Carbon dioxide footprint reduction by retrofitting regional heating boilers from gaseous to biogenic fuels

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Abstract. Carbon dioxide footprint reduction can be reached by switching fossil combustible consumers to biogenic fuels. Using biogenic fuels, mainly cellulosic ones, requires burner replacement and most often a complete change of burning process by introducing pre-burning chambers. The boiler response is also to be studied because the changes in flue gases composition, debits and temperatures. The paper analyses the necessary burning installation retrofitting for the case of switching a local boiler from gaseous fuel functioning to cellulosic fuel functioning (sawdust or pellets). It is presented a calculation method for boiler functioning verification (determining the functioning parameters considering an existing boiler used in other than nominal conditions) with the aim of determining the parametrical and performances variations when using other fuels. The nominal functioning regime, for which there are available experimental data, is used to validate the model and the computational program. Conclusions are drawn regarding the possibility and results for retrofitting regional heating boilers from gaseous to biogenic fuels.

1. General considerations
Considering the actual trend in the reduction of carbon dioxide emissions, the retrofitting of industrial boilers from gaseous combustible to sawdust based fuels becomes of great interest [1], [2]. It is important to mention that using any kind of biogenic material as combustible leads to the reduction (conventionally to zero) of the CO₂ footprint because the biomass combustion is in fact recirculating the CO₂ and not providing new CO₂ quantities to the atmosphere (the exact same amount of CO₂ is generated in the combustion process as the CO₂ absorbed from the atmosphere by the biomass in order to grow). It is also important to mention the ecological aspect of energetic use of waste products from wood industry.

There are numerous solutions for industrially use the biomass, mainly the sawdust from different industrial or cleaning activities. The sawdust can result either from processing the wood, either from chipping the discharged material from technological processes, from initial raw material harvesting or from seasonal cleansing of green spaces (parks, yards, highway side or central green lines etc.).

The sawdust can be used as it is or processed as pellets or stumps. It is well known the discussion about the opportunity of processing the sawdust; mainly, the pros are linked to higher quality of fuel and better handling and storage and the cons are linked to the energetic and technological needs. The guideline in choosing the way of storing and using the sawdust takes into consideration the storing capacities and the delays between getting and using the combustible, the thermal installation power and other particular constraints.

In the present paper we decided to study and present the case of direct using of sawdust.

The sawdust is considered as raw from the process that generated it, with the only treatment regarding its humidity. We chose that particular one and only treatment because it is well known the major impact or the solid fuels humidity over burning process cleanliness and over the boiler efficiency (a high level of humidity generates burning instability due to low burning temperature and high degree of inert
molecule presence and also high flue gases heat losses due to the high enthalpy of the humidity at chimney temperatures).

Considering this, we chose as burning technology the cyclonic (or multi-cyclonic) adiabatic pre-chamber. This technology will be later detailed in the paper.

The boilers considered are from the ignitubular (high volume) family due to their high thermal inertia and easiness of convective surfaces cleaning. Another reason for this choice was the large spread of this constructive solution in the thermal power range that we were interested in. We considered interesting the thermal power domain ranging from 1000 to 10000 kW because it is larger than the usually studied 100 – 500 kW of “home” or local equipments and inferior to the “large” energetic technologies (that also indispensably switch to aqua-tubular constructive solutions) ranging from 20 to 100 MW [3]. Also, for this thermal power range, the cyclonic (or multi-cyclonic) adiabatic pre-chamber is technologically adequate.

2. Setting the problem
Setting the problem reduces to the following steps:

- Choosing the constructive solution for the boiler range to be studied and the specific working conditions;
- Choosing a reference boiler for modelling validation (for which there are measured data for at least nominal conditions);
- Choosing the appropriate technology for the burning process, considering the combustible’s characteristics and particularities and the thermal power range intended;
- Modelling the boiler functioning with gaseous fuel and validating the model for the boiler which also has provided experimental data;
- Modelling the boiler functioning with biomass fuel;
- Using the computational model for the chosen range of boilers for both gaseous and biogenic fuel;
- Analysing and interpreting the output data in order to draw conclusions regarding the possibility and opportunity of using biogenic fuels instead of gaseous fuels for the studied boilers.

2.1. The constructive solution for the boiler range
The thermal power range chosen was between 2000 and 10000 kW (more eloquently 2 to 10 MW) for the reasons stated before.

The constructive solution is one largely used for the ignitubular boilers in that thermal power range, more precisely a three-way flue gas pass boiler with one cross flow furnace (input at one end and output at the other), sunk return chamber (flue gas routing from furnace in the convective pipes) and two convective flue gas passes.

The flue gases turn from the first convective pass in the second one via an external, thermally insulated chamber, fixed on the frontal tubular plate of the boiler (surrounding the burner or the connection duct of the burning pre-chamber).

There are considered no supplementary recuperative surfaces, such as burning air preheater or water preheater.

The secondary agent (water) parameters are considered to be 90 °C for the intake and 120 °C for outtake, with a mean value of 105 °C (temperature level of the secondary agent in the heat exchange calculations).
2.2. The reference boiler

We chose for reference a 13 MW (13000 kW) boiler using gaseous combustible (methane – G20).

The main constructive characteristics are:
- Furnace diameter / length : 1450 / 5790 mm;
- Sunk return chamber diam. / length : 2010 / 320 mm;
- Primary convective tubes number/diameter/length: 120 pcs. / 88,9 x 3,2 mm / 5790 mm;
- Secondary convective tubes number/diameter/length: 48 pcs./ 88,9 x 3,2 mm / 6420 mm and 40 pcs. / 76,1 x 2,9 mm / 6420 mm;
- Mantel diameter: 2650 mm.

The available measured data for this boiler referred to global functioning parameters for nominal output, such as: useful thermal power, fuel consumption, fuel characteristics, flue gas temperature and composition and secondary agent temperatures. Later in this paper those values will be numerically detailed for comparison with model results.

3. Burning technique

Of course, this part only deals with the biomass burning technique because the gaseous combustible is burned via ventilated forced inlet air (produced by a battery of fans) and specifically designed burning heads with local recirculation and swirl stabilisation mounted in burner bodies attached to the boiler front plate (classical solution for industrial burners of 5 to 10 MW).

So, for the biomass burning process, as stated before, we chose the pre-chamber burning technique.

The type of “burner” we chose for the type of boilers we intended to study, is realised in cylindrical solution, with under-grate introduction of primary burning air and tangential inlet of the secondary burning air. The biomass is introduced by the means of a variable rotation speed screw feeder. In Figure 2 are presented the general schematics for such a burner. The burner casing is cooled by burning air circulated between the casing and the inner refractory cylinder, fact that diminishes the depth of the high refractory material required and also insures a lower level of thermal loss at the burning chamber level. As an observation, the primary burning process takes place in a semi-fluidized bed generated by the biomass fed to the grate and the primary air flow through the grate and the secondary burning process is swirled by the tangential inlet of the secondary air at speeds averaging 50 to 70 m/s (fact generating a high level of stability for both the ignition and the burning process) [4], [5], [6], [7].
For thermal powers ranging from 1 to 4 MW it is economically correct to use single cylinder burners, but for larger thermal powers, up to 10 MW, a multi-cylinder solution is more appropriate, in order to maintain both diameter and height in decent domains.

Figure 2. General schematics for sawdust pre-chamber burner with semifluidized bed and swirl secondary burning air.

As a fact, even considering the high quality burning process (with low unburnt gases content and low air excess – averaging 1.6) and the good insulation realised by both solid insulation materials and the burning air flow, still it was determined a total heat loss (by unburnt species and external casing heat losses) at the burner level of 1 to 3 % from the input combustible heat flow. By the sake of a clear and focused comparison between boiler functioning with gaseous and biomass combustibles, this particular heat loss is neglected in the boiler calculus, the efficiency of the burning chamber for the biomass being considered equal to unity. Of course, in the situation of real retrofitting a gaseous combustible boiler to biomass combustible, the pre-chamber heat loss must be taken into consideration for the general...
installation combustible consumption calculus. In Table 1 are presented the main constructive and functional characteristics of real sawdust swirl pre-chamber burners. Considering the characteristic geometrical ratios and corroborating them with specific load parameters per section or per volume, indicative values of the overall pre-chamber dimensions can be calculated. Because the purpose of this paper focuses on the boiler and not on the burner, the actual detailed dimensioning of the burning chamber will not be treated but only some general gauge sizes will be determined. The pre-dimensioning results are also presented in Table 1.

Table 1. Main constructive and functional characteristics of sawdust swirl pre-chamber burners.

| Burner output nom. therm. power [kW] | Comb debit * [kg/h] | Burner diam./height ** [mm] | Sectional thermal load [kW/m²] | Volume thermal load [kW/m³] |
|-------------------------------------|---------------------|-----------------------------|-------------------------------|----------------------------|
| 100                                 | 27.6                | 500/650                     | 509                           | 783                        |
| 250                                 | 60.5                | 700/700                     | 649                           | 928                        |
| 520                                 | 115.4               | 800/800                     | 1034                          | 1293                       |
| 2269                                | 491                 | 1555/1450                   | 1202                          | 829                        |
| 4571                                | 989                 | 2100/1850                   | 1319                          | 713                        |
| 7871                                | 1703                | 2x1950/1700                 | 1317                          | 775                        |
| 11138                               | 2411                | 2x2200/2000                 | 1465                          | 732                        |

* sawdust with 10% humidity and net caloric power of 3974 kcal/kg;
** inner values of the free burning volume (insulation, air flow section and mantle width must be added);
@ values from actual tested burners;
@@ values for designed burners.

4. Modelling the boiler
The boiler modelling was realised by conceiving a computational program in BASIC language. The program was based on the logical schematics of a verification calculus of a boiler. The main steps followed were:

- Setting the boiler construction details (lengths, diameters, number of pipes, etc.);
- Setting the combustible characteristics (composition, properties from the burning process point of view, thermal properties, physical properties);
- Defining the useful thermal power and the secondary agent conditions (temperatures);
- Burning process calculus (combustible debit, flue gases debit and theoretical burning temperature) considering an initial value for the boiler efficiency (value that will be later modified after each boiler calculus iteration, till the input and output values vary with less than 0.01%);
- Heat transfer calculus for the furnace with consideration for the combined radiation heat transfer of flame and flue gases;
• Return temperature determination (from furnace into the first convective pass) and useful heat transfer calculus for the furnace [8];

\[
\begin{align*}
t_f &= \frac{T_f}{\left( M - \frac{C_o \xi T_t^3 S_R}{(1 - q_{\text{cal}}) BV_g c_{\text{pg}} f}\right)^{0.6}} - 27315 \
& \quad \text{[\degree C]} \quad (1)
\end{align*}
\]

\[
Q_F = B \cdot V_g \cdot (1 - q_{\text{cal}}) \cdot \left( c_{\text{pg}}^{T_f} I_f - c_{\text{pg}}^{T_t} I_f \right) \quad \text{[W]} \quad (2)
\]

- \( t_f \) = furnace output temperature;
- \( T_t \) = theoretical burning temperature;
- \( M \) = position factor;
- \( B \) = combustible debit;
- \( S_R \) = radiative heat exchange surface;
- \( V_g \) = specific flue gases volume;
- \( c_{\text{pg}} \) = specific heat (calorimetric);
- \( Q_F \) = furnace thermal load;
- \( C_o \) = Stefan – Boltzmann constant.

Verification computation for heat transfer in the first convective pass determining the output flue gases temperature and the useful heat flow [9], [10];

Mendeleev convective heat transfer invariant equation:

\[
\alpha_c = \varepsilon \cdot 0.0263 \cdot C_i \frac{\lambda \cdot \rho_{1.35} d_i}{d_i} \cdot Re^{0.8} \quad \text{[W/m}^2\text{K]} \quad (3)
\]

\begin{align*}
\text{for} & \quad 2300 < Re < 10000, \\
\text{with} & \quad \varepsilon = 1 - \frac{6 \cdot 10^5}{Re^{1.8}}
\end{align*}

- \( \alpha_c \) = convection coefficient;
- \( \varepsilon \) = correction coefficient;
- \( d_i \) = tube inner diameter;
- \( \lambda \) = flue gases thermal conductivity.

Radiative heat transfer coefficient:

\[
\alpha_r = 5.765 \cdot 10^{-8} \cdot \frac{a_p + 1}{2} \cdot a_g \cdot T_{gm}^3 \cdot \left[ \frac{1 - \left( \frac{T_p}{T_{gm}} \right)}{1 - \left( \frac{T_p}{T_{gm}} \right)^{3.6}} \right] \quad \text{[W/m}^2\text{K]} \quad (4)
\]

- \( \alpha_r \) = radiation coefficient;
- \( T_{gm} \) = flue gas mean temperature;
- \( T_p \) = tube wall temperature;
- \( a_p \) = tube wall emissivity;
- \( a_g \) = flue gases emissivity.
Total heat transfer to the surface:

\[ \alpha_i = \alpha_c + \alpha_t \quad [W/m^2K] \]

\[ K = \frac{1}{(1/\alpha_i + \varepsilon + 1/\alpha_{eq})} \quad [W/m^2K] \]  

\[ T_f = T_{pf} + \left( T_t - T_{pf} \right) e^{-\frac{AGF \cdot S_R}{B \cdot V \cdot C_{py} \cdot \phi}} \]

- \( \alpha_c = AGF = \) total transfer coefficient;
- \( t_f \) = flue gases exit temperature;
- \( t_t \) = flue gases inlet temperature;
- \( T_{pf} \) = tube wall temperature;
- \( S_R \) = convective heat exchange surface.

Verification computation for heat transfer in the second convective pass determining the output flue gases temperature and the useful heat flow;

Final heat balance and overall iteration parameters determination;

\[ Q_{ut} = Q_{loc} + Q_{CI} + Q_{CII} \quad [W] \]  

- \( Q_{ut} \) = total thermal power;
- \( Q_{loc} \) = furnace thermal power;
- \( Q_{CI} \) = first convective thermal power;
- \( Q_{CII} \) = second conv. thermal power;

Iteration parameters comparison (in view of deciding if the errors for the key boiler parameters are in the satisfactory range) and decision over going to final results printing or returning to the boiler calculus with the new obtained input matrix;

Final functioning parameters printing, regarding the overall boiler performances and also the detailed heat transfer behaviour for each heat exchange surface (flue gases key temperatures, heat exchange coefficients, heat fluxes).

The computational program validation was made by comparing the calculus results with the measured parameters for a reference boiler. The boiler for which we had measured values (experimental data) was the 13 MW boiler using gaseous fuel and functioning at 105 °C secondary agent mean temperature, at 13000 kW (13 MW). In Table 2 are presented for comparison the values obtained from the computation and those measured.

| Parameter            | Measure unit | Measured value | Calculated value | Error [%] |
|----------------------|--------------|----------------|------------------|-----------|
| Useful thermal load  | kW           | 13037          | 13029            | 0,06      |
| Thermal efficiency   | %            | 90,54          | 90,85            | 0,31 *    |
| Flue gas exhaust temp.| °C         | 204            | 183              | 1,26 **   |

* by respect to 100 % ;  
** by respect to the theoretical burning temp. of 1662 °C.

Table 2. Comparison of values obtained from the computation and measured.
The conclusion was that the modelling results are very satisfactory from the point of view of admissible errors (the errors were generally with 1 or 2 orders of magnitude smaller than the admissible ones for an engineering calculus).

Once the reference computational program validated, the completion of the program for the biomass combustible functioning of the boiler was implemented. The completion regarded special subroutines for the burning process (with consideration for the solid fuel way of defining the composition and particular combustion equations) and for the heat transfer by radiation (for both furnace and convective passes).

Introducing those supplementary elements did not influence the accuracy of the computation (compared with real cases functioning) because all new introduced elements were already largely verified and certified by researchers and literature (common knowledge), so there were no motives to require the computational program validation in the new, extended form.

5. Computational modelling
Once the computational modelling of the boiler realised and validated, the boilers functioning for the situation of either using gaseous fuel or biomass (sawdust based) fuel was possible.

The sawdust based combustible considered was characterised by a caloric power of 3974 kcal/kg for a humidity content of 10%. The other constituents were: Carbon 44.6%, Hydrogen 5.4%, Oxygen 38.7%, Nitrogen 0.2% and Ash 1.1%. The air excess considered was 1.6 for all the modelling cases (for biomass fuel).

| Boiler nominal output thermal power [kW] | Furnace diam.**/length [mm] | First conv. tubes diam.**/length / number [mm ; pcs.] | *Second conv. tubes 1 diam.** / length / number [mm ; pcs.] | *Second conv. tubes 2 diam.**/length / number [mm ; pcs.] |
|----------------------------------------|-----------------------------|-----------------------------------------------------|---------------------------------------------------|-----------------------------------------------------|
| 2000                                  | 850 / 2650                  | 47.6/2420/82                                        | 47.6/2910/26                                      | 42.2/2910/32                                         |
| 4000                                  | 1150 / 3150                | 57.6/2920/92                                        | 57.6/3380/36                                      | 52.2/3380/42                                         |
| 7000                                  | 1300 / 4700                | 73.6/4470/100                                       | 73.6/4960/36                                      | 64.2/4960/42                                         |
| 10000                                 | 1450 / 6010                | 82.5/5785/102                                       | 82.5/6290/40                                      | 70.3/6290/38                                         |

* the second convective has two types of tubes ;
** the diameters are for the exterior of the tubes.

It is important to state here the assumption we made that the modification of some constructive parameters, such as furnace diameter and length, convective tubes numbers, diameters and lengths and combustible debit does not affect the accuracy of the computation (versus real functioning) as long as
general constructive solution of the boiler and main key parameters for the heat exchange (temperatures and flow speeds) does not vary significantly (in order to change the invariant’s equations, the general heat fluxes distribution or the prevalent heat transfer phenomena) from the basic model. In order to prove this statement, in the parametric tables further presented for the calculated boilers solutions, elements like flue gases flow speeds, superficial and overall heat transfer coefficients, mean temperatures and mean temperature differences were analysed for comparison with the reference model, the 10 MW boiler (which was, at its turn, validated versus the 13 MW boiler).

The reference boiler’s thermal powers output considered relevant for the present research were 2000 kW, 4000 kW, 7000 kW and 10000 kW. For each one it was performed a computer modelling (calculus) using the computational program previously presented (as modelling algorithm). Of course, as stated before, each boiler was modelled, one at a time, for both gaseous fuel and biomass fuel (sawdust based fuel). In Table 3 are presented the main construction characteristics for each boiler’s thermal power.

In Table 4 are presented the main flow speeds and the Re invariant values for the flue gases and the main characteristic temperatures along the heat exchange surface (also for flue gases) will be later detailed in Table 6.

Table 4. Computational results – flue gases flow speeds and the Re numbers

\(G\) means gaseous fuel and \(S\) means sawdust.

| Boiler nominal output thermal power [kW] | First conv. flue gases flow speed [m/s] | Second conv. flue gases flow speed [m/s] | First conv. flue gases Re number | Second conv. flue gases Re number |
|----------------------------------------|----------------------------------------|----------------------------------------|---------------------------------|----------------------------------|
| 2000 G                                 | 21,9                                   | 18,92                                  | 8202                            | 20445                            |
| 2000 S                                 | 24,05                                  | 22,17                                  | 10223                           | 24593                            |
| 4000 G                                 | 27,26                                  | 19,32                                  | 11984                           | 24103                            |
| 4000 S                                 | 30,26                                  | 23,1                                   | 14929                           | 28788                            |
| 7000 G                                 | 25,85                                  | 19,83                                  | 15088                           | 35410                            |
| 7000 S                                 | 28,67                                  | 23,81                                  | 18759                           | 42004                            |
| 10000 G                                | 28,53                                  | 23,15                                  | 19321                           | 49685                            |
| 10000 S                                | 31,64                                  | 27,8                                   | 23986                           | 58744                            |
6. Analysing the output data
Table 5 presents the computational results for the boilers range from the point of view of thermal performances. First of all, by analysing the overall functioning parameters, it is easy to observe that all chimney temperatures for the flue gases and all boiler’s thermal efficiencies are in the optimal range.

We remind here that the burning process efficiency is considered to be 100% (no losses by burner’s external surfaces and negligible incomplete burning losses) in order to compare solely the boiler functioning (as heat exchange surfaces succession), but in a real case retrofitting it would be necessary to consider the 1 to 3% burner heat losses (relative to boiler’s heat output) generated by unburnt species (both chemical and mechanical) and external casing heat losses.

Table 5. Computational results – general thermal performances.

| Boiler nominal output thermal power [kW] | Fuel type | Comb. debit [m³/h] or [kg/h] | Thermal efficiency [%] | Chimney flue gas temp. [°C] |
|-----------------------------------------|-----------|-----------------------------|------------------------|-----------------------------|
| 2000 Gas                                | 226,3     | 89,83                        | 188,7                  |
| 2000 Sawdust                           | 491,3     | 88,09                        | 190,2                  |
| 4000 Gas                               | 453,7     | 89,6                         | 201,2                  |
| 4000 Sawdust                           | 989,5     | 87,48                        | 206,8                  |
| 7000 Gas                               | 782,6     | 90,9                         | 180,1                  |
| 7000 Sawdust                           | 1703,1    | 88,94                        | 186,1                  |
| 10000 Gas                              | 1109,5    | 91,60                        | 168,2                  |
| 10000 Sawdust                          | 2410,6    | 89,77                        | 174                    |

The values for the heat exchange superficial coefficients, presented in Table 6, are also in the normal range (for this constructive solution and the chosen thermal powers) and consequently the overall heat transfer coefficients are in the normal (expected) range.

The total (percentile) heat flux distribution over the boiler’s heat transfer surfaces remains quite steady, proving that the functioning regime is normal, with no overall over or under demand and no necessary redistribution due to some particular surface over or under demand.
The fact is of outmost importance because not only proves the correct boilers dimensioning and functioning at the studied thermal regimes and for the considered combustibles, but also proves that thermal output power variations are possible in real functioning without altering the normal behaviour of the boilers (if the nominal regime is under optimal conditions, the under and over charge regimes will be possible without altering the correctness of functioning parameters ranging).

7. Conclusions
The main conclusions issued from the comparison between the functioning parameters with the two combustibles situations (one at a time, for each boiler) outlines some general trends:

- The thermal efficiency of the boiler is with approx. 2% less for the biomass functioning (compared with gaseous combustible functioning);

| Boiler nominal output thermal power [kW] | Furnace to first conv. passing temp. [°C] | First conv. to second conv. passing temp. [°C] | First conv. global heat exchange coeff. W/(m²·K) | Second conv. global heat exchange coeff. W/(m²·K) |
|----------------------------------------|----------------------------------------|----------------------------------------|----------------------------------------|----------------------------------------|
| 2000 G                                 | 1145                                   | 382,2                                  | 52,12                                  | 57,79                                  |
| 2000 S                                 | 1002                                   | 364,2                                  | 58,51                                  | 65,23                                  |
| 4000 G                                 | 1168                                   | 400,7                                  | 62,02                                  | 56,33                                  |
| 4000 S                                 | 1022                                   | 396                                    | 66,77                                  | 64,13                                  |
| 7000 G                                 | 1168                                   | 349,2                                  | 57,71                                  | 55,89                                  |
| 7000 S                                 | 1022                                   | 350,6                                  | 61,87                                  | 63,68                                  |
| 10000 G                                | 1153                                   | 319,4                                  | 60,68                                  | 62,02                                  |
| 10000 S                                | 1011                                   | 322,7                                  | 65,13                                  | 70,64                                  |
• Considering the pre-chamber thermal efficiency (non-adiabatic in real cases), the overall efficiency of the burner-boiler system will fall with 3% to 5% for sawdust functioning compared with gaseous fuel functioning;
• The specific surfaces thermal loads, both by specific values (overall heat exchange coefficients) and percentile contribution to nominal thermal power of the boiler, are in the normal range, suggesting a functioning in the frame of optimal values of parameters (flow speeds, hydraulic charge losses, necessary heat exchange surfaces, thermal elasticity for non-nominal functioning regimes etc.);
• Using the pre-chamber semi-fluidized bed with swirl secondary air inflow as burning technique for the sawdust (or sawdust based) biomass combustible is feasible from all points of view (technological, economical and functional) and can be easily implemented;
• For the hot water boilers ranging 2000 to 10000 kW with ignitubular constructive solution (furnace and two convective passes), it is possible to retrofit the functioning from gaseous combustible (G20) to biogenic combustible (sawdust or sawdust based) by simply modifying the combustor and with no necessary changes to the boiler body or annexes;
• Maintaining the thermal power output is easy by overcharging the boiler (compared with the calculated nominal charge) with approx. 5% (the maximum overall thermal efficiency drawback at combustible change).

So, the retrofitting of the hot water boilers ranging 1000 to 10000 kW with ignitubular constructive solution (furnace and two convective passes) is highly recommended if there is an interest in diminishing the CO2 footprint of the boiler. Considering the market price (per caloric unity) of the sawdust combustible, that is significantly smaller than for the gaseous combustible, it is very likely to obtain also economical gains along with the ecological improvement of the boiler’s functioning.

As the urge of reducing the CO2 footprint of all activities, including (or even mostly) the energetical field, the study will be further developed for high thermal power boilers (over 20 MW), with aqua-tubular constructive solutions and internal grate burners.

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