INVESTIGATION OF PRE-DRYING LIGNITE IN AN EXISTING GREEK POWER PLANT

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The application of lignite pre-drying technologies in next generation of lignite power plants by utilizing low pressure steam as a drying medium instead of hot recirculated flue gas - combined with thermal utilization of the vaporized coal moisture - is expected to bring efficiency increase of 2-4 percentage points in future lignite power plants compared with today’s state of the art.

The pre-drying concept is of particular importance in Greek boilers firing lignite with a high water and ash content. The combustion of Greek pre-dried lignite has been investigated experimentally and via numerical simulations in our previous research. This study focuses on the potential integration of a lignite pre-drying system in an existing Greek power plant with dry lignite co-firing thermal share of up to 30%. The radiative and convective heat fluxes to the boiler and the overall boiler heat balance is calculated for reference and dry lignite co-firing conditions by an in-house calculation code. The overall plant’s thermal cycle is then simulated using commercial thermal cycle calculation software. The net plant efficiency is in this way determined for reference and dry coal co-firing conditions. According to the simulation results the integration of a pre-drying system and the implementation of dry lignite co-firing may bring an efficiency increase of about 1.5 percentage points in existing Greek boilers. It is therefore considered as an important measure towards improving plant efficiency and reducing specific CO2 emissions in existing plants.

Key words: pre-drying, dry lignite co-firing, thermal cycle calculation

1. Introduction

The development of future lignite-fired power plant technology will be driven by some of the main goals and challenges regarding climate protection, energy savings and resource conservation as well as competitiveness. Therefore, one of the major tasks in power plant engineering is efficiency enhancement.

An essential parameter for increasing efficiency is the drying process. The lignite drying method applied has critical influence on the efficiency of the power plant, as lignite drying is an energy-intensive process due to the high moisture content and energy can, therefore, be saved by optimizing the process. The optimization of the process in future brown coal plants is calculated to lead to an efficiency increase of 2 to 4 percentage points[1,3].

The pre-drying method in particular, is predicted to play an important role for all future types of lignite-based power plants discussed today. Among these are the steam power plant with optimized efficiency, as well as the IGCC with and without carbon capture, and the oxy-fuel plant. Lignite pre-
drying is based on the idea of utilizing low temperature heat for drying purposes and recovering the waste heat from the drying process. [2]

The fluidized bed drying concept with waste heat utilization (WTA-Drying) is one of the most promising brown coal pre-drying technologies, currently under development. WTA pre-drying is a technology that can be integrated with all the aforementioned future routes of lignite based electricity generation - whether to increase efficiency or to effectively capture \( \text{CO}_2 \) or also in gasification routes or coal-to-liquid processes, where pre-drying is a vital element.

The moisture content of raw lignite is high, exceeding 50%. Thus, in conventional lignite-fired power plants a portion of the fuel’s heat is consumed in the boiler during combustion and milling / drying to evaporate coal-inherent water in a very high temperature level, leading in high exergy losses. Additionally, this heat cannot be used in the plant process and is lost since this coal-inherent water leaves the power plant as steam contained in the flue gas. (Fig 1a) [2]

With the WTA technology, drying is made exergetically more efficient by performing it at a low temperature level in a separate operating unit upstream of the steam generator and utilizing the energy of evaporated coal-inherent water in the power plant process, either in the feed water preheaters or in the WTA dryer for evaporation (Fig. 1b) [2]

The “WTA fine grain” drying concept has been successfully demonstrated in a pilot plant in Frechen, Germany and a prototype fluidised bed dryer has been constructed and integrated to the steam cycle of the 1000 MWe brown coal power plant in Niederaussem, Germany.

Fig. 1a.b: Conventional drying system and fluidised bed drying concept

Some of the previous pre-drying investigations performed so far at the Laboratory of Steam Boilers and Thermal Plants in the National Technical University of Athens (LSBTP/NTUA) involve the comparison of different pre-drying technologies based on thermodynamic calculations in which the WTA-concept has been proven as one of the most promising concepts, drying tests of Greek lignite in a lab scale fluidised bed dryer in which the equilibrium curves have been determined, and combustion tests with Greek dry lignite in a semi-industrial scale facility in which, among others, temperature and emission measurements, fly ash sampling and investigation of slagging and fouling tendency have been carried out. Recent work includes the examination of possible oxy-fuel configurations for future Greek power plants burning Greek dried lignite in terms of boiler and steam cycle optimization and the possibilities of upgrade and economic impact of pre-drying on a state of the art Greek lignite power plant through the integration of lignite pre-drying technology.

The present work is involved with the numerical investigation of the potential integration of the WTA pre-drying concept in an existing Greek brown coal power plant and the impact of its implementation on the boiler performance and the overall plant efficiency. In this case, design data of the 339 MW, Agios Dimitrios V unit in Kozani are used for modeling the project. Thermal performance and behavior of the plant for raw and dry lignite co-firing cases is evaluated and predicted through an in-house built code in the NTUA/LSBTP. The raw lignite firing case is used as a reference case for confirmation and comparison and thereafter, some cases of thermal share substitution of the boiler
input through co-firing dry lignite are modeled. The cases of thermal share substitution examined are those of 10, 20 and 30% of the initial input. The steam extracted from lignite pre-drying in each case is used as a heating medium in the power plant’s feed water pre-heaters. The overall plant’s thermal cycle is afterwards simulated by using commercial thermal cycle calculation software.

2. Methodology

2.1. Description of the 339MWe Agios Dimitrios unit V facility

Agios Dimitrios V in Kozani is a unit with a boiler consisting of the furnace where the pulverized fuel is burnt and the circulated water is evaporated, and of six more gas and water/steam heat exchangers, forming altogether the economizer, superheater and reheater surfaces. The superheater is constituted of three parts, while the reheater of two and the economizer of one. The set-up of the facility and the order of the heat exchanger surfaces after the furnace, according to the gas flow, as seen in Fig. 2, is:

1. SH2
2. SH3-RH2
3. SH1
4. RH1
5. ECO

The last parts of the superheater and reheater (SH3 & RH2) come along as a pair. There is also a hot gas recirculation for drying the raw lignite.

Fig. 2: Layout of the furnace and the heat exchanger surfaces of Agios Dimitrios V unit.

The unit also includes 3 steam turbines (High and Intermediate Pressure-1 HP / 2 IP) and 7 water preheaters. Two streams of water downstream of the preheaters are used as cooling spray water between the different parts of the superheater and another one downstream of the pump for the reheater. The full water/steam cycle with all the auxiliary units is shown in Fig. 3.
2.2. In-house built code

The code developed in the LSBTP/NTUA for the calculation of the boiler’s thermal characteristics uses the theory according to the German FDBR regulation. [5]

2.2.1. Furnace calculations

For the furnace calculations, the combustion chamber is separated into three sections, according to Fig. 4. Section 1 includes the bottom of the furnace (region below the flame), section 2 is the flame region and section 3 includes the upper part of the furnace and the combustion chamber “neck”.

The total heat flux radiated from the flame and the hot gas to the sections 1, 2 and 3 of the furnace, is:

\[
\dot{Q}_{\text{int}} = \dot{Q}_1 + \dot{Q}_2 + \dot{Q}_3 + \dot{Q}_G3
\]  

(2.1)

where:

\( \dot{Q}_1, \dot{Q}_2, \dot{Q}_3 \): radiated heat flux from the flame towards sections 1, 2 and 3 respectively and

\( \dot{Q}_G3 \): radiated heat flux from the gas that occupies the space above the flame section (section 3)

This total heat flux is equal to the thermal power offered by the gas in the furnace:

\[
\dot{Q}_{\text{gas}} = \dot{Q}_G = \dot{m}_G \cdot (h_{G\text{ex}} - h_{G\text{in}})
\]  

(2.2)
where:

\( m_G \): gas mass flow

\( h_{G_{\text{m}}} \): enthalpy in adiabatic flame temperature

\( h_{G_{r,\text{p}}} \): enthalpy corresponding to the combustion chamber outlet temperature

The radiated heat flux from the flame towards sections 1, 2 and 3 are calculated as follows:

\[
\dot{Q}_1 = C_s \cdot \varepsilon_1 \cdot A_{\text{w}} \left( \tilde{T}_f \right)^4 - \left( \tilde{T}_w \right)^4
\]

\[
\dot{Q}_2 = C_s \cdot \varepsilon_2 \cdot A_{\text{w}} \left( \tilde{T}_f \right)^4 - \left( \tilde{T}_w \right)^4
\]

\[
\dot{Q}_3 = C_s \cdot \varepsilon_3 \cdot A_{\text{w}} \left( \tilde{T}_f \right)^4 - \left( \tilde{T}_w \right)^4
\]

where:

\( C_s \): \( =5.6697 \cdot 10^{-8} \text{ Wm}^{-2} \text{ K}^{-4} \) Stefan-Boltzmann constant

\( \tilde{T}_f \): average flame temperature

\( \tilde{T}_w \): average wall temperature

\( \varepsilon_1, \varepsilon_2, \varepsilon_3 \): emissivity for sections 1, 2 and 3 respectively

\( A_{\text{w1}}, A_{\text{w2}}, A_{\text{w3}} \): equivalent radiated surface of sections 1, 2 and 3 respectively

while the radiated heat flux from the gas that occupies section 3 is:
\[ \hat{Q}_{G3} = C_s \cdot \varepsilon_s \cdot v_3 \cdot A_f \cdot \left( (T_f)^4 - (T_w)^4 \right) \]

where:

- \( A_f = l_p \cdot b_p \) : area of the flame as seen in Fig. 5
- \( v_3 \) : a coefficient depending on the average gas temperature of section 3

The total radiated heat flux, the sum of the aforementioned heat fluxes, is only partly absorbed by the evaporating water, as there is heat loss in the combustion chamber. The absorbed part of the heat flux is:

\[ \hat{Q}_w = \hat{Q}_{tot} \cdot n_w \]

where

\[ n_w = \frac{\hat{Q}_{tot} - \hat{Q}_a}{\hat{Q}_{tot}} : \text{relative thermal coefficient that indicates the heat loss to the environment} \]

\( \hat{Q}_a \) : heat loss of the combustion chamber

The absorbed heat flux is also calculated as the heat flux received from the water in the furnace during vaporizing:

\[ \hat{Q}_D = \dot{m}_D \cdot (h_{D,F} - h_{W,s}) \]

where:

- \( \dot{m}_D \) : steam production mass flow
- \( h_{D,F} \) : furnace outlet steam enthalpy
- \( h_{W,s} \) : furnace inlet water enthalpy

### 2.2.2. Heat exchanger calculations

The total amount of thermal power transferred at each heat exchanger is the amount of heat power received from the water/steam which is equal to the amount of heat power removed from the gas passing through the heat exchanger:

\[ \hat{Q}_{exch,i} = \hat{Q}_{D,i} = \hat{Q}_{g,i} \]

where

- \( \hat{Q}_{D,i} = \dot{m}_D \left( h_{D,F}^{out} - h_{D,F}^{in} \right) : \text{heat flux received from the water/steam at each heat exchanger i} \)
- \( \hat{Q}_{g,i} = \dot{m}_g \left( h_{g}^{in} - h_{g}^{out} \right) : \text{heat flux removed from the gas at each heat exchanger i} \)
- \( \hat{Q}_{exch} = K \cdot A \cdot A\theta_m : \text{total amount of thermal power exchanged} \)
- \( A \) : total heat exchanger surface
- \( A\theta_m \) : average logarithmic temperature

\[ K = f \cdot K_{th} \quad \text{(corrected) total heat transfer coefficient of the heat exchanger} \]

\[ K_{th} : \quad \text{(theoretical) total heat transfer coefficient of the heat exchanger} \]

\( f \) : heat exchanger pollution factor

The average logarithmic temperature is expressed by the inlet and outlet temperatures of gas and water/steam through the following equation:
\[
\Delta \theta_n = \begin{cases} 
\frac{(\theta_{in}^{g} - \theta_{D}^{in}) - (\theta_{out}^{g} - \theta_{D}^{out})}{\ln \frac{\theta_{in}^{g} - \theta_{D}^{in}}{\theta_{out}^{g} - \theta_{D}^{out}}}, & \text{for co-current flow} \\
\frac{(\theta_{in}^{g} - \theta_{D}^{out}) - (\theta_{out}^{g} - \theta_{D}^{in})}{\ln \frac{\theta_{in}^{g} - \theta_{D}^{out}}{\theta_{out}^{g} - \theta_{D}^{in}}}, & \text{for counter-current flow}
\end{cases}
\]

while the theoretical total heat transfer coefficient is expressed through the following:

\[
K_{th} = \frac{1}{\frac{1}{a_g} + \frac{d_o}{2 \cdot \lambda} \cdot \ln \left( \frac{d_o}{d_i} \right) + \frac{d_o}{d_i} \cdot \frac{1}{a_w}}
\]

where:

- \(a_g = a_{g, \text{conv}} + a_w\) heat transfer coefficient by convection and radiation of gas
- \(a_w\): heat transfer coefficient of water/steam
- \(\lambda\): thermal conductivity of the material of pipes
- \(d_o\) and \(d_i\): external and internal diameter of the pipeline, respectively

### 2.2.3. Case study

The fuel used as a reference case is raw lignite with 54 % moisture.

A certain amount of the fuel is dried to a point of 12 % moisture and part of the thermal input is substituted by dry lignite. Apart from the raw-lignite-firing case, which is considered as a reference case, during the co-firing simulation, three cases of thermal substitution are considered:

- 10 % substitution of the boiler’s initial thermal input,
- 20 % substitution of the boiler’s initial thermal input and
- 30 % substitution of the boiler’s initial thermal input

In each case, the mixture of raw and dry lignite is considered as an equivalent fuel with a composition as shown in Table 1.

<table>
<thead>
<tr>
<th>Table 1: Fuel composition</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Fuel ultimate analysis</strong></td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td>C</td>
</tr>
<tr>
<td>H</td>
</tr>
<tr>
<td>N</td>
</tr>
<tr>
<td>O</td>
</tr>
<tr>
<td>S</td>
</tr>
<tr>
<td>ash</td>
</tr>
<tr>
<td>water</td>
</tr>
<tr>
<td>CO₂</td>
</tr>
<tr>
<td>Hu (MJ/kg)</td>
</tr>
</tbody>
</table>

Some of the various other modeling parameters are the fuel, air, gas and working medium (water/steam) conditions (Table 2). The code calculates the furnace first and then the heat exchangers in the direction of water/steam flow, setting initial values for gas and water/steam temperatures. The
procedure is then repeated until the values of the temperatures and heat fluxes converge. A parametric investigation was first carried out, so that the results of the code at the reference case can accurately match the data from the technical report of the facility. In the co-firing cases, cooling water mass flow was adjusted to keep the temperature of the steam below a threshold, which is 540 °C for both the last part of the superheater (SH3) and the reheater (RH2) and 520 °C for the intermediate part of the superheater (SH2). Mass flow, temperature and pressure of the feed water entering the economizer were also held constant in each case, as well as the temperature and pressure of the steam entering the reheater (Table 2).

<table>
<thead>
<tr>
<th>Table 2</th>
<th>Thermal share of substitution (% thermal)</th>
<th>0 % (Reference)</th>
<th>10 %</th>
<th>20 %</th>
<th>30 %</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fuel conditions</td>
<td>Fuel mass flow (kg/s)</td>
<td>167.5</td>
<td>157.94</td>
<td>148.33</td>
<td>138.69</td>
</tr>
<tr>
<td></td>
<td>Fuel mass flow (tn/h)</td>
<td>603</td>
<td>569</td>
<td>534</td>
<td>499</td>
</tr>
<tr>
<td></td>
<td>Ambient Temperature $T_a$ (°C)</td>
<td>20</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Temperature of preheated air (°C)</td>
<td></td>
<td>268.2</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Oxygen Mass content in air (kg/kg)</td>
<td></td>
<td>0.2321</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Humidity (kg/kg dry air)</td>
<td></td>
<td>0.01</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Air ratio</td>
<td></td>
<td>1.2</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Air conditions</td>
<td>Air mass flow (kg/s)</td>
<td>448.89</td>
<td>440.03</td>
<td>431.17</td>
<td>422.31</td>
</tr>
<tr>
<td></td>
<td>Total mass flow (kg/s)</td>
<td>643.39</td>
<td>623.52</td>
<td>603.65</td>
<td>583.79</td>
</tr>
<tr>
<td></td>
<td>Main mass flow to the heat exchangers (kg/s)</td>
<td>590.56</td>
<td>572.320</td>
<td>554.083</td>
<td>535.85</td>
</tr>
<tr>
<td>Gas conditions</td>
<td>Mass flow to the gas recirculation ducts (kg/s)</td>
<td>52.83</td>
<td>51.201</td>
<td>49.570</td>
<td>47.938</td>
</tr>
<tr>
<td></td>
<td>Steam Boiler (economizer) inlet mass flow (kg/s)</td>
<td></td>
<td>249.87</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Water/Steam conditions</td>
<td>Steam boiler (economizer) inlet temperature (°C)</td>
<td></td>
<td>236.3</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Steam boiler inlet pressure (bar)</td>
<td></td>
<td>233.3</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Reheater inlet temperature (°C)</td>
<td></td>
<td>287.7</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Reheater inlet pressure (bar)</td>
<td></td>
<td>34.2</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

The HP and IP turbine steam extraction mass flows were considered constant in each case for the extractions 4-7 (Fig.3), but a greater amount of steam is extracted from the steam extraction 3 for lignite drying, while the extraction 1 is blocked and isolated in the 10% substitution case, and both the extractions 1 and 2 likewise in the other cases (20% and 30%) as the preheaters which used those extractions are now heated from the dried lignite vapour.

The overall net efficiency of the plant is then calculated manually using the known mass flow and enthalpy of steam through the steam turbines, the steam extraction mass flows and the efficiency of the generator.

### 2.3. Commercial code (IPSE PRO) model and assumptions

During the commercial software IPSE PRO modelling, the same co-firing cases as before are examined and the fuel composition is the same as mentioned in Table 1. In this case though, the full water/steam cycle is modelled, including every part of the facility (steam turbines, pre-heaters and extractions, condenser etc.).

Based on the reference case given plant data, the heat exchanger transfer coefficients are calculated. The particular coefficients are taken as constant in the dry coal co-firing cases. In this model, feed water mass flow entering the steam boiler (economizer inlet) is also considered constant while cooling spray water mass flow is adjusted according to the needs in order for the temperature of the superheaters and reheater to be kept below the same threshold in each case (540°C) and subsequently feed water through the water pump is also adjusted and it is not held. The steam boiler inlet water temperature is also different in each case and it is determined by the previous upstream preheaters.
This model also includes lignite drying which is modeled through a heat exchanger. The high temperature stream in the heat exchanger is the steam extraction from the steam turbine, and the low temperature stream is the water content of lignite that is evaporated in the drying process. Lignite is dried through this extra stream of steam extracted from extraction 3 (Fig. 3) and extractions 1 and 2 used to heat the first and second preheater are replaced by the evaporated moisture of lignite, as mentioned previously.

The layout of the facility is represented in Fig. 5, for the reference case and in Fig. 6, for the 30% thermal substitution case.
Fig. 5: Flowsheet diagram of Agios Dimitrios V plant: Reference case
Fig. 6: Flowsheet diagram of Agios Dimitrios V plant: 30% thermal substitution
3. Results

The predicted results for the reference case during calibration are quite accurate for both methods. In the following Q-T diagrams (Fig. 7), a comparison between the technical specifications manual data and the calculated results is demonstrated.

The heat fluxes on the boiler heat exchanger surfaces for the reference and the dry coal co-firing cases are given in Fig. 8. An increase of the total useful heat and the useful heat produced in the furnace section is predicted for the dry-coal firing cases, due to the higher adiabatic temperature and the increase of radiative heat flux. The total useful boiler heat flux increases from 824 to 840 MWth.

Fig. 7: Q-T diagram comparison of the reference case as calculated and predicted by the two methods.
Fig. 8: Prediction of the produced useful heat at each heat exchanger

Fig. 9: Calculated flue gas temperature at furnace and boiler exit
Taking into account that the total thermal input remains constant in all cases considered, the increasing total useful heat leads to increased boiler efficiency. The boiler efficiency calculated by the direct method increases accordingly 1.5 absolute percentage points (Fig. 11). This improvement is also confirmed by the lower flue gas temperature values at the boiler exit. The Q-T diagram (Fig. 12) illustrates more vividly the changes in temperatures and useful heat due to thermal substitution. The calculated plant efficiency rate also increases 0.8 absolute points from 35.4% to 36.2% (Fig. 11). A detailed analysis of thermal and electric power self-consumption is given in Table 3. Finally the calculated specific CO₂ emissions decrease from 1272.048 kg/MWe in the reference case to 1173.326 kg/MWe in the 30% co-firing case (Fig. 13).

### Table 3: Thermal and electric power self consumption

<table>
<thead>
<tr>
<th>Thermal share of substitution (% thermal)</th>
<th>ref.case</th>
<th>10%</th>
<th>20%</th>
<th>30%</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mills (kW)</td>
<td>14.446</td>
<td>13.001</td>
<td>11.557</td>
<td>10.112</td>
</tr>
<tr>
<td>Feed water circulation (pumps) (kW)</td>
<td>11.332</td>
<td>11.444</td>
<td>11.526</td>
<td>11.612</td>
</tr>
<tr>
<td>Total (kW)</td>
<td>25.778</td>
<td>24.446</td>
<td>23.082</td>
<td>21.724</td>
</tr>
<tr>
<td>Heat</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Drying steam heat (kw)</td>
<td>0.000</td>
<td>5.752</td>
<td>11.514</td>
<td>17.267</td>
</tr>
</tbody>
</table>
Fig. 12: Q-T diagram for the reference case and the 10% and 30% substitution cases

Fig. 13: Specific CO\textsubscript{2} emissions

4. Conclusions
Dry lignite firing will play an important role in the next generation of lignite power plants. The pre-drying concept is of particular importance in Greek boilers firing lignite with a high water and ash content. In the current work, the integration of a WTA dryer in an existing Greek power plant is evaluated through thermodynamic cycle calculations. The moisture evaporated from the drying process is used as heating medium and partially replaces the steam bleed utilised for the first low pressure water pre-heaters. The steam cycle of Agios Dimitrios V unit is taken into account for the examination. According to the calculation results, increasing the share of dry coal in the total thermal input, an increase in plant efficiency of up to 0.8 absolute percentage points is predicted, and, therefore, reduced CO$_2$ emissions can be achieved.

To sum up, the overall evaluation of the investigations performed indicates that the application of the WTA pre-drying concept is feasible in an existing Greek power plant without great risks to the operational behaviour of the boiler. The installation of dry lignite burners in an existing Greek power plant by replacing the start up oil burners with dedicated dual burners firing oil and dry lignite dust could be a next step towards the application of the particular technology in Greek power plants. Detailed CFD simulations on the impact of such a co-firing concept on combustion related aspects of the furnace may be carried out. These investigations should focus on the evaluation of the ash slagging propensity in the walls near the dry coal burners and the estimation of the NO$_x$ emissions increase tendency due to the adiabatic temperature increase after the implementation of dry coal co-firing. Moreover, the design of the new generation of lignite boilers, where firing 100% of pre-dried lignite will be implemented, is the further development step required for the full exploitation of the pre-drying technology.

5. References

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