Coupling waste heat extraction by phase change materials with superheated steam generation in the steel industry

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ABSTRACT

To allow a better exergy exploitation than the current state-of-the-art waste heat to power solutions in the steel industry, a new type of energy recovery system based on Phase Change Materials is proposed. In particular, the use of high temperature PCMs evolves from simply smoothing off gas temperature, as in the most recent studies for energy recovery from electric arc furnaces, to generating constant superheated steam able to feed the downstream turbine nearly at nominal load. This result is achieved by introducing an auxiliary section between the PCM Section and the steam generation one, which provides the auxiliary heat needed to level the thermal content of off gas. The auxiliary heat is extracted from the PCM units by a heat transfer fluid flowing across the inner tube of each PCM container.

Different models to properly size and simulate the operations of the proposed energy recovery system have been developed and integrated. Results show how the size of the steam generator and the turbine can be reduced of about 41% with respect to traditional solutions, while increasing electric power production by 22% thanks to the reduced fluctuation in steam parameters at the turbine inlet, which leads to a greater overall efficiency.

KEYWORDS

Energy recovery, Phase change material, thermal power fluctuation, superheated steam, steel industry

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1 INTRODUCTION

Waste heat recovery in steel industry represents one of the greatest opportunity to reduce the consumption of primary energy while increasing the sustainability of the steelmaking process [1–4]. One of the most important and challenging source of waste heat is represented by the off gas emitted by the Electric Arc Furnace (EAF), which accounts for about 30% of the total energy provided to the process [5]. However, due to the dynamic of the EAF process, off gas temperature fluctuates intensively causing large variations in thermal power and an inefficient energy recovery, consequently [6,7].

Current available technologies can be classified depending on the final use of the recovered heat as performing a direct or an indirect recovery (refer to [8] for a review). In direct recovery, off gas thermal energy is recuperated and directly used to preheat the scrap before its charging into the furnace. In indirect recovery, instead, off gas thermal energy is carried out by a heat transfer fluid (HTF), mainly steam, and used for several applications such as district heating, vacuum degasser, and power production [9]. However, a thermal energy storage system (TESS) is generally required in the attempt of providing a constant supply of heat downstream [6]. Recently, innovative TESS based on the capacity of phase change materials (PCMs) to store and release latent heat have been proposed also for high temperature conditions such those encountered in the steel industry [7,10–13]. In particular, an innovative solution for indirect energy recovery in a discontinuous charge EAF process [14] and a conjoint direct–indirect recovery in a continuous charge one [8] have been developed. The features of metal PCMs have been exploited not only for latent heat storage, but mainly to smooth the variability of off gas temperature and feed the downstream energy conversion technology more constantly at around 300 °C.

The aim of this paper is to further increase the efficiency of indirect energy recovery from EAF off gas by providing superheated steam with not only low thermal variability, but also with higher temperature to a steam turbine for power production. Therefore, in the following section 2 the potential evolution of steam-based energy recovery in steel industry is depicted, departing from current technologies and moving to the novel energy recovery approach proposed in this study. In section 3, the new PCM-coupled energy recovery system is described, while in section 4 the models created for its development and performance analysis are reported. In section 5 results are discussed, while in section 6 conclusions are derived.

2 MANAGING THE EVOLUTION OF STEAM-BASED ENERGY RECOVERY FROM EAF

State-of-the-art solutions for EAF waste heat to power (WHTP) systems can be considered as composed by three main components:

1. a waste heat recovery boiler (WHRB);
2. a thermal energy storage system (TESS);
3. an energy conversion technology (ECT).

Concerning the WHRB, two technologies are commonly used: the saturated steam generator (Sat-SG) and the hot water boiler (HWB). The choice between saturated steam and hot water usually depends on their adoption also as a cooling medium in off gas ducts. The use of thermal oil boilers is generally excluded by steel plant operators due to safety issues related to fire risk.

When a saturated steam generator is used, Ruth's steam accumulators (RSA) are employed as TESS, while two type of energy conversion technologies can be used: the saturated steam turbine (Sat-ST) [6] and the Organic Rankine Cycle (ORC) turbine [15]. When a hot water boiler is used as WHRB, instead, only a hot water tank (HWT) and an ORC turbine can be used as TESS and ECT, respectively [16].

The design of the whole waste heat to power system normally starts from the WHRB, followed by the TESS, and finally by the ECT (see the grey coloured portion in Figure 1a) in a sequential approach.

Each system is sized individually and the output parameters are used as input parameters of the following component (i.e. maximum steam flow rate and temperature exiting TESS become the design parameters of the downstream ECT). As shown in Figure 1b, such state-of-the-art solutions are able to manage thermal power fluctuations, but the ECT feeding temperature is rather low. Since the average temperatures of the off gas are usually around 600 °C, in order to reduce exergy losses a better solution could be the generation of superheated steam at higher temperature to directly feed a superheated steam turbine (Sh-ST). In [17], two solutions have been proposed: 1) applying a duct burner before the super heater and 2) providing a common WHRB by combining the off gas ducts of two EAFs. The former solution implies the use of additional fossil fuel, while the latter leads to manage a double production process. An innovative solution has been proposed in [18], where a heat recovery boiler based on molten salt is coupled with superheated steam generation. In this case, molten salt acts both as a heat transfer fluid and as a thermal storage system, thus facing the same issues of solar plants, such as fast draining of the liquid and freezing prevention during EAF shutdown.

High temperature phase change materials (PCMs) can be employed to reduce the fluctuating thermal power entering the steam generator. Previous findings have shown that the introduction of a PCM-based smoothing (PCMS) device is capable to smooth off gas temperature profiles, significantly affecting the design of the whole waste heat to power system [14]. The PCMS has been further developed in [8], with the introduction of a heat transfer fluid (HTF) to overcome overheating issues in PCM containers. The change in the design approach is reported in Figure 1a (green coloured portion). TESS features are integrated into the PCMS, which becomes the first component of the waste heat to power system to be sized, followed by the WHRB. In this case, the WHRB is a superheated steam generator (Sh-SG) and its size depends on the performance of the upstream PCMS. Referring to Figure 1b, the PCMS allows to enhance the steam temperature, but the control of thermal fluctuations is less effective than current state-of-the art solutions.

In this paper, the chance of actively managing the heat transfer fluid flowing through the PCM-based device in order to improve steam parameters (i.e. temperature and mass flow rate) is investigated. The basic idea is to make the whole waste heat to power system evolving towards the left upper side of Figure 1b, thus providing both higher steam temperature and near-zero thermal power fluctuations to the downstream energy recovery section.

To this end, a new type of heat recovery system called PCM-Coupled Steam Generator (PCMCSG) is developed by integrating the PCMS features (TESS and temperature smoothing) with the superheated steam generator (Sh-SG) ones, as described in the following section 3. The effects of combining the off gas temperature smoothing and the latent heat extraction from PCM by a HTF are analysed with regards to steam parameters and the related performance of the downstream energy conversion system. Since the PCMCSG is the single component of the waste heat recovery section integrating different features (see the purple coloured part of Figure 1a), its sizing should rely on both PCMS and Sh-SG performances, thus further changing the design approach of waste heat to power energy systems. Integrated models are needed to develop and analyse the performance of the novel PCMCSG, as described in section 4.

3 THE PCM-COUPLED STEAM GENERATOR

The proposed new PCM-coupled steam generator is composed by three sections: the PCM Section, the Steam Generation Section, and the Auxiliary Section (see Figure 2).

The PCM Section (red coloured in Figure 2) is composed by a set of Heat Exchange and Storage Units (HESUs), which exchange thermal energy with the off gas and the auxiliary heat transfer fluid (HTF) during charging or discharging phases of the PCM. When the off gas enters the PCM Section, in facts, its temperature can be increased or decreased by the action of the HESUs. When off gas temperature is lower than the PCM melting point, in particular, each HESU releases latent heat from the PCM to the off gas increasing its temperature. On the contrary, when the off gas temperature is higher than the PCM melting point, the HESU absorbs heat from the off gas decreasing its temperature, by storing latent heat in the PCM. The combination of these two effects leads to smooth the off gas temperature, which tends to stabilise at the PCM phase transition temperature.

After the PCM Section, the off gas enters the Steam Generation Section (blue coloured in Figure 2), where it exchanges heat with the superheater, the evaporator, and the economizer, thus generating superheated steam. The Steam Generation Section is sized taking into account the maximum thermal power of the off gas exiting the PCM Section. The Auxiliary Section (purple coloured in Figure 2) connects the PCM Section and the Steam Generation Section by providing the additional heat supply required to achieve constant steam parameters at the turbine inlet. In order to obtain a constant mass flow rate of steam, an auxiliary evaporator located downstream the evaporator of the Steam Generation Section is employed. To achieve a constant temperature of the superheated steam, instead, an auxiliary superheater is installed downstream the superheater of the Steam Generation Section. Both the auxiliary evaporator and the auxiliary superheater are connected to the PCM Section by a closed loop circuit with the gaseous heat transfer fluid flowing through the HESUs.

As reported in [19], several systems for containing PCM employ cylindrical pipes [20,21]. The use of cylindrical pipes allows increasing heat exchange surfaces thereby optimizing the process both from the point of view of recovery efficiency and of cost, as well as preventing breakage and crack formation [14]. Based on the previous considerations, the Heat Exchange and Storage Unit (HESU) is composed by two coaxial pipes; the gap between the inner pipe and the external one is filled with the PCM (Figure 3). The external pipe is in direct contact with the off gas, whereas the inner pipe contains the heat transfer fluid feeding the heat exchangers of the Auxiliary Section. At the bottom of each HESU insulation is provided in order to dampen the generation of a thermal bridge between the external wall and the bottom.

Since all the HESUs within a given row (see Figure 3b) undergo the same off gas thermal load, the same thermal power can be extracted from each of them by the HTF. Thus, HESUs within the same row are connected in parallel by using a single inlet and outlet manifold (see Figure 3a), so that the same HTF control parameters can be applied to all of them.

The design of the PCM Section has to take into account the very high concentration of dust in off gas (typically 20 g/Nm^3 [15]), therefore a vertical arrangement of the HESU should be chosen in order to reduce dust accumulation on external pipe surface. Since dust accumulation among HESUs can create a barrier to off gas flow, an aligned layout of the HESUs (Figure 3b) should be preferred to a staggered one, as analysed in [14]. Furthermore, an easy dust removal can be enabled by hanging the HESUs to the upper wall of the waste heat boiler, as can be seen in Figure 3a. In order to minimize the manufacturing cost of the customized PCM Section, the geometry of the HESUs has been kept constant for all of them.

Concerning PCM selection, as underlined in [22] and [7], metals have greater potential as high temperature PCMs than molten salts, since they are generally characterised by higher thermal conductivity and smaller volume expansion. In particular, Al–Si alloys [23–25] represent the most promising high temperature PCMs for waste heat recovery from EAF, due to their suitable melting temperature, high latent heat of fusion and high thermal conductivity [26]. However, as pointed out by Fukahori et al. in [25, 27], Al–Si alloys have highly corrosive nature to metals such as steel based-materials. They investigated the use of ceramics as shell materials for high-temperature metallic PCMs and they found out that Al_2O_3 is the most suitable ceramic to employ, even though ceramics are quite fragile and expensive for industrial application.

A suitable alternative could be coating the steel walls in contact with the PCM with a thin film of Al_2O_3, which can be obtained by means of an aluminizing and oxidation treatment [28].

Given the above considerations, in this study alloy Al-12%Si (in mass %) has been chosen, since it presents one of the highest heat of fusion (560 kJ/kg) and thermal conductivity (160 W/m K at solid state) among Al-Si alloys. Furthermore, its temperature of fusion is about 576 °C, which is very close to the average temperature of the off gas emitted by an Electric Arc Furnace, thus enhancing the smoothing potential.

4 INTEGRATED MODELS FOR SYSTEM ANALYSIS AND OPTIMISATION

The optimisation of system configuration and management assuring the best performance of the novel PCM-coupled steam generation has required the development and integration of different models, which in turn rely on different software packages, as shown in Figure 4.

The first step to be faced is the process characterisation, which should generate a reference profile of the off gas temperature resembling the real behaviour in an electric arc furnace plant. In the second step, which leads to system sizing, the main components of the PCM-device are designed, the size of the steam boiler and the turbine are identified, and a proper HTF is selected. Once this stage is completed, the design of the whole system is verified and system operations are modelled and simulated, adopting a simplified approach, in order to assess the power production of different configurations. In the following, each modelling phase is analysed in more details.

4.1 Process characterization

In order to fully exploit waste heat, the most energy intensive cycle occurring during an EAF process should be identified as the reference cycle for the design of the related energy recovery system. Then, a generalisation of the reference cycle should be performed to create an off gas temperature profile characterised by the same highest temperature, total energy, and critical temperature gradient of the original reference cycle, which can be easily processed by models. In this study, a generalized off gas temperature profile has been generated from analyses carried out on several EAF processes; the mass flow rate of the off gas is assumed to be constant. Figure 5 compares the reference cycle (solid black line) to the generalised cycle (red dotted line).

4.2 System design modeling

The design of the whole PCMCSG involves a series of energy balances between the PCM Section and the Steam Generation Section, based on the temperature profile of the generalised cycle at the inlet of the PCM Section, as described in the following.

4.2.1 Steam Generation

The PCM Section should be sized in order to guarantee that the thermal energy stored in the PCM is able to completely smooth thermal power fluctuations when feeding the steam turbine. The reduction of the maximum off gas temperature exiting the PCM Section (see the smoothed profile of the green solid line in Figure 6a) is linked to the decrease of the maximum off gas thermal power entering the downstream Steam Generation Section. Thus, as reported in Figure 6b, the design point thermal power q_{des} of the Steam Generation Section is calculated as the difference between the maximum off gas thermal power exiting the PCM Section and the thermal power at the Steam Generation Section outlet. The latter is usually assigned by considering the minimum off gas temperature admissible at the stack T_{st} (see Table 1). From the design thermal power q_{des} the design mass flow rate $\dot{m}_{s,des}$ of the steam can be calculated as in Eq. (1):

$$\dot{m}_{s,des} = \frac{q_{\text{des}}}{h(T_{SH,des}, p_s) - h(T_{ECO,des}, p_s)} \tag{1}$$

where $T_{SH,des}$ and $T_{ECO,des}$ are the design temperatures of the superheated steam and feedwater, respectively, and p_s is the design pressure of the steam; Table 1 reports the values considered in our study. Given the design parameters of the steam cycle reported in Table 1, the distribution ratio χ of the design thermal power q_{des} among the three main heat exchangers of the Steam Generation Section (i.e. economizer - ECO, evaporator - EVA and superheater - SH) can be calculated as:

$$\chi_{ECO} = \frac{\dot{m}_{s,des} \cdot \left[h_{sat,l}(p_s) - h\left(T_{ECO,des}, p_s\right)\right]}{q_{\text{des}}} \tag{2}$$

$$\chi_{EVA} = \frac{\dot{m}_{s,des} \cdot \left[h_{sat,v}(p_s) - h_{sat,l}(p_s)\right]}{q_{\text{des}}} \tag{3}$$

$$\chi_{SH} = \frac{\dot{m}_{s,des} \cdot \left[h(T_{SH,des}, p_s) - h_{sat,v}(p_s)\right]}{q_{\text{des}}} \tag{4}$$

Where $h_{sat,l}(p_s)$ and $h_{sat,v}(p_s)$ are the enthalpies of the saturated liquid and saturated vapour of the steam at the design pressure p_s, respectively. The thermal power entering each heat exchanger can then be quantified as:

$$q_{ECO}(t) = \chi_{ECO} \cdot q_{SG,tot}(t) \tag{5}$$

$$q_{EVA}(t) = \chi_{EVA} \cdot q_{SG,tot}(t) \tag{6}$$

$$q_{SH}(t) = \chi_{ESH} \cdot q_{SG,tot}(t) \tag{7}$$

being $q_{SG,tot}(t)$ is the total thermal power entering the Steam Generation Section at any time t.

The purple area in Figure 6b represents the required auxiliary energy $E_{aux_{req}}$ to be provided by the Auxiliary Section to guarantee constant steam parameters at the turbine inlet. It can be calculated as shown in Eq. (8).

$$E_{aux_{req}} = \int_{0}^{TTT} q_{aux_{tot}}(t)\, dt \tag{8}$$

where the auxiliary thermal power $q_{aux_{tot}}(t)$ is the difference between the design point q_{des} and the off gas thermal power exiting the PCM Section at time *t*, to be accounted for a whole tap-to-tap cycle (TTT).

The auxiliary thermal power $q_{aux_{tot}}(t)$ provided to steam by the Auxiliary Section should be distributed between the auxiliary evaporator and the auxiliary superheater. The control systems that can be implemented are shown in Figure 7. In both the auxiliary heat exchangers, a fan is used to deliver the required mass flow rate of the heat transfer fluid. The control system of the auxiliary evaporator (see Figure 7a) measures the temperature of the HTF at points (6) and (7) and the actual mass flow rate of the steam $\dot{m}_s$ at point (5). The mass flow rate $\dot{m}_{F1}$ to be provided by the HTF fan 1 can then be calculated as

$$\dot{m}_{F1} = (\dot{m}_{s,des} - \dot{m}_s)\frac{h_{sat,v}(p_s) - h_{sat,l}(p_s)}{h(T_6,p_s) - h(T_7,p_s)} \tag{9}$$

In the control system of the auxiliary superheater (see Figure 7b), the temperature of the steam entering the heat exchanger at point (8), the temperature of the heat transfer fluid at points (10) and (11) can be used to calculate the mass flow rate $\dot{m}_{F2}$ to be provided by the HTF fan 2 as

$$\dot{m}_{F2} = \dot{m}_s \frac{h(T_{SH,des}, p_s) - h(T_8, p_s)}{h(T_{10}, p_s) - h(T_{11}, p_s)} \tag{10}$$

When the auxiliary thermal power required is null, the required mass flow is equal to zero and the fan is shut down.

4.2.2 PCM energy storage and release

In first instance, the heat exchange within the PCM Section can be characterized by considering the off gas temperature with respect to the PCM melting point. In particular, two important heat exchange phases can be identified: the hot phase, occurring when the off gas temperature is greater than the phase change temperature, and the cold phase, occurring when the off gas temperature is lower than the phase change one.

During the hot phase, a HESU absorbs and stores the thermal energy $E_{hot}(R)$ from the off gas, while during the cold phase it releases the thermal energy $E_{cold}(R)$ to the off gas. Both energies depend on the row R where each HESU is located, since the thermal load of the off gas decreases while moving from the first to the last rows of HESUs in the PCM Section. Figure 8a reports the profile of the heat transfer rate between the off gas and a HESU; the red area represents the hot phase energy, while the light blue area represents the cold phase energy.

The PCMSG design model involves the identification of the external diameter D for a HESU so that each HESU is able to store, in the form of the latent heat E_{sto}, the hot phase energy it can absorb during the generalised cycle. Moreover, the number of HESU rows n_R should be chosen in order to enable the whole PCM Section to store all the required auxiliary energy $E_{aux_{req}}$. For a given diameter, by increasing n_R, the available auxiliary energy $E_{aux_{av}}$ in the PCM Section increases, while the required auxiliary energy $E_{aux_{req}}$ decreases.

When the external diameter and the total number of rows balancing required and available energies have been selected, the thermal power to be provided to the auxiliary evaporator $q_{aux_{EVA}}$ and to the auxiliary superheater $q_{aux_{SH}}$ can be identified. The total number of rows of the PCM Section can be further split into the auxiliary evaporator rows $n_{R_{EVA}}$ and the auxiliary superheater rows $n_{R_{SH}}$.

Figure 8b shows the total thermal power exchanged by the HESU (black dash-dot line), which is the sum of the thermal power exchanged with the off gas (black solid line) and the heat transfer fluid (purple solid line). The total thermal power identifies the charging (orange area) and discharging (green area) phases occurring within each HESU. During the charging phase, the stored thermal energy within the HESU increases, while during the discharging phase it is released. While the thermal power exchanged with the off gas depends on the geometrical configuration of the HESU and the off gas thermodynamic properties, the thermal power exchanged with the heat transfer fluid can be controlled by varying HTF thermodynamic parameters (inlet temperature, velocity and pressure). Thus, the profile of the thermal power exchanged with the auxiliary HTF should guarantee that the energy stored by the HESUs during the charging phase is equal to the thermal energy to be released during the discharging one. The selection of the proper HTF involves managing the trade-off between minimising pressure drop and lowering circuit pressure.

4.2.3 Fortran implementation of the PCM-coupled steam generator design model

Based on the above considerations, the overall PCM coupled steam generator design model has been developed. The flow chart in Figure 9 summarizes the related algorithm, which has been implemented in Fortran 90, using the open-source integrated development environment Force [29]. The workflow encompasses three main iterative loops representing the three main steps of the design process, which are: 1) the identification of the external pipe diameter D of the HESUs (green coloured portion in Figure 8), 2) the determination of the number of HESUs' rows n_R (orange coloured portion in Figure 8), and 3) the selection of the heat transfer fluid (grey coloured portion in Figure 8).

In the first step, starting from the input design parameters (off gas temperatures and flow rates), by means of a first guess of the diameter D of a HESU, the cross-flow area of the whole system is calculated by taking into account the maximum inlet velocity of the off gas [30]. A maximum inlet velocity of 12 m/s has been chosen in order to limit the erosion effect due to the high dust content in the off gas. The choice of the inner pipe diameter d, instead, relies on acceptable thermo-structural stresses generated by the difference between the thermal linear expansion coefficients of PCM and tube material [31]. Given the geometric configuration and the PCM characteristics, the amount of energy storable E_{sto} as latent heat in each HESU can be easily calculated. At the same time, the HESUs geometry deeply affects the heat transfer between off gas and PCM. Since during normal operating conditions the PCM is kept in phase change at its melting temperature, thus behaving as a body of infinite heat capacity, both the PCM and the cylindrical outer surface of the external tubes can be assumed as isothermal at the PCM melting temperature. Accordingly, the problem can be dealt with as an external forced convection analysis in a tube bundle with constant temperature on the outer surface of the tubes. The heat transfer rate $q_{PtO}(t, R)$ from PCM to off gas (positive or negative) at time t and for row R (with R ranging from 1 to a first guess value of n_R) can be evaluated as

$$\mathrm{q}_{PtO}(t, R) = U_{PtO}\, A_{PtO} (\Delta T_{ml})_{PtO} \tag{11}$$

where U_{PtO} is the overall heat transfer coefficient (which basically corresponds to the convective heat transfer coefficient α_{PtO}), A_{PtO} is the heat exchange area, and $(\Delta T_{ml})_{PtO}$ is the log mean temperature difference between PCM and off gas. The value of α_{PtO} is calculated using the Grimson correlation for aligned tube bundles [32].

Off gas properties are estimated using an iterative procedure, where the outlet off gas temperature for each row is first guessed and then corrected using the heat balance for the off gas. The inlet temperature for a given row corresponds to the outlet temperature of the previous one. All the thermo-physical properties have been evaluated at the mean off gas temperature and operating pressure using the REFPROP 8.0 routines [33], conveniently interfaced with the Fortran code.

The thermal energy $E_{hot}(R)$ absorbed by a HESU during the hot phase in the row R is calculated according to Eq. (12), while the maximum thermal energy $E_{hot_{max}}$,which should be equal to the storable energy E_{sto}, is identified according to Eq. (13).

$$E_{hot}(R) = \sum_{t\,:\,q_{PtO}(i,R)\leq 0} q_{PtO}(t,R)\cdot\Delta t \tag{12}$$

$$E_{hot_{max}} = \max_{1\leq R\leq n_R}[E_{hot}(R)] \tag{13}$$

The maximum thermal energy $E_{hot_{max}}$ allows the further identification of the design reference row, which it corresponds to. When this condition is fulfilled, it is possible to move to the following second step of the design procedure.

At the beginning, the available auxiliary energy $E_{aux_{av}}$ is calculated as the integral over time of the heat transfer rates $q_{PtO}(t,R)$ for a first guess of the number of rows n_R. $E_{aux_{av}}$represents the maximum amount of energy the off gas is able to exchange with the PCM and should be equal to the energy $E_{aux_{req}}$ (see Eq.(8)) required by the Auxiliary Section in order to ensure the desired constant steam parameters. If the available auxiliary energy $E_{aux_{av}}$ is less than the required one $E_{aux_{req}}$, the number of rows should be increased and vice-versa.

The thermal powers to be provided to the auxiliary evaporator $q_{aux_{EVA}}$ and the auxiliary superheater $q_{aux_{SH}}$ can be estimated by means of the distribution ratios χ calculated in Eqs. (2)÷(4). In the proposed configuration of the PCM-coupled steam generator, the total auxiliary thermal power is distributed as follows:

$$q_{aux_{EVA}}(t) = (\chi_{ECO} + \chi_{EVA})\cdot q_{aux_{tot}}(t) \tag{14}$$

$$q_{aux_{SH}}(t) = \chi_{SH}\cdot q_{aux_{tot}}(t) \tag{15}$$

The total number of rows of the PCM Section can then be split into the number of rows $n_{R_{EVA}}$ and $n_{R_{SH}}$ devoted to the auxiliary evaporator and the superheater, respectively.
The third and final step of the overall model involves the selection of a proper gaseous HTF able to extract the desired amount of heat from the PCM at the right time with low pressure drops. For any suitable fluid, the velocities v at time *t* and for any row *R* can be inferred from the auxiliary thermal power $q_{aux}(t,R)$. The problem can be modelled as internal forced convection in a tube of inner diameter d with constant temperature equal to the PCM phase change temperature. Similarly, the heat transfer rate $q_{aux}(t,R)$ at the time *t* and for the row *R* can be expressed as:

$$q_{aux}(t,R) = U_{PtH}\, A_{PtH}(\Delta T_{ml})_{PtH} \tag{16}$$

where symbols have the same meaning as those in Eq. (11), but here they refer to the heat exchange between the PCM and the HTF.
Based on the value of q_{aux} and guessing the outlet temperature from the inner tube, the value of U_{PtH} , which essentially corresponds to the convective heat transfer coefficient α_{PtH}, can be easily calculated. From the value of α_{PtH}, using the Gnielinski formula [34], the HTF velocity v can be inferred through the Reynolds number. Iteratively, the heat balance for the duct is used to provide updated values of the tube outlet temperature. Pressure losses are finally calculated via the Petukhov correlation [35].
The thermo-physical properties of the HTF are evaluated at its mean temperature and operative pressure using the REFPROP software interfaced with the Fortran code. HTF velocities and their rates versus operative pressure dv/dp are analyzed together with pressure drops Δp in the row undergoing the maximum thermal load, in order to manage the trade-off between minimising pressure drop and lowering circuit pressure.

4.3 Design verification and System Operations Assessment

Once the PCM-Coupled Steam Generator has been properly sized as described in the sub-section 4.2, the behaviour of the whole waste heat recovery system needs to be verified.
Firstly, the PCMCSG design should be verified, which means ensuring that the configuration of PCM Section, obtained at the end of the optimisation process (sec. 4.2.2), enables the Auxiliary Section (sec. 4.2.1) to completely smooth thermal power fluctuations. Furthermore, the modelling of the whole energy recovery system should allow the comparison of the power production of the proposed system and the traditional one.

The energy recovery system can be modelled by means of ad-hoc codes implemented in Matlab/Simulink such as in [36,37]. Alternatively, the energy recovery can be simulated by means of commercial software such as: the advanced processing simulation software Aspen Plus Dynamics [38]; Dymola, which is a commercial modelling and simulating environment based on the Modelica language [39,40] and the Advanced Process Simulation Software APROS [41]. Since this study involves innovative plant components, thus characterised by intrinsic uncertainty and lack of reliable data for deep and complex analyses, a simplified model has been considered as adequate for our aims. The PCM-Coupled Steam Generator enables the steam turbine to work almost at a fixed point (fixed inlet steam flow, pressure and temperature) and, at this stage, we were not interested in startup procedures, strong irregular and discontinuous operations or unexpected load changes. Furthermore, during normal operating conditions, the PCM is kept in phase change at its melting temperature, thus speeding up the dynamic response of heat exchangers. For all the above reasons, the developed model is approximated: no dynamic effects are taken into account except for temperatures. In the steam cycle, heat exchangers are considered as a simple heating load imposed on a flow of steam and the steam condensed pump power is inferred from a compressed liquid calculation taking into account an overall pump efficiency. Steam mixing valves and steam drums are governed by an overall energy balance (assuming that each of the inlet streams adiabatically expands to the pressure of the lowest pressure inlet fluid). The steam turbine simulation relies on an isentropic efficiency approach to calculate the performance given the steam inlet conditions and the turbine back-pressure (which are, in our plant, constant in time). Specific PID controls, described in the following, are added to the model when needed.

Two models for operations of the waste heat to power system based on steam generation have been developed by TRNSYS 17.0 package [42], which provides a high level programming language and an easy to use graphical interface..

The first model simulates a traditional Sh-SG with and without a PCM-based smoothing-only device (PCMS) installed upstream (see Figure10). Such condition is simulated applying two different off gas thermal power profiles to the Sh-SG: the reference profile and the smoothed profile due to PCMS insertion. The traditional Sh-SG system is controlled by means of two PID controllers: the former controls the mass flow rate of the feed pump in order to maintain a constant temperature of the superheated steam entering the turbine, the latter controls the heat flow provided to the condenser in order to guarantee constant thermodynamic conditions at the extraction pump. The design of the PCMS refers to the technical solutions described in [8], where a heat transfer fluid is used to control the PCM temperature in order to overcome overheating issues in PCM containers. The PCMS has been sized so that the same smoothing effect (i.e. same thermal power fluctuation) of the PCM Section of the novel system is achieved. Therefore, the geometry and the layout of the PCMS are the same of the PCM Section of the PCM-coupled steam generator.

The output files generated by the Fortran code, concerning heat loads at the heat exchangers, are used as input in the TRNSYS simulation (see Figure 11). In the traditional Sh-SG model, a control loop is implemented in order to provide a constant steam temperature at the turbine inlet by varying the flow rate of the feed pump.

The flow rate of the feed pump in the PCMCSG is, instead, fixed at the design value; the control on the steam parameters is obtained by providing the auxiliary evaporator and the auxiliary superheater with the auxiliary thermal powers from the PCM Section. In both models, the turbine power calculation relies on an isentropic efficiency, which is a common approach in literature (e.g. [43,44]) if the maps of the specific component are not available. The performances of the turbine at off-design steam mass flow rates have been modelled by using an efficiency curve for partial loads, which has been created thanks to interactions with a local supplier (see Figure 12).

The part-load efficiency curve represents the isentropic efficiency ratio $\epsilon = \frac{\eta_{is,a}}{\eta_{is,d}}$ (where $\eta_{is,d}$ and $\eta_{is,a}$ are the design and actual isentropic efficiency, respectively) as a function of the mass flow ratio $\chi = \frac{\dot{m}_a}{\dot{m}_d}$, where $\dot{m}_a$ and $\dot{m}_d$ are the actual and design mass flow rate of the steam, respectively. Typically, steam turbines are designed to work at mass flow ratios higher than 0.4, since lower values lead to a drastic reduction of the turbine performance and a significant increase of the maintenance costs. In this study, it has been assumed that the steam turbine can work also in the low efficiency zone in order to highlight the effects of thermal power fluctuations.

5 RESULTS AND DISCUSSION

The off gas temperature profile at the PCM inlet section adopted to optimise the proposed novel system is represented by the red dotted line in Figure 6a and has been derived by analysing several EAF plants. The flow rate is equal to 130,000 Nm^3/h. Al-12%Si has been used as PCM material, with a phase change temperature of about 576 °C.

Table 2 reports the optimal geometric configuration of the PCM Section, provided by the Fortran code; it consists of 41 rows of HESUs, with external *D* and inner *d* pipe diameters equal to 128 mm and 88 mm, respectively.

In Figure 13a, the PCM state evolution over time is shown for three specific HESU rows: the first one, the reference one, namely the row undergoing the maximum thermal load, and the last one facing the minimum thermal load. It is worth noting how the reference row is not the first one, because in the first rows of a bank of tubes the turbulent flow is not completely developed. In this paper, the correlation for aligned tube bundles proposed by Grimson [32] has been used. As expected, the PCM state goes from completely liquid to completely solid only in the reference row, proving the effectiveness of the sizing procedure.

Concerning HTF selection, three different fluids have been investigated: air, for its low cost and large availability, carbon dioxide, which is an inert gas with high specific thermal capacity, and argon, since it is commonly used in steel industry. Different values of the operating pressure *p* in the range between 0.1 and 1 MPa have been analysed. Small variations of the HTF velocity, and of pressure drops consequently, are gained by increasing operative pressure over 0.4 MPa. Therefore, the latter has been set as the reasonable operating condition, in order to lower the cost of fans and their energy consumption. Figure 13b shows that the lowest velocities are achieved when using carbon dioxide; this means that the use of the CO_2 guarantees to obtain the lowest pressure drop. In addition to the lowest pressure drop, it is worth noting that carbon dioxide allows improving system safety in comparison to air, which may lead to safety issues (i.e. fire risk) due to its oxygen content. For these reasons, the carbon dioxide is considered to be the best choice among the considered HTF.

In Table 3the main performances of the PCM-Section are summarised. Thermal fluctuation decreases from 32.4 % to 12.7 %; the maximum thermal power of the off gas exiting the PCM Section is reduced by 34.4 %.

Figure 14a reports the steam temperature profile at the outlet of the PCMSG components; a constant temperature of about 400°C is achieved at the outlet of the auxiliary superheater, which demonstrates that the Auxiliary Section completely smooths the thermal power fluctuation entering the steam turbine. It can be noted that at the outlet of the components of the Steam Generation Section a temperature fluctuation still exists. This is due to the fact that after the PCM-Section the thermal fluctuation of the off-gas is reduced but not completely smoothed. Thus, all the components of the Steam Generation Section (i.e. economizer, evaporator and superheater) undergo a variable thermal load. In the economizer and the superheater this fluctuating thermal load is converted in a fluctuation of the steam temperature, while in the evaporator it is converted in a fluctuation of the steam quality.

Figure 14b shows the profile of the electric power production for the three waste heat to power systems simulated in TRNSYS: the traditional steam generator with and without the PCMS, and the PCMCSG. It can be noted that the WHTP system employing a traditional Sh-SG without a PCMS has a huge fluctuation in electric power production (near-zero electric power production is reached), much larger than the fluctuation in the off gas thermal profile. This behaviour is due to the intrinsic limits of the turbine, whose performance drastically falls down when a too low steam flow rate is provided (see Figure 12).

Table 4 compares the performance of the three possible configurations of the WHTP system. The introduction of a PCM-based device reduces the size of the waste heat recovery components (steam generation boiler and turbine) of about 41%. However, when a PCM-based device is used for smoothing only, the production of electric energy undergoes a slight reduction of about 2% with respect to the traditional configuration. This reduction can be ascribed to the working principle of the PCMS, which dissipates part of the recovered energy to control the temperature of the PCM and avoid overheating. When the PCM-coupled steam generator is adopted, the electric production increases of 22% with respect to the traditional configuration. These results are achieved thanks to the reduced fluctuation in the steam parameters at the turbine inlet, which leads to a greater overall efficiency of the turbine.

Table 5 reports the cost-benefit analysis of the PCM-coupled steam generator. Investments have been estimated by interactions with local suppliers of each system component. As concerns operational costs, maintenance costs are set as a percentage of investment cost on the basis of the common practice for each component as suggested by potential service suppliers: for the novel PCM section a percentage significantly higher than mature component one has been cautiously adopted to take into account uncertainty.

An electrical energy generation of 48,166 MWh/year can be estimated by assuming that the EAF operates 7500 h/year and the WHTP system availability is 75%. Considering an energy supply cost of 50 €/MWh_e, currently granted to the steel industry by the Italian energy market, a cost saving of about 2,408,290 €/year can be achieved. Furthermore, a positive cash flow of 3,631,349 € can be estimated for the first 5 years taking into account an up-to-date value of 120 €/TOE [45] of the Italian energy efficiency certificates. Thus, a pay-back period of about 3 years can be expected, which underlines the profitability of the proposed novel WHTP system.

6 CONCLUSIONS

In this paper, the evolution of current waste heat to power systems for the steel industry towards the production of high temperature superheated steam with limited variability is enabled by coupling PCM-based heat extraction with steam generation. In particular, the use of high temperature PCMs evolves from smoothing off gas temperature only, as in previous studies [8,14] , to generating constant superheated steam able to feed the downstream turbine nearly at nominal load. This result is achieved by introducing an auxiliary section between the PCM Section and the Steam Generation one, which provides the auxiliary heat needed to further level the thermal content of the off gas. The auxiliary heat is extracted from the PCM units by a gaseous heat transfer fluid flowing across the inner tube of each PCM container.

Different models to properly size the proposed energy recovery system and simulate its operations have been developed and integrated, adopting both commercial tools and ad hoc elaborated codes.

Results show how the new system is able to prevent the steam turbine from working at partial loads, thus drastically increasing its efficiency. Furthermore, the size of the steam generator and of the turbine, required to exploit the same amount of waste heat, can be significantly reduced in comparison to current solutions with benefits on related investments.

The proposed PCM-coupled steam generation can actually trigger effective waste heat recovery in such an energy intensive sector as the steel industry, thus fostering the evolution towards more sustainable industrial processes.

Future developments of the present research will be the build up of a more accurate model of the whole waste heat recovery system, including the dynamics of each component, and the development of a system control strategy which enables to smooth off the power production even for off-design thermal load conditions.

NOMENCLATURE

Acronyms

Aux	Auxiliary
CON	Condenser
EAF	Electric Arc Furnace
ECO	Economizer
ECT	Energy Conversion Technology
EVA	Evaporator
ExPump	Exctaction pump
HESU	Heat Exchange and Storage Unit
HTF	Heat Transfer Fluid
HWB	Hot Water Boiler
HWT	Hot Water Tank
ORC	Organic Rankine Cycle
PCM	Phase Change Material
PCMCSG	PCM Coupled Steam Generator
PCMS	PCM-based Smoothing device
RSA	Ruth's Steam Accumulators
Sat-SG	Saturated Steam Generator
Sat-ST	Saturated Steam Turbine
SH	Superheater
Sh-SG	Superheated Steam Generator
Sh-ST	Superheated Steam Turbine
TESS	Thermal Energy Storage System
WHR	Waste Heat Recovery
WHRB	Waste Heat Recovery Boiler
WHTP	Waste Heat To Power

Latin letters

A	heat exchange area [m^2]
D	outer diameter [m]
d	inner diameter [m]
E	thermal energy [J]
H	chamber height [m]
h	enthalpy [kJ/kg]
L	chamber length [m]
$\dot{m}$	mass flow rate [kg/s]
n_R	total number of rows
p	pressure [Pa]
q	heat transfer rate [W]
R	row number
S	pitch [m]
T	temperature [°C]
t	time [s]
U	overall heat transfer coefficient [W/m^2 K]
v	velocity [m/s]
W	chamber width [m]

Greek letters

α	convection heat transfer coefficient [W/m^2 K]
Δ	properties difference
χ	thermal power distribution ratio

Subscripts

aux	auxiliary
av	available
cold	cold phase
des	design point
ECO	economizer
EVA	evaporator
F	Fan
hot	hot phase
L	longitudinal
l	liquid
ml	mean log
PtH	PCM to HTF
PtO	PCM to off-gas
req	requested
s	steam
set	set point
SH	superheater
st	stack
sto	storable
T	transversal
TTT	tap to tap
v	vapour

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