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Technical, economic, and environmental analysis of integrating a high-temperature heat pump into a pulp and paper industry in Italy

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Abstract

This study presents a comprehensive technical, economic, and environmental analysis of integrating a high-temperature heat pump (HTHP) coupled with a mechanical vapor recompression (MVR) system into the steam generation process of a pulp and paper industry in Italy. The HTHP is designed to extract 600 kW of thermal energy as waste heat at 90 °C, producing saturated steam at 115 °C, which is further compressed by the MVR to 162 °C and 6.5 bar(a), meeting the plant's steam header requirements. The system is expected to operate for 7,000 hours annually, delivering 5.8 GWh of heat and consuming 2.0 GWh of electricity, with a system COP of 2.85 and second-law efficiency of 49%. The techno-economic analysis reveals a total capital expenditure (CAPEX) of 0.8 M€ and annual operational expenditures (OPEX) savings of 94.0 k€/y compared to a conventional gas boiler, resulting in a simple payback period of 8.4 years, a levelized cost of heat of 0.09 €/kWh, and a return on investment of 13%. Environmentally, the proposed system achieves a 43% reduction in CO_2 emissions, saving 725.0 tons of CO_2 annually, equivalent to 69.0 k€/y in emission costs. Moreover, a parametric study was conducted to highlight the sensitivity of performance to various source and sink temperatures, flow rates, working hours, and emission taxes. The results demonstrate the viability of HTHP+MVR systems as a sustainable alternative for industrial heat upgrading, contributing to the decarbonization and energy efficiency goals.

Keywords: *HTHP, Steam generation, Pulp and paper industry, Techno-economic study, TEWI*

1. Introduction

Global warming and its awareness are transforming the energy sector. Europe has set a target to become carbon neutral by 2050, and the Strategic Energy Technology Plan 2025-2027 includes specific actions, such as enhancing energy efficiency in the industry and increasing the integration of renewable energies. The industry is a key sector that should be tackled, since it is responsible for 21% of the CO_2 emissions [1].

Process heat is particularly energy demanding. There is a great potential to replace conventional fossil-fuel burners with different technologies. For instance, in the process heat temperature range of 100-200 ºC, which accounts for 26% of the industrial energy demand [1], HTHPs can play a crucial role in contributing to the energy transition and decarbonization targets. On the one side, they can increase the system efficiency, among other reasons, by recovering heat from low-temperature heat sources (which are often wasted). On the other hand, the compressor is electrically powered, and in many countries, due to the weight of renewable energies in the energy mix, the indirect CO_2 emissions are low. Recent studies in Europe have shown the potential to reduce the industrial CO_2 emissions by up to 70-80% in district heating networks, or by a maximum of 90% depending on the country [2,3]. In China, where industrial heat is often produced from coal, the potential is also very high.

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The technology has been reviewed in recent studies, for instance by Hamid et al. [4] and Arpagaus et al. [5]. These studies demonstrate that the technology has undergone significant enhancements over the past decade, thereby opening the door to potential applications of process heat at higher temperature levels.

In 2024, the International Energy Agency (IEA) documented a total of 300 HTHPs. HTHPs have a relatively high technology readiness level of 4-9, and several manufacturers already offer systems for pressurized water or steam generation in the range from 100-150 ºC. The range from 150-200 ºC is expected to be reached in the next years [6]. However, according to the IEA, only 15 cases had been applied in the industry. Among the different reasons justifying the gap between technology and its industrial application are high investment costs, the uncertainty in regulations, or the fact that the technology is not a plug-and-play product. Its integration requires an in-depth analysis of waste heat availability, the required process heat, energy costs, and other relevant factors.

Nevertheless, recent studies, for instance, from the European project PUSH2HEAT [7], are showing that payback periods of less than 4 years can be achieved in some industries. Given the energy context and technological status, HTHPs are consequently expected to increase their market penetration in the upcoming years. The IEA's HPT TCP Project 68 [8] is currently screening the technology and aiming to increase its integration in the industry, with many applications on the horizon.

This work is conducted within the framework of the PUSH2HEAT project and focuses on the design of a HTHP for integration into a paper industry in Italy, Cartiera di Guarcino (CDG). The design of the HTHP is explained, along with an analysis of its expected nominal performance from technical, economic, and environmental perspectives. Moreover, a parametric study is conducted to understand the sensitivity of the results to key technical and economic parameters.

2. Heat Source Analysis and HTHP Design and Integration

The performance of HTHPs depends critically on the temperature of the heat source; therefore, the most common approach is to operate them in conjunction with a source of waste heat available within the plant.

2.1. Description of the heat source circuit

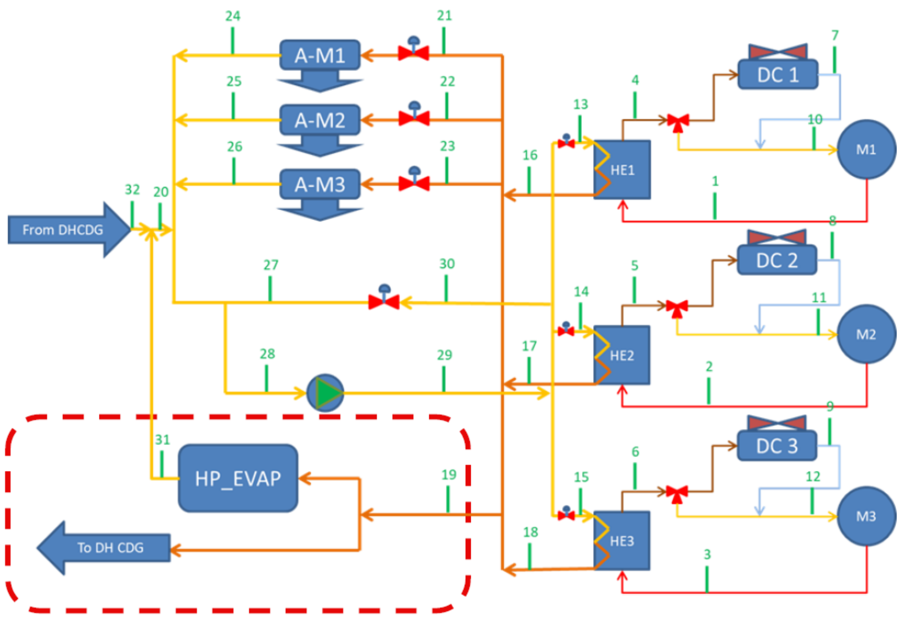

Figure 1. Schematic of the heat source circuit and waste heat extraction.

The source of waste heat at CDG comes from the cooling water circuit of the cogeneration (CHP) plant operated by Bio Energia Guarcino (BEG). Figure 1 shows a scheme of the CHP plant, which consists of three biomass engines (M1, M2, and M3), each rated at 6.87 MWel. The engines' cooling water circuit is the primary hot water circuit. The cooling water for these engines enters at ≈ 70 °C (points 10, 11, and 12) and exits at ≈92-94 °C (points 1, 2, and 3) with a nominal flow rate of 135 m^3/h per engine.

Under normal operation, the cooling water transfers part of its heat through three primary heat exchangers (HE1, HE2, and HE3) to a secondary hot water circuit, which uses this heat for preheating the air required for the combustion process (A-M1, A-M2, and A-M3) and for the plant district heating (point 19). Any excess heat in the primary water circuit is dissipated to the ambient through dry coolers (DC1, DC2, and DC3).

The HE1, HE2, and HE3 were originally sized to 2MW, while historical data analysis demonstrated that the heat duty of the mentioned heat exchangers could be increased to 2.7 MW. So, as a first change, these heat exchangers have been upgraded to 2.7 MW, and the proposed HTHP is planned to be installed in parallel to the plant's district heating (DH CDG) (between points 19 and 31), which will recover this waste heat from the secondary side of the heat exchangers, as shown in Figure 1. This configuration minimizes the system modifications and utilizes the existing secondary hot-water circuit between BEG and CDG.

2.2. Expected nominal working conditions

The HTHP is designed to extract approximately ≈ 600 kWth from the secondary hot water circuit (heat source) by cooling the inlet stream at $T_{src,in,EVAP}$= 90 °C, and a flow rate of $\dot{m}_{src}$= 14.3 kg/s. It will preheat and then evaporate the water that comes from the deaerator at $\dot{m}_{w,in,COND}$= 0.33 kg/s, $T_{w,in,COND}$= 105 °C and $p_{w,in,COND}$= 1.7 bar(a), producing saturated at $T_{st,COND}$= 115 °C. A MVR unit is coupled with the HTHP to pressurize the steam further up to $p_{st,MVR}$= 6.5 bar(a) and $T_{st,MVR}$= 162 °C, as shown in Figure 2. The employed MVR is not a typical oil-cooled one. It is a patented technology where subcooled, or saturated, water is injected into the steam line at the suction side, which, apart from helping to cool different stages of the screw compressor, also guarantees that the outlet steam is saturated, or close to saturation. A seen in Figure 2, the water is injected at the rate of $\dot{m}_{w,inj,MVR}$= 0.03 kg/s and $T_{w,inj,MVR}$= 105 °C (similarly to the condenser inlet).

The outlet pressure matches the main process steam header supplying the paper machines of CDG. This integration allows the HTHP+MVR system to partially replace fossil-fuel steam generation from the site's gas boilers. Overall, the system is expected to provide approximately steam with $\dot{m}_{st,MVR}$= 0.36 kg/s (≈ 1.3 t/h), corresponding to around 5.7% of CDG's total steam demand, and to operate for up to 7,000 h/y based on waste heat availability.

2.3. HTHP design and main components

The proposed HTHP+MVR system is being designed and assembled by the project partner BONO, employing the low-GWP refrigerant R1233zd(E). As seen in Figure 2 (left), the HTHP is a single-stage configuration that comprises a variable-speed twin screw compressor (COMP), H5PH720F model by Palladio, with a total displacement of 4577 cm^3, a shell-and-tube evaporator (EVAP) with a nominal capacity of 600 kW, a shell-and-tube condenser (COND) with a nominal capacity of 740 kW, and an electric expansion valve (EXV). The MVR is a twin-screw steam compressor, GP-SC02 model by G-Plus, with a nominal power consumption of 160 kW.

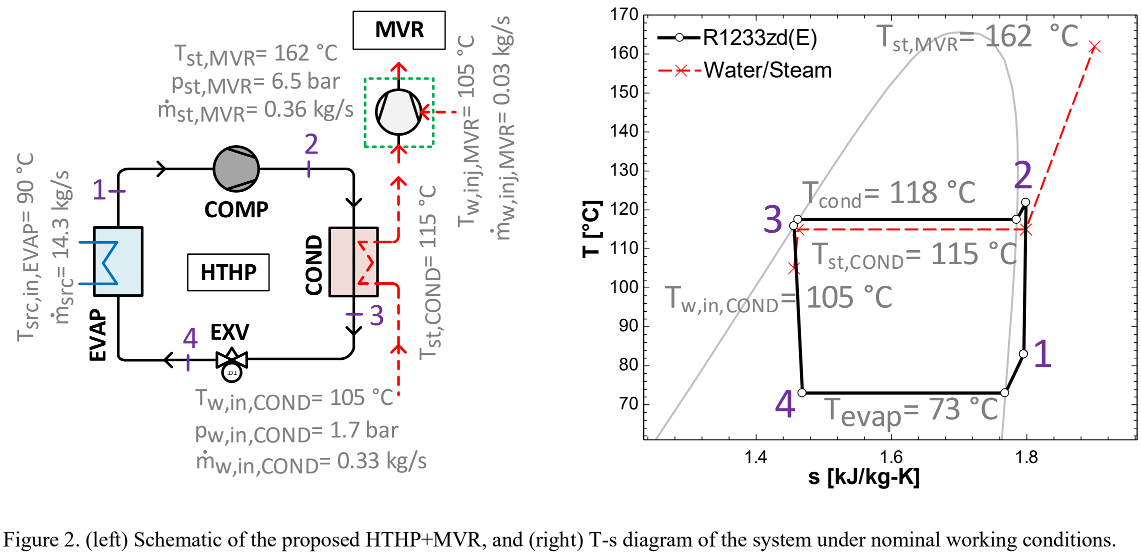

Figure 2. (left) Schematic of the proposed HTHP+MVR, and (right) T-s diagram of the system under nominal working conditions.

3. System Modelling Methodology

3.1. Methodology for technical analysis

The HTHP+MVR system, shown in Figure 2, was modelled using the Engineering Equation Solver software [9]. The evaporator and condenser were modelled using the pinch point method [10]. The pinch point in the evaporator and condenser was fixed at 6.5 K and 2.5 K, respectively. The compressor efficiency η_{comp} versus pressure ratio Pr polynomials, provided by the manufacturers, were used to estimate the power consumption of the HTHP's compressor ($P_{el,HTHP}$) in kW, as shown in Eq. (1). The MVR is treated as a black box model; its power consumption ($P_{el,MVR}$) is estimated similarly to the HTHP's compressor using Eq. (1). The impact of water injection and evaporation process on its power consumption and heat balance is also considered in the system modelling. To estimate the isentropic efficiency η_{is} and subsequently the discharge conditions, a heat loss ratio φ_{loss} of 20% of the power consumption was assumed, which is considered a typical average value for such high-temperature applications [11]. Eq. (2) shows the relation between η_{comp} and η_{is} through the φ_{loss}. The volumetric efficiency η_v versus Pr polynomial was used to estimate the refrigerant flow rate $\dot{m}_{ref}$, in kg/s, as shown in Eq. (3). The main performance indicators for the proposed system are the COP of the HTHP COP_{HTHP} (Eq. (4)), the equivalent COP of the system COP_{SYS} (Eq. (5)), and the efficiency of the second law η_{II} (Eq. (6)), using the COP of Lorentz cycle $COP_{LZ,SYS}$ as reference (Eq. (7)). It is worth mentioning that the auxiliary electrical consumption, such as circulation pumps, is not included in the current technical analysis.

$$\eta_{comp} = f(Pr) = \frac{\dot{m}_{ref}\Delta h_{is,comp}}{P_{el}} \tag{1}$$

$$\eta_{is} = \frac{\Delta h_{is,comp}}{\Delta h_{comp}} = \frac{\eta_{comp}}{1-\varphi_{loss}} \tag{2}$$

$$\eta_v = f(Pr) = \frac{\dot{m}_{ref}v_{suc}}{\dot{V}_{ideal}} \tag{3}$$

$$COP_{HTHP} = \frac{\dot{Q}_{COND}}{P_{el,HTHP}} \tag{4}$$

$$COP_{SYS} = \frac{THC}{P_{el,HTHP} + P_{el,MVR}} \tag{5}$$

$$\eta_{II} = \frac{COP_{SYS}}{COP_{LZ}} \tag{6}$$

$$COP_{LZ,SYS} = \frac{LMTD_{snk}}{LMTD_{snk} - LMTD_{src}} \tag{7}$$

Where $\Delta h_{is,comp}$ and Δh_{comp} are the isentropic and real enthalpy difference, in kJ/kg, across the compressor, respectively. v_{suc} is the suction specific volume, m^3/kg. $\dot{V}_{ideal}$ is the ideal flow rate, m^3/s. $\dot{Q}_{COND}$ and THC are, respectively, the heating capacity provided only by the HTHP and the total heating capacity provided by both the HTHP and MVR, in kW. $LMTD_{snk}$ and $LMTD_{src}$ are the logarithmic mean temperature difference, in K, for sink (snk) and source (src), respectively. $LMTD_{snk}$ was evaluated based on inlet and outlet sink enthalpy and entropy values, rather than temperatures alone, because it involves a phase change process.

3.2. Methodology for the economic and environmental analysis

The economic analysis begins by estimating the capital expenditures of the proposed system ($CAPEX_{SYS}$) and the first-year operating expenditures ($OPEX_{year1}$), which include annual energy costs, maintenance, and CO_2 emission costs. The maintenance cost is always evaluated as a percentage of the CAPEX of the technology. The simple payback period (SPP) is simply the relation between the total CAPEX of the new technology and the OPEX savings, or the difference between the OPEX values of the HTHP+MVR system and a reference technology, which is, in the current study, a conventional gas boiler (GB).

Applying a discount method, the yearly costs increase due to inflation, and the future expenses are traced back to the net present value of money (NPV) by considering the market discount (d) and inflation (i) rates. Eq. (8) shows the overall OPEX paid at the end of year Y_n:

$$OPEX_{Y_n} = OPEX_{year1}\left[\frac{(1+i)^{Y_n-1}}{(1+d)^{Y_n}}\right] \tag{8}$$

Eq. (9) presents the NPV_{SYS}, in €, of introducing a HTHP+MVR. The inflation and discount rates are assumed to be the same for energy prices, maintenance, and CO_2 emissions costs.

$$\begin{aligned} NPV_{SYS} = {} & OPEX_{year1}\left(\frac{1}{d-i}\right)\left[1-\left(\frac{1+i}{1+d}\right)^{n}\right]\Bigg|_{GB} \\ & - OPEX_{year1}\left(\frac{1}{d-i}\right)\left[1-\left(\frac{1+i}{1+d}\right)^{n}\right]\Bigg|_{SYS} - CAPEX_{SYS} \end{aligned} \tag{9}$$

The discounted payback period (DPP) is evaluated iteratively using the SOLVER tool in Microsoft Excel to determine the exact year in which the NPV_{SYS} equals zero (the breakeven point), with typical, fixed market discount and inflation rates. The return on investment (ROI) percentage is also evaluated iteratively by fixing the inflation rate and varying the discount rate to reach zero NPV_{SYS}. Finally, the levelized cost of heat (LCOH), in €/kWh, is evaluated as shown in Eq. (10).

$$LCOH = \frac{CAPEX_{SYS} + \sum_{n=0}^{20} OPEX_{Y_n}}{\sum_{n=0}^{20} \frac{DHP}{(1+d)^{Y_n}}} \tag{10}$$

Where DHP is the discounted annual heat production from the HTHP+MVR system in kWh/y.

Regarding the environmental analysis, the total equivalent warming impact (TEWI) method is adopted to calculate the total direct and indirect CO_2 emissions in kg-CO_2, as shown in Eq. (11).

$$\begin{aligned} TEWI = {} & \left[GWP \times \left(M_{ref} \times L_{yr}\right) \times N\right] + \left[GWP \times M_{ref} \times (1-\alpha_{rec})\right]\big|_{direct} \\ & + \left[E_{yr} \times \beta \times N\right]\big|_{indiret} \end{aligned} \tag{11}$$

Where GWP is the global warming potential of the refrigerant used, in kg-CO_2/kg_{ref}. M_{ref} is the total refrigerant charge, in kg. L_{yr} is the annual leakage percentage. N is the lifetime of the system, in years. α_{rec} is the recuperation factor for the refrigerant after the lifetime. E_{yr} is the annual energy consumption, in kWh/y. β is the carbon emission intensity, in kg-CO_2/kWh. Finally, Table 1 shows the main fixed parameters used in techno-economic analysis of the proposed system.

Table 1. Fixed parameters used in the techno-economic analysis of the HTHP+MVR system.

Parameter	Value	unit
GB's specific CAPEX*	100	€/kWt
GB's efficiency*	90	%
Gas cost*	0.053	€/kWh
HTHP+MVR's specific CAPEX (including installation)*	950.0	€/kWt
Electricity cost*	0.148	€/kWh
Refrigerant charge (M_{ref})*	155	kg
GWP (R1233zd(E)) [12]	1	kg-CO_2/kg$_{ref}$
Annual refrigerant leakage (L_{yr}) [13]	10	%
Refrigerant recuperation factor (α_{rec}) [13]	95	%
Maintenance cost percentage (from the CAPEX)	2	%
Carbon emission intensity for gas (β_{gas}) [14]	0.216	kg-CO_2/kWh$_t$
Carbon emission intensity for electricity (β_{el}) [15]	0.330	kg-CO_2/kWh$_{el}$
CO_2 emission taxes*	95	€/tCO_2
Annual working hours for HTHP+MVR/GB*	7,000/8,424	h/y
Lifetime (N)	20	years
Market discount rate (d)	5	%
Average inflation rate (i)	3	%

*Data provided directly by the CDG.

4. Results and Discussion

4.1. Techno-economic+environmental results under nominal point

Table 2 shows the summary of the techno-economic+environmental analysis of the proposed HTHP+MVR system working under nominal working conditions (Section 2.2), in comparison with a traditional GB with the same THC, working almost all year round (8,424 h/y). The system delivers a THC of 833 kW and consumes 292 kWe with a COP_{SYS} of 2.85. This corresponds to 49% of the $COP_{LZ,SYS}$. Figure 2 (right) shows that the HTHP works between evaporation and condensation temperatures of 73 and 118 °C, respectively. This corresponds to a temperature lift of 45 K, as seen in Figure 2 (right). Under these conditions, the HTHP consumes 143 kWe with an expected COP_{HTHP} of 5.22.

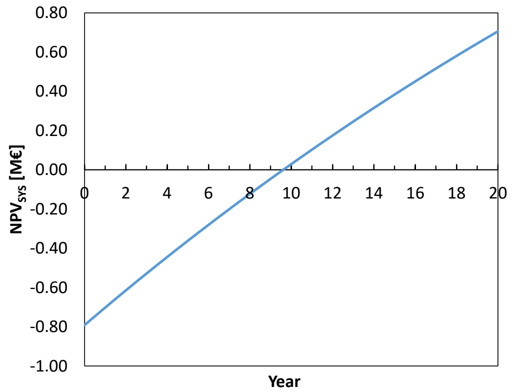

Figure 3. The evolution of the NPV_{SYS} within the expected lifetime.

However, due to the limitation of waste heat availability (7,000 h/y), the HTHP+MVR system has a heat demand coverage (HDC) of 83% (5,831,280.0 kWh/y), while the GB serves as a backup gas boiler (BGB) to cover the remaining 17% (1,186,249.0 kWh/y). The BGB was assumed to operate at its full load capacity of 833 kW when the HTHP+MVR system is active, adjusting its working hours to cover any annual heat deficit.

The technical analysis indicates that the proposed system has a total CAPEX of 791,388.0 € and results in an OPEX saving, compared with the GB, of 93,805.0 €/y. This finally results in a SPP of 8.4 years, indicating a long-term investment scenario. Figure 3 shows the evolution of the NPV_{SYS} through the expected lifetime of the system. The breakeven point is at ≈9.6 years, which corresponds to the DPP.

Table 2. Techno-economic analysis of the HTHP+MVR system under the nominal point.

HUT	Parameter	Value	Unit
HTHP+MVR	Total heating capacity (THC)	833.0	kW
	Total power consumption	292.0	kW
	$COP_{SYS}/COP_{LZ,SYS}/\eta_{II}$	2.85/5.83/0.49	–
	Annual energy consumption	2,044,700.0	kWh/y
	Annual heat production	5,831,280.0	kWh/y
	HDC	83	%
	CAPEX	791,388.0	€
	$OPEX_{year1}$	382,546.0	€/y
Backup GB (BGB)	Annual energy consumption	1,318,054.0	kWh/y
	Annual heat production	1,186,249.0	kWh/y
	HDC	17	%
	CAPEX	–	€
	$OPEX_{year1}$	98,569.0	€/y
(HTHP+MVR)+BGB	$OPEX_{year1}$	481,116.0	€/y
	NPV_{SYS} (at N= 20 years)	706,182.0	€
	OPEX $saving_{year1}$	93,805.0	€/y
	SPP	8.4	years
	DPP	9.6	years
	ROI	13	%
	LCOH	0.09	€/kWh
	Annual CO_2 emissions	959.0	tCO_2/y
	Annual CO_2 emission reduction (percentage)	725.0 (43%)	tCO_2/y
	Annual CO_2 emission saving	68,850.0	€/y

The ROI of the proposed system is 13%, which indicates the profitability of the investment relative to its cost. Finally, the LCOH is a useful indicator for estimating the cost of heat generated by the entire system throughout its life cycle, considering both initial investment, operation & maintenance, and CO_2 emissions costs. The results show a LCOH of 0.09 €/kWh, which is a competitive value compared with other heat upgrade technologies (HUTs) used in industry.

Regarding the environmental analysis, the (HTHP+MVR)+BGB system results in ≈43% reduction in annual CO_2 emissions compared to the GB. Specifically, it saves up to 725 tCO_2/y, corresponding to 68,850.0 €/y. This significant reduction in CO_2 emissions is one of the key reasons to replace the fossil-fuel-driven HUTs with electricity-driven or green alternatives, such as HTHPs or absorption heat transformers (AHTs).

4.2. Impact of key techno-economic parameters on the results

The waste heat availability is a crucial parameter that determines the working hours of the HTHP+MVR system, its HDC, and the percentage of reduction in CO_2 emissions. Figure 4 (left) shows that reducing working hours below 5,000 h/y results in a significant increase in SPP, making the investment unprofitable, as well as a significant decrease in CO_2 emissions reduction and HDC percentages, reaching 12% and 24%, respectively, at 2,000 h/y. Increasing working hours to 8,424 h/y results in complete coverage of the heat demand, a lower SPP (6.7 years), and a significant reduction in CO_2 emissions (52%) compared with the GB.

CO_2 emission taxes are another important parameter that significantly impacts the OPEX savings and, consequently, the SPP of the investment. Reducing it to 57 €/tCO_2 results in a reduction in OPEX savings by 29.5% and an increase in SPP by 3.5 years, as seen in Figure 4 (right). Increasing taxes (e.g., 133 €/tCO_2) would result in a 29% increase in OPEX savings and a 1.9-year decrease in the SPP.

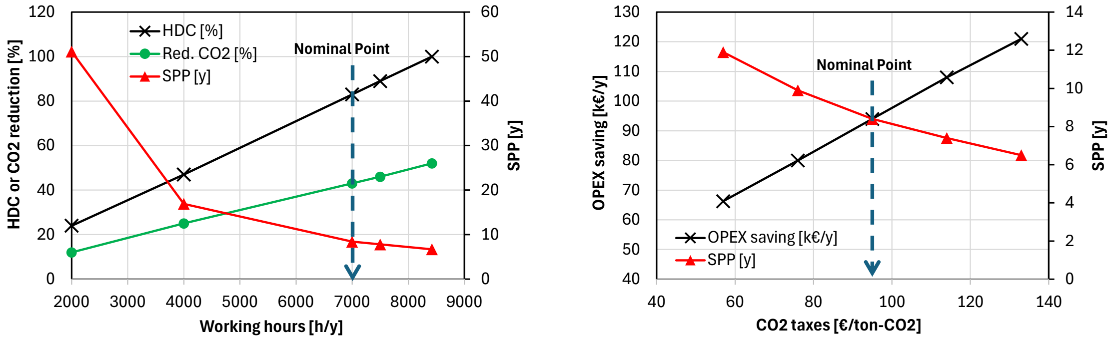

Figure 4. Impact of working hours (left) and CO_2 emission taxes (right) on the techno-economic+environmental results.

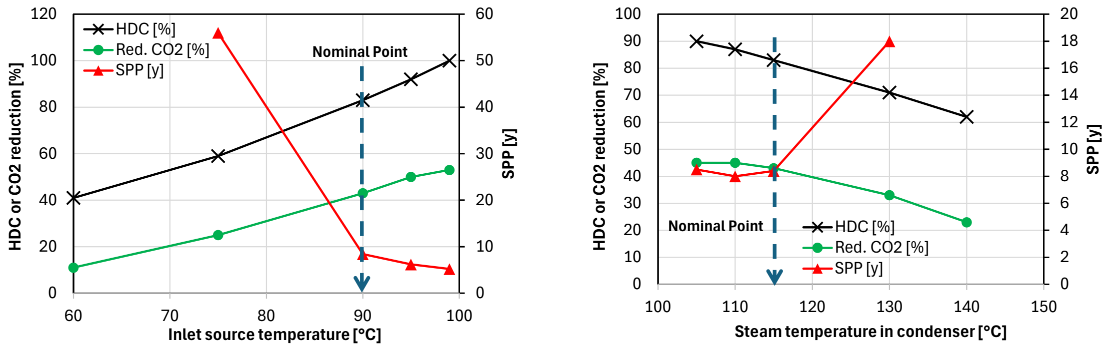

Figure 5. Impact of inlet source temperature (left) and saturated steam temperature in condenser (right) on the techno-economic+environmental results.

The inlet source temperature is a working parameter that affects the performance of the HTHP. Decreasing it to 60 °C results in a significant reduction in THC and COP_{SYS}, reaching 407 kW and 1.92, respectively. This results in negative values of NPV_{SYS} and SPP, indicating that this investment is unprofitable. Increasing the inlet source temperature to 99 °C results in an increase in THC and COP_{SYS}, reaching 1003 kW and 3.1, respectively. This, in turn, has a positive impact on the SPP and the reduction percentage of CO_2, as seen in Figure 5 (left), since under these working conditions, the HTHP+MVR system covers 100% of the heat demand. The SPP is reduced to 5.2 years, while the CO_2 reduction percentage is increased to 56%. Instead of increasing the heat source temperature to cover the whole demand, the opportunity of increasing the heat source flow rate was also investigated. Increasing the flow rate from 53 to 179 m³/h, while maintaining a temperature of 90 °C, yields a SPP of 5.3 years and a CO_2 reduction percentage of 55%.

Figure 5 (right) shows the impact of changing the saturated steam temperature in the condenser on the techno-economic+environmental results. Its reduction to 105 °C results in an increase of THC to 902 kW, which results in a HDC of 90% and an increase of the reduction percentage of CO_2 up to 45%; however, the SPP increased slightly to 8.5 years due to the decrease of the COP_{SYS} by 2.5% as a result of the increase of $P_{el,MVR}$. The optimum value is 110 °C, where the SPP reaches its lowest value of 8.0 years; however, the system consumes more source heat by 12.5%. Increasing the temperature to 140 °C results in a significant increase in the $P_{el,HTHP}$, resulting in a COP_{SYS} of 2.2 and negative values of NPV_{SYS}.

5. Conclusions

- This study shows the feasibility of designing and integrating a HTHP+MVR system in a paper plant in Italy that can work with an inlet source temperature of 90 °C and an outlet sink temperature of 162 °C, using waste heat from cooling biomass heat engines as a heat source.
- Under nominal working conditions, it can work up to 7,000 h/y, delivering a total heat of 5.8 GWh/y, while consuming electricity of 2.0 GWh/y, with COP_{SYS} of 2.85 and η_{II} of 49%.
- However, due to the limit of waste heat availability (7,000 of 8,424 h/y), a BGB is used as a backup system to cover 17% of the total required heat demand throughout the year.

- The whole system (HTHP+MVR)+BGB has a total CAPEX of 0.8 M€ and OPEX savings of 94.0 k€/y, compared with a conventional GB with the same heating capacity. This results in a SPP, DPP, and ROI of 8.4 years, 9.6 years, and 13%, respectively.
- The proposed system results in a reduction in CO_2 emissions up to 725.0 tCO_2/y or 43%, compared with the GB, which indicates the importance of adopting electricity-driven HUTs as potential alternatives to conventional fossil-fuel-driven ones.
- The parametric study showed that the HTHP+MVR system can cover 100% of the heat demand by increasing the working hours to 8,424 h/y. This results in a 20% increase in the CO_2 reduction and a decrease in the SPP of 1.7 years.
- The CO_2 emissions taxes are crucial in the analysis. Increasing the taxes by 40% can increase the OPEX savings by 29% and reduce the SPP of the investment by 1.9 years.
- Increasing the heat source temperature up to 99 °C or increasing its flow rate up to 179 m^3/h results in an HDC of 100%, a reduction in the SPP by ≈3.0 years, and an increase in the CO_2 reduction percentage by 28%, compared with the nominal point. However, these scenarios should be investigated in more detail to ensure that they do not impact the district or air heating demand.
- Finally, decreasing the steam generation temperature inside the condenser to 110 °C results in an increase in the THC and CO_2 reduction by 5%, making the HTHP+MVR system capable of covering 87% of the total heating demand, with 0.4 years less in the SPP, compared to the nominal working conditions.

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