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Primena analize škripca na projektovanje gasifikacionog kotla na pelet

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U radu je određena analiza škripca (minimalne temperaturske razlike) za razmenu toplote u gasifikacionom kotlu na pelet. Cilj je bio da se na adekvatan način pronađu osnovni parametri (temperature i snage) konvektivno zračnog (ložište) i konvektivnog (druga i treća promaja) dela gasifikacionog kotla na pelet. Na osnovu ovih parametara moguće je, relativno lako, isprojektovati ložište i razmenjivače toplote na kotlu. Kotao je posmatran kao sistem više toplih i hladnih struja. U tople struje (tokove koji se hlade) spada dimni gas. U hladne struje (tokove koji se greju) spadaju biomasa, vazduh za gasifikaciju, vazduh za sagorevanje, gorivi gas i voda. Za sve tokove izračunata su toplotna opterećenja u zavisnosti od početnih i krajnjih temperatura, masenih protoka i specifičnih toplotnih kapaciteta. Pri projektovanju kotla u kome se 25% toplote razmeni u ložištu a ostatak 75%, najčešće u dobošastom dimovodnom razmenjivaču toplote, za slučaj sagorevanja drvnog peleta sa 7% kiseonika u suvim produktima, polazni parametri za projektovanje pomenutog razmenjivača toplote bili bi: temperatura dimnog gasa na ulazu u razmenjivač 710 °C, na izlazu 100 °C; temperature vode na ulazu 60 °C i izlazu 75 °C. Na osnovu dijagrama datog u radu i odnosa konvektino zračne razmene toplote lako je pronaći parametre za projektovanje toplovodnog kotla na pelet. Izložena metodologija može se primeniti na različite vrste kotlova i različita goriva.

Ključne reči: Analiza škripca; Pelet; Gasifikacioni kotlovi; Razmenjivači toplote.

1. UVOD

Dosadašnja metodologija dizajniranja kotlova zasnivala se na klasičnim pristupima pri čemu se vodilo računa da budu ispunjeni uslovi koji se odnose na sigurnost u radu, jedostavnost u rukovanju, male dimenzije i visok stepen korisnosti koji odgovara vrednostima sadržanim u Pravilniku o energetskoj efikasnosti. U cilju smanjenja troškova, a samim tim i cene kotlova, savremena kotlogradnja je uslovljena, kako ekonomskom i energetskom krizom tako i zaštitom životne sredine, pa današnje kotlove karakteriše povećanje stepena korisnosti raznim konstrukcionim unapređenjima i tehničkim rešenjima koji se odnose prvenstveno na iskorišćenje toplote izlaznih gasova [1]. Upravo tehnologija škripca predstavlja sastavnu metodologiju za očuvanje i uštedu energije u raznim procesima koji je troše [2-3]. Koristi se za određivanje energetskih troškova i projektovanja mreže razmenjivača toplote u procesu, pri čemu se zasniva na osnovnim zakonima termodinamike. Cilj koncepta je poboljšanje energetske efikasnosti prilikom razmene toplote unutar gasifikacionog kotla, odnosno u razmenjivaču toplote. Studija je rađena za različite vrednosti kiseonika u suvim produktima sagorevanja: za 7%, 10% i 13%, ali će ovde biti prikazan slučaj sa 7% kiseonika u suvim produktima. Koncept obuhvata iskorištenje toplote tzv. „Toplih struja“ za zagrevanje „Hladnih struja“, odnosno uočena je mogućnost za što bolje iskorišćenje toplote otpadnih gasova za zagrevanje vode. Ovo omogućava da se izvrši proračun dobošastog razmenjivača toplote, sa suprotnosmernim strujanjem i dvostrukim prolazom [4].

Pri izradi:

1. Koncepta sveobuhvatnog termotehničkog rešenja kotla i
2. Proračuna dobošastog razmenjivača toplote,

vodilo se računa da su rešenja:

- sveobuhvatna, tj. da integrišu različite slučajeve koji se javljaju u praksi, a odnose se na količinu kiseonika u suvim produktima sagorevanja;
- projektovana tako da omoguće maksimalno iskorištenje dobijene energije;
- tehnički savremena i energetski efikasna;
- usmerena na smanjenje potrošnje električne energije;
- ekonomski prihvatljiva i isplativa i
- ekološki prihvatljiva: da ne doprinose povećanju emisije zagađujućih materija iz kotla [5-7].

2. PRIMER ODREĐIVANJA ANALIZE ŠKRIPCA ZA RAZMENU TOPLOTE U GASIFIKACIONOM KOTLU NA PELET

2.1. Mogućnost integracije procesa grejanja i hlađenja energetskih tokova unutar kotla

U okviru ovog dela koncepta, zbirno se daje materijalni i toplotni bilans i analizira mogućnost povezivanja različitih energetskih tokova unutar kotla. Na osnovu analize škripca, proverava se mogućnost optimalnog povezivanja energetskih tokova u kotlu. Naime, unutar kotla postoji više uočenih tokova koji istovremeno zahtevaju grejanje i hlađenje. Ovom analizom se pokušava otkriti potencijal da se toplota koja se dobija hlađenjem otpadnih gasova iskoristi za grejanje vode [8-10]. Kao što je u uvodu već rečeno, integracija sistema grejanja i hlađenja energetskih tokova unutar kotla je srž ovog koncepta.

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2.2. Materijalni i toplotni bilans

Za dato gorivo čija je organska masa data sledećim hemijskim sastavom:
C=50.75%
H=7%
O=42%
N=0.2%
S=0.03%

i čija je tehnička analiza takva da je sastav vlage u gorivu:
W=8.5%
A=1.05%
OM=90.45%

određene su minimalna potrebna količina kiseonika za sagorevanje i minimalna potrebna količina vazduha za sagorevanje [11].

Minimalna potrebna količina kiseonika za sagorevanje:

$$O_{min} = 1.867 \cdot g_C + 5.6 \cdot g_H + 0.7 \cdot g_S - \frac{22.4}{32} \cdot g_O \quad (1)$$

$$O_{min} = 0.957 \; {m_N^3}/{kg}$$

Minimalna potrebna količina vazduha za sagorevanje:

$$L_{min} = \frac{O_{min}}{0.21} \quad (2)$$

$$L_{min} = 4.56 \; {m_N^3}/{kg}$$

Formula datog goriva glasi:

$$\mathrm{CH_{1.66}O_{0.62}N_{0.0034}S_{0.00022}} + 0.12 \cdot \mathrm{H_2O} + 0.0045 \cdot \mathrm{S_iO_2}$$

pri čemu je poznata donja toplotna moć, i iznosi:

$$H_d = 17485.1 \; {kJ}/{kg}$$

2.3. Nezavisne hemijske jednačine

Iz nezavisnih hemijskih jednačina:

$$C + \frac{1}{2} \cdot O_2 \rightarrow CO$$
$$C + O_2 \rightarrow CO_2$$
$$C + 2 \cdot H_2 \rightarrow CH_4$$
$$H_2 + \frac{1}{2} \cdot O_2 \rightarrow H_2O$$

Gibsove funkcije stvaranja:

$$\overline{g_{fi}^{\bullet}} = \left(\overline{g_f^{\circ}}\right)_i + \overline{C_{pi}} \cdot (T - T^{\circ})_i - T \cdot \left(\overline{S^{\circ}} + \overline{C_p} \cdot ln\frac{T}{T^{\circ}}\right) + T^{\circ} \cdot \left(\overline{S^{\circ}} \cdot T^{\circ}\right)_i \quad (3)$$

temperature i univerzalne gasne konstante, dobija se konstanta hemijske ravnoteže:

$$k_e = exp\left\{\frac{\sum_R \nu_i \cdot \overline{g_0^{\circ}} - \sum_P \nu_i \cdot \overline{g_0^{\circ}}}{\mathcal{R} \cdot T}\right\} \quad (4)$$

$$k_e = 1.179$$

Odnosno:

$$k_e = \frac{r_{CO2} \cdot r_{H2}}{r_{CO} \cdot r_{H2O}} \cdot \left(\frac{p_m}{p_0}\right)^{2-2} = \frac{r_{CO2} \cdot r_{H2}}{r_{CO} \cdot r_{H2O}} \quad (5)$$

na osnovu čega se računaju zapreminski udeli komponenata u smeši gorivog gasa dobijenog u procesu gasifikacije peleta:

$$r_{CO} = 0.19 \; ; \; r_{H2O} = 0.062 \; ; \; r_{CO2} = 0.093; \; r_{H2} = 0.1493$$

$$r_{N2} = 0.4857$$

2.4. Sagorevanje peleta

Pri potpunom sagorevanju peleta biće:

$$\mathrm{CH_{1.66}O_{0.62}N_{0.0034}S_{0.00022}} + 0.12 \cdot \mathrm{H_2O} + 0.0045 \cdot \mathrm{S_iO_2} + 1.105 \cdot [\mathrm{O_2} + 3.76 \cdot \mathrm{N_2}] \rightarrow \mathrm{CO_2} + 0.95 \cdot \mathrm{H_2O} + 0.00022 \cdot \mathrm{SO_2} + 4.16 \cdot \mathrm{N_2} + 0.0045 \cdot \mathrm{S_iO_2}$$

Minimalna potrebna količina kiseonika za sagorevanje:

$$O_{min} = 0.957 \; {m_N^3}/{kg}$$

Minimalna potrebna količina vazduha za sagorevanje:

$$L_{min} = 4.56 \; {m_N^3}/{kg}$$

Pri sagorevanju peleta sa 7% O2 u suvim produktima biće:

$$\mathrm{CH_{1.66}O_{0.62}N_{0.0034}S_{0.00022}} + 0.12 \cdot \mathrm{H_2O} + 0.0045 \cdot \mathrm{S_iO_2} + 1.6468 \cdot [\mathrm{O_2} + 3.76 \cdot \mathrm{N_2}] \rightarrow \mathrm{CO_2} + 0.95 \cdot \mathrm{H_2O} + 0.00022 \cdot \mathrm{SO_2} + 6.1939 \cdot \mathrm{N_2} + 0.0045 \cdot \mathrm{S_iO_2} + 0.5418 \cdot \mathrm{O_2}$$

Koeficijent viška vazduha iznosi:

$$\lambda = 1.49$$

Stvarno potrebna količina kiseonika za sagorevanje:

$$O_{st} = \lambda \cdot O_{min} \quad (6)$$

$$O_{st} = 1.427 \; {m_N^3}/{kg}$$

Stvarno potrebna količina vazduha za sagorevanje:

$$L_{st} = \lambda \cdot L_{min} \quad (7)$$

$$L_{st} = 6.79 \; {m_N^3}/{kg}$$

2.5. Gasifikacija biomase

Prilikom gasifikacije biomase:

$$30.3 \cdot \left[\mathrm{CH_{1.66}O_{0.62}N_{0.0034}S_{0.00022}} + 0.12 \cdot \mathrm{H_2O} + 0.0045 \cdot \mathrm{S_iO_2} + 0.348 \cdot [\mathrm{O_2} + 3.76 \cdot \mathrm{N_2}]\right] \rightarrow 19 \cdot \mathrm{CO} + 9.3 \cdot \mathrm{CO_2} + 2 \cdot \mathrm{CH_4} + 6.2 \cdot \mathrm{H_2O} + 14.93 \cdot \mathrm{H_2} + 48.57 \cdot \mathrm{N_2}$$

Koeficijent viška vazduha iznosi:

$$\lambda = 0.315$$

dok je potrebna količina vazduha za gasifikaciju:

$$L_{st} = 1.44 \; {m_N^3}/{kg}$$

Da bi se dobio gorivi gas datog sastava potrebno je da sagori 30.3 kg peleta.

Proizvodnja gorivog gasa iznosi:

$$\dot{V}gg = 2.83 \ {m^3_{Ngg}}\Big/{kg_{bm}}$$

2.5.1. Sagorevamje gorivog gasa

Pri potpunom sagorevanju gorivog gasa biće:

$$0.19 \cdot \mathrm{CO} + 0.093 \cdot \mathrm{CO_2} + 0.02 \cdot \mathrm{CH_4} + 0.1493 \cdot \mathrm{H_2} + 0.4857 \cdot \mathrm{N_2} + 0.062 \cdot \mathrm{H_2O} + 0.2097 \cdot [\mathrm{O_2} + 3.76 \cdot \mathrm{N_2}] \rightarrow 0.303 \cdot \mathrm{CO_2} + 0.2513 \cdot \mathrm{H_2O} + 1.2742 \cdot \mathrm{N_2}$$

Minimalno potrebna količina kiseonika za sagorevanje gorivog gasa iznosi:

$$O_{min} = 0.2097 \ {m^3_N}\Big/{m^3_{Ngg}}$$

Dok minimalno potrebna količina vazduha za sagorevanje gorovog gasa iznosi:

$$L_{min} = 0.999 \ {m^3_N}\Big/{m^3_{Ngg}}$$

Pri sagorevanju gorivog gasa sa 7% O2 u suvim produktima biće:

$$0.19 \cdot \mathrm{CO} + 0.093 \cdot \mathrm{CO_2} + 0.02 \cdot \mathrm{CH_4} + 0.1493 \cdot \mathrm{H_2} + 0.4857 \cdot \mathrm{N_2} + 0.062 \cdot \mathrm{H_2O} + 0.3752 \cdot [\mathrm{O_2} + 3.76 \cdot \mathrm{N_2}] \rightarrow 0.303 \cdot \mathrm{CO_2} + 0.2513 \cdot \mathrm{H_2O} + 1.8963 \cdot \mathrm{N_2} + 0.1655 \cdot \mathrm{O_2}$$

Koeficijent viška vazduha iznosi:

$$\lambda = 1.79$$

Stvarno potrebna količina vazduha za sagorevanje gorivog gasa biće:

$$L_{st} = \lambda \cdot L_{min} \cdot \dot{V}gg \quad (8)$$

$$L_{st} = 5.07 \ {m^3_N}\Big/{kg_{bm}}$$

2.5.2. Temperatura sagorevanja

Temperatura sagorevanja se određuje pomoću sledeće formule:

$$T_{sag} = T^{\circ} + \frac{\sum_R r_i \cdot \left(\left(\overline{h_f^{\circ}}\right)_i + \overline{C_{pi}} \cdot \Delta t\right) + \dot{m}_v \cdot \overline{C_{pv}} \cdot (t_{pr} - 25) - \sum_P r_i \cdot \left(\overline{h_f^{\circ}}\right)_i}{\sum_P r_i \cdot \overline{C_{pi}}} \quad (9)$$

i iznosi:

Za λ=1

$$T_{sag} \approx 2045\ °C$$

Za λ=1.79

$$T_{sag} \approx 1722\ °C$$

2.5.3. Toplotna opterećenja

Za izračunate masene protoke i odgovarajuće specifične toplotne kapacitete izračunavaju se toplotna opterećenja za sve struje u kotlu [12-13]. Detaljan proračun je rađen u Excel-u a rezultati su prikazani u tabeli u sledećem odeljku.

Toplotna opterećenja se računaju po formuli:

$$H = \dot{m} \cdot \overline{C_p} \cdot \Delta t \quad (10)$$

odnosno:

$$H = \dot{m} \cdot \Delta t \cdot \sum r_i \cdot \overline{C_{pi}} \quad (11)$$

za smešu gasova čije zapreminske udele znamo.

3. ENERGETSKI BILANS

Materijalni bilans energetskih tokova se može dobiti iz protočnog toplotnog kapaciteta. Objašnjenja podataka uzetih u tabeli daju se u nastavku teksta.

Tabela 1:Materijalni bilans

Tokovi koji se greju (hladne struje)	Temperatura [°C]	Protočni toplotni kapacitet	Prosečna maksimalna toplotna snaga
Naziv	početna-krajnja	[KJ/KgK]	[KJ/Kg]
Biomasa	25-100	1.4	105
	100		213
	100-300	1.475	295
	300-800	2.838	1419
Vazduh za gasifikaciju	25-800	2.001	1551
Vazduh za sagorevanje	25-1722	7.52	12761
Gorivi gas	800-1722	5.469	5042
Voda	60-80	808.7	16174
Tokovi koji se hlade (tople struje)			
Dimni gas	1722-100	8.516	13813
Biomasa	100		231
	100-300	1.475	295
	300-800	2.838	1419
Vazduh za gasifikaciju	25-800	2.001	1551
Vazduh za sagorevanje	25-1722	7.52	12761
Gorivi gas	800-1722	5.469	5042

U podacima za analizu škripca, koji su prikazani u tabeli 1 izračunato je da je za zagrevanje biomase, prilikom gasifikacije, sa 25 °C na temperaturu od 100 °C, potrebna prosečna toplotna snaga od 105 kJ⁄kg_bm . Na temperaturi od 100 °C, toplota ispravanja vode iz biomase iznosi 213 kJ⁄kg_bm. Prilikom sagorevanja biomase, od 100 °C do 300 °C, sagorevaju biomasa, vodene pare i pepeo iz bimase, prosečna toplotna snaga iznosi 295 kJ⁄kg_bm. Dalje, od 300 °C do 800 °C, dakle do temperature gasifikacije, 75% preostale biomase, bez vlage, prešlo je u gas čiji sastav znamo, tako da sada imamo sagorevanje preostalog ugljenika iz biomase, smeše gasova, pepela i vodene pare. Prosečna toplotna snaga iznosi 1419 kJ⁄kg_bm.

Za izračunati protok vazduha za gasifikaciju od 1.44 mN^3⁄kg_bm, temperatursku razliku od 775 °C i srednji specifični toplotni kapacitet koji iznosi 1.3896 kJ⁄kgK, izračunato toplotno opterećenje iznosi 1551 kJ⁄kg.

Za izračunati protok vazduha za sagorevanje od 5.07 mN^3⁄kg_bm, temperatursku razliku od 1697 °C i srednji specifični toplotni kapacitet koji iznosi 32.52 kJ⁄kmolK, izračunato toplotno opterećenje iznosi 12761 kJ⁄kg.

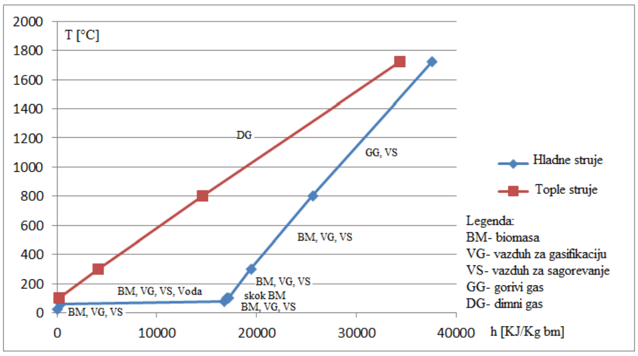
Za izračunati protok gorivog gasa od 2.83 mN^3⁄kg_bm, temperatursku razliku od 922 °C i srednji specifični toplotni kapacitet smeše gasova, izračunato toplotno opterećenje iznosi 5042 kJ⁄kg.

Toplotno opterećenje vode je izračunato tako što smo pretpostavili da 92.5 % donje toplotne moći biomase ide na zagrevanje vode, pri čemu smo dobili vrednost od 16174 kJ⁄kg i maseni protok vode od 0.418 kg⁄s.

Za izračunati protok dimnog gasa od 7.4 mN^3⁄kg_bm, temperatursku razliku od 1697 °C i srednji specifični toplotni kapacitet smeše gasova, izračunato toplotno opterećenje iznosi 13813 kJ⁄kg.

Kao što se vidi iz tabele u tokove koji se greju (hladne struje) spadaju biomasa, vazduh za gasifikaciju, vazduh za sagorevanje, gorivi gas i voda dok u tokove koji se hlade (tople struje) spada dimni gas, sa dodavanjem biomase, vazduha za gasifikaciju, vazduha za sagorevanje i gorivog gasa, koji se ovde uzimaju u obzir zbog tačnijeg proračuna.

Kumulativne krive svih tokova koji se greju i hlade u kotlu su prikazane na sledećoj slici:



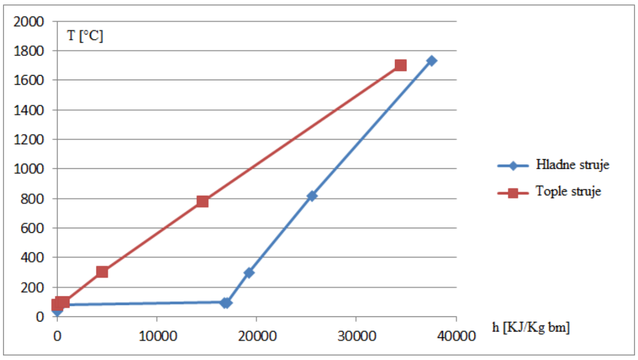
Slika 1: Kumulativne krive

Na apscisi je uneta ukupna (zbirna) specifična toplotna snaga svih tokova koji zahtevaju grejanje i hlađenje. Na ordinati su date temperature zagrevanja i hlađenja. Npr. za zagrevanje biomase, vazduha za gasifikaciju i vazduha za sagorevanje sa 25 °C na 60 °C, ukupna potrebna snaga je 382 kJ⁄kg, za slučaj sa 7% kiseonika u suvim produktima sagorevanja.

Prethodna tabela i slika se odnosi na slučaj kada je temperaturska razlika između tople i hladne struje na hladnom kraju razmenjivača jednaka nuli. Za modifikovane temperature, kada se usvoji da je $\Delta T\_min=40$ °C dobijaju se sledeći podaci:

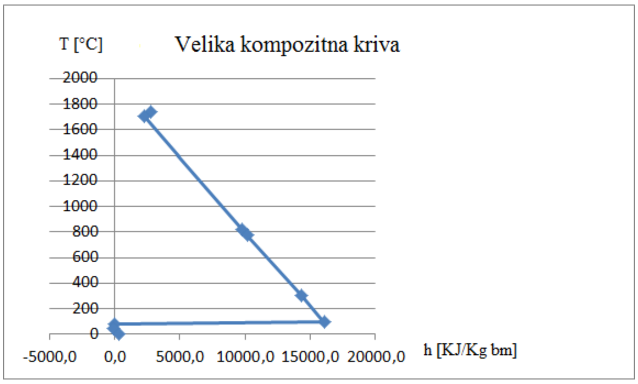
Tabela 2:Materijalni bilans za modifikovane temperature

Tokovi koji se greju (hladne struje)	Temperatura [°C]	Protočni toplotni kapacitet	Prosečna maksimalna toplotna snaga
Naziv	početna-krajnja	[KJ/KgK]	[KJ/Kg]
Biomasa	45-100	1.909	105
	100		213
	100-300	1.475	295
	300-820	2.729	1419
Vazduh za gasifikaciju	45-820	2.001	1551
Vazduh za sagorevanje	45-1742	7.52	12761
Gorivi gas	800-1742	5.469	5042
Voda	80-100	808.7	16174
Tokovi koji se hlade (tople struje)			
Dimni gas	1702-80	8.516	13813
Biomasa	100		231
	100-300	1.475	295
	300-780	2.956	1419
Vazduh za gasifikaciju	5-780	2.001	1551
Vazduh za sagorevanje	5-1702	7.52	12761
Gorivi gas	780-1702	5.469	5042



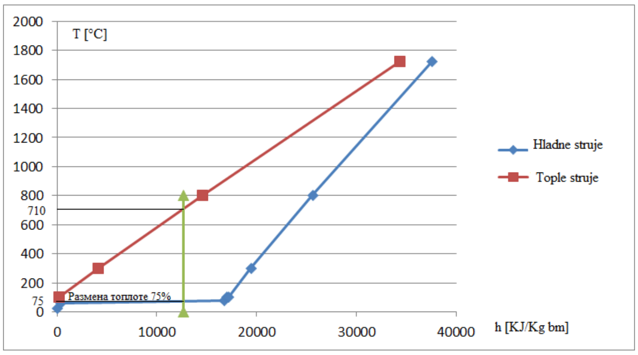
Slika 2: Kumulativne krive za modifikovane temperature

Velika kompozitna kriva pokazuje koliko je minimalno potrebne količine energije potrebno odvesti od tokova koji se hlade i koliko je minimalno potrebne količine energije potrebno dovesti tokovima koji se greju u kotlu.
Na sledećoj slici prikazana je velika kompozitna kriva.



Slika 3: Velika kompozitna kriva

Sa slika može da se uoči da toplotna energija toplih struja, odnosno dimnog gasa, može da se iskoristi za grejanje hladnih struja, odnosno vode. Sa dijagrama takođe mogu da se očitaju i temperature u svakom trenutku, što je neophodno za dalji proračun razmenjivača toplote u kotlu.



Slika 4: Temperature na ulazima i izlazima razmenjivača toplote

Sa slike 4 se vidi da za prikazani slučaj, kako dobošasti razmenjivač treba da oduzme 75% toplote od dimnog gasa i ostatak preda vodi, za slučaj sa 7% kiseonika u suvim produktima, dobijaju se sledeće temperature na ulazima i izlazima razmenjivača toplote [14-15]:
- temperatura dimnog gasa na ulazu iznosi 710 °C a na izlazu 100 °C;
- temperatura vode na ulazu iznosi 60 °C a na izlazu 75 °C.

4. ZAKLJUČAK

U cilju smanjenja potrošnje i racionalizacije korištenja energije, ulažu se veliki napori kako bi se isti ostvarili.“Pinch“ analiza, odnosno analiza škripca, jedna je od metoda energetske optimizacije koja se koristi kao alat za analizu i sintezu energetski iskoristivih delova procesa, kako pri projektovanju i izgradnji novih, tako i na postojećim postrojenjima.

Faze koje odlikuju metod škripca su:
- sakupljanje podataka;
- predprojektovanje;
- projektovanje inicijalnog rešenja i
- optimizacija.

Sama metoda se realizuje u T/H dijagramu, u kome se uočava škripac, određuje se tačka škripca hladne i tople kompozitne krive, prostor ispod i iznad škripca i potrebe za toplotnim ponorom i izvorom, dok je sama tačka škripca kritična tačka u kojoj je najmanji potencijal za razmenu toplote.

Primenom metode na prikazanom primeru, može se zaključiti da je:

- moguće i energetski efikasno međusobno povezivanje procesa hlađenja i grejanja toplih i hladnih struja u kotlu, odnosno moguće je da se iskoristi toplotno opterećenje neophodno za hlađenja toplih struja, za grejanje hladnih struja.

Pri projektovanju kotla u kome se 25% toplote razmeni u ložištu a ostatak 75%, najčešće u dobošastom dimovodnom razmenjivaču toplote, za slučaj sagorevanja drvnog peleta sa 7% kiseonika u suvim produktima, polazni parametri za projektovanje pomenutog razmenjivača toplote bili bi: temperatura dimnog gasa na ulazu u razmenjivač 710°C, na izlazu 100 °C; temperature vode na ulazu 60 °C i izlazu 75 °C. Na osnovu dijagrama datog u radu i odnosa konvektino zračne razmene toplote lako je pronaći parametre za projektovanje toplovodnog kotla na pelet. Izložena metodologija može se primeniti na različite vrste kotlova i različita goriva.

Projektovani gasifikacioni kotao je moguće izvesti kao kondenzacioni kotao, čime bi se snizila temperatura dimnog gasa ispod 100 °C i iskoristilo više toplote za grejanje hladnih struja, ali to izlazi iz okvira ovog rada zbog komplikovanijeg proračuna samog razmenjivača toplote.

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Pinch Analysis Application in Designing Gasification Pellets Boiler

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In this paper was determined the analysis of the pulley (minimum temperature difference) for the exchange of heat in the gasification pellet boiler. The aim was to adequately track the main parameters (temperature and power) of convective radiant (combustion chamber) and convective (second and third draft) parts of gasification pellet boiler. Based of these parameters relatively easy can be designed combustion chamber and heat exchangers in the boiler. The boiler system is observed as more warm and cold currents. The warm currents (flows to be cooled) consist of the flue gas. The cold currents (flows that are heated) include biomass, gasification air, combustion air, fuel gas and water. Loads are calculated for all the heat flows depending on the initial and final temperatures, mass flow and specific heat capacity. In the design of the boiler in which 25% of the heat is exchanged in the firebox and the remaining 75%, mostly in a shell and tube heat exchanger with gas pipes, in case of burning wood pellets with 7% oxygen in dry products, the initial parameters for the design of the mentioned heat exchanger would be: the temperature of the flue gas at the entrance to the heat exchanger 710 °C, at the outlet 100 ° C; the input water temperature 60 °C and outlet 75 °C. Based on the diagrams given in paper and ratio of convective radiant heat exchange it is easy to find the parameters for the design of hot water pellet boiler. The presented methodology can be applied to different types of boilers and different fuels.

Keywords: Pinch Analysis; Pellets; Gasification boiler; Heat exchangers.

1. INTRODUCTION

The previous methodology of designing boilers was based on classical approaches which were taking into account the conditions to be met relating to the safety, simple handling, small size and high efficiency, which corresponds to the values contained in the Regulations on energy efficiency. In order to reduce costs, and consequently the price of the boilers, modern boiler industry is conditioned by both economic and energy crisis and environmental protection, so today's boilers are characterized by the increase of efficiency degry due to improvements such as different construction and technical solutions that relate primarily to the utilization of the heat of exhaust gases [1]. Pinch technology presents an integral methodology for conservation and energy saving in various processes which spents it [2-3]. This technology is used to determine the energy cost and in designing the heat exchanger network in the process, while being based on the basic laws of thermodynamics. The goal of this concept is to improve energy efficiency in the exchange of heat in the gasification boiler, or in the heat exchanger. The study was performed for different values of oxygen in the dry flue gases: 7%, 10% and 13%, but you will be shown the case with the 7% of oxygen in dry products. The concept includes the utilization of the so-called "warm currents" to heat "cold currents", that was seen as an opportunity for better use of waste gas heat to heat water. This allowed us to perform the calculation of shell and tube heat exchanger, with counter-flow and double-aisle [4].

While making:

1. The concept of a comprehensive technical solution for thermal boiler and
2. Calculation od shell and tube heat exchanger, was taken into consideration that the solutions are:

- comprehensive, ie. that they integrate various cases that occur in practice, and refer to the amount of oxygen in the dry flue gases;
- designed to allow maximum utilization of energy yield;
- technically modern and energy efficient;
- focused on reducing electricity consumption;
- economically feasible and cost-effective and
- environmentally friendly: they do not contribute to increased pollutant emissions from the boiler [5-7].

2. EXAMPLE OF PINCH ANALYSIS DETERMINATION FOR HEAT EXCHANGE IN GASIFICATION PELLETS BOILER

2.1. The possibility of the integration of heating and cooling energy flows process inside the boiler

In this part of the concept are collectively given material and heat balance and analysis of the possibility of connecting different energy flows inside the boiler. Based on the pinch analysis, we examine the possibility of optimum connectivity of energy flows in the boiler. Inside of the boiler there are more perceived streams that simultaneously require heating and cooling. This analysis attempts to discover the potential of the heat that is obtained by cooling the waste gases used for heating water [8-10]. As already mentioned in the introduction, integration of heating and cooling energy flows inside the boiler is at the core of this concept.

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2.2. Material and heat balance

For a given fuel whose organic mass is given by following chemical composition:
C=50.75%
H=7%
O=42%
N=0.2%
S=0.03%

and whose technical analysis of composition is such that the content of the moisture in the fuel is:
W=8.5%
A=1.05%
OM=90.45% ,

minimum required amount of oxygen for combustion and minimum required amount of combustion air were determined [11].
The minimum required amount of oxygen for combustion:

$$O_{min} = 1.867 \cdot g_C + 5.6 \cdot g_H + 0.7 \cdot g_S - \frac{22.4}{32} \cdot g_O \quad (1)$$

$$O_{min} = 0.957 \ {m_N^3}/{kg}$$

The minimum required amount of combustion air:

$$L_{min} = \frac{O_{min}}{0.21} \quad (2)$$

$$L_{min} = 4.56 \ {m_N^3}/{kg}$$

Formula of our fuel is:

$$\mathrm{CH_{1.66}O_{0.62}N_{0.0034}S_{0.00022} + 0.12 \cdot H_2O + 0.0045 \cdot S_iO_2}$$

while the lower heating value is known:

$$H_d = 17485.1 \ {kJ}/{kg}$$

2.3. Independent chemical equations

From independent chemical equations:

$$C + \frac{1}{2} \cdot O_2 \rightarrow CO$$
$$C + O_2 \rightarrow CO_2$$
$$C + 2 \cdot H_2 \rightarrow CH_4$$
$$H_2 + \frac{1}{2} \cdot O_2 \rightarrow H_2O$$

Gibbs functions:

$$\overline{g_{fi}^{\bullet}} = \left(\overline{g_f^{\circ}}\right)_i + \overline{C_{pi}} \cdot (T - T^{\circ})_i - T \cdot \left(\overline{S^{\circ}} + \overline{C_p} \cdot ln\frac{T}{T^{\circ}}\right) + T^{\circ} \cdot \left(\overline{S^{\circ}} \cdot T^{\circ}\right)_i \quad (3)$$

temperature and universal gas constant, chemistry balance constant is:

$$k_e = exp\left\{\frac{\sum_R \nu_i \cdot \overline{g_0^{\circ}} - \sum_P \nu_i \cdot \overline{g_0^{\circ}}}{\mathcal{R} \cdot T}\right\} \quad (4)$$

$$k_e = 1.179$$

apropos:

$$k_e = \frac{r_{CO2} \cdot r_{H2}}{r_{CO} \cdot r_{H2O}} \cdot \left(\frac{p_m}{p_0}\right)^{2-2} = \frac{r_{CO2} \cdot r_{H2}}{r_{CO} \cdot r_{H2O}} \quad (5)$$

based on which volume of the components in the mixture of combustible gas obtained in the process of gasification of pellets was calculated and is:

$$r_{CO} = 0.19\ ;\ r_{H2O} = 0.062\ ;\ r_{CO2} = 0.093;$$
$$r_{H2} = 0.1493$$
$$r_{N2} = 0.4857$$

2.4. Pellets combustion

While complete pellets combustion:

$$\mathrm{CH_{1.66}O_{0.62}N_{0.0034}S_{0.00022} + 0.12 \cdot H_2O + 0.0045 \cdot S_iO_2 + 1.105 \cdot [O_2 + 3.76 \cdot N_2] \rightarrow CO_2 + 0.95 \cdot H_2O + 0.00022 \cdot SO_2 + 4.16 \cdot N_2 + 0.0045 \cdot S_iO_2}$$

The minimum required amount of oxygen for combustion:

$$O_{min} = 0.957 \ {m_N^3}/{kg}$$

The minimum required amount of combustion air:

$$L_{min} = 4.56 \ {m_N^3}/{kg}$$

While burning pellets with 7% of O2 in dry products:

$$\mathrm{CH_{1.66}O_{0.62}N_{0.0034}S_{0.00022} + 0.12 \cdot H_2O + 0.0045 \cdot S_iO_2 + 1.6468 \cdot [O_2 + 3.76 \cdot N_2] \rightarrow CO_2 + 0.95 \cdot H_2O + 0.00022 \cdot SO_2 + 6.1939 \cdot N_2 + 0.0045 \cdot S_iO_2 + 0.5418 \cdot O_2}$$

Excess air coefficient:

$$\lambda = 1.49$$

The reall required amount of oxygen for combustion:

$$O_{st} = \lambda \cdot O_{min} \quad (6)$$

$$O_{st} = 1.427 \ {m_N^3}/{kg}$$

The reall required amount of combustion air:

$$L_{st} = \lambda \cdot L_{min} \quad (7)$$

$$L_{st} = 6.79 \ {m_N^3}/{kg}$$

2.5. Gasification of biomass

While gasification of biomass:

$$\mathrm{30.3 \cdot \left[CH_{1.66}O_{0.62}N_{0.0034}S_{0.00022} + 0.12 \cdot H_2O + 0.0045 \cdot S_iO_2 + 0.348 \cdot [O_2 + 3.76 \cdot N_2]\right] \rightarrow 19 \cdot CO + 9.3 \cdot CO_2 + 2 \cdot CH_4 + 6.2 \cdot H_2O + 14.93 \cdot H_2 + 48.57 \cdot N_2}$$

Excess air coefficient is:

$$\lambda = 0.315$$

And required amount of gasification air is:

$$L_{st} = 1.44 \ {m_N^3}/{kg}$$

To obtain a fuel gas of a given composition 30.3 kg of pellets need to be burned.
Production of the fuel gas is:

$$\dot{V}gg = 2.83 \ {m_{Ngg}^3}/{kg_{bm}}$$

2.5.1. Fuel gas combustion

While complete fuel gas combustion:

$$0.19 \cdot \mathrm{CO} + 0.093 \cdot \mathrm{CO_2} + 0.02 \cdot \mathrm{CH_4} + 0.1493 \cdot \mathrm{H_2} + 0.4857 \cdot \mathrm{N_2} + 0.062 \cdot \mathrm{H_2O} + 0.2097 \cdot [\mathrm{O_2} + 3.76 \cdot \mathrm{N_2}] \rightarrow 0.303 \cdot \mathrm{CO_2} + 0.2513 \cdot \mathrm{H_2O} + 1.2742 \cdot \mathrm{N_2}$$

The minimum required amount of oxygen for the fuel gas combustion is:

$$O_{min} = 0.2097 \ {m_N^3}\Big/{m_{Ngg}^3}$$

While the minimum required amount of combustion air is:

$$L_{min} = 0.999 \ {m_N^3}\Big/{m_{Ngg}^3}$$

While fuel gas combustion with 7% of O2 in dry products:

$$0.19 \cdot \mathrm{CO} + 0.093 \cdot \mathrm{CO_2} + 0.02 \cdot \mathrm{CH_4} + 0.1493 \cdot \mathrm{H_2} + 0.4857 \cdot \mathrm{N_2} + 0.062 \cdot \mathrm{H_2O} + 0.3752 \cdot [\mathrm{O_2} + 3.76 \cdot \mathrm{N_2}] \rightarrow 0.303 \cdot \mathrm{CO_2} + 0.2513 \cdot \mathrm{H_2O} + 1.8963 \cdot \mathrm{N_2} + 0.1655 \cdot \mathrm{O_2}$$

Excess air coefficient:

$$\lambda = 1.79$$

The real required amount of combustion air is :

$$L_{st} = \lambda \cdot L_{min} \cdot \dot{V}gg \qquad (8)$$

$$L_{st} = 5.07 \ {m_N^3}\Big/{kg_{bm}}$$

2.5.2. Combustion temperature

Combustion temperature is determined using the following formula:

$$T_{sag} = T^{\circ} + \frac{\sum_R r_i \cdot \left(\left(\overline{h_f^{\circ}} \right)_i + \overline{C_{pi}} \cdot \Delta t \right) + \dot{m}_v \cdot \overline{C_{pv}} \cdot (t_{pr} - 25) - \sum_P r_i \cdot \left(\overline{h_f^{\circ}} \right)_i}{\sum_P r_i \cdot \overline{C_{pi}}} \qquad (9)$$

and amounts :
For λ=1

$$T_{sag} \approx 2045\ °C$$

For λ=1.79

$$T_{sag} \approx 1722\ °C$$

2.5.3. Thermal loads

For the calculated mass flows and the corresponding specific heat capacity, we have calculated the thermal loads of all the currents in the boiler [12-13]. A detailed calculation is done in Excel and the results are shown in the table in the next section.
Thermal loads have been calculated by the formula:

$$H = \dot{m} \cdot \overline{C_p} \cdot \Delta t \qquad (10)$$

apropos:

$$H = \dot{m} \cdot \Delta t \cdot \sum r_i \cdot \overline{C_{pi}} \qquad (11)$$

for the mixture of gases whose volume ratios are known.

3. ENERGY BALANCE

Material balance of energy flows can be obtained from the flow heat capacity. Explanations of the data collected in the table are given below.

Table 1: Material balance

Flows to be heated (cold currents)	Temperature [°C]	Instantaneous heat capacity	Average maximum heat output
Name	start-end	[KJ/KgK]	[KJ/Kg]
Biomass	25-100	1.4	105
	100		213
	100-300	1.475	295
	300-800	2.838	1419
Gasification air	25-800	2.001	1551
Combustion air	25-1722	7.52	12761
Fuel gas	800-1722	5.469	5042
Water	60-80	808.7	16174
Flows to be cooled (hot currents)			
Flue gas	1722-100	8.516	13813
Biomass	100		231
	100-300	1.475	295
	300-800	2.838	1419
Gasification air	25-800	2.001	1551
Combustion air	25-1722	7.52	12761
Fuel gas	800-1722	5.469	5042

In the data for pinch analysis, which are shown in Table 1 was calculated that for heating the biomass, during gasification, from 25 °C at a temperature of 100 °C, the required average heat power is 105 kJ/kg _bm. At a temperature of 100 °C, the heat power for water evaporation from the biomass is 213 kJ/kg_bm. While biomass combustion, from 100 °C to 300 ° C, biomass, steam and ash are burning from biomass, and the average thermal power is 295 kJ/kg_bm. Next, from 300 °C to 800 °C, to a temperature gasification, 75% of remaining biomass, without moisture, turned into gas whose composition we know, so that we now have a burning residual carbon from biomass, gas mixtures , ash, and water vapour. The average thermal power is 1419 kJ/kg_bm.

For calculated gasification air flow of 1.44 mN ^ 3 /kg_bm, temperature difference of 775 °C and mean specific heat capacity, which amounts 1.3896 kJ/kgK, calculated thermal load is 1551 kJ/kg .

For calculated flow of combustion air of 5.07 mN ^ 3 /kg_bm, temperature difference of 1697 °C and the mean specific heat capacity of 32.52 kJ/kmolK, calculated thermal load is 12761 kJ/kg.

For calculated flow of combustible gas of 2.83 mN ^ 3 /kg bm, temperature difference of 922 °C and the mean specific heat capacity of the gas mixture, calculated thermal load is 5042 kJ/kg.

Heat load of water is calculated in a way that we assumed that 92.5% lower heating value of the biomass goes to water heating, where we get a value of 16174 kJ/kg and mass flow rate of 0.418 kg/s.

For calculated flow of flue gas of 7.4 mN^ 3 /kg bm, temperature difference of 1697 °C and mean specific heat capacity of the gas mixture, calculated thermal load is 13813 kJ/kg.

As it can be seen from the table, flows that are heated (cold currents) include biomass, gasification air, combustion air, fuel gas and water while flows that are cooled (hot currents) is the flue gas, with adding biomass, gasification air, combustion air and fuel gas, which have been taken into account for more accurate calculation.

Cumulative curves of all the flows that are heated and cooled in the boiler are shown in the following figure:

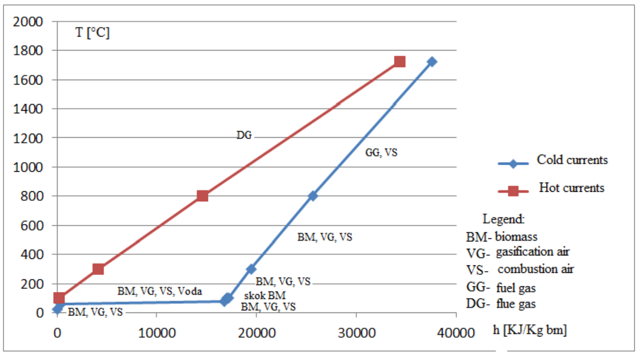

Figure 1: Cumulative curves

On the abscissa is entered the total (cumulative) specific heat output of all streams requiring heating and cooling. On the ordinate are given heating and cooling temperatures. E. g, for heating of biomass, gasification air and combustion air from 25 °C to 60 °C, the total required power is 382 kJ/kg, in case of 7% of oxygen in the dry flue gases.

Previous table and image are related to the case when the temperature difference between warm and cold currents in the cold end of the exchanger is zero. For modified temperature, when we adopt that ΔTmin = 40 °C the following data were gain:

Table 2: Material balance for modified temperatures

Flows to be heated (cold currents)	Temperature [°C]	Instantaneous heat capacity	Average maximum heat output
Name	start-end	[KJ/KgK]	[KJ/Kg]
Biomass	45-100	1.909	105
	100		213
	100-300	1.475	295
	300-820	2.729	1419
Gasification air	45-820	2.001	1551
Combustion air	45-1742	7.52	12761
Fuel gas	800-1742	5.469	5042
Water	80-100	808.7	16174
Flows to be cooled (hot currents)			
Flue gas	1702-80	8.516	13813
Biomass	100		231
	100-300	1.475	295
	300-780	2.956	1419
Gasification air	5-780	2.001	1551
Combustion air	5-1702	7.52	12761
Fuel gas	780-1702	5.469	5042

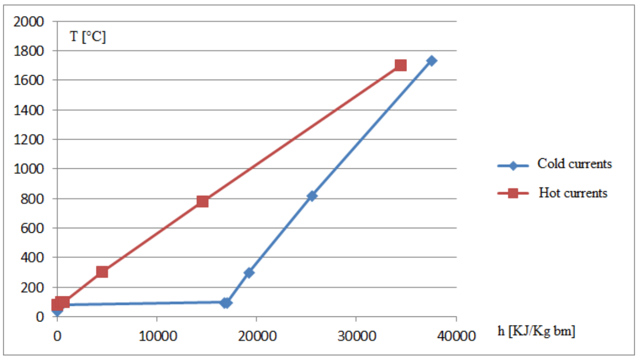

Figure 2: Cumulative curves for modified temperatures

Large composite curve shows the minimum amount of energy which is necessary to be taken from the hot currents and the minimum amount of energy which is necessary to be brought to cold currents in the boiler.

The following illustration shows the large composite curve.

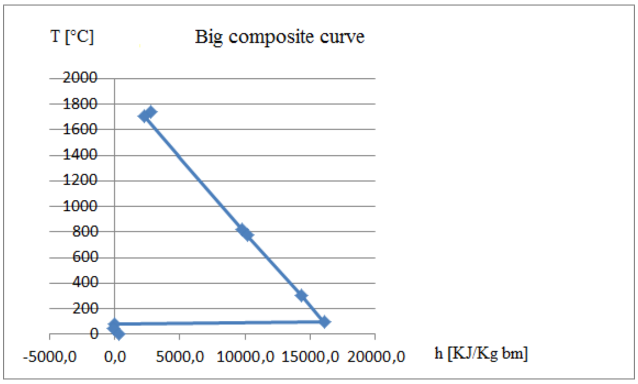

Figure 3: Big composite curve

From the pictures can be seen clearly that the thermal energy of warm current, that is the flue gas, can be used for heating cold currents, that is water. From the diagram can also be read temperatures at any time, which is necessary for the further calculation of the heat exchanger in the boiler.

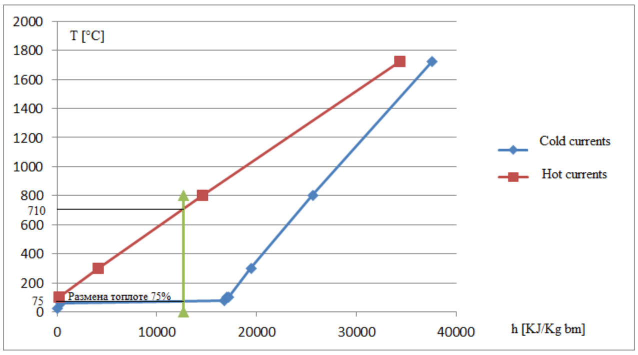

Figure 4: Temperatures at the entrances and exits of the heat exchanger

As it can be seen in figure 4 that in this case, as the shell and tube heat exchanger should confiscate 75 % of heat from the flue gas and surrender the rest to the water, in case of 7 % of oxygen in dry products, following temperatures at the entrances and exits of the heat exchanger are [14-15]:

- The temperature of the flue gas at the entrance is 710 °C and at the exit 100 °C;

- Water temperature at the entrance is 60 °C at the output 75 °C.

4. CONCLUSION

Every effort is made in order to reduce and rationalize the use of energy. Pinch analysis is one of the methods of energy optimization that is used as a tool for analysis and synthesis of energy usable parts of the process, both in design and construction of new ones, as well as on existing installations.

The phases which distinguish the pinch method are:

- data collection;
- front design;
- initial design solutions and
- optimization.

The method itself is implemented in T / S diagram, where the pinch point can be observed, pinch point is determined for cold and hot composite curve, space above and below the pinch point and the need for a heat sink and the heat source, while the pinch point itself is the critical point where we have the smallest potential for heat exchange.

By applying the method to our example, it can be concluded that it is:

- Possible and energy efficient to interconnect cooling and heating processes of hot and cold currents in the boiler, apropos it is possible to use the thermal load necessary for the cooling of the warm currents, for heating cold currents, when taken into consideration that the exchanger takes away 75% heat from the flue gas for water heating.

In the design of the boiler in which 25% of the heat is exchanged in the firebox and the remaining 75%, mostly in a shell and tube heat exchanger with gas pipes, in case of burning wood pellets with 7% oxygen in dry products, the initial parameters for the design of the mentioned heat exchanger would be: the temperature of the flue gas at the entrance to the heat exchanger 710 °C, at the outlet 100 °C; the input water temperature 60 °C and outlet 75 °C. Based on the diagrams given in paper and ratio of convective radiant heat exchange it is easy to find the parameters for the design of hot water pellet boiler. The presented methodology can be applied to different types of boilers and different fuels.

Designed gasification boiler can be designed as condensing boiler, which would lower the flue gas temperature below 100 °C and use more heat for heating cold currents, but it goes beyond the scope of this paper because of more complicated calculation of the heat exchanger.

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