DISTRICT HEATING BY RADIANT HEAT RECOVERY FROM CEMENT KILNS

Antonio C. Caputo, caputo@ing.univaq.it
Mario Palumbo, palumbo@ing.univaq.it
Pacifico M. Pelagagge, pelmar@ing.univaq.it
Paolo Salini, salini@ing.univaq.it
Department of Mechanical, Energy and Management Engineering, University of L'Aquila, Monteluco, L'Aquila, Italy

Abstract. *Heat loss from rotary kilns may represent a significant percentage of the total energy input especially in high energy-intensive industrial sectors such as cement production. In this paper the technical and economic feasibility of recovering radiant heat lost through the kiln surface, by means of a secondary external shell acting as a heat exchanger for a transfer fluid, is evaluated for district heating purposes. At first the system architecture is outlined and a technical and economical model addressing both the performances and cost estimation for the heat exchanger and the district heating network is developed. Subsequently, a parametric profitability analysis is carried out with reference to some relevant parameters characterizing the available recoverable waste heat and the size of the heat distribution network, namely the distance between kiln and user area and extension of the district heating network. This is made to obtain a mapping of the conditions were the proposed heat recovery system is economically feasible. In the paper it is demonstrated that the relevant heat consumption of cement production may make the district heating option for heat recovery a viable one even in case of low density of inhabitants in the surroundings of the plant. Furthermore significant fuel savings and emission reductions are achieved respect the adoption of traditional residential boilers.*

Keywords*: heat recovery, cement kiln, district heating, feasibility study.*

1. INTRODUCTION

District Heating (DH) is a practice in which the thermal energy produced in a centralized facility is distributed to remote users by means of a proper energy carrier fluid. DH systems may deliver heat to both residential users (for household heating), public facilities (i.e. hospitals) and industrial districts for process requirements.

DH allows large energy savings as a centralized heat generation plant shows higher energy conversion efficiency (80-85%) respect the alternative of many smaller-sized and lower efficiency (60-65%) distributed household boilers. Another advantage of DH is the possibility of feeding the heat distribution network by means of a multi-fuel combustor, enabling to use the most economical fuel available at the time, or exploiting non traditional heat sources, such as refuse incineration, geothermal energy, and waste from industrial processes. Combined Heat and Power (CHP) generation makes DH systems even more convenient, allowing global efficiencies greater than 90% (Diamant, 1970). Furthermore, DH systems provide environmental benefits as the control and limitation of pollutant emissions in a large combustion plant is easier and more effective than in many small sized boilers. On the other hand, DH requires large investments, due to the construction of the centralized facility and the installation of a distribution network. Therefore, though the convenience from social and environmental point of views is granted, the economical profitability will have to be assessed in a site specific manner.

Among industrial processes, the Portland cement production is one of the most energy intensive (Bye, 1999) with a thermal energy consumption of 3100 to 7300 MJ/t, while electric energy consumption is about 147 KWh/t. As a consequence, a number of energy saving measures, like recovering sensible heat from fumes and from product cooling, have been proposed in the literature and widely applied (Engin and Ari, 2005; Ghosh and Yadav, 1996; Weinert, 1990; Worrel and Galitsky, 2004; Worrel et al., 2000). Some architectures of cement production plants even allow for the installation of CHP systems which may dramatically enhance their overall energy efficiency (Khurana et al., 2002), while the use of cement plants for DH purposes has been also proposed (Rinaldi, 1994).

However, an interesting opportunity, although rarely practiced, is the recovery of thermal energy wasted from the rotary kiln external surface by heat transfer to the surrounding environment. In fact, the kiln external surface easily reaches temperatures in the order of 400°C. Due to the large dimensions of these kilns (the length may reach 150 m, and diameter 5-6 m), the losses from kiln walls are very high, about 8-15% of the total heat input. One manner to reduce this heat loss is to add a secondary insulating shell to the original kiln surface as suggested by Engin and Ari (2005). However, more interesting is the adoption of external heat recovery exchangers which can be applied around the kiln and arranged in order to form a secondary shell. Weinert (1990) was one of the first authors to suggest such a practice also discussing a prototype application. A number of different embodiments of this concept have been instead presented by Vorebechikove (1996). Weinert (1990) also reported that the addition of the outer shell determined an increase of kiln shell outer temperature from 290 °C to 340 °C. This circumstance might also imply that such an energy recovery measure can contribute to a saving in kiln fuel consumption. In this paper the technical and economic feasibility of recovering radiant heat lost through the kiln surface, by means of a secondary external shell acting as a heat exchanger for a transfer fluid, is evaluated for district heating purposes. At first the system architecture is outlined and a technical and economical model addressing both the performances and cost estimation for the heat exchanger and the district heating network is developed. Subsequently, a parametric profitability analysis is carried out with reference to some relevant parameters characterizing the available recoverable waste heat and the size of the heat distribution network, namely the distance between kiln and user area and extension of the district heating network. This is made to obtain a mapping of the conditions were the proposed heat recovery system is economically feasible.

2. TECHNICAL MODEL OF THE HEAT RECOVERY AND DISTRICT HEATING SYSTEM

In the previous section it was pointed out that a major source of waste heat is the rotary kiln surface exchanging heat with the surrounding air. Heat losses to the environment from the kiln surface may be expressed, per unit mass of clinker passing through, by means of the following equation (AITEC, 1964 and 1970)

$$Q_S = \frac{A_{tot} \quad S \quad (T_w - T_a)}{M} \tag{1}$$

where Q_S (kcal/kg$_{clinker}$) is the heat lost per unit mass of clinker, A_{tot} is the overall heat transfer coefficient (kcal/h m^2 °C), S is the external surface area of kiln shell (m^2), T_w is the surface temperature of the kiln (°C), T_a is the absolute mean temperature of environment (°C), M is the mass flow of clinker in the kiln (kg$_{clinker}$/h). Equation (1) is valid for a shell at uniform temperature. In the real situation this temperature assumes decreasing values from kiln burner end to the combustion gases exit end. The overall heat transfer coefficient may be evaluated as the sum of radiative transfer coefficient (A_{rad}), natural convective transfer coefficient (A_{nc}) and forced convective transfer coefficient (A_{fc}) (all expressed as Kcal/h °C m^2).

$$\boldsymbol{A_{tot} = A_{rad} + A_{nc} + A_{fc}} \tag{2}$$

$$A_{rad} = C \quad \frac{(T_w/100)^4 - (T_a/100)^4}{T_w - T_a} \tag{3}$$

$$A_{nc} = 80{,}33 \quad T_m^{-0{,}724} \quad (T_w - T_a)^{0{,}333} \tag{4}$$

$$A_{fc} = 28{,}03 \quad T_m^{-0{,}351} \quad V^{0{,}805} \quad D_K^{-0{,}195} \tag{5}$$

where C is the surface emissivity factor depending on the physical nature of the radiating body, which in this case is assumed equal to 4 kcal/h °C^4 m^2 (surface not coloured, with dusty shell), T_m is the absolute mean temperature between radiating surface and environment temperature (°C), V is the mean velocity of air flow (m/s), and D_K is the outside diameter of kiln shell (m). Equations (1) to (5) are empirically derived and are here reported adopting the formulation and measurement units of the original source (AITEC, 1964 and 1970). An order of magnitude estimate of the available recoverable heat is thus straightforward to carry out, in order to verify whether in the cement plant surroundings potential users suitable for a district heating application exist.

Although different configurations for the kiln heat recovery equipment may be conceived, here it is assumed that a tubular heat exchanger is utilized. As an example the heat exchanger may be configured as an array of pressurized water carrying tubes arranged in longitudinal pattern on the surface of a cylindrical outer shell coaxial with the rotary kiln (Fig.1) and located in the region where the wall temperature is the highest.

The overall architecture of the considered DH plant is shown instead in Fig. 2, where the kiln heat recovery exchanger (HRE), here indicated as HE1, is connected to the DH network by means of the interposition of a secondary heat exchanger HE2. Superheated water is circulated in the HRE allowing for the recovery of a part of dissipated power at temperatures in excess of 100 °C. The function of HE2 instead is to decouple the heat recovery section from the users side to avoid sudden temperature variations of the kiln surface due to the change or interruption of flow in the recovery exchanger. In case of flow interruption (or recirculation of hot water inside the recovery exchanger), in fact, the temperature inside the exchanger would rapidly increase, leading to the rise of temperature of kiln shell, thus upsetting the cement production process. As far as the district heating network is concerned, the selection of energy carrier fluid is important because it influences system regulation, type of user substation, materials, investment and operating costs. The characteristics of the examined plant rule out the use of steam. This fluid, in fact, is used at temperatures and pressures not compatible with the available source and, furthermore, would require higher costs which would make not profitable the investment. Cost and possible environmental impact concerns also rule out diathermic fluids. The utilization of hot water has been thus assumed. Heat is delivered to users by the so called “indirect connection”, where the transfer fluid is not utilized in the users' equipment but exchanges heat with a secondary transfer fluid in a final exchanger at the user location. Indirect connection has the advantage of a simpler regulation, which can be carried out acting both on temperature and flow.

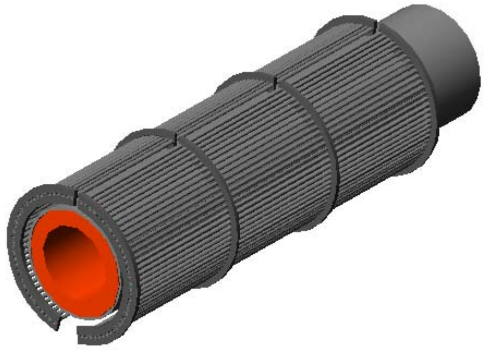

Figure 1. Radiant heat recovery exchanger (axial tubes arrangement).

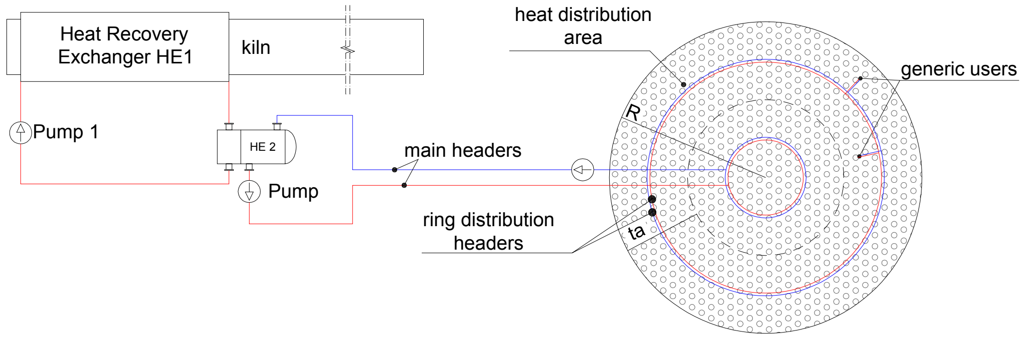

Figure 2. Scheme of the heat recovery and district heating system.

Operating conditions of DH networks generally require that the maximum water temperature leaving the HE2 exchanger is 95°C while the inlet water temperature will be around 55°C. For sake of simplicity it is assumed that the heat distribution area is circular with a uniform density of users and that within this area the distribution network headers are arranged in a concentric rings layout. Each user then connects to the nearest ring header through a separate pipe branch. A main duct conveys the water flow rate between the distribution rings and the heat recovery plant.

While the secondary heat exchanger HE2 is a traditional equipment (i.e. a shell and tube design for instance) which can be sized utilizing traditional procedures, the kiln HRE (HE1) is a non-traditional unit requiring a specialized approach. In a previous paper (Caputo et al., 2008) a detailed computer model was developed for sizing and performance modelling of tubular HRE in rotary kilns having the configuration shown in Fig. 1. In the model the heat transfer surfaces arrangement is schematised as that of two coaxial cylinders having infinite length. This allows to consider unitary view factors between the two surfaces. Heat transfer from tubes to the external environment is assumed to be zero. These simplifying assumption are valid if the tubes array circumference has a diameter similar to that of the kiln shell and if the external tubes surface is insulated. Furthermore, the kiln wall temperature is assumed to be constant. The model allows to compute water and air temperature profiles along the exchanger length for a defined equipment geometry, kiln wall temperature, and water flow rate, thus enabling to compute the recovered heat for a given equipment size or to define the required exchanger length if the amount of recovered heat is imposed. The overall thermal power subtracted to the kiln is the sum of radiation heat transfer to exchanger tubes and the convective power transferred by kiln to air, being both of the same order of magnitude. However, as detailed by Caputo et al. (2008), a complex convective and radiant exchange interplay occurs between water flowing in the exchanger tubes, kiln walls and the air stream flowing in the annulus between the coaxial cylinders, so that the thermal power dissipated by the kiln walls can not be entirely recovered by the water stream. In fact, while air and water temperature increase when proceeding from exchanger inlet to the outlet section, both the convective and radiative power exchanged with the kiln walls decrease owing to a reduction of the driving force. This means that a progressive reduction of the overall heat transfer along the kiln length occurs and that the maximum thermal power absorption takes place at the inlet section, while subsequent sections exchange gradually lower heat flows. Nevertheless, the thermal power transferred to the water stream, which is the useful effect (i.e. the recovered heat) is mainly constituted of the kiln radiated power, and on a much lower percentage, if any, of the convective heat transferred from the air stream to the water stream (in the exchanger sections where air temperature is lower than water temperature, the air stream actually penalizes heat recovery because the heated water transfers back heat to the colder air stream). It should be pointed out that the amount of heat that the HRE can extract from the kiln surface depends from its design and operational parameters and in particular from the water inlet temperature and flow rate. Therefore, when high water flow rates and/or low inlet temperatures are adopted, it could happen that the HRE subtracts from the kiln a greater thermal power than that dissipated by the bare kiln in the ambient air. From a heat recovery perspective this is not acceptable as it would imply that the additional heat input should be supplied by an increased fuel consumption if the temperature level within the kiln is to be maintained and the wall temperature is not to be lowered. Therefore, the HRE designer is constrained by the requirement that the HRE subtracts form the kiln at most the same power that the kiln would dissipate in the ambient air when the HRE is not applied. Furthermore, while this constraint is to be satisfied in global terms, it is also to be satisfied locally in any kiln section. This imposes an even more restrictive upper bound to the amount of recoverable thermal power which, in practice, is further reduced respect the thermal power dissipated by the kiln. Finally, it was also demonstrated that a cocurrent arrangement should be preferred to a countercurrent one, even if the latter is more efficient, as it reduces the heat uptake from the air stream and lowers the average water temperature thus increasing the radiant exchange driving force.

In order to show the order of magnitude of the available recovered heat respect the power dissipated by a bare kiln, and to exemplify the output of the computerized design tool, Fig. 3 shows the plots of transferred and recovered heat as a function of exchanger length and kiln wall temperature (ranging from 250 °C to 400 °C).

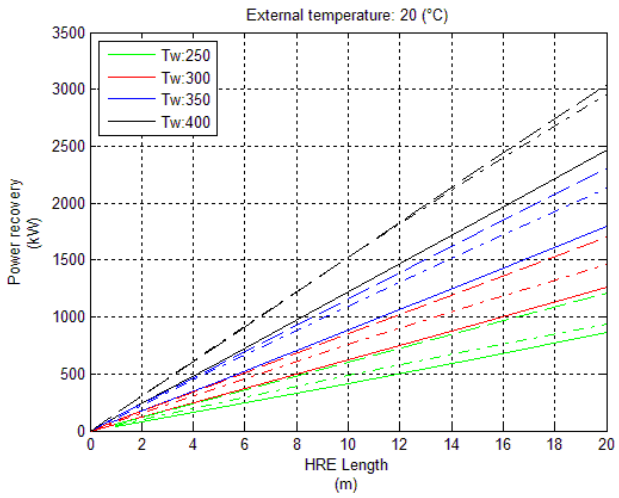

Figure 3. Transferred and recovered heat
Legend: solid lines = net recovered heat; dash-dot lines = thermal power subtracted to the kiln (by both water and air streams); dashed lines = heat dissipated in ambient air by a bare kiln.

Computations were made assuming an ambient air temperature of 20°C, an exchanger/kiln diameter ratio of 1.25, and a kiln diameter of 4 m. Such values are representative of typical cement plant conditions. From the figure it can be observed, for instance, that at 400 °C wall temperature, a 12 m long exchanger is required if a net recovery of 1.5 MW is desired. While the recovered heat falls to 0.5 MW if the same size exchanger is utilized with a kiln having a 250 °C wall temperature.The kiln wall temperature, the selected values for the geometrical parameters of the HRE, the imposed water flow rate in the primary circuit and the inlet temperature $T_{in,HE1}$ in the HRE allow the computer model to compute the outlet temperature $T_{out,HE1}$ and the recovered heat P_{rec} (kW).

$$P_{rec} = m_1 \quad C_p \quad \left(T_{in,HE1} - T_{out,HE1}\right) \tag{6}$$

The temperature requirements of the users in the secondary circuit determine instead the water flow rate in the secondary circuit through the following energy balance, where m_1 and m_2 are the water flow rates (kg/s) in the primary and secondary circuit respectively, while C_p is the constant pressure specific heat (kJ/kg K). $T_{in,HE1}$, $T_{out,HE1}$, $T_{in,HE2}$, $T_{out,HE2}$, are the inlet and outlet temperatures of exchangers HE1 and HE2.

$$m_2 = \frac{P_{rec}}{C_p \quad \left(T_{out,HE2} - T_{in,HE2}\right)} \tag{7}$$

In a similar manner, by defining the inlet and outlet temperature levels (T_{inUs}, T_{outUs}) and thermal power (P_{Us}) requested by the users in the secondary circuit (a typical value for domestic users is P_{Us} = 15 kW) the water flow rate requirement of each user follows

$$m_{Us} = \frac{P_{Us}}{C_p \quad (T_{inUs} - T_{outUS})} \tag{8}$$

It should be noted that heat losses along the distribution network make the users inlet and outlet temperatures different from the secondary exchanger inlet and outlet temperatures.

The surface area of the secondary heat exchanger S_{HE2} (m^2) is computed as

$$S_{HE2} = \frac{P_{rec}}{U \quad MLTD \quad Ft} \tag{9}$$

being U the overall heat transfer coefficient (for water-water exchangers is about U=1kW/m^2 K), while MLTD is the mean log temperature difference and Ft the temperature difference corrective factor.

The maximum number of users that can be fed by the DH network is then computed as

$$N_{Us} = \frac{m_2}{m_{Us}} \tag{10}$$

The users can be assumed as uniformly distributed with a surface density ρ_{Us} (user/km^2) within a circular geographical area of radius R (km)

$$\rho_{Us} = \frac{N_{Us}}{\pi \cdot R^2} \tag{11}$$

As the urbanized area is served by a network of concentric pipes, it is assumed to be subdivided into a predefined number N_a of annular areas where each j-th annulus, having surface area A_j (km^2) includes a number of users $N_{us,j}$ and requires an overall water flow rate m_j.

$$N_{Us,j} = \rho_{Us} \cdot A_j \tag{12}$$

$$m_j = N_{Us,j} \cdot m_{Us} \tag{13}$$

Each distribution header ring is located in the centreline of the annulus it serves and it is made up of two distinct pipes, one carrying hot water to the user and the other recovering water to the header. Each header is fitted with a circulation pump. Each branch feeding a single user from the local distribution ring includes two pipes and has an average length equal to one half of the radial width t_a of the annulus. Even the main duct connecting the heat recovery plant to the distribution network is made up of two distinct pipes each fitted with a separate pump. Knowledge of the water flow rate in the various pipe branches and headers allows to size the pipes once a delivery pressure level is selected.

3. ECONOMIC MODEL

3.1. Capital investment estimation

The capital investment of the HRE (CI_{HRE}) includes the supporting structure cost (CI_S) and the tubes cost (CI_T) and has been modeled on the basis of equipment weight.

$$CI_{HRE} = CI_S + CI_T \tag{14}$$

$$CI_S = Ns \quad \frac{D_K \quad t_s \quad d_s \quad \alpha}{2} \quad \rho \quad CS \tag{15}$$

$$Ns = \max\left(2;\left(\frac{L}{Sp}+1\right)\right) \tag{16}$$

$$CI_T = Nt \quad v \quad L \quad \rho \quad Cu \tag{17}$$

$$Nt = \frac{D_{HRE}}{D_T \quad (1+l)} \quad \frac{\alpha}{2} \tag{18}$$

In Eq. 15 to 18 Ns is the number of tubes support plates, t_s and d_s are respectively the support plates thickness (m) and width (m), CS the steel cost (€/kg), while α is the geometric angle represented in Fig. 4. The HRE, in fact, has been conceived as composed of two separate symmetric half-shells allowing partial or total removal of the tubes banks from the kiln walls. L is the HRE length (m), Nt the tubes number, v the tube material volume per unit length (m^3/m), ρ the steel density (kg/m^3), Cu the tube cost per unit weight (€/kg), Sp the support plates spacing (m), D_{HRE} the HRE shell diameter (m), D_T the tubes diameter (m) and *l* the tubes percent spacing referred to D_T.

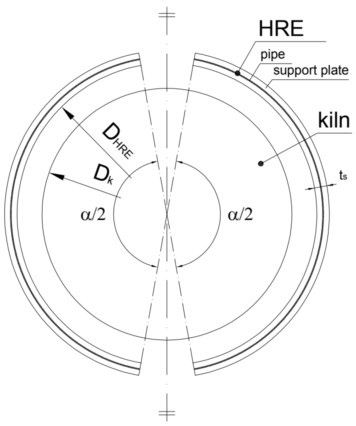

Figure 4. Cross section schematization of the exchanger.

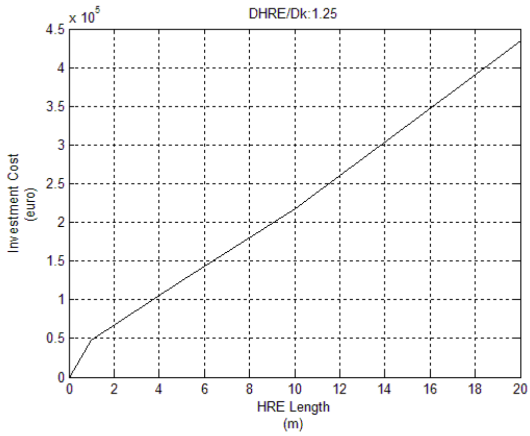

Figure 5. Recovery exchanger capital investment.

The CI_{HRE} cost function is plotted in Fig. 5 as a function of exchanger length in case the following parameters values are assumed: D_K = 4 m, D_{HRE} = 5 m, Sp =3 m, t_s = 0.1 m, d_s = 0.1 m, α = 320°, ρ = 8000 kg/m^3, D_T = 20 mm, *l* = 0.3, CS = Cu = 10 €/kg. The capital investment of the secondary exchanger is instead computed simply as a function of the exchanger surface adopting Hall's correlation (Taal, et al., 2003) valid for exchangers made with stainless steel for both shell and tubes.

$$C_{HE2} = 10000 + 324 \quad S_{HE2}{}^{0.91} \tag{19}$$

Capital investment of the distribution network basically includes the piping cost and the excavation cost for pipes burial. The overall cost of the terminal branches connecting the end user to the local *j-th* ring distribution header in each annular area is

$$C_{c,j} = \left(\frac{t_a}{2} \quad Cu \quad w_p + \frac{C_0}{2}\right) \; 2 N_{Usj} \tag{20}$$

where w_p is the pipe weight per unit length (kg/m) and C_0 the fixed user connection cost. Each *j-th* ring header investment cost CI_{Rj} is

$$CI_{Rj} = L_{rj} \quad w_p \quad Cu \tag{21}$$

where L_{rj} is the *j-th* circular header length (m). The investment cost CI_{MH} of the main headers conveying water to and from the secondary heat exchanger is computed in a similar manner changing L_{rj} to the delivery pipe length L_D between kiln and distribution area. However, the specific pipe weight changes according to pipe diameter and thickness. Assuming that pipes are buried at depth d_{Trench} = 1 m within a trench having a width 30% greater than their diameter, the excavation volume can be computed. For instance in case of each ring header the trench volume is

$$V_{trench} = L_{rj}(2 D_T \; 1.3) d_{Trench} \tag{22}$$

and naming Ct the specific excavation cost (€/m^3) the total excavation cost is

$$C_{trench} = Ct \cdot V_{trench} \tag{23}$$

The same holds for the terminal branches and the main duct. The overall investment cost CI_{tot} (€) includes the main header, the ring distribution headers, the user branch connections, the ring headers circulation pumps and the primary and secondary exchangers

$$CI_{tot} = 2 \quad \sum_{j=1}^{Na} CI_{Rj} + 2 \quad \sum_{j=1}^{Na} CP_{Rj} + \sum_{j=1}^{Na} C_{cj} + 2 \quad CI_{MH} + CI_{HE2} + CI_{HRE} \tag{24}$$

3.2. Operating costs and revenues estimation

Operating costs include electrical energy consumed to overcome friction losses in the distribution piping. For each *i-th* branch of length L_{bi} in the distribution network it can be computed as

$$Ce_i = \frac{8 \quad m_i^{\,3}}{\rho_w^{\,2} \quad \pi \quad D_T^5} \quad L_{bi} \quad f \quad \frac{Nh}{\eta_{pump}} \quad C_{EE} \tag{25}$$

where m_i is the water flow rate in the *i-th* branch (kg/s), ρ_w (kg/m^3) is the water density, f is the friction factor, Nh is the annual operating hours (hr/yr), η_{pump} is the pump efficiency and CEE (€/kWh) is the electricity cost.

Pumping power in the primary circuit and in the users branches is considered negligible. The overall operating cost Ce_{tot} (€/yr) includes the pumping cost in the main headers Ce_{MH} (computed as shown in Eq. 25), those in the distribution ring headers Ce_i, maintenance costs computed as a percent of capital investment and workforce cost computed as the operator annual wage (W_O, €/yr) times the number of employees N_W

$$Ce_{tot} = 2 \quad \sum_{i=1}^{Na} Ce_i + 2Ce_{MH} + 0.05 CI_{tot} + W_O \, N_W \tag{26}$$

Defining C_{TE} (€/GJ) the energy price and AF the annual fixed fee charged to the users (€/user yr), *i* (%/yr) the interest rate and Nyr the plant life (yr), yearly revenues (Rev, €/yr) from sale of thermal energy and the investment net present value (NPV, €) are

$$Rev = N_{Us} P_{Us} (3600 \ Nh) \frac{C_{TE}}{10^6} + N_{Us} AF \tag{27}$$

$$NPV = -CI_{tot} + \sum_{k=1}^{Nyr} \frac{(Rev - Ce_{tot})_k}{(1+i)^k} \tag{28}$$

4. PARAMETRIC ANALYSIS

A parametric analysis has been carried out to assess the impact of the distribution network characteristics on the investment profitability. In particular, assuming a value of the recovered heat, the NPV of the investment has been computed when changing both the delivery distance L_D from the cement plant to the urban distribution area and its radius R. Obviously the number of users has been maintained constant as it depends from the recovered heat and the fixed thermal requirements of each user. Figure 6 show the obtained results. Computations were made over a 15 years time span assuming the following data: water flow rate in the primary circuit m_I=10 kg/s, T_{inHE1}=105 °C, T_{outHE1}=141 °C, T_{outHE2} = 90 °C, T_{inUs} = 85°C, T_{outUs} = 60 °C, T_{inHE2} = 55 °C. Pipe sizes have been selected with the optimal economic diameter criterion resulting in nominal pipe size ½" SCH40 for the terminal branches. The urban area has been divided into two annuli. The heat recovery exchanger is 15 m long and recovers 1.5 MW. Other utilized data are as follows: *i* = 10%, AF = 20 €/yr, C_{TE} = 7.5 €/GJ, Nh = 4400 hr/yr, P_{Us} = 15 kW, N_W = 1, W_O = 20000 €/yr, C_{EE} = 0.11 €/kWh, η_{pump} = 0.8, Ct = 10 €/m^3, Cu = 10 €/kg, C_0 = 100 €/connection, N_{Us} = 75. Results show that for the given recovered heat a profitable service occurs when the users are concentrated within a radius of about 100 – 200 m and the urban area is not farther than 0.5 – 1 km from the heat recovery site. However, if greater recoverable heat is available, and the delivery distance is moderate, as shown in Fig. 7, then a profitable operation occur even for a distribution radius of up to 300-400 m. Such results show that DH from recovering radiant heat from rotary kilns may be economically feasible, even in the absence of state subsidies, in case concentrated users at moderate distances from the hest recovery site are found. This may happen for single large scale users like hospitals and schools, or for small rural villages nearby a cement plan or for single neighbourhoods within a city. Moreover, in case such values are considered unattractive for private investors they may be acceptable for a public no-profit investment, also considering the social benefits coming from the community fuel saving and the reduced amount of pollutant emissions from household boilers.

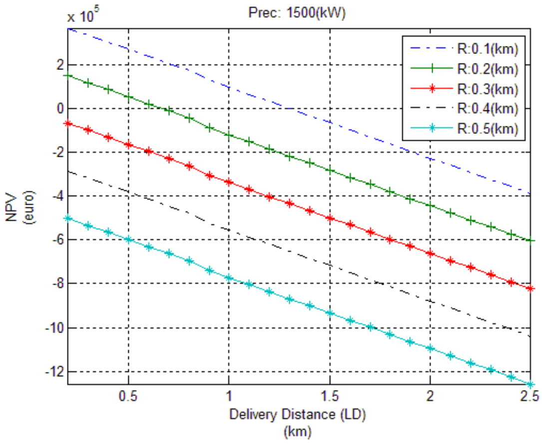

Figure 6. NPV as a function of delivery distance and urban area radius.

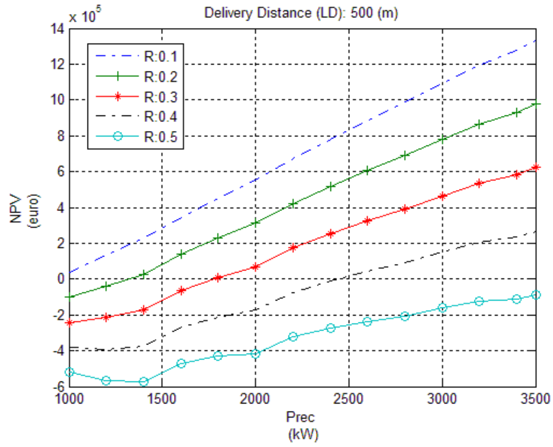

Figure 7. NPV as a function of recovered heat and urban area radius (delivery distance of 500 m).

5. CONCLUSIONS

The paper demonstrates the technical feasibility of recovering dissipated heat from rotary kiln walls and shows as the relevant heat consumption of cement production makes district heating option for heat recovery a feasible one even in case of low density of inhabitants in the surroundings of the plant, especially when no-profit institutions are involved. Furthermore, significant fuel savings and emission reductions are achieved respect the adoption of traditional residential boilers which should be accounted for in a cost-benefit analysis further justifying the examined investment.

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