Article

Techno-Economic Design of Flue Gas Condensers for Medium-Scale Biomass Combustion Plants: Impact of Heat Demand and Return Temperature Variations

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Received: 20 May 2019; Accepted: 7 June 2019; Published: 18 June 2019

Abstract: Despite their obvious benefit in terms of energy efficiency and their potential benefit on pollutant emissions, Flue Gas Condensers (FGCs) are still not widely spread in biomass combustion plants. Although their costs have significantly decreased during the last decade, the economic viability of FGC retrofits is not straightforward and their return on investments is mainly dependent on the temperature of the available heat sink and the moisture content of the fuel. Based on a new techno-economic model of a FGC validated with recent industrial data, this paper presents a methodology to assess the economic viability of an FGC retrofitting in a medium-scale biomass combustion plant. The proposed methodology is applied to the case of a typical District Heating plant for which real data was collected. For the first time, the usual assumptions of constant process data generally used are challenged by considering the variability of the return temperature and heat demand over the year. Furthermore, a new concept of optimal configurations in terms of energy savings is introduced in this paper and compared to a strictly economic optimum. The economic feasibility is mainly evaluated by means of the Net Present Value (NPV), Discounted Payback Period (DPP), and the Modified Internal Rate of Return (MIRR). As expected, results show that the higher the humidity level and the lower the return temperature, the higher the economic profitability of a project. The NPV is, however, increased when considering variable inputs: Even with an average return temperature of 60 °C, a mixed operation of the FGC as a condenser and an economizer along the year is predicted, which results in an increased profitability assessment. Considering a constant return temperature over the year can lead to a 20% underestimation of the project NPV. An alternative averaging method is proposed, where two distinct temperature zones are considered: above and below the flue gas dew point. The discrepancy with a detailed temperature variation is reduced to a few percents. Our results also show that increasing the FGC surface beyond the highest NPV can lead to substantial energy savings at a reasonable cost, up to a certain level. The energetic optimum we defined can lead to an increase in energy savings by 17% for the same relative decrease of the NPV.

Keywords: flue gas condenser; retrofitting; biomass combustion; district heating network; hourly demand and return temperature; optimum energetic; economic analysis

1. Introduction

In a conventional boiler, a great amount of energy is lost to the environment due to the heat released by the exhaust Flue Gas (FG). This energy loss consists of sensible heat and water vapor [1].
By adding a Flue Gas Condenser (FGC) to a conventional boiler island, it is possible to recover a significant part of the sensible and latent heat and, therefore, to significantly increase the thermal efficiency of the plant [2]. Several authors have investigated the technical performance of FGC’s, both from a numerical and an experimental perspective [1,3–10]. It was found that the amount of waste heat that can be recovered by a FGC depends mainly on the temperature of the available heat sink, the excess air in the flue gas, and the moisture content of the biomass. The higher the moisture content of the fuel and the lower the heat sink temperature and oxygen content in the FG, the more heat can be recovered from the condensation process. The boiler efficiency (LHV basis) can be improved by 10 to 15% by equipping the boiler with a flue gas condensing unit [9].

Despite their obvious benefit in terms of energy efficiency and their potential benefit on pollutant emissions, FGC’s are still not widely spread in biomass combustion plants with the noticeable exception of Scandinavian countries [11]. One reason for this is related to corrosion issues arising from acid compounds condensation. Although their costs have been significantly reduced during the last decade, these issues still result in high material costs for FGC compared to conventional heat exchangers. Another reason is that the recovery of the latent and sensible heat for the waste energy requires a heat sink with a sufficiently low temperature [7]. Therefore, the economic viability of FGC retrofits is not straightforward and their return on investment is mainly dependent on the temperature of the heat sink available and the moisture content of the fuel.

Existing studies from the literature References [6,7,12,13] assume a single input for the heat sink temperature and fuel moisture content. However, to evaluate the full potential of a FGC retrofit for a given installation, it is necessary to account for their variations with time along a typical year. They also assumed constant heat demands, which is not representative enough to assess the economic viability, even to draw general conclusions. Furthermore, their economic models were based on historical data that are no longer current. Few significant costs have recently been published in the literature. It is, therefore, very difficult to find reliable and accurate data, information which, nonetheless, is essential to carry out an economic analysis.

Based on a new technico-economic model of an FGC, this paper presents a methodology to assess the economic viability of an FGC retrofitting in a medium-scale biomass combustion plant integrated in a District Heating (DH) network. The usual assumptions of constant process parameters generally used are challenged by considering the variability of the return temperature and the heat demand. The impact of these latter assumptions on the viability of a project is highlighted by comparing the results obtained when constant temperature and/or constant heat demand are considered. Furthermore, the concept of an energetic optimum is introduced and compared to the pure economic optimum.

This methodology is based on a combination of the following:

- A state-of-the-art one-dimensional model of an FGC validated against experimental data from the literature;
- A thermodynamic model of the DH network, including the variation of the heat demand and the return temperature in a typical year;
- An economic model, including the cost estimation of capital expenditure (CAPEX) and operational expenditure (OPEX) validated with recent data collected from the industry.

The proposed methodology is applied to the case of a typical DH plant with a total installed power of 35 MWth. The considered process data are derived from actual industrial cases. In order to investigate the sensibility of the results, several scenarios in terms of fuel moisture and average return temperature will be investigated. The ranges of these parameters for which a FGC retrofit is economically viable for a typical medium-scale biomass-fired power plant will be deduced. A comparison between the most economical design with the optimal design in terms of energy savings will be given for the different configurations investigated. The feasibility is mainly evaluated...
by means of the Net Present Value (NPV), Discounted Payback Period (DPP), and the Modified Internal Rate of Return (MIRR).

Firstly, the thermodynamic model of the FGC and biomass boiler will be described (see Section 2). Then, the economic model developed in this study will be presented (see Section 3). In Section 4, the results of the case study will be presented and discussed (see Section 4). Finally, a discussion on the limitations of the model and future works will be outlined (see Sections 5 and 6).

2. Thermodynamic Model

2.1. Flue Gas Condenser

The type of FGC considered in this study is an indirect contact condenser which consists of counter-cross flow U-shaped bare tube bundles with an in-line arrangement. Compared to a staggered arrangement, the bare tube in-line arrangement minimizes the likelihood of an erosion and the trapping of ash and allows for easier cleaning, e.g., using sootblowers [14]. The cooling water circulates inside the tubes while the flue gas flows in horizontal tube bundles from down to up. This kind of FGC configuration is commonly used in the industry and has been previously studied by different authors [1,4,5,12].

During the condensation process in the presence of inert gases, the temperature of the vapor–gas mixture and the water vapor fraction decrease along the condensation line, reducing therefore the driving force of the heat and mass transfer. The usual methods for designing a heat exchanger (i.e., Log Mean Temperature Difference (LMTD) or effectiveness-NTU [15]) become deficient in the condensing part of an FGC, and a step-by-step calculation method needs to be foreseen in order to assess the local interfacial temperature $T_i$ and the local heat and mass transfer [12,16]. In this study, a one-dimensional finite difference method based on References [1,4,16] is used to design the FGC. The heat exchanger is discretized by dividing the tube areas into cells on which governing equations are integrated. The step-by-step procedure is conducted along the flow direction of FG in the FGC where the outlet parameters are calculated as a function of the inlet. The main assumptions of this model are as follows:

- steady-state, one-dimensional flow;
- no heat transfer between the system and its surroundings;
- FG is a mixture of water vapor ($H_2O$) and inert gases. Inert gases are restricted to nitrogen ($N_2$), oxygen ($O_2$), and carbon dioxide ($CO_2$);
- FG is considered as an ideal gas mixture;
- no chemical reactions;
- film condensation only occurs on the heat transfer surface;
- thermodynamic equilibrium is assumed at the condensate film interface;
- continuous laminar condensation film.

At a high inert gas concentration, Osakabe et al. [1] observed several dry spots on the heat transfer surface during condensation in their experimental FGC setup. This implies that the latter assumption taken in this study is not appropriate in the strict sense. However, in the condensate region, the influence of the film condensation on the heat transfer is relatively small compared to the convection and diffusion of the mixture gas side, and so, the non-continuities due to the dry spots can be neglected [1].

The developed model was validated against the experimental data published by Osakabe et al. [1] and Jeong et al. [3–5].

2.2. Biomass Boiler

The amount of energy that can be recovered from the FG not only depends on its own flow rate, moisture content, and temperature, but also on the temperature and the mass flow rate of the heat sink. The latter parameters vary with the operating conditions of the DH system and the behavior
of customers, while the former parameters depend on the moisture content of the fuel as well as the 
operating conditions of the boiler (mainly the excess air) and its efficiency.

The composition of the FG and its mass flow rate per kilogram of dry fuel, taking into account the 
excess air fed to the combustion chamber, can be assessed as suggested by References [11,17]. The fuel 
composition can be restricted to the simplified expression $C_mH_nO_xN_y$. Ash is considered as inert for 
combustion reaction.

In a condensation process, inert gases act as an extra mass transfer resistance which affect the 
performance of a FGC. The amount of inert gases in the FG is obviously correlated to the excess 
air coefficient applied during combustion. In order to ensure a complete and stable combustion, 
the excess air is usually not constant over the entire operating range and it increases when load 
decreases. For a given boiler, the excess air can be assessed experimentally by directly measuring 
the oxygen content in the FG. It is also possible to approximate this value by using typical data. For 
instance, Mermoud et al. [18] reported in their experimental study that, above 50% load, the excess air 
is between 1.5 and 2. According to them, this range is in the usual recommended values [18]. In this 
model, the oxygen content of the flue gas is correlated to the boiler load and the fuel moisture content. 
Two different operating modes are assumed here: low and high fuel moisture content. In both modes, 
a given excess air is set when the boiler runs at a high load and it increases linearly in the low load 
operating range. In general, the higher the moisture content, the higher the excess air.

Grate furnace is a mature and market proven technology. The typical efficiency values reported 
in the open literature and announced by manufacturers range between 80 to 90% without FG 
condensation [18,19]. In their experimental study carried out on two wood boilers (2 MW and 0.65 MW), 
Mermoud et al. [18] showed that the efficiency was barely influenced by the load. This instantaneous 
efficiency can be expressed in terms of thermal and chemical losses. Using the appropriate methods 
and assumptions to evaluate all the other terms [18,20], the FG temperature can be deduced from the 
efficiency of the boiler or vice versa.

3. Economic Model of a Flue Gas Condenser

In an economic analysis, capital expenditure (CAPEX) and operational expenditure (OPEX) are 
usually segregated. CAPEX includes the costs related to the initial investments such as materials, 
manufacturing, testing, shipping, and installation. The costs incurred during the running of the 
equipment are associated with the OPEX, e.g., fluid pumping power, insurance, and operating and 
maintenance costs [7].

The most accurate way to estimate CAPEX and OPEX for a specific project is to obtain quotations 
from manufacturers and return on experience from similar existing plants [21]. Although it is crucial 
to assess the economic viability of FGCs, such information is however hardly found in the literature [7]. 
In the past years, few authors attempted to develop economic models to estimate the different costs 
linked to an FGC.

Che et al. [13] evaluated the price of the heat exchanger for different materials by multiplying the 
weight of the tubes by a material price. The piping, the valves, and the casing of the heat exchanger 
were considered additional fixed prices. Costs linked to manufacturing and installation were taken 
into account by multiplying the material cost by specific factors. The investigated surface range was 
between 8 and 145 $m^2$. The OPEX was evaluated by estimating the extra power consumption due to 
the additional pressure drops and the expenses related to maintenance and repair. The latter were 
taken as three times the initial investment spread over the lifetime period.

The FG Condenser investigated in the economic analysis conducted by Terhan and Comakli [12] 
was a cross flow tube bundle heat exchanger with a size of 80 $m^2$. The CAPEX associated with the FGC 
was evaluated using a fixed price per tube length for material and manufacture costs. In addition to 
the extra electricity consumption for the pump and fan, additional costs for the recurring maintenance 
were taken into account.
Hazell [22] estimated the price of a tube bundle heat exchanger for different materials. They focused their work on FGCs with a large surface area (more than 9000 m$^2$). It was suggested that, for these sizes and when expensive materials are used, such as stainless steel and nickel alloys, CAPEX is dominated by the tube material cost. The manufacturing and plant installation expenses were based on the labor costs associated with the manufacturing of carbon steel shell and tubes heat exchanger of the same size. An additional 30% was added for any unaccounted element, leading to a total factor, for the manufacture and installation, of 3.90 times higher than the carbon steel tube material cost.

In their study, Chen et al. [7] calculated the initial investment for a shell and tubes heat exchanger. The CAPEX was correlated with historical data costs on which corrective factors are applied to account for design or material changes from the reference equipment. The OPEX was based on the extra power needed for the utilities and on a fixed price for the condensate chemical treatment. In addition, an estimated value of 6% of the initial investment cost per year was used to account for other maintenance expenses.

The model developed in this study is intended to be as flexible and accurate as can be, using a state-of-the-art economic model validated against recent industrial data. The costs of FGC and all the ancillary materials (i.e., instrumentation, water pump, valves, etc.) were estimated with the Bare Module (BM) method proposed in Reference [23], which is ultimately based on parametric correlations such as the one suggested by Chen et al. [7] for FGCs.

### 3.1. Capital Expenditure

In the Bare module method, the main pieces of equipment of the system are estimated separately. In the case of an FGC, three main parts can be distinguished: the heat exchanger, the condensate water treatment, and potentially an extra FG fan. Since the additional pressure losses in the FG side for the range of heat exchanger surfaces investigated in this study are low, it is assumed that no extra fan is needed.

For the two remaining parts, the Free On Board cost, $C_{FOB}$, which is the cost of equipment ready for shipment from supplier, is firstly assessed. The $C_{FOB}$ is usually expressed in terms of a base cost $C_0$ validated for an equipment size $S_o$ multiplied by a scaling factor $(S/S_o)^n$. Then, the fully installed and functioning unit, the Bare Module cost $C_{BM}$, is deduced by multiplying $C_{FOB}$ by factors that account for all associated equipment (e.g., concrete, piping, electrical, instrumentation, insulation, etc.) and installation labor required in a radius of about 1 m out from the sides of the equipment. The imaginary space described by the circle is called the module. The total module cost, $C_{TM}$, is evaluated by imputing extra costs for the contractors fees and contingency during construction to $C_{BM}$. $C_{TM}$ does not account for land, spare parts, working capital, startup expenses, etc. However, in the frame of this study, the CAPEX is assumed to be limited to $C_{TM}$. Finally, the CAPEX of each subparts are summed up to obtain the total CAPEX of the system [23,24].

The reference cost is generally based on data collected from existing industrial cases, i.e., actual offers prepared by manufacturers. To account for the escalation rate over the years, reference costs need to be indexed. In the industry, several indices exist to adjust the price from one period to another [25]. In this study, the Chemical Engineering Plant Cost Index (CEPCI) was used, as suggested in Reference [23]:

$$C_{TM_{current}} = C_{TM} \frac{CEPCI}{CEPCI_{Ref}} \quad (1)$$

As highlighted by Vatavuk [25], although the CEPCI is an excellent substitute if there is no access to current costs, CEPCI is merely a model that should be cautiously applied. In particular, in order to reduce the uncertainties, it should not be used to escalate costs for periods greater than five years. Recent industrial data on the cost of FGCs was therefore retrieved and analysed to validate the economic model developed in this study. To the best knowledge of the authors, it is the first time that an economic model of an FGC has been confronted with real data.
3.1.1. Flue Gas Condenser

In this study, the different factors and installed instruments costs are taken from Reference [23], while the base cost is adjusted in order to minimize the least squares error between the model and the data retrieved from the industry. A summary of the different parameters used are depicted in Table 1. The same notations as in Reference [23] were used. The base cost \( C_o \) corresponds to the cost of an FGC with a heat transfer area of 407 m\(^2\), which is the average surface of the retrieved industrial cases. The FGC is a counter-cross flow U-shaped bare tube bundle with a square tube layout. It works at atmospheric pressure and is made of stainless steel. The scaling factor is \( n = 0.71 \) and the indicated range of validity is set according to the retrieved industrial data (i.e., 150 to 750 m\(^2\)). The cost is for the basis CEPCI = 1000. The value of the CEPCI Index for the year 2017 was 558.3, meaning that the costs reported in this study are around 1.8 times the value in 2017 [23].

Table 1. A summary of the different parameters of the Bare Module (BM) model proposed by Reference [23]. The correction factors and installed instruments costs are taken from Reference [23], while the base cost is adjusted in order to minimize the least squares error between the model and the data retrieved from the industry.

<table>
<thead>
<tr>
<th>Characteristic</th>
<th>Values</th>
<th>Ref</th>
</tr>
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<tbody>
<tr>
<td>Base cost</td>
<td>( C_o = 227,700 ) €</td>
<td></td>
</tr>
<tr>
<td>Reference area</td>
<td>( A_o = 407 ) m(^2)</td>
<td></td>
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<tr>
<td>Range of validity</td>
<td>150 to 750 m(^2)</td>
<td></td>
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<tr>
<td>Scaling factor</td>
<td>( n = 0.71 )</td>
<td>[23]</td>
</tr>
<tr>
<td>Design and operational factors</td>
<td>( f_d = f_{op} = 1 )</td>
<td>[23]</td>
</tr>
<tr>
<td>Labor and Material factor</td>
<td>( f_{LM} = 1.3 )</td>
<td>[23]</td>
</tr>
<tr>
<td>Installed instruments cost</td>
<td>( C_{Instr} = 23,417 ) €</td>
<td>[23]</td>
</tr>
<tr>
<td>Taxes, freight and insurance factor</td>
<td>( f_{Freight} = 0.15 )</td>
<td>[23]</td>
</tr>
<tr>
<td>Off-sites + indirects for home office and field expenses factor</td>
<td>( f_{Eng} = 0.20 )</td>
<td>[23]</td>
</tr>
<tr>
<td>Contractors fees factor</td>
<td>( f_{Fees} = 0.05 )</td>
<td>[23]</td>
</tr>
<tr>
<td>Contingency for unexpected delays factor</td>
<td>( f_{Delays} = 0.1 )</td>
<td>[23]</td>
</tr>
<tr>
<td>Design contingency for changes in scope during construction factor</td>
<td>( f_{Mod} = 0.1 )</td>
<td>[23]</td>
</tr>
<tr>
<td>Chemical Engineering Plant Cost Index of reference</td>
<td>( CEPCI_{Ref} = 1000 )</td>
<td>[23]</td>
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The Installed instruments cost suggested by Reference [23] is translated in Euro at the prevailing exchange rate on 17 July 2018 [26].

Figure 1 compares the proposed model with those described earlier and with the retrieved industrial data. The actual costs were indexed with the CEPCI and are related to a heat exchanger made of stainless steel. As already discussed in Reference [27], the various models significantly diverge. These gaps can be explained by the questionable assumptions made and the limited range of validity of the models found in the literature. As a general rule, a global accuracy of \(+/-30\%\) of the investment cost should be considered for this level of detail, corresponding to a pre-feasibility study stage [23,28].

3.1.2. Condensate Water Treatment System

Condensate is a mixture of water and different components such as solids, salts, and heavy metals. These impurities come from dust particles that were not removed by the FG treatment and the absorption of certain gaseous components. The quality of the condensate is therefore determined by the FG properties, the upstream FG treatment technologies, and the reduction efficiency of FGC [10,11].
In this study, condensate is considered to be released into the environment after an appropriate treatment. Condensate water must be purified in order to comply with the regulations in force before being discharged. The most suitable condensate treatment process depends on the amount and nature of impurities in the condensate and the level of quality required [10]. The limit values set in the Best Available Techniques-associated emission levels (BAT-AELs) for direct discharges to a receiving water body from a flue–gas treatment can be used as reference [29]. As medium-scale biomass combustion plants are generally equipped with efficient dedusting systems (e.g., fabric filters), a sedimentation tank and ion exchangers are assumed to be sufficient to reach the acceptable limits in terms of heavy metals and salt concentrations. pH is controlled by adding a chemical additive. This configuration is the same as the one recommended by Reference [8].

Based on data from the industry, the CAPEX for the sedimentation tank is assumed to be 18 € for a condensate flow of 1 L/h. The ion exchange cost including chemical storage, associated piping and valves, and instrumentation and controls is taken as 15.8 € for a condensate flow of 1 l/h. The CAPEX should be calculated according to the maximum system capacity [8].

### 3.2. Operational Expenditure

The OPEX is assumed to be the sum of the cost due to the maintenance of the FGC (e.g., fouling removal) and all the subsystems associated, the condensate chemical treatment, and the extra electrical power consumption.

A cost of 1 € per cubic meter of condensate is considered for the chemical treatment, and 6% of the initial investment per year was used to account for the maintenance expenses [7,8]. Chen et al. [7] suggests a cost of $0.45 per cubic meter of condensate, and Reference [8] assumed the operations and maintenance costs for ion exchange as $0.003 per U.S. gallon of condensate. The chemical treatment
cost was assumed as the sum of these prices. They were adjusted by using the Produced Price Index for water treating compounds [30] and then translated in Euro at the prevailing exchange rate on 17 July 2018 [26]. The extra power consumption was directly related to the additional pressure losses in the FG and water side. They can be evaluated, in both side, with Equation (2). $P_{\text{elec}}$ is the extra electrical power, $V$ is the volumetric flow, and $\Delta P$ is the extra pressure drop. The pump and fan global efficiencies, $\eta$, are taken as 80% [12].

$$P_{\text{elec}} = \frac{V\Delta P}{\eta}$$  \hspace{1cm} (2)

### 3.3. Profitability and Economic Viability of a Project: Assessment Tools

The selection of the optimum FGC surface needs to be based on objective criteria. In this study, Net Present Value (NPV), Discounted Payback Period (DPP), and the Modified Internal Rate of Return (MIRR) are used to outline the project with the higher economic performance [31].

The NPV is defined as the sum of all the discounted incomes minus costs over the project life time, noted as $N$ (Equation (3)). $C_0$ is the initial investment. $I_n$ and $C_n$ are respectively the incomes and OPEX in year $n$, while $r$ is the discount rate.

$$\text{NPV} = -C_0 + \sum_{n=1}^{N-1} \frac{I_n - C_n}{(1 + r)^n}$$  \hspace{1cm} (3)

$$I_n = Q_{\text{saved}}^{\text{bio}} C_{\text{bio}} + Q_{\text{saved}}^{\text{gas}} C_{\text{gas}}$$  \hspace{1cm} (4)

$$C_n = E_{\text{elec}} C_{\text{elec}} + Q_{\text{cond}} C_{\text{cond}} + C_{\text{extra}}$$  \hspace{1cm} (5)

The incomes $I_n$ are directly linked to fuel saving due to the improved efficiency when a biomass boiler is retrofitted with an FGC. We indeed consider here the case of a retrofit, where the heat demand remains constant. For a new built or when the heat demand can be extended, the income should be related to the revenues of the extra heat delivered to the clients. In Equation (4), $Q_{\text{saved}}^{\text{bio}}$ and $Q_{\text{saved}}^{\text{gas}}$ refer to the quantity of biomass and gas saved and the corresponding cost per unit of fuel are given by $C_{\text{bio}}$ and $C_{\text{gas}}$. The OPEX, noted as $C_{\text{n}}$, can be expressed by Equation (5). $E_{\text{elec}}$ is the extra power consumed and is computed by integrating $P_{\text{elec}}$ over the year (Equation (2)). $C_{\text{elec}}$ is the price of the electricity. $Q_{\text{cond}}$ refers to the total amount of condensate in a year, and $C_{\text{cond}}$ is the cost per unit of condensate associated with the chemical treatment, while $C_{\text{extra}}$ corresponds to the maintenance of the FGC systems.

The MIRR (Equation (6)) is defined as the discount rate that makes the investment equal to the future value of the cash flows from the investment (i.e., NPV = 0). As with the Internal Rate of Return (IRR), it is an image of the maximum rate that a project can support and still break even by the end of the project life. A project should be considered as financially attractive when MIRR exceeds the project’s hurdle rate, and the higher the MIRR, the more profitable the project. However, contrary to the IRR, the MIRR accounts for the difference between the reinvestment and investment rates [32].

$$\text{MIRR} = \left( \frac{\text{FV}_{\text{in}}}{\text{PV}_{\text{out}}} \right)^{\frac{1}{n}} - 1$$  \hspace{1cm} (6)

In Equation (6), $n$ is the number of periods, $\text{PV}_{\text{out}}$ is present value of negative cash flows at the investment rate, and $\text{FV}_{\text{in}}$ is future value of positive cash flows at the reinvestment rate.

DPP is the number of years needed by the project in recovering the investment on the present value basis. The time period is determined by finding the value of “$N$” in Equation (3) such that NPV = 0. Based on this criterion, a given project will be approved only if the DPP is less than a targeted period previously determined.
None of these parameters alone reflects all the dimensions of profitability of a project which are relevant to capital expenditure decisions. They should, therefore, be considered altogether when assessing the profitability of a given project [33,34].

Given the specific operating mode and fuel moisture content, the larger the FGC heat transfer surface, the more fuel can be saved. On the other hand, a higher surface implies higher CAPEX and OPEX. A positive NPV means that the discounted incomes exceed costs and that, therefore, the proposed project can be considered as viable.

In this work, a distinction is made between the FGC surfaces corresponding to the purely economic optimum FGC surface (\(A_{\text{NPV}}\)) and the energetically optimal FGC surface (\(A_{\text{E}}\)). The former is the one resulting in the highest NPV, noted NPV\(_{\text{max}}\), and the amount of energy recovered by the FGC in this case is noted E\(_{\text{NPV}}\). The latter is defined as the surface for which the relative decrease of NPV beyond its maximum value equals the relative increase in recovered energy (again compared to the maximum NPV) as expressed by the following equation:

\[
\frac{(\text{NPV}_{\text{max}} - \text{NPV}_{\text{E}})}{\text{NPV}_{\text{max}}} = \frac{(\text{E}_{\text{max}} - \text{E}_{\text{NPV}})}{\text{E}_{\text{NPV}}}
\]

\(\text{E}_{\text{max}}\) and \(\text{NPV}_{\text{E}}\) correspond to the FGC surface for which an extra energy recovery is not worth the required investment. This interpretation will become clearer when illustrated by the relevant graph.

4. Case Study: A Representative District Heating Plant

The proposed methodology outlined in this study is applied to a typical medium-scale DH plant. The economic viability of an FGC retrofitting is assessed as a function of the surface of the heat exchanger, the fuel moisture, the return temperature, and the variability of the heat demand during the year by means of the NPV, DPP, and MIRR.

The input data required for the techno-economic model are based on process data retrieved from an existing DH plant located in France and on key values taken from the literature. In particular, the heat demand and return temperature variations around their mean values are retrieved from the considered plant. The fuel characteristics are assumed to be constant over the year. In order to study the sensitivity of the results, this latter parameter and the average return temperature will be varied around their actual values in different simulations. The ranges of these parameters for which an FGC retrofit is economically viable for a typical DH plant will be deduced. The results will then be compared to the ones obtained when the usual assumptions of constant process data over the year are considered.

This section is divided in three distinct parts. Firstly, a brief description of the considered typical DH is given. An analysis of the process data in terms of relative shares of the different fuels in the heat production is outlined. Secondly, a calibration of the model and simplifications made in the frame of this case study are presented. Finally, the results of this techno-economic study are discussed.

4.1. Description of the Plant

The DH plant comprises gas boilers with a total output thermal power of 22.5 MW, a 4 MW\(_e\) Combined Heat and Power (CHP) gas engine, a water tank storage of 400 m\(^3\), and two biomass boilers with output thermal powers of 3.2 MW and 5.3 MW (see Figure 2). This biomass boiler’s configuration allows more flexibility in the thermal power production by lowering the turn-down ratio. The nominal load is the sum of the two, while the minimum load corresponds to the minimum of the smaller biomass boiler.

The operating data were sampled over 1 year every minute and then averaged on an hourly basis. In a typical year, between mid-May and mid-September, the heat demand is low and then it increases and reaches a peak around mid-January before decreasing; see Figure 3. Biomass boilers work most efficiently when they can run at a nominal load for long periods, and it is recommended not to go below 30% of their maximum capacity. Therefore, when connected to a DH network, the
biomass boilers are usually sized to supply the base load while other heating systems cover the peak and the low load [35]. During the low heat demand season, the load is mainly covered by biomass boilers. In the remaining period of the year, both fuels are used and the base loads are ensured by the CHP unit. In the considered representative case, 56% of the heat load was covered by the biomass boilers, 21% was covered by the gas boilers, and 23% was covered by the CHP unit.

The variability of the return temperature is given in Figure 4. The yearly mean return temperatures, noted $T_r$, is 55 °C while the variance is equal to 22 °C. The return temperatures in the low heating season are typically higher.

Figure 2. A drawing of the district heating plant on which the proposed methodology is applied. The DH plant comprises gas boilers with a total output thermal power of 22.5 MW, a 4 MW, CHP gas engine, a water tank storage of 400 m$^3$, and two biomass boilers with output thermal powers of 3.2 MW and 5.3 MW.

Figure 3. The total heat load supplied by the biomass and gas boilers over a typical year retrieved from an existing DH plant. The DH plant provides 55.7 GWh of heat among which 56% was ensured by biomass boilers, 21% was ensured by the gas boilers, and 23% was ensured by the CHP unit.
4.2. Hypothesis and Model Parameters

As already stated, the performance of a FGC is very sensitive to the heat sink temperature available (i.e., return temperature), the fuel moisture, and the oxygen content in the FG.

The return temperature depends on different parameters such as the DH operating conditions, the network characteristics, the outside temperature, and the load demand, while the fuel moisture content is affected by its origin, its handling, and storage conditions and by a possible drying process. The fuel characteristics are kept constant over the year. The return temperature and heat demand are fitted to the one of the DH network considered. The influence of the fuel moisture content on the viability of a specific project is assessed by considering different values (MC = 30, 40, and 50%) while the return temperature data are shifted around their mean value ($T_r = 50, 55,$ and $60 $ °C). In total, 9 different scenario are investigated, noted $T_{rx}MC_{yy}$, where $T_{r55}MC_{40}$ is the base case (bc). The objective here is to assess for which ranges of biomass moisture content and return temperature such a typical medium-scale system is suitable for an FGC retrofit.

The return temperature duration curves studied are illustrated on Figure 5. The biomass FG’s dew points, evaluated at nominal load, are 62.6 °C, 58.7 °C, and 53.7 °C for MCs of 30%, 40%, and 50%, respectively. The intersections between the curves and these three dew points are illustrated by dots. These dots highlight the period of time where the return temperatures are low enough to allow condensation for various MC. For instance, in the base case scenario (55 °C average return temperature and 40% moisture content), the return temperature is 71% of the time below the dew point.

The oxygen content of the flue gas is correlated to the boiler load and the fuel moisture content. In the case where the moisture content is lower than 35%, the oxygen content is set to 6.5% when the boiler load goes from 100 to 50% and then it increases linearly from 6.5 to 8% in the power range between 50% and 30%. For a higher moisture content, the first and second set points are respectively replaced by 7.5 and 9%.

The FG temperature at the inlet of the FGC is fixed at 180 °C. The boiler efficiency can be deduced from this value. When the moisture content goes from 30 to 50%, the boiler efficiency decreases from 87 to 85%.

It is assumed that the total heat demand and thermal power provided by the CHP unit remain unchanged such that the energy recovered from the FGC surrogates the thermal power provided by the biomass and the gas boilers, hence resulting in fuel savings. When additional heat is provided by the FGC, the gas boiler output is decreased first.
As mentioned in Section 2.1, the FGC considered in this study consists of a counter-cross flow U-shaped bare tube bundle with a square tube layout. The design parameters shown in Table 2 were selected based on the different recommendations given in References [6,7,36]. The number of transversal tubes $N_t$ is calculated to reach the desired water velocity. Knowing $N_t$, the tube length is selected so that the desired FG velocity is reached. The number of longitudinal tubes $N_l$ is the design variable that is adjusted to modify the total heat transfer surface. In this case study, the total cooling water volume flow rate $\dot{V}_w$ is assumed to be 100 m$^3$/h. The number of transversal tubes is calculated as 70, and tube length is approximately 1.9 m.

Table 2. The Flue Gas Condenser (FGC) design parameters considered in the representative DH plant case study. They are selected based on the different recommendations exposed in References [6,7,36].

<table>
<thead>
<tr>
<th>Characteristic</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tube outside diameter, $D_o$</td>
<td>25 mm</td>
</tr>
<tr>
<td>Tube inside diameter, $D_i$</td>
<td>22.4 mm</td>
</tr>
<tr>
<td>Pitch, $S_t/D_o$</td>
<td>1.4</td>
</tr>
<tr>
<td>Cooling water velocity, $V_{cw}$</td>
<td>1 m/s</td>
</tr>
<tr>
<td>Maximum FG velocity, $V_{max}^{FG}$</td>
<td>5 m/s</td>
</tr>
<tr>
<td>Total water volume flow rate, $\dot{V}_w$</td>
<td>100 m$^3$/h</td>
</tr>
</tbody>
</table>

Specific market conditions such as the wood, electricity, and natural gas prices also have an effect on the feasibility of an FGC retrofitting. These prices often fluctuate from year-to-year and per country [34]. Furthermore, wood prices vary with the moisture content, the fuel conditioning, the delivery size, and the type of lorry acceding the site. Biomass with a higher moisture content and a larger particle size have a lower price [37]. Electricity and natural gas prices are fixed to 0.2 €/kWh and 0.04 €/kWh, respectively [34]. The price for wood chips is assumed to be 0.02 €/kWh [38]. It should be noted that this price does not include the transport cost, meaning that the incomes are most probably higher than those presented in the next section. All the economic parameters are shown in Table 3. CAPEX is calculated based on the model described in Section 3.1, and the most unfavorable case is considered (“+30%” curve in Figure 1).
Biomass composition is taken as $\text{CH}_{1.44}\text{O}_{0.66}$ by mass on dry ash free base, and the ash content is assumed to be 2.7% on a dry base. The lower heating value of the fuel is estimated as 18.54 MJ/kg daf [11].

Table 3. Economic factors assumed in the representative DH plant case study.

<table>
<thead>
<tr>
<th>Economic Factors</th>
<th>Value</th>
<th>Ref</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wood price</td>
<td>0.02 €/kWh</td>
<td>[38]</td>
</tr>
<tr>
<td>Gas price</td>
<td>0.02 €/kWh</td>
<td>[34]</td>
</tr>
<tr>
<td>Electricity price</td>
<td>0.2 €/kWh</td>
<td>[34]</td>
</tr>
<tr>
<td>Chemical treatment price</td>
<td>1.03 €/m$^3$ <em>cond</em></td>
<td>[7,8]</td>
</tr>
<tr>
<td>Fixed maintenance cost</td>
<td>6% of the CAPEX €/year</td>
<td>[7]</td>
</tr>
<tr>
<td>Investment rate (discount rate)</td>
<td>10%</td>
<td></td>
</tr>
<tr>
<td>Reinvestment rate</td>
<td>7%</td>
<td></td>
</tr>
<tr>
<td>Project life time</td>
<td>20 years</td>
<td></td>
</tr>
</tbody>
</table>

4.3. Simulation Results

Firstly, the simulation outcomes for the base case scenario $T_r=55 \, ^\circ C$ and $\text{MC} = 40\%$ are analyzed in details. In particular, the selection of the optimal surfaces (i.e., $A_{\text{Eco opt}}$ and $A_{\text{E opt}}$) based on the objective criteria presented in Section 3.3 and the impact on the energy savings are discussed. The results are then compared to the ones obtained when the usual assumptions of constant process data are used. In a second step, an overview of the results for other boundary conditions is given in order to derive the range of return temperatures and moisture contents for which an FGC retrofit is economically viable for a typical medium-scale biomass-fired power plant.

4.3.1. Base Case: $T_r=55 \, ^\circ C$ and $\text{MC} = 40\%$

Profitability and Economic Viability of the Project

The discounted cash flow streams over the project life time for different FGC areas are depicted in Figure 6. The initial investment is the present value at $t = 0$, the DPP is the project duration for which the present value $= 0$, and the NPV is the present value at the end of the project (here for $t = 20$ years). The MIRR is determined by solving Equation (6) and is also given in Figure 6. The results obtained with four different FGC areas are shown: $A = 50, 147, 493,$ and $615 \, m^2$. The NPV with a surface area equal to $615 \, m^2$ is negative, and therefore, this option should not be considered. In the other cases, the NPVs are positive, meaning that these projects are economically viable. However, the economically optimal FGC surface is the one resulting in the highest NPV. In this scenario, the optimal surface is $147 \, m^2$, which leads to an NPV of $299 \, k€$, a DPP of $4.8$ years, and an MIRR of $14.7\%$. Although the initial investment is higher, the DPP is barely longer than in cases where the surface is smaller (less than one year difference). On the other hand, this period is much shorter than the one related to $A = 493 \, m^2$. By comparing the slope of the different curves, one can observe that, up to a certain FGC’s surface, the higher the initial investment (i.e., the larger the FGC’s surface), the steeper will be the slope of the cash flow stream. A positive slope means that the difference between the income and expenses is positive and that the stiffness of the slope is positively correlated with the benefits. In general, the larger the FGC, the more heat can be recover and, therefore, the higher is the income. However, at some point, a further increment in the FGC’s area will only result in a slight increase in the heat recovery which will lead to a situation in which the additional income will not be able to overcome the extra expenses. This situation can be observed by comparing the curves for $A = 493 \, m^2$ and $A = 615 \, m^2$. Over one year, the extra energy recovered by the FGC in the optimal configuration allows savings of approximately $1757 \, MWh$ of biomass and $1653 \, MWh$ of gas corresponding to a reduction in the consumption of $5.7\%$ and $6.7\%$ respectively. From an economic point of view, around $80 \, k€$ can be saved per year.
Economic Optimum versus Energetic Optimum

The variation of the NPV as a function of the amount of energy saved is given in Figure 7. The NPV and the saved energy are expressed relative to those obtained for the maximum NPV (i.e., \( NPV_{\text{max}} = 299 \) k€ and \( E_{NPV_{\text{max}}} = 3410 \) MWh). Each point on this curve corresponds to an FGC heat exchange surface. While the NPV exhibits a maximum value, the recovered energy continuously increases with the surface. The efficiency of the heat recovery, however, decreases above a certain limit, as the curve approaches a vertical asymptote (the maximal amount of recovered energy, i.e., the total enthalpy of the flue gas). The economic and energetic optimum are highlighted by circle and rectangular dots, respectively. The energetically optimal FGC surface is approximately 267 m\(^2\), corresponding to an 82% increase compared to the economically optimal surface. The additional heat transfer surface raises the energy savings by 17.1% and, therefore, by definition (see Equation (7)), decreases the NPV by the same percentage. A further increment in the surface would lead to a situation where the relative increase of the energy savings is lower than the relative decrease of the NPV, which is considered here as the limit above which the additional saved energy in not worth the additional investment. The sharp slope of the curve beyond the energetic optimum supports this interpretation.

Figure 6. Discounted cash flow streams over the project life time for four different FGC areas and in the specific case where the yearly mean return temperature and the moisture content are 55 °C and 40%.

Figure 7. A comparison between the most economical designs and the optimal designs in terms of energy savings in the specific case where the yearly mean return temperature and the moisture content are 55 °C and 40%.
Assumption Impacts on the Profitability

The impact of the usual assumptions used on the energetic and economic optimum is highlighted by investigating four additional cases, noted as H1 to H4 (see Table 4). The results presented in the previous section are used as the reference scenario, noted as H0, in which the variability of both the return temperature and the heat demand are taken into consideration (\(Tr(t)\) and \(Q_{load}(t)\)). In H1 and H2, the assumption on the heat demand is unchanged \(Q_{load}(t)\), while in the two other scenario (i.e., H3 and H4), the average heat demand is used \(Q_{load}(t) = \bar{Q}_{load}\). The return temperature in H2 and H4 is equal the yearly mean return temperature all over the year \(Tr(t) = \bar{Tr}\). In H1 and H3, the return temperatures are splitted in two groups, below and above the dew point of the flue gas, and averaged. In each group, the mean value of the return temperatures was used in the calculation procedure \(Tr(t^{\text{nc}}) = \bar{Tr}^{\text{nc}}\) and \(Tr(t^{c}) = \bar{Tr}^{c}\). The subscripts “nc” and “n” refer to no condensation and condensation, respectively.

The variation of the NPV as a function of the FGC’s surface is given in Figure 8 for the different hypotheses. All values are expressed in relative percentages of the economic optimum computed in the reference case. The economic and energetic optimums are highlighted by circle and rectangular dots, respectively. Small differences between the reference case and H1 can be observed. The optimum of H1 is slightly overestimated (less than 2% of difference). This shows that considering two constant return temperature zones, above and below the dew point of the flue gas, is a very effective simplification of the return temperature data, as long as the considered heat load remains variable along the year. This could greatly simplify the analysis of existing installations or the prefeasibility studies for new installations. Such a good agreement between H0 and H1 can be explained by the shape of the curve representing the FGC’s recovered energy vs. return temperature that presents a discontinuity in its first derivative when condensation starts. Over the whole considered range of temperature, this curve is not linear, but it can however be efficiently approximated by two separated linear functions below and beyond the dew point.

Table 4. A summary of the hypothesis used in the additional cases investigated to highlight the impact of the usual assumptions on the energetic and economic optimum.

<table>
<thead>
<tr>
<th>Hypothesis</th>
<th>Description</th>
<th>Notation</th>
</tr>
</thead>
<tbody>
<tr>
<td>H0 (reference scenario)</td>
<td>Variable return temperature (Tr(t))</td>
<td>(Q_{load}(t))</td>
</tr>
<tr>
<td>H1</td>
<td>Constant return temperatures: below and above the dew point of the flue gas (Tr(t^{\text{nc}}) = \bar{Tr}^{\text{nc}}) (Tr(t^{c}) = \bar{Tr}^{c})</td>
<td>Variable heat demand (Q_{load}(t))</td>
</tr>
<tr>
<td>H2</td>
<td>Constant return temperature (Tr(t) = \bar{Tr})</td>
<td>Variable heat demand (Q_{load}(t))</td>
</tr>
<tr>
<td>H3</td>
<td>Constant return temperatures: below and above the dew point of the flue gas (Tr(t^{\text{nc}}) = \bar{Tr}^{\text{nc}}) (Tr(t^{c}) = \bar{Tr}^{c})</td>
<td>Constant heat demand (Q_{load}(t) = \bar{Q}_{load})</td>
</tr>
<tr>
<td>H4</td>
<td>Constant return temperature (Tr(t) = \bar{Tr})</td>
<td>Constant heat demand (Q_{load}(t) = \bar{Q}_{load})</td>
</tr>
</tbody>
</table>

The NPVs are strongly underestimated (between –10% and –20%) when the other hypothesis are applied. This is due to the fact that low return temperatures are correlated with a high heat demand during the winter. Considering the variations of the return temperature allows for the prediction of a mixed operation of the FGC as a condenser and an economizer along the year. This effect is, of course, reinforced when the variability of the demand is also taken into account, as condensation is more likely to occur when the demand is high.
However, due to the flatness of the reference curve around the economic optimum, the consideration of too restrictive assumptions would not lead to a significant discrepancy between the actual economic viability and the optimum value. Nevertheless, an underestimation of the NPV could lead to a rejection of a project while the actual NPV could be significantly higher. The energetic optimum is even more sensitive to the considered assumptions: Hypothesis H4 leads to an underestimation of the energetic optimum FGC surface of 20%. Due to the shape of the curves of Figure 8, the chosen lower surface would actually result in a higher NPV than the real energetic optimum (H0).

![Figure 8. Impact of the assumptions on the energetic and economic optimum.](image)

### 4.3.2. Sensitivity Analysis

The strong dependence of the viability on the fuel moisture and return temperature is highlighted in Figure 9. Both subfigures show the NPV as a function of the surface of the FGC for the typical medium-scale DH that we considered. On the right hand side, the average return temperature is fixed and the influence of the moisture content is illustrated while it is the opposite on the left hand side. All curves were obtained using variable heat loads and return temperatures. The higher the humidity level and the lower the return temperature, the higher the NPV. In all cases, the NPV reaches a maximum value corresponding to the economically optimal surface (i.e., circle dots on Figure 9).

The maximum NPV increases by 51% when the average return temperature is decreased from 60 to 55 °C. A further decrease to 50 °C results in an NPV increase of 46%. Depending on the boundary conditions, the energetically optimal surfaces (i.e., rectangular dots on Figure 9) are 45 to 82% larger than the economic optimum. Although lower NPVs are obtained, energy savings are increased by 12 to 17% (relative).

Figure 10 shows the optimal surface for each configuration (i.e., for a return temperature and moisture content). As expected, the optimal NPV and the corresponding heat exchange surface increase as the moisture content increases and the return temperature decreases. The NPV, DPP, MIRR, and saved energy values for the different boundary conditions are summarized in Table 5. Due to the recent decrease of the FGC cost on the market, all projects investigated in this study are economically viable in the strict sense of the NPV (i.e., positive NPV) when the appropriate FGC surface is selected. The consideration of variable return temperatures and heat loads along the year also leads to the prediction of mixed operations of the FGC as a condenser and an economizer, which further increase the assessed profitability of the projects. When condensation occurs more than 50% of the time, NPV oscillates between 260 and 698 k€. DPP and MIRR go from 3.1 to 5.4 years and from 14.2 to 16.7%, respectively. When the FGC is mainly operated as an economizer, NPV does not exceed 200 k€. MIRR is around 13%, while the DPP is over 5.8 years.
As investment decisions rely on a set of indicators rather than a single one, a positive NPV cannot be the only criterion to assess the profitability of an FGC retrofit. For instance, if a DPP lower than 5.5 years is required, all scenarios where condensation occurs during less than 50% of the time would be rejected.

Table 5. Net present values, discounted payback periods, modified internal rates of return, and saved energy for the different boundary conditions investigated in this study. Scenario where condensation occurs during less than 50% of the time are highlighted in gray.

<table>
<thead>
<tr>
<th>Return Temperature Tr = 50</th>
<th>55</th>
<th>60</th>
</tr>
</thead>
<tbody>
<tr>
<td>Moisture Content MC (%)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>30</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Optimal surface (m²)</td>
<td>157</td>
<td>137</td>
</tr>
<tr>
<td>Net present value (k€)</td>
<td>260</td>
<td>177</td>
</tr>
<tr>
<td>Discounted Payback Period (Years)</td>
<td>5.4</td>
<td>6.4</td>
</tr>
<tr>
<td>Modified Internal rate of return (%)</td>
<td>14.2</td>
<td>13.5</td>
</tr>
<tr>
<td>Saved energy (%)</td>
<td>7.7</td>
<td>6.1</td>
</tr>
<tr>
<td>40</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Optimal surface (m²)</td>
<td>172</td>
<td>147</td>
</tr>
<tr>
<td>Net present value (k€)</td>
<td>436</td>
<td>299</td>
</tr>
<tr>
<td>Discounted Payback Period (Years)</td>
<td>4.0</td>
<td>4.8</td>
</tr>
<tr>
<td>Modified Internal rate of return (%)</td>
<td>15.6</td>
<td>14.7</td>
</tr>
<tr>
<td>Saved energy (%)</td>
<td>10.3</td>
<td>7.9</td>
</tr>
<tr>
<td>50</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Optimal surface (m²)</td>
<td>203</td>
<td>195</td>
</tr>
<tr>
<td>Net present value (k€)</td>
<td>698</td>
<td>520</td>
</tr>
<tr>
<td>Discounted Payback Period (Years)</td>
<td>3.1</td>
<td>3.8</td>
</tr>
<tr>
<td>Modified Internal rate of return (%)</td>
<td>16.7</td>
<td>15.9</td>
</tr>
<tr>
<td>Saved energy (%)</td>
<td>14.5</td>
<td>11.9</td>
</tr>
</tbody>
</table>
Economic versus Energetic Optimum

A comparison between the most economical designs and the optimal designs in terms of energy savings is presented in Table 6. The results obtained for the optimal energy savings are compared to those obtained in the most economical cases. The changes are strongly dependent on the boundary conditions (i.e., return temperature and moisture content). In the most favorable conditions for heat recovery in the FGC investigated in this study (i.e., mean return temperature: 50 °C and moisture content: 50%), the energetically optimal surface increases by 63% compared to the economically optimal surface, allowing a saving of 8.7% of extra energy. By definition of the energetic optimum, the additional costs lead to a 8.7% decrease in the NPV. The DPP increases by 34%, which corresponds in absolute value to 1 extra year.

Impact of the Usual Assumptions

The impact of the assumptions generally used on the viability of a project for hypotheses H1 and H4 is summarized in Table 7 for the different moisture content scenarios. All values are expressed in relative percentages of the economic optimum for the reference case. Small differences between the reference case and H1 can be observed. The largest difference lies in the DPP, and it does not exceed 7%. The discrepancies in the NPV and MIRR oscillate between 0 and 5% and 0 and 2%, respectively. On the other hand, when constant return temperature and heat demand are considered (i.e., H4), significant divergences in the results appeared. The NPVs is underestimated by up to 22%, and the DPP is overestimated by up to 27%. The differences in the MIRRs are less pronounced (less than 7%).

Table 6. A comparison between the optimal design in terms of energy savings and the most economical design. The FGC surfaces, NPVs, and energy savings obtained for the optimal energy savings are compared to those obtained in the most economical cases. All values are expressed as relative percentage difference.

<table>
<thead>
<tr>
<th>Return Temperature T\text{r}=</th>
<th>50</th>
<th>55</th>
<th>60</th>
</tr>
</thead>
<tbody>
<tr>
<td>Moisture Content MC (%)</td>
<td>30</td>
<td>40</td>
<td>50</td>
</tr>
<tr>
<td>Optimal surface (%r)</td>
<td>+52.3</td>
<td>+52.3</td>
<td>+52.3</td>
</tr>
<tr>
<td>Net present value (%r)</td>
<td>-11.5</td>
<td>-12.4</td>
<td>-12.3</td>
</tr>
<tr>
<td>Discounted Payback Period (%r)</td>
<td>+28.5</td>
<td>+25.4</td>
<td>+23.5</td>
</tr>
<tr>
<td>Modified Internal rate of return (%r)</td>
<td>-7.4</td>
<td>-6.6</td>
<td>-6.1</td>
</tr>
<tr>
<td>Saved energy (%r)</td>
<td>+11.5</td>
<td>+12.4</td>
<td>+12.3</td>
</tr>
<tr>
<td>Optimal surface (%r)</td>
<td>+65.5</td>
<td>+81.6</td>
<td>+61.8</td>
</tr>
<tr>
<td>Net present value (%r)</td>
<td>-11.7</td>
<td>-17.2</td>
<td>-15.8</td>
</tr>
<tr>
<td>Discounted Payback Period (%r)</td>
<td>+36.5</td>
<td>+44.9</td>
<td>+34.1</td>
</tr>
<tr>
<td>Modified Internal rate of return (%r)</td>
<td>-9.1</td>
<td>-10.7</td>
<td>-8.5</td>
</tr>
<tr>
<td>Saved energy (%r)</td>
<td>+11.7</td>
<td>+17.2</td>
<td>+15.8</td>
</tr>
<tr>
<td>Optimal surface (%r)</td>
<td>+62.9</td>
<td>+60.7</td>
<td>+73.9</td>
</tr>
<tr>
<td>Net present value (%r)</td>
<td>-8.7</td>
<td>-10.4</td>
<td>-15.4</td>
</tr>
<tr>
<td>Discounted Payback Period (%r)</td>
<td>+34.1</td>
<td>+34.3</td>
<td>+41.9</td>
</tr>
<tr>
<td>Modified Internal rate of return (%r)</td>
<td>-8.5</td>
<td>-8.6</td>
<td>-10.2</td>
</tr>
<tr>
<td>Saved energy (%r)</td>
<td>+8.7</td>
<td>+10.4</td>
<td>+15.4</td>
</tr>
</tbody>
</table>
Table 7. Impact of the usual assumptions generally used on the viability of a project for the different scenario investigated in this project.

<table>
<thead>
<tr>
<th>Return Temperature T_r</th>
<th>MC (%) = 30</th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>H1</td>
<td>50</td>
<td>55</td>
<td>60</td>
<td>50</td>
<td>55</td>
</tr>
<tr>
<td>Net present value (%r)</td>
<td>+2.0</td>
<td>+4.2</td>
<td>+0.3</td>
<td>-13.2</td>
<td>-10.9</td>
<td>+0.6</td>
</tr>
<tr>
<td>Discounted Payback Period (%r)</td>
<td>-5.1</td>
<td>-4.9</td>
<td>-0.6</td>
<td>+3.3</td>
<td>-0.4</td>
<td>-2.4</td>
</tr>
<tr>
<td>Internal rate of return (%r)</td>
<td>+1.6</td>
<td>+1.6</td>
<td>+0.2</td>
<td>-1.0</td>
<td>+0.1</td>
<td>+0.7</td>
</tr>
<tr>
<td></td>
<td>H4</td>
<td>40</td>
<td>50</td>
<td>50</td>
<td>55</td>
<td>60</td>
</tr>
<tr>
<td>Net present value (%r)</td>
<td>-1.1</td>
<td>+2.1</td>
<td>+5.1</td>
<td>-13.1</td>
<td>-19.6</td>
<td>-21.0</td>
</tr>
<tr>
<td>Discounted Payback Period (%r)</td>
<td>-5.8</td>
<td>-6.1</td>
<td>-6.6</td>
<td>+11.6</td>
<td>+12.0</td>
<td>+10.1</td>
</tr>
<tr>
<td>Internal rate of return (%r)</td>
<td>+1.8</td>
<td>+2.0</td>
<td>+2.1</td>
<td>-3.3</td>
<td>-3.4</td>
<td>-2.9</td>
</tr>
<tr>
<td></td>
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<td>40</td>
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<td>55</td>
<td>60</td>
</tr>
<tr>
<td>Net present value (%r)</td>
<td>-1.5</td>
<td>-1.8</td>
<td>+3.5</td>
<td>-14.1</td>
<td>-14.2</td>
<td>-21.7</td>
</tr>
<tr>
<td>Discounted Payback Period (%r)</td>
<td>-6.5</td>
<td>-6.7</td>
<td>-6.1</td>
<td>+26.7</td>
<td>+14.0</td>
<td>+8.8</td>
</tr>
<tr>
<td>Modified Internal rate of return (%r)</td>
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<td>+2.1</td>
<td>+1.9</td>
<td>-6.9</td>
<td>-3.9</td>
<td>-2.5</td>
</tr>
</tbody>
</table>

5. Future Works

Some techniques could be applied to further improve the efficiency of FGCs and could potentially result in higher profitability. Their impact on our techno-economic results should be investigated in future works.

The return temperature can be lowered to a certain extent by changing the process control strategy and upgrading the DH network (e.g., improved insulation). Combustion air humidification and heat pumps can also be used as condensing optimization systems. Although these modifications and additional system imply an improvement in the efficiency, they also lead to extra costs. In future works, the impact of such increased performances and the related extra investment on the profitability of FGCs should be investigated.

Furthermore, a sensitivity analysis should be carried out on the other key parameters such as the prices, the heat demand, heat sink flow rate, etc. The extension of the sources of uncertainty to all these variables could, however, require the use of more advanced computational techniques from the field of Uncertainty Quantification (UQ).

In addition to their positive impact on thermal efficiency and greenhouse gases emissions, evidences also show that they can play a role in pollutant reduction. In particular, the concentrations of the smallest fractions of particulate matter could be further decreased. This effect is, however, not well documented in the literature and should be subject to further investigations. FGC could, therefore, be a way to cope with increasingly stringent regulations in terms of energy performance and pollutant emissions, especially when the size of the emitted particles, and not only their total mass flow rate, will be at stake.

6. Conclusions

A methodology to assess the economic viability of a Flue Gas Condenser (FGC) retrofitting in a medium combustion plant integrated in a District Heating (DH) network was presented. The proposed methodology was applied to the case of an existing DH plant with a total installed power of 35 MW_th. The project feasibility was assessed by means of the Net Present Value (NPV), Discounted Payback Period (DPP), and the Modified Internal Rate of Return (MIRR).

Due to the recent decrease in the costs of FGCs on the market as well as the consideration of variable heat loads and return temperatures over the year, our results show that all scenarios investigated in this study are economically viable in the strict sense of the NPV (i.e., positive NPV) when the appropriate FGC surface is selected. The higher the humidity level and the lower the return temperature, the higher the NPV. For a 40% moisture content, the maximum NPV increases by approximately 50% when the average return temperature is decreased by 5 °C in the usual range of 60 to 50 °C.
Considering variable return temperatures along the year allows for the prediction of a mixed operation of the FGC as a condenser and an economizer, which increases the assessed profitability for the highest average return temperatures. Considering a variable heat demand further adds to this effect, as low return temperatures are generally correlated with high heat demands. The economic profitability of the cases where condensation occurs more than 50% of the time in the heat exchanger are clearly higher compared to those where the FGC is mainly operated as an economizer. Assuming a constant return temperature over the year can therefore lead to a 20% underestimation of the project NPV. However, due to the flatness of the reference curve around the economic optimum, the consideration of too restrictive assumptions would not lead to significant discrepancies between the actual NPV and the optimal one. An alternative averaging method was proposed to reduce the error on the predicted thermal and economic performances, where two distinct temperature zones are considered: above and below the flue gas dew point. The discrepancy with a detailed temperature variation is reduced to a few percents.

Increasing the FGC surface beyond the highest NPV can lead to substantial energy savings at a reasonable cost, up to a certain level. In the base case scenario, the energetic optimum we defined can lead to an increase of the energy savings by 17% for the same relative decrease of the NPV.

The results of our up-to-date techno-economic model using detailed input data show that Flue Gas Condensers deserve better attention, as their profitability might be underestimated, especially when it comes to mixed operation as condensers and economizers along the year.

Author Contributions: Conceptualization, T.C. and J.B.; data curation, T.C.; formal analysis, T.C.; funding acquisition, J.B.; investigation, T.C.; methodology, T.C. and J.B.; project administration, J.B.; resources, T.C. and J.B.; software, T.C.; supervision, J.B.; validation, T.C.; visualization, T.C.; writing—original draft, T.C.; writing—review and editing, T.C. and J.B.

Funding: This research received no external funding.

Conflicts of Interest: The authors declare no conflict of interest.

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