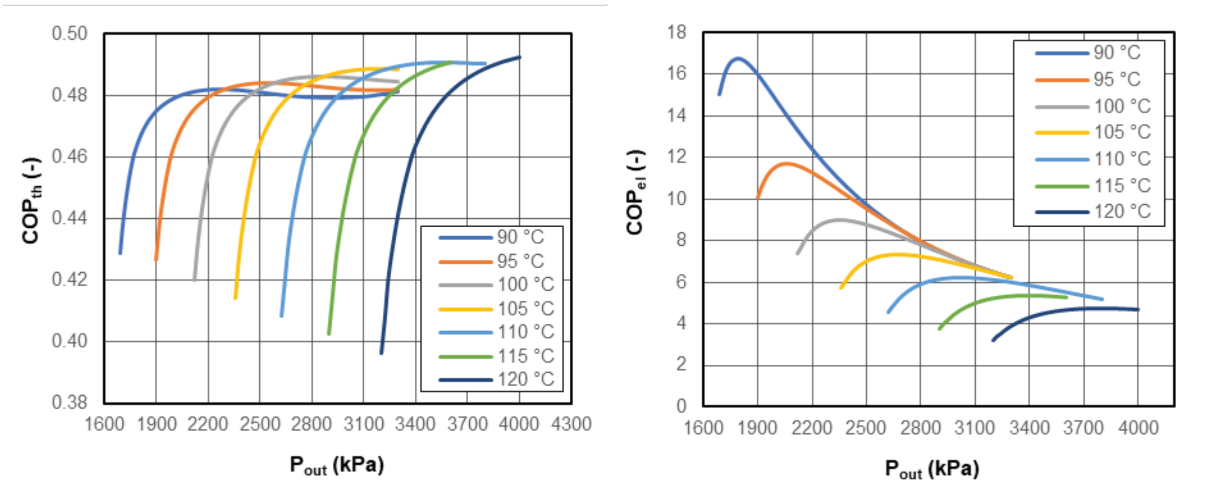

Figure 2 – COP_{th} and COP_{el} vs. Pout for $T_{H\ out}$ from 90 to 120 °C, with $T_{C\ in}$ = -10 °C and $T_{M\ in}$ = 50 °C.

3.2 Heat transformer performances

A second analysis of the cycle performances is carried out comparing four different cases, obtained through the combination of two values of $T_{C\ in}$ (0 °C and -10 °C) and two $T_{M\ in}$ (50 °C and 60 °C). The results presented in Figure 3 provide the COP_{th} and the COP_{el} as function of $T_{H\ out}$. Given the impact of P_{out} on the cycle performances, the results in Figure 3 have been obtained using for each condition the value of P_{out} which maximizes the COP_{el}. This choice can be justified considering that in the present work the use of waste heat of relatively low value is explored. Under these conditions, the minimization of the electrical input is more important than the maximization of the heat recovery. Moreover, the choice of a P_{out} optimized for the COP_{el} has little influence on the COP_{th}. In fact, as observed in Figure 2, if sufficiently far from the cut-off condition, the variation of COP_{th} with P_{out} is very limited.

The two conditions at 60 °C (orange and grey lines) begin at are higher values of $T_{H\ out}$ than the conditions at 50 °C because at low $T_{H\ out}$ the compressor is not needed and a traditional heat transformer can be used.

The results reported in figure 3 confirm that the use of a compressor allows the heat transformer to operate over the entire range of absorber outlet temperature, even in the least favorable conditions, i.e. with $T_{C\ in}$ = 0 °C and $T_{M\ in}$ = 50 °C as the low and intermediate temperature circuits. Moreover, it appears that $T_{M\ in}$ has a larger impact on the COP_{el} than $T_{C\ in}$. In fact, starting from the 0 °C and 60 °C condition, a higher COP_{el} is obtained increasing the $T_{M\ in}$ from 50 to 60 °C rather than decreasing the $T_{C\ in}$ from 0 to -10 °C.

The upper pressure of the cycle resulted below 40 bars for all the working conditions presented in Figure 3 except with $T_{C\ in}$ = 0 °C and $T_{M\ in}$ = 50 °C when $T_{H\ out}$ goes above 110 °C, when the limits move to 46 bars.

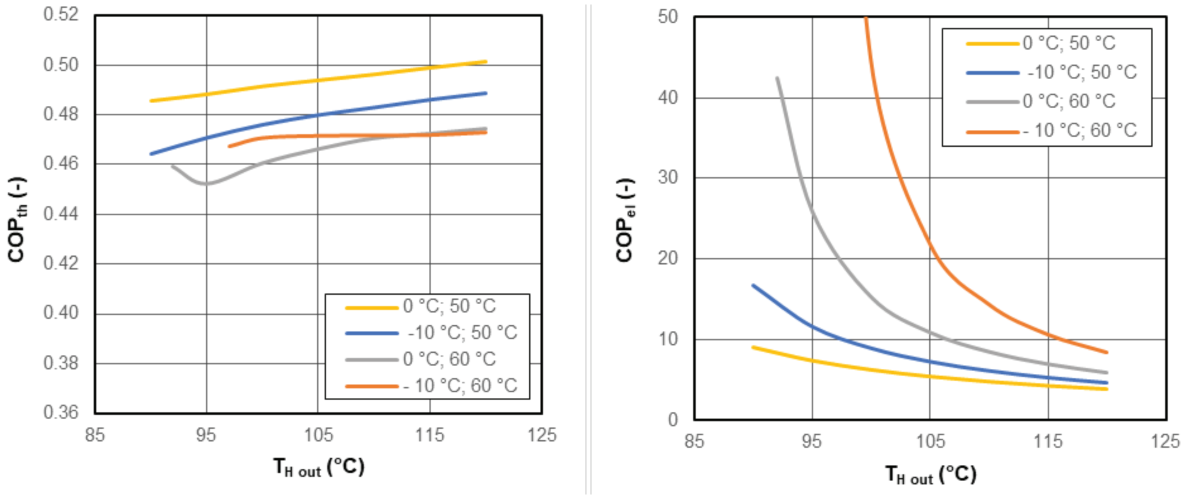

Figure 3 – COP_{th} and COP_{el} vs. absorber outlet temperatures for four combination cold and intermediate circuits temperatures.

4 Conclusions

The numerical calculation of the performances of the compression assisted heat transformer proved that it represents a suitable solution for powering district heating networks at temperatures up to 120 °C. In particular it has been observed that:

- for a given working condition the compressor outlet temperature can be adjusted to maximize the electrical COP, with minor impact on the thermal COP;
- the thermal COP is between 0.45 and 0.50 for all the investigated conditions;
- the use of a compressor increases the working range of the heat transformer allowing thermal lift which cannot be achieved with a traditional heat transformer;
- the electrical COP is well above 10 for most of the high temperature range, when the temperature of the waste heat is 60 °C.

5 List of References

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