ECOS 2012

The 25th International Conference on Efficiency, Cost, Optimization and Simulation of Energy Conversion Systems and Processes

(Perugia, June 26th-June 29th, 2012)

edited by

Umberto Desideri, Giampaolo Manfrida, Enrico Sciubba

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The 25th ECOS Conference 1987-2012: leaving a mark

The introduction to the ECOS series of Conferences states that “ECOS is a series of international conferences that focus on all aspects of Thermal Sciences, with particular emphasis on Thermodynamics and its applications in energy conversion systems and processes”. Well, ECOS is much more than that, and its history proves it!

The idea of starting a series of such conferences was put forth at an informal meeting of the Advanced Energy Systems Division of the American Society of Mechanical Engineers (ASME) at the November 1985 Winter Annual Meeting (WAM), in Miami Beach, Florida, then chaired by Richard Gaggioli. The resolution was to organize an annual Symposium on the Analysis and Design of Thermal Systems at each ASME WAM, and to try to involve a larger number of scientists and engineers worldwide by organizing conferences outside of the United States. Besides Rich other participants were Ozer Arnas, Adrian Bejan, Yehia El-Sayed, Robert Evans, Francis Huang, Mike Moran, Gordon Reistad, Enrico Sciubba and George Tsatsaronis.

Ever since 1985, a Symposium of 8-16 sessions has been organized by the Systems Analysis Technical Committee every year, at the ASME Winter Annual Meeting (now ASME-IMECE). The first overseas conference took place in Rome, twenty-five years ago (in July 1987), with the support of the U.S. National Science Foundation and of the Italian National Research Council. In that occasion, Christos Frangopoulos, Yalcin Gogus, Elias Gyftopoulos, Dominick Sama, Sergio Stecco, Antonio Valero, and many others, already active at the ASME meetings, joined the core-group.

The name ECOS was used for the first time in Zaragoza, in 1992: it is an acronym for Efficiency, Cost, Optimization and Simulation (of energy conversion systems and processes), keywords that best describe the contents of the presentations and discussions taking place in these conferences. Some years ago, Christos Frangopoulos inserted in the official website the note that “ècos” (>this) means “home” in Greek and it ought to be attributed the very same meaning as the prefix “Eco-” in environmental sciences.

The last 25 years have witnessed an almost incredible growth of the ECOS community: more and more Colleagues are actively participating in our meetings, several international Journals routinely publish selected papers from our Proceedings, fruitful interdisciplinary and international cooperation projects have blossomed from our meetings. Meetings that have spanned three continents (Africa and Australia ought to be our next targets, perhaps!) and influenced in a way or another much of modern Engineering Thermodynamics.

After 25 years, if we do not want to become embalmed in our own success and lose momentum, it is mandatory to aim our efforts in two directions: first, encourage the participation of younger academicians to our meetings, and second, stimulate creative and useful discussions in our sessions. Looking at this years’ registration roster (250 papers of which 50 authored or co-authored by junior Authors), the first objective seems to have been attained, and thus we have just to continue in that direction; the second one involves allowing space to “voices that sing out of the choir”, fostering new methods and approaches, and establishing or reinforcing connections to other scientific communities. It is important that our technical sessions represent a place of active confrontation, rather than academic “lecturing”. In this spirit, we welcome you in Perugia, and wish you a scientifically stimulating, touristically interesting, and culinarily rewarding experience. In line with our 25 years old scientific excellency and friendship!

Umberto Desideri, Giampaolo Manfrida, Enrico Sciubba
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A comparison of optimal operation of residential energy systems using clustered demand patterns based on Kullback-Leibler divergence

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Abstract:
Residential energy demand varies widely in terms of time-series behaviors, amounts consumed between families, and even within one family. Residential energy demand profiles have a high degree of uncertainty in their essentials because the demand profile is entirely based on the occupant-driven load. When evaluating residential energy systems like co-generation systems, hot water and electricity demand profiles are critical. In this paper, in order to clarify rational energy system selection guidelines and rational operation strategies, authors aim to extract basic demand time-series patterns from two kinds of measured demand (electricity and domestic hot water), measured over 26307 days of data in Japan. Authors also aim to reveal the relationship between primary energy consumption and demand patterns. Demand time-series data are categorized by means of a kind of "unsupervised" learning, which is a hierarchical clustering method using a statistical pseudo-distance. The statistical pseudo-distance is calculated from the generalized Kullback-Leibler divergence with the Gaussian mixture distribution fitted to the demand time-series data. The classified demand patterns are built using a hierarchical clustering and then a comparison is performed between the optimal operation of the two systems (a polymer electrolyte membrane fuel cell co-generation system, and a CO\textsubscript{2} heat pump system) and the operation of a reference system (a conventional combination of a condensing gas boiler and electricity purchased from the grid) using the demand profiles appropriately built.

Our results show that basic demand patterns are extracted by the proposed method. The demand patterns, the amount of daily demand and heat-to-power ratio of demand affect the primary energy reduction ratio of the polymer electrolyte membrane fuel cell co-generation system.

Keywords:
Co-generation, Demand Pattern, Gaussian Mixture Model, Hierarchical Clustering, KL-divergence, Optimal Operation.

1. Introduction

A variety of water heaters are commercially available in Japan. Condensing gas boilers and heat pump water heaters, which are operated with CO\textsubscript{2} as a working fluid, and co-generation systems (CGS), which have polymer electrolyte fuel cells, are available. Residential energy consumption has been increasing slowly but surely in Japan. Residential energy consumption accounted for 81.3\% of all residential energy consumption by domestic hot water (DHW) and electricity (excluding heating, ventilation and air conditioning (HVAC)) demand [1]. There is a need for introducing high-efficiency equipment. Residential energy demand varies widely in terms of time-series behaviors, amounts consumed between families, and even within one family. Residential energy demand profiles have a high degree of uncertainty in their essentials because of the occupant-driven load. When evaluating residential energy systems, demand profiles are critical. In order to reduce the primary energy consumption when introducing an energy supply system, we need rational energy system selection guidelines and rational operation strategies that consider a variety of demand profiles. Therefore, it is important to clarify the matches between the characteristics of energy systems and the characteristics of demand profiles.
Hashimoto et al. [2] carried out a comparative evaluation on both a CO\textsubscript{2} heat-pump system (HP-S) and two kinds of polymer electrolyte membrane fuel cell co-generation system (PEFC-CGS). At that time, because there was no commercial PEFC-CGS, they estimated the model parameters with assumptions based on some ideal physics model.

In our previous study [3], the daily optimal operations of the PEFC-CGS and the HP-S were analyzed using measured operated system data and measured demand. It showed that primary energy consumption is reduced when an energy system is introduced with characteristics matching the characteristics of the demand profiles. Thus, one of the selection criteria of introducing an energy system is the amount of daily demand. The primary energy consumption of the PEFC-CGS shows large differences in spite of the similarity in the amount of daily demand and the similarity of the heat-to-power ratio of non-HVAC electricity and DHW demand. We guess that other factors affect the primary energy consumption of each system. Here we hypothesize that demand patterns have an effect on the primary energy consumption. We make an analysis of the demand patterns.

In this paper, in order to clarify the rational energy system selection guidelines and the rational operation strategies, our main purposes are

1. To extract basic demand time-series patterns from the measured operated system data and two kinds of demand (non-HVAC electricity and DHW), measured over 26307 days, and
2. To reveal the relationship between demand patterns and the primary energy consumption of each system.

![Fig. 1. Analysis framework.](image)

Figure 1 shows the analysis framework of this paper. It does not use demographic data but uses only demand time-series data for the extraction of demand patterns. Demand time-series data are categorized by means of a kind of "unsupervised" learning, which is a hierarchical clustering method using a statistical pseudo-distance. The statistical pseudo-distance is calculated from the generalized Kullback-Leibler (KL) divergence [4] with the Gaussian mixture distribution (GMD) fitted to the demand time-series data. The method was proposed by Shen et al. [5]. The classified
clusters are evaluated by the optimal operation of each energy system. The main consideration is the relationship between the clustered demand time-series data and the primary energy consumption of each energy system, one PEFC-CGS, and one HP-S. Each system is compared to a reference system (C-S): a conventional combination of a condensing gas boiler and electricity purchased from the grid.

2. Hot water and non-HVAC electricity demand

Figure 2 shows demand maps measured in 40 households in detached houses and 32 households in residential buildings. Data were gathered for a total of 26307 days, measured with a sampling interval of 30 minutes. As shown in Fig. 2 (a), the annual non-HVAC electricity demand is 15.49 GJ/year (about 11.79 kWh/day), and the annual DHW demand is 13.05 GJ/year (about 9.93 kWh/day) on average for the 72 households. As shown in Fig. 2 (b), the modal value of daily non-HVAC demand, $E$, is around 8 kWh/day, and the modal value of daily DHW demand, $Q$, is around 4 kWh/day. Daily demand varies widely because DHW demand accrues to over 60 kWh/day, which is 15 times more than the modal value. Since the operational strategy might be implemented in cogeneration systems on daily, time-series data should be analyzed on daily time scale.

As shown in Fig. 2 (b), measured demand data are separated from the daily heat-to-power ratio, $R$, and DHW demand, $Q$, by red lines. The 26307 days are divided into 6 sets of groups, defined by each zone of heat-to-power ratio, $R$, for descriptive purposes as shown in Table 1. Measured demand data are separated based on information obtained from our earlier study [3]. This showed that the targeted PEFC-CGS increases primary energy consumption compared with the simple condensing gas boiler and grid electricity system in the zone under 6 kWh/day of DHW demand, $Q$. We do a clustering analysis for each group.

![Fig. 2. Measured demand: a) yearly, b) daily.](image)

<table>
<thead>
<tr>
<th>Group name</th>
<th>Number of Elements, days</th>
<th>DHW demand $Q$, kWh/day</th>
<th>Heat-to-power ratio $R$</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>9807</td>
<td>$Q &lt; 6.0$</td>
<td>---</td>
</tr>
<tr>
<td>B</td>
<td>971</td>
<td>$6.0 \leq Q$</td>
<td>$R &lt; 0.5$</td>
</tr>
<tr>
<td>C</td>
<td>6958</td>
<td>$6.0 \leq Q$</td>
<td>$0.5 \leq R &lt; 1.0$</td>
</tr>
<tr>
<td>D</td>
<td>4920</td>
<td>$6.0 \leq Q$</td>
<td>$1.0 \leq R &lt; 1.5$</td>
</tr>
<tr>
<td>E</td>
<td>1926</td>
<td>$6.0 \leq Q$</td>
<td>$1.5 \leq R &lt; 2.0$</td>
</tr>
<tr>
<td>F</td>
<td>1725</td>
<td>$6.0 \leq Q$</td>
<td>$2.0 \leq R$</td>
</tr>
</tbody>
</table>
3. Clustering Model

Demand time-series data are categorized by a hierarchical clustering method using a statistical pseudo-distance. A statistical pseudo-distance is the distance between two points in a pseudo-metric, which is defined by identity of indiscernibles, symmetry and triangle inequality. The statistical pseudo-distance is calculated from the generalized KL divergence with the GMD fitted to the demand time-series data. Actually, this process fits the GMD to some peaks of daily non-HVAC electricity and daily DHW demands. The GMD is represented by three parameters: the mean, the covariance, and the coefficient in the linear combination. 48-dimensional vectors, which are 48 terms accumulated every 30 minutes through a day, are translated to vectors with up to 12 dimensions. The demand time-series data are categorized by the hierarchical clustering method using the distance, which is calculated with information per day represented by vectors with up to 12 dimensions. The KL divergence, which represents the dissimilarity measure between two distributions, is often used for cluster analyses. The Bregman divergence [6] is a pseudo-distance for measuring the discrepancy between two functions. The generalized KL divergence is extended by the framework of the Bregman divergence, and can handle distributions and others. In this paper, the generalized KL divergence between two biased distributions, which represents the dissimilarity measure of the histogram of daily demand time-series data, is used as the pseudo-distance of the clustering. Please refer [5] for details.

3.1. Gaussian Mixture Model

For a \( d \)-dimensional vector \( \mathbf{x} = (x_1, \ldots, x_d)^T \) of continuous variables, the Gaussian distribution (GD) is defined by

\[
\mathcal{N} (\mathbf{x} | \mu, \Sigma) = \frac{1}{d \pi d^2} \exp \left\{ -\frac{1}{2} (\mathbf{x} - \mu)^T \Sigma^{-1} (\mathbf{x} - \mu) \right\},
\]

where \( \mu \) is a \( d \)-dimensional mean vector, and \( \Sigma \) is a \( d \times d \) covariance matrix. A superposition of \( K \) Gaussian densities of the form

\[
p (\mathbf{x} | \boldsymbol{\theta}) = \sum_{k=1}^{K} \pi_k \mathcal{N} (\mathbf{x} | \mu_k, \Sigma_k),
\]

is called a mixture of Gaussians. The parameters \( \pi \) are called mixing coefficients. If we integrate both sides of (2) with respect to \( \mathbf{x} \), and note that both \( p (\mathbf{x}) \) and the individual Gaussian components are normalized, we obtain

\[
\sum_{k=1}^{K} \pi_k = 1.
\]

The form of the GMD is governed by the parameters \( \pi \equiv \{\pi_1, \ldots, \pi_K\}, \mu \equiv \{\mu_1, \ldots, \mu_K\}, \Sigma \equiv \{\Sigma_1, \ldots, \Sigma_K\} \). Here, the unknown parameters \( \boldsymbol{\theta} \) are defined by

\[
\boldsymbol{\theta} = \{\pi_k, \mu_k, \Sigma_k\}_{k=1}^K.
\]

We determine values for the unknown parameters \( \boldsymbol{\theta} \) in the GMD by maximizing the likelihood function. Here, the maximum likelihood solution for the parameters no longer has a closed-form analytical solution. We use the expectation-maximization (EM) algorithm [7], which is a method of approximating inference, to estimate the parameters of the Gaussian mixture model. Mclust function [8] with mclust packages in R language, which is an open source programming language and software environment for statistical computing, is utilized. The numbers of the Gaussian mixture elements are chosen by the Bayesian information criterion (BIC) from one to four. In other words, the GMD fits the demand patterns, which might have up to four peaks in a day, at morning, noon, evening and midnight. In this process, 48-dimensional daily demand vectors are represented by a maximum of \( 3 \times 4 = 12 \)-dimensional vectors of the GMD, because the GMD is represented by the three parameters, namely the mean, the covariance, and the coefficient in the linear combination.
3.2. Kullback-Leibler Divergence

Statistical pseudo-distances between all days are represented by the KL divergence, which is known as the “relative entropy”. The KL divergence between two probability density functions, \( f(x) \) and \( g(x) \), is given by

\[
D_{\text{KL}}(f, g) = \int f(x) \ln \frac{f(x)}{g(x)} \, dx. 
\]

The KL divergence between \( d \)-dimensional GDs \( \tilde{f}(x) \) and \( \tilde{g}(x) \) is given by

\[
D_g(\tilde{f}, \tilde{g}) = \frac{1}{2} \left[ \log \frac{|\Sigma_{\tilde{g}}|}{|\Sigma_{\tilde{f}}|} + \text{tr} \left( \Sigma_{\tilde{g}}^{-1} \Sigma_{\tilde{f}} \right) - d + (\mu_{\tilde{g}} - \mu_{\tilde{f}})^T \Sigma_{\tilde{g}}^{-1} (\mu_{\tilde{g}} - \mu_{\tilde{f}}) \right]. 
\]

The KL divergence between two mixture of Gaussian \( p(x|\theta) \) and \( q(x|\theta') \) is approximated by

\[
D_{\text{gm}}(p, q) = \sum_{a=1}^{K_a} \pi_a \log \frac{\sum_{a=1}^{K_a} \pi_a \exp \left( -D_g(p_a, p_{a'}) \right)}{\sum_{b=1}^{K_b} \pi_b \exp \left( -D_g(p_a, q_{a'}) \right)}, \tag{7}
\]

where Gaussian, \( p_a(x) \), is the \( a \)th component of the GMD \( p(x|\theta) \). Specifically, in this context, \( x \) represents the measured demand of a day, \( p(x|\theta) \) represents the characteristics of one day of the waveform regarding demand time-series data as a histogram, and \( p_a(x) \) represents the characteristics of a peak fitted to the GD. \( q(x) \) represents the characteristic of the waveform of the other day. Until this point, we have considered the KL divergence as the difference between the shapes of the histogram of daily demand time-series data. In this paper, it is important to consider both the amount of daily total demand and time-series behaviors, because the balance of electricity and heat demand is critical for CGS performance. Here, we consider a biased GMD, \( \tilde{p}(x) \) for one day and \( \tilde{q}(x) \) for the other day, which are multiplied by daily total non-HVAC electricity demand, \( E_p, E_q \) kWh/day. The KL divergence of non-HVAC electricity demand is explained in the following context:

\[
\tilde{p}(x) = \frac{E_p}{1000} p(x),
\]

\[
\tilde{q}(x) = \frac{E_q}{1000} q(x). \tag{8}
\]
For calculating the KL divergence from the biased GMD, (8), the generalized KL divergence, $D_{\text{gen}}$, which is extended by the framework of the Bregman divergence, is given by

$$D_{\text{gen}}(\tilde{p}, \tilde{q}) = \int_x \left( \tilde{p}(x) \log \frac{\tilde{p}(x)}{\tilde{q}(x)} - \tilde{p}(x) + \tilde{q}(x) \right) dx,$$

$$= \frac{E_p}{1000} D_{\text{gm}}(p, q) + \frac{E_p}{1000} \log \frac{E_p}{E_q} + \frac{E_p}{1000} + \frac{E_q}{1000}. \quad (9)$$

The KL divergence is not a symmetrical quantity, that is to say $D_{\text{gen}}(\tilde{p}, \tilde{q}) \neq D_{\text{gen}}(\tilde{q}, \tilde{p})$. In order to use the KL divergence for a distance measure in cluster analysis, we adopt the symmetrized KL divergence, $D_{\text{symm}}^{\text{ELE}}$, given by

$$D_{\text{symm}}^{\text{ELE}}(\tilde{p}, \tilde{q}) = \frac{D_{\text{gen}}(\tilde{p}, \tilde{q}) + D_{\text{gen}}(\tilde{q}, \tilde{p})}{2}. \quad (10)$$

A distance matrix $D_{\text{symm}}^{\text{ELE}}$ for the clustering is composed the symmetrized KL divergences. The symmetrized KL divergence for DHW demand $D_{\text{symm}}^{\text{HW}}$, and a distance matrix $D_{\text{symm}}^{\text{HW}}$ are also calculated in the same manner.

### 3.3. Hierarchical Clustering

The hierarchical clustering analysis uses a distance matrix $D$, with the KL divergence of both non-HVAC electricity and DHW demands as the distance measure between clusters by Ward’s method [9]. Because both distance matrices $D_{\text{symm}}^{\text{ELE}}$ and $D_{\text{symm}}^{\text{HW}}$ are calculated independently until this point, they are normalized by dividing by the median of each $D_{\text{symm}}^{\text{ELE}}$ and $D_{\text{symm}}^{\text{HW}}$ in order to compute the sum of the two distance matrices. The distance matrix, $D$, which sums the two kinds of symmetrized KL divergence, is given by:

$$D = \frac{D_{\text{symm}}^{\text{ELE}}}{\text{Median}(D_{\text{symm}}^{\text{ELE}})} + \frac{D_{\text{symm}}^{\text{HW}}}{\text{Median}(D_{\text{symm}}^{\text{HW}})}, \quad (11)$$

where “Median” represents the median of the matrix. Demand is classified into 16 clusters for each group by the hierarchical clustering method. The reason for 16 clusters is because we assume DHW demand patterns vary widely while non-HVAC electricity demand patterns do not vary widely. In other words, $2^4=16$ clusters represent the combination of the four existing or non-existing DHW demand peaks, which are morning, noon, evening and midnight.

### 4. Optimization Model

The clusters, which are classified in the previous section, are evaluated relative to the optimal operation of each energy system. The main consideration is the relationship between the clustered demand time-series data and the primary energy consumption of each energy system, the PEFC-CGS and the HP-S. Each system is compared to the reference system: a conventional combination of a condensing gas boiler and electricity purchased from the grid. A optimal operational planning method for cogeneration systems with thermal storage was proposed by Yokoyama et al. [10]. Schematic diagrams with the specifications [3] of the PEFC-CGS, the HP-S and the C-S are shown below. The parameters in the model are taken up from catalog values. If measured values are available, model parameters are identified from them. This problem is formulated as a Mixed Integer Linear Programming (MILP) problem. The models were coded by the algebraic modeling language AMPL version 12.1 [11] as MILP, and were solved by the general optimization solver CPLEX version 12.1 [12].

#### 4.1. Objective Function

The objective function to be minimized is the daily primary energy consumption calculated from summation of purchased electricity and gas consumption multiplied by each primary energy
conversion factor. In particular, the primary energy consumption of purchased gas is converted by higher heating value of 45 MJ/m$^3$ [13]. The primary energy conversion factors of electricity mean all of the conversion efficiency of input fuel between thermal power plants and end users. In other words, they mean transmission loss, the efficiency of thermal power plant and so on. The primary energy conversion factor of purchased electricity in the daytime is 9.97 MJ/kWh [14], and the primary energy conversion factor of purchased electricity in the nighttime is 9.28 MJ/kWh [14]. The objective function $P$, which is the daily operating cost at the viewpoint of the primary energy consumption, is given by

$$\text{Minimize } P = \sum_{t=1}^{T} f(x_t) \delta t,$$

where $t = 1, \ldots, T$ represents the time index, $T$ is the number of time periods. The sampling period, $\delta t$, is 30 minutes. One day is discretized to $T = 48$ terms. We stack optimal solutions and operations for each day to evaluate the characteristics of the energy systems.

4.2. Constraints

A $T$-dimensional vector $x \equiv (x_1, \ldots, x_T)$ of continuous variables represents the energy flow, a $T$-dimensional vector $z \equiv (z_1, \ldots, z_T)$ of binary variables represents the start-stop status of each device in each term. The linear equality, $h$, corresponds to the energy balance of the system. The inequality, $g$, corresponds to a special case for mitigating the strict energy balance. This inequality means loss of heat due to transport, and is introduced in order to relax the computational load.

Subject to $h(x_{t-1}, x_t, z_{t-1}, z_t) = 0,$

$$g(x_t, z_t) > 0,$$

$$z_t \in \{0,1\}. \quad (13)$$

4.3. CO$_2$ Heat Pump System (HP-S)

As shown in Fig. 4, the HP-S consists of two main parts: the heat pump (HP) unit, which is operated with CO$_2$ as the heat-transfer medium, and the thermal storage tank (HW tank), which has a capacity of 370 liters. The rated hot-water output of the HP unit is 4.5 kW. The Coefficient of Performance (COP) of the HP unit is assumed to be a function of the hot water outlet temperature and ambient temperature [15]. Hot water is stored by the HP-S in the thermal storage tank at nighttime, using the cheaper electricity available then. The thermal storage tank has to match the heat quantities at the beginning and end of the day.

![Fig. 4. HP-S.](image-url)
4.4. Polymer Electrolyte Membrane Fuel Cell Co-Generation System (PEFC-CGS)

As shown in Fig. 5, the PEFC-CGS consists of four main parts: the polymer electrolyte membrane fuel cell (PEFC) unit, the thermal storage tank, which has a capacity of 200 liters, the auxiliary boiler, which has an 83% conversion efficiency (lower heating value of the fuel: LHV), and the electric heater (H), which has a 95% conversion efficiency to hot water. Electricity demand is supplied from the grid and the PEFC unit. Because reverse flow of electricity from a CGS to the grid is not allowed in Japan, surplus electricity produced by the CGS is supplied to the electric heater to prevent reverse flow to the grid. The rated hot water output of the PEFC unit is 1.0 kW (100% load), which is smaller than the output of the HP unit. The DHW demand is supplied from the auxiliary boiler, in case of supply shortages of the PEFC unit and the thermal storage tank. The relationship between the output of the PEFC unit and gas consumption identified piecewise-linear function based on measured data, as shown in Fig. 6. Table 2 shows the partial load efficiency of the PEFC unit. The rated electricity output is 700 W. At rated load, the overall efficiency is 50.0+35.0=85.0%, the heat-to-power ratio of the rated output is 50.0/35.0=1.43. The minimum electricity output is 0.25 kW (35.7% load), and the heat-to-power ratio of the output is 30.0/30.0=1.0. When starting up, the PEFC-CGS has the following three requirements: 60 minutes, 0.5 kWh of electricity consumption, and 0.04 Nm$^3$ of gas consumption.

Table 2. Partial load performance of PEFC unit.

<table>
<thead>
<tr>
<th>Load factor %</th>
<th>35.7</th>
<th>71.4</th>
<th>100</th>
</tr>
</thead>
<tbody>
<tr>
<td>Electricity power output kW</td>
<td>0.25</td>
<td>0.50</td>
<td>0.70</td>
</tr>
<tr>
<td>Electricity power efficiency (LHV) %</td>
<td>30.0</td>
<td>34.0</td>
<td>35.0</td>
</tr>
<tr>
<td>Thermal efficiency (LHV) %</td>
<td>30.0</td>
<td>45.0</td>
<td>50.0</td>
</tr>
</tbody>
</table>

![Fig. 5. PEFC-CGS.](image)
4.5. Conventional System (C-S)

Figure 7 shows the reference system: a condensing gas boiler, which has a 92% conversion efficiency (LHV), with electricity from the grid. It has no thermal storage tank. DHW demand is supplied from the boiler just in time. Electricity demand is supplied from the grid only.

5. Numerical Results

5.1. Optimal Operation

In order to compare the PEFC-CGS and the HP-S, we introduce an index $\phi_{X}^{ene}$, representing the primary energy reduction ratio.

$$\phi_{X}^{ene} = \frac{p_{cs}-p_{X}}{p_{cs}} \times 100,$$

where subscript X represents the primary energy consumption of each system, and subscript cs represents the primary energy consumption of the C-S. This index $\phi_{X}^{ene}$, which is called the primary energy reduction ratio, represents the difference between the two optimal solutions, the primary energy consumption of each system. Figure 8 shows the distribution of the daily primary energy reduction ratio. The peak of the distribution for the PEFC-CGS, which is the blue line, is shifted to the negative side relative to the peak of the distribution of the HP-S, which is the red line. As shown in Fig. 8, for the average of all 26307 days, the primary energy reduction ratio of the HP-S is 9.88%, which is shown by the red dotted line. The primary energy reduction ratio of the PEFC-CGS is 5.64%, which is shown by the blue dotted line. One of the dominant factors, the negative primary energy reduction ratio, is the hot water supplied by the auxiliary boiler on the PEFC-CGS. This is because the efficiency of the auxiliary boiler of the PEFC-CGS is inferior to that of the condensing gas boiler of the C-S. There are two possible cases: the PEFC-CGS operation is ineffective because
of lower DHW demand, or higher DHW demand is supplied from the auxiliary gas boiler in addition to the PEFC unit. In these cases, the characteristics of the PEFC-CGS don’t match the demand, because of lower DHW demand for the operations of the PEFC-CGS over an entire day.

\begin{figure}[h]
\centering
\includegraphics[width=0.5\textwidth]{fig8.png}
\caption{Distribution of primary energy reduction ratio in optimal operation.}
\end{figure}

5.2. Extraction of Demand Patterns

Six sets of groups of demand were defined in terms of heat-to-power ratio. Because of space limitations, the following section shows the numerical results of Group-D, which matches the characteristics of the PEFC-CGS output. Figures 9 to 10 show the demand patterns clustered into 16 groups from Group-D. The bar charts, which are shown in Figs. 9 and 10, represent the average demand of each term in each cluster. The dotted lines shown in Figs. 9 and 10 represent the demand ± standard deviation on average of each term.

It is confirmed that the basic demand patterns are extracted by the hierarchical clustering with the generalized KL divergence. As shown in the second row and the 1st, 4th, 5th, and 7th columns in Table 3, four basic demand patterns, which have about 10% of the elements in each cluster, are recognized. As shown in Figs. 9 (a), (d), (e), and (g), clusters show the peak of DHW demand at evening. Others, Cluster 12 shows the two peaks of DHW demand: at morning and evening, as shown in Fig. 10 (l). Cluster 9 shows the three peaks of DHW demand: at morning, evening, and midnight, as shown in Fig. 10 (i).

On average in Group-D, the primary energy reduction ratio of the PEFC-CGS is 11.07%, and that of the HP-S is 13.01%. From the viewpoint of the primary energy reduction ratio of the PEFC-CGS, there is 11.07-5.64=5.43% difference between the average in the 4920 days of Group-D and the average of all 26307 days. Therefore, it is confirmed that the heat-to-power ratio defined by the relationship between non-HVAC electricity and DHW demand has an influence on the primary energy reduction ratio of the PEFC-CGS.

It is confirmed that the distributions of the primary energy reduction ratio differ for each cluster, as shown in Figs. 9 and 10. It means the amount of daily demand and demand patterns have an effect on the primary energy reduction ratio of each system. As shown in the sixth-to-last and third-to-last rows and the 1st, 2nd, 3rd, 9th, 11th, 12th, 13th, and 16th columns in Tables 3 and 4, the primary energy reduction ratio of the PEFC-CGS is better than that of the HP-S at Clusters 1, 2, 3, 9, 11, 12, 13, and 16, on average in each cluster. As shown in the second row and those same columns in Tables 3 and 4, these clusters have 10.28+4.04+4.49+2.93+5.63+4.65+6.04+2.46=40.52% of the elements of Group-D.
As shown in Figs. 9 and 10, DHW demand and the standard deviations vary widely by hour on average for each term in each cluster. On the other hand, as shown in Figs. 9 and 10, there is a certain amount of non-HVAC electricity demand throughout the day. As shown in the hot water demand row and the electricity demand row in Tables 3 and 4, DHW demand standard deviations are larger than that of non-HVAC electricity, on average in each cluster.

As shown in the third-to-last row and the 7th column in Table 3, 6.05% of Cluster 7, which has the minimum daily non-HVAC electricity and DHW demand, is at the worst primary energy reduction ratio of the PEFC-CGS on average in each cluster for Group-D. As shown in the last row and the 16th column in Table 4, 15.15% of Cluster 16, which has the maximum daily non-HVAC electricity and DHW demand, is at the best primary energy reduction ratio of the PEFC-CGS on average in each cluster for Group-D.

Thus, in overall, because of a high degree of uncertainty of demand, it is difficult to exercise better performance of the PEFC-CGS than that of the HP-S.

6. Conclusion

Demand time-series data are categorized by means of a hierarchical clustering method using a statistical pseudo-distance. The statistical pseudo-distance is calculated using the generalized KL divergence with the GMD fitted to the demand time-series data of non-HVAC electricity and DHW demand from 26307 days of data, measured in Japan. The demand patterns are useful means to compare the performance of conventional and non-conventional systems. We formulated an analytical framework of the characteristics of the energy systems, and of the characteristics of the demand profiles. The following main results were obtained.

1. Basic demand patterns are extracted by the proposed method.
2. Factors which are at least associated with the primary energy reduction ratio of the PEFC-CGS, are heat-to-power ratio, demand patterns, and the amount of daily demand.
3. The average primary energy reduction ratio of the PEFC-CGS is better than that of the HP-S at Clusters1, 2, 3, 9, 11, 12, 13, and 16.
4. These clusters contain 40.52% of the elements of Group-D.
5. The primary energy reduction ratio of the PEFC-CGS varied from 6.05% to 15.15% on average for each cluster of Group-D.
Fig. 9. Clusters in Group-D (1/2).
Fig. 10. Clusters in Group-D (2/2).
Table 3. Clustering data of Group-D (1/2).

<table>
<thead>
<tr>
<th>Group-D Cluster</th>
<th>Total elements</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total elements</td>
<td>4920</td>
<td>506</td>
<td>199</td>
<td>221</td>
<td>532</td>
<td>491</td>
<td>296</td>
<td>566</td>
<td>392</td>
</tr>
<tr>
<td>%</td>
<td>10.28</td>
<td>4.04</td>
<td>4.49</td>
<td>10.81</td>
<td>9.98</td>
<td>6.02</td>
<td>11.50</td>
<td>7.97</td>
<td></td>
</tr>
<tr>
<td>Hot water demand</td>
<td>Mean Wh/30min</td>
<td>307.64</td>
<td>371.36</td>
<td>365.04</td>
<td>254.54</td>
<td>269.04</td>
<td>305.82</td>
<td>207.81</td>
<td>216.61</td>
</tr>
<tr>
<td></td>
<td>Median Wh/30min</td>
<td>0.00</td>
<td>0.00</td>
<td>0.00</td>
<td>0.00</td>
<td>0.00</td>
<td>0.00</td>
<td>0.00</td>
<td>0.00</td>
</tr>
<tr>
<td></td>
<td>Maximum Wh/30min</td>
<td>11408.58</td>
<td>8649.48</td>
<td>11542.30</td>
<td>10354.05</td>
<td>10242.69</td>
<td>10458.39</td>
<td>9688.55</td>
<td>10262.62</td>
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<tr>
<td></td>
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<td>0.00</td>
<td>0.00</td>
<td>0.00</td>
<td>0.00</td>
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<td>Standard deviation</td>
<td>887.42</td>
<td>967.17</td>
<td>987.21</td>
<td>799.92</td>
<td>823.45</td>
<td>952.94</td>
<td>701.63</td>
<td>764.48</td>
</tr>
<tr>
<td>Electricity demand</td>
<td>Mean Wh/30min</td>
<td>258.28</td>
<td>304.46</td>
<td>307.80</td>
<td>207.96</td>
<td>226.84</td>
<td>255.88</td>
<td>176.05</td>
<td>182.39</td>
</tr>
<tr>
<td></td>
<td>Median Wh/30min</td>
<td>231.46</td>
<td>265.35</td>
<td>264.51</td>
<td>170.44</td>
<td>179.47</td>
<td>203.58</td>
<td>142.14</td>
<td>154.57</td>
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<tr>
<td></td>
<td>Maximum Wh/30min</td>
<td>1651.77</td>
<td>1329.62</td>
<td>1909.49</td>
<td>1253.07</td>
<td>1188.22</td>
<td>1326.76</td>
<td>1347.63</td>
<td>1036.94</td>
</tr>
<tr>
<td></td>
<td>Minimum Wh/30min</td>
<td>29.88</td>
<td>38.78</td>
<td>22.67</td>
<td>11.09</td>
<td>28.11</td>
<td>33.34</td>
<td>7.62</td>
<td>15.28</td>
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<tr>
<td></td>
<td>Standard deviation</td>
<td>147.01</td>
<td>178.61</td>
<td>192.89</td>
<td>134.92</td>
<td>141.08</td>
<td>162.45</td>
<td>112.70</td>
<td>108.97</td>
</tr>
<tr>
<td>Primary energy reduction ratio of the HP-S</td>
<td>Mean %</td>
<td>12.49</td>
<td>11.66</td>
<td>10.90</td>
<td>13.71</td>
<td>13.50</td>
<td>13.46</td>
<td>13.82</td>
<td>14.14</td>
</tr>
<tr>
<td></td>
<td>Minimum %</td>
<td>3.09</td>
<td>-1.33</td>
<td>2.35</td>
<td>0.50</td>
<td>1.99</td>
<td>0.46</td>
<td>2.14</td>
<td>4.64</td>
</tr>
<tr>
<td>Primary energy reduction ratio of the PEFC-CGS</td>
<td>Mean %</td>
<td>12.86</td>
<td>13.37</td>
<td>13.15</td>
<td>8.46</td>
<td>10.59</td>
<td>11.28</td>
<td>6.05</td>
<td>7.16</td>
</tr>
<tr>
<td></td>
<td>Maximum %</td>
<td>19.55</td>
<td>18.52</td>
<td>19.45</td>
<td>18.21</td>
<td>18.76</td>
<td>19.46</td>
<td>17.32</td>
<td>18.17</td>
</tr>
<tr>
<td></td>
<td>Minimum %</td>
<td>-0.43</td>
<td>5.67</td>
<td>1.62</td>
<td>-7.92</td>
<td>-1.78</td>
<td>-0.76</td>
<td>-9.43</td>
<td>-6.43</td>
</tr>
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Table 4. Clustering data of Group-D (2/2).

<table>
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<tr>
<th>Group-D Cluster</th>
<th>Total elements</th>
<th>9</th>
<th>10</th>
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<td>318</td>
<td>277</td>
<td>229</td>
<td>297</td>
<td>163</td>
<td>168</td>
<td>121</td>
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<tr>
<td>%</td>
<td>2.93</td>
<td>6.46</td>
<td>5.63</td>
<td>4.65</td>
<td>6.04</td>
<td>3.31</td>
<td>3.41</td>
<td>2.46</td>
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<tr>
<td>Hot water demand</td>
<td>Mean Wh/30min</td>
<td>484.62</td>
<td>274.72</td>
<td>349.65</td>
<td>334.48</td>
<td>350.71</td>
<td>313.06</td>
<td>312.74</td>
<td>552.09</td>
</tr>
<tr>
<td></td>
<td>Median Wh/30min</td>
<td>50.00</td>
<td>0.00</td>
<td>0.00</td>
<td>0.00</td>
<td>0.00</td>
<td>0.00</td>
<td>0.00</td>
<td>137.94</td>
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<td></td>
<td>Maximum Wh/30min</td>
<td>10841.97</td>
<td>10046.06</td>
<td>9302.10</td>
<td>11358.01</td>
<td>9333.45</td>
<td>9282.30</td>
<td>9819.48</td>
<td>9711.63</td>
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<tr>
<td></td>
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<td>0.00</td>
<td>0.00</td>
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<tr>
<td></td>
<td>Standard deviation</td>
<td>1020.72</td>
<td>933.00</td>
<td>822.97</td>
<td>815.69</td>
<td>833.86</td>
<td>871.56</td>
<td>849.59</td>
<td>952.34</td>
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<td>Electricity demand</td>
<td>Mean Wh/30min</td>
<td>391.88</td>
<td>234.04</td>
<td>285.96</td>
<td>280.69</td>
<td>294.44</td>
<td>252.33</td>
<td>261.09</td>
<td>439.16</td>
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<tr>
<td></td>
<td>Median Wh/30min</td>
<td>361.68</td>
<td>200.75</td>
<td>259.31</td>
<td>251.81</td>
<td>251.20</td>
<td>224.63</td>
<td>237.76</td>
<td>415.29</td>
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<tr>
<td></td>
<td>Maximum Wh/30min</td>
<td>1643.08</td>
<td>1310.84</td>
<td>1493.83</td>
<td>2148.11</td>
<td>1429.03</td>
<td>1297.27</td>
<td>1469.93</td>
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<tr>
<td></td>
<td>Minimum Wh/30min</td>
<td>8.85</td>
<td>20.58</td>
<td>35.68</td>
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<td>26.16</td>
<td>28.84</td>
<td>35.33</td>
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<td>Standard deviation</td>
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<td>137.40</td>
<td>151.16</td>
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<td>168.51</td>
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<td>Primary energy reduction ratio of the HP-S</td>
<td>Mean %</td>
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<td>12.81</td>
<td>13.93</td>
<td>12.55</td>
<td>11.64</td>
<td>14.22</td>
<td>14.43</td>
<td>10.14</td>
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<tr>
<td></td>
<td>Maximum %</td>
<td>20.52</td>
<td>21.73</td>
<td>22.38</td>
<td>21.87</td>
<td>21.48</td>
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<td>-0.95</td>
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<tr>
<td>Primary energy reduction ratio of the PEFC-CGS</td>
<td>Mean %</td>
<td>14.66</td>
<td>11.34</td>
<td>14.81</td>
<td>13.77</td>
<td>13.68</td>
<td>11.84</td>
<td>13.84</td>
<td>15.15</td>
</tr>
<tr>
<td></td>
<td>Maximum %</td>
<td>19.87</td>
<td>18.09</td>
<td>20.34</td>
<td>20.28</td>
<td>19.41</td>
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</tr>
<tr>
<td></td>
<td>Minimum %</td>
<td>5.01</td>
<td>-1.14</td>
<td>-0.62</td>
<td>0.38</td>
<td>6.68</td>
<td>-3.96</td>
<td>0.58</td>
<td>7.18</td>
</tr>
</tbody>
</table>

Acknowledgement

The authors acknowledge Mr. H.Shen and Associate Professor H.Hino of Murata Laboratory of Waseda University for technical suggestions about implementation using the R language. Part of this work is supported by a Strategic Research Foundation Grant-aided Project for Private Universities grant from MEXT(2010). The authors would like to acknowledge the support of the "Distributed Autonomous Urban Energy Systems for Mitigating Environmental Impact" project of
Osaka University. This research was conducted under Optimal Planning of Energy Supply Systems for Various Buildings, 09P05 at RISE, Waseda University.

Nomenclature

\begin{align*}
\text{\textit{d}} & \quad \text{number of dimensions} \\
\text{\textit{D}} & \quad \text{distance measure} \\
\text{\textit{D}} & \quad \text{distance matrix} \\
\text{\textit{E}} & \quad \text{non-HVAC electricity demand, kWh/day} \\
\text{\textit{f}}, \text{\textit{g}} & \quad \text{a probability density function} \\
\text{\textit{\bar{f}}}, \text{\textit{\bar{g}}} & \quad \text{Gaussian probability density function} \\
\text{\textit{k}} & \quad \text{discretized index} \\
\text{\textit{K}} & \quad \text{number of discrete value} \\
\text{\textit{P}} & \quad \text{primary energy consumption, MJ/day} \\
\text{\textit{p}}, \text{\textit{\tilde{p}}}, \text{\textit{q}}, \text{\textit{\tilde{q}}} & \quad \text{Gaussian mixture distribution} \\
\text{\textit{Q}} & \quad \text{DHW demand, kWh/day} \\
\text{\textit{R}} & \quad \text{heat-to-power ratio} \\
\text{\textit{t}} & \quad \text{discretized time index} \\
\text{\textit{T}} & \quad \text{number of time index} \\
\text{\textit{x}} & \quad \text{vector of continuous variables} \\
\text{\textit{z}} & \quad \text{vector of binary variables} \\
\end{align*}

Greek symbols

\begin{align*}
\text{\textit{\delta t}} & \quad \text{sampling period, hour} \\
\text{\textit{\theta}} & \quad \text{unknown parameters} \\
\text{\textit{\mu}} & \quad \text{mean vector} \\
\text{\textit{\pi}} & \quad \text{mixing coefficient} \\
\text{\textit{\Sigma}} & \quad \text{covariance matrix} \\
\text{\textit{\varphi}} & \quad \text{primary energy reduction ratio, \%} \\
\end{align*}

Subscripts and superscripts

\begin{align*}
\text{\textit{a}}, \text{\textit{b}} & \quad \text{component index of the Gaussian mixture distribution} \\
\text{\textit{CS}} & \quad \text{conventional system} \\
\text{\textit{ELE}} & \quad \text{non-HVAC electricity demand} \\
\text{\textit{ene}} & \quad \text{primary energy consumption} \\
\text{\textit{\bar{f}}}, \text{\textit{\bar{g}}}, \text{\textit{p}}, \text{\textit{q}} & \quad \text{a probability density function} \\
\text{\textit{g}} & \quad \text{Gaussian distribution} \\
\text{\textit{gen}} & \quad \text{generalized} \\
\text{\textit{gm}} & \quad \text{Gaussian mixture distribution} \\
\text{\textit{HW}} & \quad \text{hot water demand} \\
\text{\textit{KL}} & \quad \text{KL divergence} \\
\text{\textit{symm}} & \quad \text{symmetrized} \\
\text{\textit{X}} & \quad \text{each system} \\
\end{align*}
References

A Model for Simulation and Optimal Design of a Solar Heating System with Seasonal Storage

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Abstract:
A thermo-economic model for the simulation and optimization of a Central Solar Heating Plant with Seasonal Storage (CSHPSS) is presented. The model, written in Matlab, is used to investigate the effects of different design variables on thermal performance and cost. Daily and seasonal variations of solar irradiation at different latitudes are considered, and an original approximate model for thermal stratification is included. The simulation model has been also integrated with a non-linear constrained optimization procedure, in order to determine the optimal choice of design variables for different locations and operating conditions.

Keywords:
Solar Thermal Plant, Seasonal Storage, Model, Optimization

1. Introduction

The perspective depletion of fossil fuels and the concerns about climate changes due to carbon dioxide emission in the atmosphere are strongly stimulating the recourse to renewable energy sources and to energy savings [1]. European Community [2] and governments are committed to achieve significant improvements in terms of renewable share of final energy (Fig. 1).

![Fig. 1. EU Renewable Shares, Targets for 2020](attachment:image.png)

Thermal uses represent a relevant fraction of the energy consumption. Solar thermal power is largely used in many countries, particularly in China, to cover this energy demand (Fig. 2). At the end of 2010, a thermal installed power of 185 GW for solar collectors for hot water/space heating was already in existence, and other 30 GW were installed during 2010 [1]. Solar plants can provide part of the thermal power needed for space heating, but they have an intrinsic drawback, since most of the solar power would be available during summer, whereas the demand for thermal power is at its minimum level. In this case, most of the thermal power produced during summer would be wasted.
The usage of solar-cooling may partly overcome this problem, since the exceeding thermal power could be used for refrigeration and air-conditioning through absorption cooling plants. This solution, anyway, is limited by the relatively high cost and low efficiency of such plants. Moreover, there are many cases in which air-conditioning is not needed: significant examples are represented by schools, normally closed during summer. In these cases, a possible solution is resorting to Central Solar Heating Plant with Seasonal Storage (CSHPSS).

In next chapter, a review of the systems for seasonal heat storage is presented, and the modeling approaches are discussed. Then, a model for simulation and optimization of a thermal solar plant with a seasonal water storage is presented, and some results are discussed.

2. Seasonal Heat Storage Systems

In the last decades, several pilot projects on CSHPSS plants have been developed, mainly in Central and Northern Europe. Attention was particularly given to the costs involved in heat storage, since economic feasibility is the most critical factor for this technology.

2.1. Literature review

A technical assessment of the different technologies for solar thermal energy storage is presented in [4]. Both sensible heat and latent heat storage technologies can be adopted. Storage systems are usually classified as “Water storage”, “Earth storage”, “Ground diffusive storage” and “Aquifer storage” (Fig. 3) [5] [6] [7].

Water Storage use tanks constructed from concrete, steel or fiberglass. The tank is located underground to benefit from the insulating and structural properties of the surrounding ground, and to minimize the above-ground space requirements. Insulation is applied to any above-ground surface of the tank [6][8].

Earth Storage (or Pit) Systems are essentially large artificially-dug holes usually filled with water and gravel. They are one of the most popular type of seasonal storage systems, due to low cost and
ease of construction. The gravel provides structural support but reduces the effective storage capacity compared to water alone. An insulated floating cover completes the storage unit.

In Ground Diffusive Storage (or Borehole Systems), heat is stored directly in the ground. Heat exchangers are installed in boreholes drilled in ground that is suitable for heat storage [9]. These bores can be between 30-100 m in depth and 100-150 mm in diameter. The heat exchangers are U-shaped tubes, providing an inlet and outlet for the heat transfer fluid, which is usually water. Insulation is installed at the ground level to minimize heat losses from this top surface. Studies have also been carried out on use of ground source heat pumps [10].

In Aquifer Storage Systems, a naturally-occurring water-saturated media (usually sand) is used as the storage medium. Because the natural occurrence of such bodies in the right location is uncommon, the system is not as diffused as the previous three types.

![Water storage](image1)
![Earth storage](image2)
![Ground diffusive storage](image3)
![Aquifer storage](image4)

**Fig. 3. Schemes of different type of storage systems [14]**

For space heating applications, usually low temperature (less than 100°C) sensible heat storage is generally used. Water is the most suitable thermal storage liquid in such temperature range, due to its high thermal capacity, large availability and low cost. Rocks can also be used as a storage medium.

Since the ’90s, several pilot and demonstration CSHPSS plants have been built in Germany, within a governmental R&D program. The monitoring of such plants has proved to be well matched with the simulation performed during the design phase. Moreover, no major problems during their construction and operation occurred. In 2003, the cost for solar heating with such systems were, at maximum, twice as high as the conventional heat cost [5]. An accurate study on construction techniques and costs of water seasonal storage, based on three pilot projects in Germany, has been published in 1977 [11]. The cost sharing between the component was studied, and different solutions analyzed and discussed. A unit cost of about 90 €/m$^3$, including cost for insulation, were estimated for large volume storage (10.000 m$^3$), while the long term goal was set to about 50 €/m$^3$. Studies on the techniques to prevent oxygen penetration in the storage were performed, since the storage would be used in direct connection with district heating network. Use of steam cushion and of nitrogen atmosphere was discussed [11].

The thermal performance and economic feasibility of two types of central solar heating systems with seasonal storage in Turkey have been investigated by Ucar and Inalli [12] [13]. The effects of different ground types were studied with a finite element analysis. Pay-back time ranging from 19
to 34 years has been found. A study on a CSHPSS under construction in Cheju Island (Korea) is also available [8]. The plant has been simulated using TRNSYS to predict thermal performances and economic outcomes for two different types of solar collectors (flat plate and vacuum tube). Return of investment ranging from 18 to 30 years resulted by these studies.

2.2. Modelling approaches
In most papers, a simulation approach has been followed, also with a parametric analysis of the main design variables: particularly, storage volume and solar panels area. In some cases, thermal stratification has been considered, also with FEM analysis. In most cases, the simulation has been performed with TRNSYS simulation model [15].

Although TRNSYS includes an optional optimization tool (TRNOPT) that would allow to minimize a cost function [15], there are no studies in literature on CSHPSS performed via optimization analysis. A study on the optimization of a near-zero energy solar home via Genetic Algorithms has been presented [16], but seasonal storage was not investigated. In order to limit the computational time required by the optimization analysis, a simplified (one-zone) modeling approach was used. In case of a seasonal storage, a multi-year simulation is needed in order to reach steady operation. Therefore, there is a need for simplified models realizing a good compromise between precision and computational time, in order to allow their use in an optimization tool. A simplified model of a thermal solar plant with seasonal storage, also considering thermal stratification, is presented in next chapter.

3. Model of Thermal Solar Plant with Seasonal Storage
A simplified analytical model of solar irradiation has been adopted, able to describe seasonal and daily variations of irradiance, and the effects of latitude, also considering real sky conditions.

Two different types of solar collectors have been considered, flat plate and vacuum tube. Their efficiency curves are shown in Fig. 4.

![Solar Collectors Efficiency Curves](image)

**Fig. 4. Solar collector efficiency curves.**
A parabolic efficiency curve has been implemented in the model:

\[ \eta = \eta_0 + U_1 T^* + U_2 T^{*2} \]  

(1)

where:

\[ T^* = \frac{t_m - t_{amb}}{T_{rr}} \]  

(2)

The parameters for the two collectors are presented in Table 1. It can be observed that the quadratic terms are zero (Flat Plate) or almost negligible (Vacuum Tube).

<table>
<thead>
<tr>
<th>Collector type</th>
<th>( \eta_0 )</th>
<th>( U_1 )</th>
<th>( U_2 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flat Plate</td>
<td>0.7486</td>
<td>-3.7465</td>
<td>0</td>
</tr>
<tr>
<td>Vacuum Tube</td>
<td>0.8131</td>
<td>-2.16</td>
<td>-0.001</td>
</tr>
</tbody>
</table>

In order to precisely estimate solar collector performance, an analytical model for the time varying value of ambient temperature has been developed, considering sinusoidal variations between daily minimum and maximum temperatures. Daily minimum and maximum temperatures are estimated by linear interpolation from monthly average values, available on-line for many locations [17]. The equations of the solar collector simplified performance model are reported in the Appendix.

Thermal losses are estimated by the following equation:

\[ P_{loss} = \frac{\lambda}{L} A_{st} (t_{mean} - t_g) \]  

(1)

as a function of insulation thickness \( L \), thermal conductivity \( \lambda \), storage area \( A_{st} \), mean storage temperature \( t_{mean} \) and ground temperature \( t_g \). This latter is computed as the yearly average value of ambient temperatures, for each location [17].

### 3.1. Stratification effects

Thermal stratification occurring in water storage plays an important role in the thermal plant management. Stratification has been studied both in experimental way and numerically [18][19]. Usually, rather complex models are used to describe thermal stratification [15]. Although these models are adequate to perform detailed design analysis of the water storage system, they are not suitable to be integrated in optimization studies, sometimes requiring hundreds or thousands of iterations. Therefore, a different approach is pursued in this paper, consisting in the development of a simple model based on the synthesis of physical data describing the most relevant aspects of thermal stratification (grey-box approach). The model has been developed starting from a detailed study on thermal stratification in water thermal storage of different aspect ratio in static mode, published by Khalifa et al. [18]. The results have confirmed that thermal stratification is maximized at higher aspect ratio (i.e. AR=2), while it is less pronounced at lower aspect ratio (AR=0.5). AR is in this case defined as ratio between height and diameter.

The results obtained with AR=1 are shown in Fig. 5. Temperature values in four subsequent times (a, b, c, d) are shown. For each time frame, the difference \( \Delta T \) between maximum temperature \( t_{top} \) and minimum temperature \( t_{bottom} \), and the mean temperature \( t_{mean} \) (computed as the mean value between \( t_{top} \) and \( t_{bottom} \)) have been computed and plotted in Fig. 6. It can be observed that the relationship between these variables is remarkably linear.
Fig. 5 – Thermal stratification in static conditions in a water storage (Aspect Ratio=1). From [18].
The linear model has been then integrated by further relations, in order to take into account that thermal stratification tends to zero when water mean temperature approaches external temperature $t_{\text{min}}$, or in case it approaches maximum allowed water temperature $t_{\text{max}}$. The model, synthesized in the following equations (2)-(5), is represented in Fig. 7.

\[ t_{\text{top}} = t_{\text{mean}} + \frac{\Delta t}{2} \quad (2) \]
\[ t_{\text{bottom}} = t_{\text{mean}} - \frac{\Delta t}{2} \quad (3) \]
\[ \Delta t = \min\left[(k_1 + k_2 t_{\text{mean}});\ 2(t_{\text{max}} - t_{\text{mean}});\ 2(t_{\text{mean}} - t_{\text{min}})\right] \quad (4) \]
\[ \Delta t \geq 0 \quad (5) \]

The parameters of the linear model $k_1$ and $k_2$ in (4) have been identified by linear regression techniques: their values are respectively -7.7039 and 0.3329.

Fig. 7. Thermal stratification model
The model has been validated over the measured data available for a water reservoir of a CSHPSS in Germany [21]. The mean temperature has been computed as the mean of top and bottom measured temperature (red and blue lines). Then, the estimated top and bottom temperatures (red and blue dotted lines) have been computed by means of the model (2)-(5). It can be observed that the matching between measured and computed data is quite satisfactory, suggesting that most of thermal stratification occurring in a typical water reservoir for CSHPSS can be explained considering static effects. Some significant deviations occur in first starting phase, where transient effects may be more relevant. Although this conclusion cannot be generalized, it comes out that the model can be used to make an approximate estimate of thermal stratification in water if a more detailed model is unavailable or cannot be used due to excessive computational time (i.e. for optimization studies).

![Fig. 8 – Comparison of measured and estimated high and low temperatures in a water storage.](image)

The model is integrated starting from given initial conditions for a number of years, until convergence is reached, in terms of difference between initial and final value of the water storage temperature. The thermal and economic performance is then evaluated with reference to the last year. A variable step 4th order Runge-Kutta method, implemented in the “ode45” routine of Matlab [22], has been used. Preliminary numerical studies were performed in order to set proper values of maximum allowable integration step and of termination criteria, in order to find the best compromise between numerical precision and stability, and computational time. This aspect is particularly relevant, since the model has to be integrated within an optimization procedure, where hundreds of iterations may occur. Computational time is about 50 [s] for a year simulation (CPU Intel® Core™ i3, 4 GB RAM, 3.07 GHz).

4. Simulation results

A parametric analysis has been performed, by varying storage volume \( V \) and solar panel area \( A \) in a large range of values (\( A=25\pm1000 \text{ m}^2, V=25\pm1000 \text{ m}^3 \)). The values assumed for the other variables are reported in Table 2. Insulation unit cost is referred to the volume of insulating material. Total insulation cost is then computed considering insulation thickness and area. When reservoir volume increases, the incidence of insulation cost decreases, since the ratio between insulating area and reservoir volume decreases.
Table 2. Data for simulation analysis

<table>
<thead>
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<th>Component</th>
<th>Value</th>
<th>Unit Costs</th>
</tr>
</thead>
<tbody>
<tr>
<td>Solar panel type</td>
<td>Flat plate</td>
<td>Insulation</td>
</tr>
<tr>
<td>Solar panel tilt</td>
<td>40 [deg]</td>
<td>Solar panels</td>
</tr>
<tr>
<td>Insulation thickness</td>
<td>0,25 [m]</td>
<td>Natural gas</td>
</tr>
<tr>
<td>Thermal conductivity</td>
<td>0,04 [W/(mK)]</td>
<td>Storage (except insulation)</td>
</tr>
</tbody>
</table>

Fig. 9. Simulation analysis. Reservoir temperature [°C] vs time [hours]. Solar surface=100-300 m². Reservoir volume=1000 m³. Collector type: Flat plate. Location: Naples.

Thermal power utilized by the user was assumed to be 100 kW, distributed from 8 AM to 1 PM for six days in a week. The plant is supposed to be off from June 1st to October 31st. A total thermal load of 91,5 MWh/year resulted. More articulated load schedules, also taking into account external temperature effects, can be adopted with minor changes. It is assumed that low temperature space heaters are used. Input and output water temperatures can be varied, to account for different heating systems. In the following computations, the values of 35°C and 30°C were assumed for water flow in and out, respectively.

The graphs presented in Fig. 9 show the storage temperature versus time (in hours) for three different cases, characterized by the same storage volume (1000 m³) and decreasing solar panel area (300-200-100 m²). The black, red and blue lines represent respectively the mean, top and bottom water temperature, estimated by the stratification model. Temperature trajectories of both starting year and steady solution are represented in the graphs.
The following figures report the solar fraction (defined as the fraction of the thermal energy demand covered by solar energy) versus storage volume and panel area for all the computed cases. The contour plots of payback (Fig. 10), mean solar collector efficiency (Fig. 11) and storage thermal losses (Fig. 12) are also shown. The solutions at the left of the purple line are characterized by heat dissipation during summer, indicating that storage volume is undersized with regard to solar panel area. This case occurs for the lower graph of Fig. 9 (300 m$^2$) in fact, during summer part of solar heat must be dissipated, to avoid that the storage temperature exceeds the maximum allowed temperature (90°C). It can be observed that temperature stratification is not present during such phase, since it is assumed that heat from solar panel is added until all the storage is at its maximum allowable temperature, to maximize heat storage. Solar fraction is 100%, in this case (Fig. 10). In the second case (A=200 m$^2$) maximum temperature reaches about 80°C and no dissipation occurs, while a solar fraction of about 92% is reached. In the third case (A=100 m$^2$) the storage is oversized and therefore underutilized: maximum temperature is about 60°C, and solar fraction stops at about 59%. The best payback is reached in the second case (A=200 m$^2$) (Fig. 10), while the third case (A=100 m$^2$) is characterized by higher average solar collector efficiency (Fig. 11) and lower storage losses (Fig. 12), due to lower storage temperatures.

The study of the results (Fig. 10, Fig. 11, Fig. 12) shows that the effect of solar collectors area over solar fraction is non-linear: an increase in solar area from 100 m$^2$ to 200 m$^2$ produces a significant improvement in solar fraction (from 43% to 68% about, at the minimum storage volume), while passing from 300 m$^2$ to 400 m$^2$ results in a much lower improvement (from 78% to 85% about). Also the effect of storage volume is non-linear: the slope of the constant solar area curves is positive and almost constant until the saturation conditions (purple lines) are reached, tending to assume an asymptotic behavior afterwards. The observed effects of storage volume and panel area over solar fraction and the pay-back time values, ranging from 15 years up, are consistent with other studies available in literature [9] [12] [13].

![Figure 10](image_url)  
**Fig. 10.** Solar fraction vs reservoir volume, for different solar panel area. Dotted lines: pay-back time [years]. Collector type: Flat plate. Location: Naples.
Fig. 11. Solar fraction vs reservoir volume, for different solar panel area. Dotted lines: solar collector efficiency [%]. Collector type: Flat plate. Location: Naples.

Fig. 12. Solar fraction vs reservoir volume, for different solar panel area. Dotted lines: reservoir thermal losses [%]. Collector type: Flat plate. Location: Naples.
5. Optimization approach

Non-linear effects occur for most of the variables affecting plant performance. In this case, the best set of design variables cannot be determined solely by analyzing each variable independently of other variables, since they are interdependent. Therefore, the adoption of a non-linear optimization approach is suitable [24]. The plant model has been then integrated within a nonlinear constrained optimization algorithm, in order to determine the optimal combination of design and operating variables corresponding to the best values of the performance indices. The mathematical problem is formulated in the following way:

\[ \min_x f(x) \quad (6) \]

\[ G(x) \leq 0 \quad (7) \]

\[ E(x) = 0 \quad (8) \]

\[ LB \leq x \leq UB \quad (9) \]

The objective function (6) is represented by the simple pay-back time, defined as the ratio between the plant cost and the yearly savings. The equality constraint (8) may express the condition that the solar fraction must be equal to a given value (i.e. 100%), while inequality constraints (7) may express the condition that no dissipation occurs, and therefore storage temperature is always below the maximum allowed value.

All the design variables are assumed to be positive, i.e. \( LB=0 \) in (9). A classical 2\(^{nd}\) order Quasi-Newton approach is used [24], as implemented in the routine “fmincon” of the optimization toolbox of Matlab [22].

Preliminary tests have been performed, to verify the functionality of the procedure. The graphs in Fig. 13 show the values of design variables (storage volume and panel area) and of the objective function (pay-back time) versus the iterations, for two different cases (Flat Plate and Vacuum Tube). For this computation, lower values for storage cost (40 €/m\(^3\)) and for insulation cost (40 €/m\(^3\)) were assumed with respect to the values reported in Table 2.

![Optimization results: Flat Plate](image1)

![Optimization results: Vacuum Tube](image2)

Fig. 13. Optimization results: design variables and objective function vs iterations.
It can be observed that, starting from arbitrary initial values \((A=600 \text{ m}^2, V=600 \text{ m}^3)\), optimal solutions are found in both cases in approximately 60 iterations. Very similar optimum pay-back time were obtained (about 17.9 years). The optimal storage volume \(V\) is also quite similar in the two cases (about 1130 \text{ m}^3). In case of vacuum tube, however, lower surface is suggested for solar panels, as expected (Table 3).

Computational time for each test is less than two hours, and therefore compatible with the use of model for design purposes.

<table>
<thead>
<tr>
<th>Solar collector type</th>
<th>(V \text{ [m}^3])</th>
<th>(A \text{ [m}^2])</th>
<th>Pay-back time [year]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flat plate</td>
<td>1125</td>
<td>239</td>
<td>17.8</td>
</tr>
<tr>
<td>Vacuum tube</td>
<td>1136</td>
<td>161</td>
<td>17.9</td>
</tr>
</tbody>
</table>

### Conclusions

Thermal solar plants with seasonal water storage have been considered as an environmentally valuable, but expensive, solution to cover thermal energy demand in buildings. In recent years their economic feasibility is improving, due to the parallel increase of fuel cost and decrease of solar collectors cost. A thermo-economic model of a thermal solar plant with seasonal water storage has been presented. The model includes also a simple sub-model for the estimation of thermal stratification in water storage, successfully validated by literature data. The results of a parametric analysis on the effects of storage volume and panel area over solar fraction, pay-back time, average collector efficiency and thermal losses have been presented and compared with other studies available in literature, and non-linear effects have been evidenced.

The model has been integrated within a non-linear constrained optimization procedure. Preliminary tests have demonstrated that the results are sound and that computational time is compatible with practical uses of this tool for design purposes.

In future research, the model will be used to make a systematic study of the effects of design variables, as tilt angle, insulation thickness, panel type, as well as of operating and control variables. The effects of plant location over optimal design variables will be also investigated by means of the optimization procedure.

### Acknowledgments

The contributions given to the present analysis by Claudià Poto and Vito Della Corte during their Master Thesis in Mechanical Engineering at the University of Salerno are gratefully acknowledged.

### Appendix

The following equations describe the solar power absorbed by a solar collector with tilt angle \(\beta\) and azimuth \(\psi\), in a location of latitude \(\varphi\). The model predicts the direct solar irradiance with sunny sky \(I_{rr}\). The reduction factor \(f_{SUN}\) is introduced in (17) to predict irradiance under real sky conditions. This factor has been identified for different locations (Fig. 14) [25], starting from average real monthly solar data [23].
\[ \delta = 0.4093 \sin \left( \frac{2n \frac{284 + d}{365}}{2} \right) \]  
\[ \omega = 2\pi \frac{12 - h}{12} \]  
\[ z = \arccos (\sin(\delta) \cdot \sin(\varphi) + \cos(\delta) \cdot \cos(\varphi) \cdot \cos(\omega)) \]  
\[ AM = \min \left[ \frac{1}{\sin \left( \frac{\pi}{2} - z \right)} \right] \]  

\[ Irr = 1.1 \cdot I_0 \cdot 0.7^{AM^{0.678}} \]  

\[ \psi_{SUN} = \arccos \left( \frac{(\sin(\delta) \cdot \cos(\varphi) - \cos(\omega) \cdot \cos(\delta) \cdot \sin(\varphi))}{\cos \left( \frac{\pi}{2} - z \right)} \right) \]  
\[ \theta = \arccos \left( \cos \left( \frac{\pi}{2} - z \right) \cdot \cos (\psi_{SUN} - \Psi_{SUN}) \cdot \sin(\varphi) \right) \]  
\[ P_{SUN} = Irr \cdot f_{SUN} \cdot \cos(\theta) \]  

**Fig. 14. Solar reduction factor for different locations**
Nomenclature

\[ A \] Solar collector area \([m^2]\)
\[ AM \] Air Mass \([/]\)
\[ A_{w} \] Water storage area \([m^2]\)
\[ d \] Day index (1-365) \([/]\)
\[ E \] Equality constraint
\[ f \] Objective function
\[ f_{SUN} \] Solar radiation reduction factor \([/]\)
\[ G \] Inequality constraint
\[ h \] Hour (1-24) \([/]\)
\[ I_{0} \] Solar constant=1366 \([W/m^2]\)
\[ Irr \] Irradiance \([W/m^2]\)
\[ k_{1}, k_{2} \] Parameters in the stratification model
\[ L \] Insulation thickness \([m]\)
\[ LB \] Lower bound
\[ P_{loss} \] Reservoir thermal losses \([W]\)
\[ P_{SUN} \] Solar power \([W/m^2]\)
\[ T^{*} \] Variable in collector efficiency model
\[ t_{amb} \] Ambient temperature \([^\circ C]\)
\[ t_{bottom} \] Bottom temperature in the water reservoir \([^\circ C]\)
\[ t_{g} \] Ground temperature \([^\circ C]\)
\[ t_{m} \] Solar collector mean temperature \([^\circ C]\)
\[ t_{max} \] Maximum allowed temperature in the water reservoir \([^\circ C]\)
\[ t_{mean} \] Mean temperature in the water reservoir \([^\circ C]\)
\[ t_{top} \] Top temperature in the water reservoir \([^\circ C]\)
\[ U_{1}, U_{2} \] Parameters of the collector efficiency model
\[ UP \] Upper bound
\[ V \] Reservoir volume \([m^3]\)
\[ x \] Decision variables
\[ z \] Zenith angle \([rad]\)

Greek Symbols

\[ \eta \] Solar collector efficiency \([/]\)
\[ \eta_{0} \] Parameter in the collector efficiency model
\[ \Delta t \] Difference between top and bottom temperature in the water reservoir
\[ \beta \] Tilt angle \([rad]\)
\[ \delta \] Declination \([rad]\)
\[ \theta \] Incidence angle \([rad]\)
\[ \lambda \] Thermal conductivity \([W/mK]\)
\[ \varphi \] Latitude \([rad]\)
\[ \psi \] Azimuth \([rad]\)
References


A thermodynamic and economic comparative analysis of combined gas-steam and gas turbine air bottoming cycle

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Abstract:
An increase in the efficiency of energy systems can be achieved by the development of combined cycles. Examples of high efficiency cycles are combined cycle power plants (CCPP) and gas turbine air bottoming cycles (GT-ABC), which are a combination of a gas turbine and air turbine cycle coupled by means of a heat exchanger referred to as the Air Heat Exchanger (AHX). The main feature of the GT-ABC is a low water consumption. For this reason, it can be used in gas transport and storage systems.

In this paper the GT-ABC and the combined gas-steam cycle designed for the same class of application are compared. An example of the considered technological structures is presented. The calculations include thermodynamic characteristics and a preliminary economic analysis determining the capital expenditure for each installation. In particular, the GT-ABC (either a simple system or a version with an intercooled compressor) and the combined gas-steam cycle with a single-pressure Heat Recovery Steam Generator (HRSG) are compared. Due to a limited access to cooling water at a potential place of application, the combined gas-steam cycle with an air fan cooling tower and an air-cooled condenser are used.

Keywords:
Air turbine, ABC (Air Bottoming Cycle), GT (Gas Turbine), HRSG (Heat Recovery Steam Generator), efficiency, combined gas-steam cycle, heat exchanger, AHX (Air Heat Exchanger), cogeneration, economic analysis, thermodynamic analysis.

1. Introduction
Gas turbines are one of the basic technologies used to produce electricity and power working machinery. The popularity of the technology results from its advantages, the most important of which are: the fast start-up, high efficiency, low pollutant emissions, the short time needed for the installation to be constructed and a reasonable size. Gas turbines are becoming increasingly important in new power installations [1-8]. They find application in hierarchical power systems, e.g.
- gas-steam systems,
- combined multi-fuel systems,
- pressure fluidised bed boiler systems,
- partial and complete gasification systems,
- gas-air systems [9-12].

So far, cycles with steam or an organic medium have been the ones employed in combined systems with gas turbines [13-16]. The main factor that decided about the popularity of these systems was their high power efficiency.
Another option is to couple the gas turbine with an air turbine (GT-ABC) by means of an air heat exchanger (AHX). The construction of this type of systems may turn out to be energy-effective due to the advancement in flow machinery construction, especially in the field of improvement to blade profiles and sealing. Other features that make gas-air systems interesting are the following:

- an increase in the efficiency of power installations with gas turbines,
- the potential to meet the peak demand for power,
- mobility,
- no demand for water,
- no toxic substances,
- a harmless working medium,
- lower investment costs compared to gas-steam systems,
- fuel diversification for a given coal-fired power plant.

The GT-ABC systems can find application in the food industry (industrial bakeries, powdered milk factories and as a source of hot air in glass melting furnaces), in cogenerative systems with air as the working medium or in high-temperature furnaces where the pre-heated air comes from the ABC. Gas-air systems can also be used as a potential improvement to the efficiency of simple power units with gas turbines operating at locations without access to large amounts of water. It seems that these systems could be applied in other technologies, such as power engineering of renewable sources, energy recovery from gases, especially where water availability is limited. The potential applications of gas-air systems in coal-fired plants as a source of heat feeding the carbon dioxide capture installations, as well as in heat engineering are also considered [11,13]. The mechanical power obtained from the turbine can be used either to support the gas turbine system or to generate electricity. Due to the short start-up time of the air turbine, the ability to meet the peak demand for power may also be significant.

Air turbine systems are simple in terms of operation. This results from the fact that there is no combustion process and there are no toxic mediums or mediums causing erosion or needing to be topped up.

2. Systems under analysis

It is assumed that the source of heat for gas-air and gas-steam systems is the turbine ABB GT10, whose basic data are listed in Table 1 [17-19].

<table>
<thead>
<tr>
<th>Table 1. Basic data of ABB GT10 gas turbine</th>
</tr>
</thead>
<tbody>
<tr>
<td>Model</td>
</tr>
<tr>
<td>Power $N_{GT}$, MW</td>
</tr>
<tr>
<td>Turbine inlet temperature $t_4$, °C</td>
</tr>
<tr>
<td>Net efficiency $\eta_{GT}$, %</td>
</tr>
<tr>
<td>Pressure ratio $\beta_{GT}$, -</td>
</tr>
<tr>
<td>Heatrate $q_{GT}$, kJ/kWh</td>
</tr>
</tbody>
</table>
The following composition of flue gases at the outlet from the gas turbine expander is assumed for the modelling:

- $\text{N}_2$: 75.76%
- $\text{O}_2$: 13.56%
- $\text{CO}_2$: 3.28%
- $\text{H}_2\text{O}$: 7.40%

### 2.1. Gas-air systems

The simplest model of an air turbine with a gas turbine system is shown in Fig. 1. The system is composed of a simple gas turbine, an air turbine and a compressor.

The crucial element is the structure of the air heat exchanger (AHX) [20-26], which has a decisive impact on the efficiency of the entire system. A rationally designed system must take account of the differences in the medium temperature which determine its size, as well as the pressure drops which determine the efficiency of both the air and the gas turbine. A high efficiency of the system is obtained for small temperature differences in the heat exchanger. Therefore the heat exchanger will have a large size. Shell-and-tube heat exchangers or plate heat exchangers can be applied. Preliminary calculations show that lower pressure drops are obtained for plate structures. Also, the area of a plate heat exchanger is significantly smaller.

![Schematic diagram of a simple gas-air system](image)

The air turbine system is composed of compressor $C_1$, an air heat exchanger (AHX) and expander $T_1$. The main advantage of this type of system is its simplicity; the downside, however, is the fact that the air temperature at the compressor outlet is high, which causes that a relatively small amount of heat is exchanged in the AHX. Due to the low values of the upper heat source temperature, this system achieves high efficiency values only if the internal efficiency of the compressor and the turbine is also high.

In order to improve the efficiency of air turbine installations, it is necessary to employ more complex system configurations. An example of a complex system of the air turbine is the installation shown in Fig. 2. In this system two compressors and an intercooler are used. The outlet air from compressor $C_3$ has a lower temperature compared to the simple system, which allows a
more intense cooling of the gas turbine flue gases and a reduction in the driving operation of the compressors. In the calculations of this system it is assumed that the air in the intercooler is cooled to the temperature of 40°C.

![Diagram of a complex gas-air system with an air intercooler](image)

**Fig. 2. Complex gas-air system with an air intercooler**

### 2.2. Gas-steam system

In order to compare gas-air and gas-steam systems, the same gas turbine (ABB-GT10) is used in it. The flowchart of the gas-steam system under consideration is presented in Fig. 3. In the analysed system the heat recovery steam generator (HRSG) is a single-pressure boiler composed of three different heat exchange surfaces: the economiser (ECO), the evaporator (EVAP) and the steam superheater (SH). The deaerator (DEA) is fed with steam from the steam turbine (ST) bleed. The task of the heat exchanger (HX) after the condensate pump (CP) is to stabilise the boiler feed water temperature at a constant level, which is a standard solution in gas-steam power plants. The system electricity generation efficiency reaches 48.4%. A downside of a system with a single-pressure heat recovery boiler is the large exhaust loss (approx. 180°C).

![Diagram of the gas-steam system with single pressure HRSG](image)

**Fig. 3. The flowchart of the gas-steam system with single pressure HRSG**
A subcooling temperature at the level of $t_{sc} = 10^\circ C$ is assumed in the economiser. The pinch point in the evaporator is $t_{pp} = 6^\circ C$, and the live steam temperature is $t_{l} = 535^\circ C$. 

In order to minimise the demand for water, and thereby ensure a better comparison of the gas-steam cycle to gas-air systems, an air-cooled condenser with an air fan cooling tower (CND) is used. The air-cooled condenser is composed of bays; in each bay several fans can be placed. Each bay is fed with steam from a common collector. Due to the fact that the condensate is also carried away to the common collector, the whole can be treated as a group of coolers. The facility is available in two variants:

- condensation,
- cooling.

In the former case, the air sucked in from the environment is used to cool the steam to saturation temperature. In the latter, the sucked-in air is used to cool water to a set temperature value (enthalpy). Fig. 4 schematically presents the two modes of the air condenser operation [18,27].

![Fig. 4. Two variants of air cooled condenser operation (a. – condensation, b. – cooling)](image)

### 3. Calculation results

In the case of the standalone operation of a gas turbine, the important parameter is electricity generation efficiency defined by the formula:

$$\eta_{elGT} = \frac{N_{elGT}}{m_f \cdot LHV}$$

(1)

The target of a gas-air system analysis should be the maximisation of the ratio of the amount of generated electricity to the chemical energy of the fuel. In this case, the power efficiency of the system can be defined as follows:

$$\eta_{elAT-elGT} = \frac{N_{elGT} + N_{elAT}}{m_f \cdot LHV}$$

(2)

The system efficiency can be evaluated using the following definition of power efficiency:

$$\eta_{elAT} = \frac{N_{elAT}}{\dot{Q}_4}$$

(3)

where: $\dot{Q}_4$ – the heat of cooling flue gases to the reference temperature.
In order to simplify the analyses, it is assumed in the calculations presented in this paper that $Q_4$ corresponds to the cooling of flue gases to temperature $t_5 = 15^\circ\text{C}$, while the water vapour contained in flue gases is not condensed.

The cycle efficiency is defined by following dependence:

$$\eta_{\text{cAT}} = \frac{N_{\text{eAT}}}{I_4 - I_5}$$  \hspace{1cm} (4)

where:

$I_4$ and $I_5$ are values of enthalpy corresponding to point 4 and 5 (Fig.1 and Fig.2).

The electricity generation efficiency of the gas-steam power plant is defined as follows:

$$\eta_{\text{GCCPP}} = \frac{N_{\text{eGT}} + N_{\text{elST}}}{m_f \cdot \text{LHV}}$$  \hspace{1cm} (5)

The air system is optimized in terms of efficiency. The pressure ratio and the air mass flow are chosen as decision variables. The other values are selected according to the parameters of the machines and equipment operating in gas turbine systems. For set parameters of the heat exchanger, pressure value $p_{2a}$ is varied within the range of 0.15-0.6 MPa. The procedure is repeated for different mass flow values of the air sucked in by compressor $C_2$.

It is important to choose an appropriate pressure value at the outlet of the compressor’s first stage. The optimum pressure value which minimizes the power consumption to drive the compressor can be determined by minimizing the following objective function:

$$\frac{1}{\eta_t} \left[ T_{lw}(\xi_1 - 1) + T_{3a}(\xi_3 - 1) \right] \rightarrow \min$$  \hspace{1cm} (6)

where: $\xi_1 = \frac{T_{2a}}{T_{lw}}$, $\xi_3 = \frac{T_{4a}}{T_{3a}}$.

### 3.1. Air Heat Exchanger calculations

Parameters which have a significant impact on the efficiency of the entire system are overall heat transfer coefficient value and pressure drop in the air heat exchanger (AHX). The higher the pressure drop in AHX, the lower efficiency $\eta_{\text{cAT}}$. On the other hand, a bigger pressure drop makes it possible to obtain high heat transfer coefficients, which leads to a reduction in the heat exchange area and, consequently – to a smaller size of the device. In order to determine the heat transfer coefficient many physical properties of air and flue gas were taken into account:

$$\alpha = f(\eta, \lambda, c_p, \rho, \nu, \Delta t, a...)$$  \hspace{1cm} (7)

The calculation algorithm is based on the LMTD method. [22-24]. To determine the heat surface area the Nusselt number should be found. The plate heat exchanger was taken into account. The Nusselt number for this type of heat exchangers can be defined by the formula:

$$Nu = 0.022 \cdot \sqrt{\frac{1}{\xi_0}} \cdot \beta \cdot \beta_t \cdot Re^{0.825} \cdot Pr^{0.54}$$  \hspace{1cm} (8)

where:

$\beta$ – turbulence damping ratio,

$\beta_t$ – forced turbulence ratio,

$\xi_0$ – flow resistance ratio.

In order to obtain reduced surface of heat exchange with relatively high values of the heat transfer coefficient, higher velocities of the working mediums are required. This, however, results in a bigger flow resistance and a higher pressure drop. Finally the low velocity values (especially on the air side) were chosen. The loss in the AHX also depends on the equivalent diameter of the duct and
on the heat radiation losses. In calculation the radiation losses were assumed at a level equal \( \dot{Q}_{\text{rad}} = 1\% \). The velocity of flue gases and equivalent diameter of ducts (flue gas side) has been also assumed. Assumption of flue gas velocity equal 15m/s resulted in a very low air speeds, but also very low pressure drop value (especially in the air side).

The overall heat transfer coefficient was between 23-30 W/(m²K). The example results of the analysis performed for the plate heat exchanger are given in Table 2. In the case of simple system an air inlet temperature was at a level of about \( t_{\text{ain}} = 170^\circ\text{C} \) with the isentropic efficiency of compressor equal \( \eta_{iC2} = 85\% \) (in the case of complex system \( t_{\text{ain}} = 105^\circ\text{C} \) and \( \eta_{iC2} = 85\% \) respectively).

Determining of the pressure drop value is done by calculating the sum of hydraulic resistance (local resistance) as well as longitudinal resistance.

It is assumed that the wall which separates mediums in AHX are made of P235GH, 16Mo3 and 14CrMo4-5 steel, and its heat conductivity depends on temperature [22].

Table 2. Example of AHX calculation results

<table>
<thead>
<tr>
<th></th>
<th>Simple system</th>
<th>Complex system</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat transfer surface, m²</td>
<td>approx. 58700</td>
<td>approx. 58200</td>
</tr>
<tr>
<td>Pressure drop (flue gas side), %</td>
<td>approx. 1.65</td>
<td>approx.1.65</td>
</tr>
<tr>
<td>Pressure drop (air side), %</td>
<td>approx. 0.3</td>
<td>approx 0.3</td>
</tr>
<tr>
<td>Overall heat transfer coefficient, W/(m²K)</td>
<td>approx 25</td>
<td>approx 25</td>
</tr>
<tr>
<td>Equivalent diameter of flue gas duct, m</td>
<td>0.04</td>
<td>0.04</td>
</tr>
<tr>
<td>Effectiveness, %</td>
<td>95</td>
<td>95</td>
</tr>
</tbody>
</table>

The heat transfer surface area is strongly dependent on overall heat transfer coefficient and assumed equivalent diameter. Determination of AHX geometry is broad issue of thermoeconomic optimization. A trade-off between the system energy efficiency and the size of the AHX will be required (minimization of investment expenditures and maximizing the energy efficiency). It should be mentioned that also plate fin heat exchanger will be considered as an element which coupled topping and bottoming cycle.

### 3.2. Gas Turbine Air Bottoming Cycle - Results and discussion

Most often, the optimum parameters of the air turbine cycle with respect to the power efficiency of cycle \( \eta_{eAT} \) and the efficiency of cycle \( \eta_{eAT} \) are different, which is shown in the example chart in Fig. 5, which presents the dependence of the air turbine system efficiency as a function of the compressor outlet pressure. Pressure drop in AXH is \( \Delta P_a = \Delta P_{fg} = 1\% \), isentropic efficiency of compressor and air turbine is \( \eta_{iC2} = \eta_{iT} = 0.9 \), \( t_{\text{cpt}} = 10^\circ\text{C} \). Flue gas temperature which is a function of pressure ratio is equal \( t_4 = 500^\circ\text{C} \).

Nomenclature used:

\[
\begin{align*}
\Delta P_a & \quad \text{– relative pressure drop (air side),} \\
\Delta P_{fg} & \quad \text{– relative pressure drop (flue gas side),} \\
t_{\text{cpt}} & \quad \text{– cold pinch temperature,} \\
\eta_{iCi} & \quad \text{– compressors internal efficiency,} \\
\eta_{iT} & \quad \text{– turbine internal efficiency.}
\end{align*}
\]
It’s should be noted that every point of each characteristic in Fig 5-8 represents other engine. That’s why the analysis was conducted with constant isentropic efficiency.

Energy efficiency as a function of compressor outlet pressure for simple cycle with ABB GT10 is presented in Fig 6. Isentropic efficiency of compressor and air turbine is $\eta_{iC2} = \eta_{iT} = 0.85$ and $t_{cp} = 10^\circ$C.

The factor which most affects the installation efficiency is the temperature of flue gases. A drop in the flue gas temperature results in a considerable decrease in efficiency, with other optimum values, such as the compressor outlet pressure, changed at the same time. In ABB GT10 the turbine outlet temperature is relatively high ($t_4 = 540^\circ$C).

Table 3 presents the results of the power efficiency of a simple gas-air system depending on the efficiency of the gas turbine system for a given temperature difference between the flue gas outlet and the air inlet in the AHX (three values of isentropic efficiency of the turbomachinery in the air system are distinguished). The pressure drop on part of both flue gases and air is 4%.

The impact of the pressure drop in the AHX on the efficiency of the simple gas-air system was also analysed. The difference in temperatures between the air inlet and the flue gas outlet is 10K. The results are listed in Table 4.
Table 3. Power efficiency of a simple gas-air system for different values of the difference in the flue gas and air temperature in the AHX ($t_{fg}=540^\circ C$, $\Delta P_a = \Delta P_{fg} = 4\%$)

<table>
<thead>
<tr>
<th>$\Delta T$, K</th>
<th>$\eta_{GT}$, $%$</th>
<th>$\eta_{C2} = \eta_{IT}$, $%$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>82.50%</td>
<td>85.00%</td>
</tr>
<tr>
<td>30</td>
<td>35.03</td>
<td>36.27</td>
</tr>
<tr>
<td>10</td>
<td>39.67</td>
<td>40.82</td>
</tr>
<tr>
<td>40</td>
<td>44.31</td>
<td>45.37</td>
</tr>
<tr>
<td>20</td>
<td>34.77</td>
<td>35.97</td>
</tr>
<tr>
<td>35</td>
<td>39.43</td>
<td>40.54</td>
</tr>
<tr>
<td>40</td>
<td>44.09</td>
<td>45.12</td>
</tr>
<tr>
<td>30</td>
<td>34.52</td>
<td>35.69</td>
</tr>
<tr>
<td>35</td>
<td>39.19</td>
<td>40.28</td>
</tr>
<tr>
<td>40</td>
<td>43.87</td>
<td>44.88</td>
</tr>
</tbody>
</table>

Table 4. Power efficiency of a simple gas-air system for different values of the pressure drop in the AHX ($t_{fg}=540^\circ C$)

<table>
<thead>
<tr>
<th>$\Delta P$, $%$</th>
<th>$\eta_{GT}$, $%$</th>
<th>$\eta_{C2} = \eta_{IT}$, $%$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>82.50%</td>
<td>85.00%</td>
</tr>
<tr>
<td>3</td>
<td>35.22</td>
<td>36.45</td>
</tr>
<tr>
<td>35</td>
<td>39.84</td>
<td>40.99</td>
</tr>
<tr>
<td>40</td>
<td>44.47</td>
<td>45.53</td>
</tr>
<tr>
<td>5</td>
<td>34.85</td>
<td>36.09</td>
</tr>
<tr>
<td>35</td>
<td>39.39</td>
<td>40.65</td>
</tr>
<tr>
<td>40</td>
<td>44.16</td>
<td>45.22</td>
</tr>
<tr>
<td>7</td>
<td>34.48</td>
<td>35.72</td>
</tr>
<tr>
<td>35</td>
<td>39.16</td>
<td>40.31</td>
</tr>
<tr>
<td>40</td>
<td>43.84</td>
<td>44.90</td>
</tr>
</tbody>
</table>

In order to improve the efficiency of air turbine installations, it is necessary to employ more complex system configurations. In the system two compressors and an air intercooler are used. The outlet air from compressor $C_3$ has a lower temperature compared to the simple system, which allows a bigger cooling of the gas turbine flue gases and a reduction in the driving operation of the compressors. In the calculations of this system it is assumed that the air in the intercooler is cooled to the temperature of $40^\circ C$.

Energy efficiency as a function of compressor $C_3$ outlet pressure for complex cycle with ABB GT10 is presented in Fig 7. Isentropic efficiency of compressor and air turbine is
$\eta_{C2} = \eta_{T} = 0.85$ and, $t_{cpt} = 10^\circ$C. Compressor $C_2$ outlet pressure value was assumed according to dependence 6.

Fig. 7. Dependence of efficiency as a function of compressor $C_3$ outlet pressure for complex cycle with ABB GT10 (results for different value of pressure drop in AHX)

The impact of the isentropic efficiency of the compressors and of the turbine on the power efficiency was also analysed (Fig. 8). A decrease in the internal efficiency values results in a reduction in the power efficiency and in a drop in the optimum values of compressor $C_3$ outlet pressure. Pressure drop in AHX is $\Delta P_a = \Delta P_{fg} = 1\%$, isentropic efficiency of compressors and air turbine is $\eta_{iC2} = \eta_{iC3} = \eta_{iT}$, $t_4 = 550^\circ$C, $t_{cpt} = 10^\circ$C.

Fig. 8. Dependence of efficiency $\eta_{\text{AT}}$ as a function of the compressor outlet pressure for different values of the isentropic efficiency of the compressors and the turbine ($\eta_{iC2} = \eta_{iC3} = \eta_{iT}$)

Table 5 presents the results of the power efficiency of a complex gas-air system depending on the efficiency of the gas turbine system for a given temperature difference between the flue gas outlet and the air inlet in the AHX (three values of isentropic efficiency of the turbomachinery in the air system are distinguished). The pressure drop on part of both flue gases and air is 4%.

The impact of the pressure drop in the AHX on the efficiency of the complex gas-air system was also analysed. The difference in temperatures between the air inlet and the flue gas outlet is 10K. The results are listed in Table 6.
The gas-steam cycle was analysed with regard to the determination of the highest value of electricity generation efficiency for a given pressure of generated steam. The analysis was conducted parametrically for various pressure values in the air condenser (Fig. 9). In the gas-steam cycle the steam turbine featured internal efficiency at the level of $\eta_{\text{ST}} = 90\%$.

Table 5. Power efficiency of a complex gas-air system for different values of the difference in the flue gas and air temperature in the AHX ($t_{fg}=540^\circ C$)

<table>
<thead>
<tr>
<th>$\Delta T$, K</th>
<th>$\eta_{\text{GT}}, %$</th>
<th>$\eta_{\text{C2}}=\eta_{\text{C3}}=\eta_{\text{IT}}, %$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>30</td>
<td>36.01</td>
</tr>
<tr>
<td>10</td>
<td>35</td>
<td>40.57</td>
</tr>
<tr>
<td></td>
<td>40</td>
<td>45.15</td>
</tr>
<tr>
<td>20</td>
<td>30</td>
<td>35.68</td>
</tr>
<tr>
<td></td>
<td>35</td>
<td>40.27</td>
</tr>
<tr>
<td></td>
<td>40</td>
<td>44.87</td>
</tr>
<tr>
<td>30</td>
<td>30</td>
<td>35.36</td>
</tr>
<tr>
<td></td>
<td>35</td>
<td>39.97</td>
</tr>
<tr>
<td></td>
<td>40</td>
<td>44.59</td>
</tr>
</tbody>
</table>

Table 6. Power efficiency of a complex gas-air system for different values of the pressure drop in the AHX ($t_{fg}=540^\circ C$)

<table>
<thead>
<tr>
<th>$\Delta P$, %</th>
<th>$\eta_{\text{GT}}, %$</th>
<th>$\eta_{\text{C2}}=\eta_{\text{C3}}=\eta_{\text{IT}}, %$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>30</td>
<td>36.16</td>
</tr>
<tr>
<td>3</td>
<td>35</td>
<td>40.72</td>
</tr>
<tr>
<td></td>
<td>40</td>
<td>45.28</td>
</tr>
<tr>
<td>5</td>
<td>30</td>
<td>35.84</td>
</tr>
<tr>
<td></td>
<td>35</td>
<td>40.43</td>
</tr>
<tr>
<td></td>
<td>40</td>
<td>45.01</td>
</tr>
<tr>
<td>7</td>
<td>30</td>
<td>35.52</td>
</tr>
<tr>
<td></td>
<td>35</td>
<td>40.12</td>
</tr>
<tr>
<td></td>
<td>40</td>
<td>44.73</td>
</tr>
</tbody>
</table>
Fig. 9. Electricity generation efficiency depending on steam pressure for different pressure values in the air cooled condenser

Even if the condenser pressure is $p_s = 10$ kPa, by selecting an appropriate steam pressure value, it is possible to achieve the system electricity generation efficiency at the level exceeding $\eta_{CCPP} = 48\%$. Considering the above, the gas-steam cycle turns out to be thermodynamically better than the gas-air cycle (both simple and complex with interstage cooling).

4. Economic analysis

The execution of an investment involves incurring certain investment expenditures. The determination of the size of the outlays is based on the information obtained from design departments. This information is often supplemented with feasibility studies, concerning for example:

- the costs related to the installation and start-up of the machinery and equipment,
- the costs related to the purchase and preparation of the site intended for the investment,
- the costs related to the construction works, etc.

The design of a consolidated statement of costs for a power engineering investment consists of:

- Studies, documentation and site preparation:
  - research works and related project documentation,
  - site acquisition and preparation.
- Basic and auxiliary facilities:
  - basic production facilities,
  - auxiliary and service facilities,
  - energy management facilities,
  - transport and communication management facilities,
  - external networks and structures related to them.
- Temporary structures together with the construction site equipment:
  - infrastructure for the contractor's needs,
  - temporary structures for the investor's needs.
- Investment-related services, training, start-up:
  - maintenance of investment staff,
  - preparation of the operational staff,
  - start-up costs.
Reserve for incidental works and expenses.

This paper is focused on the estimation of the purchase costs of the basic machinery and equipment operating in the gas-air and the gas-steam installations. Two variants are distinguished in the case of the former installation:

- a simple system,
- a complex system.

### 4.1. Gas turbine purchase cost

The same gas turbine is used in each of the installation type mentioned above. The cost of generation of 1kWh was taken from [28] and adjusted with appropriate economic indicators for year 2011 [29].

The cost of the gas turbine can be determined from the dependence [30,31]:

\[
C_{GT} = 10764 \cdot N_{aTG}^{0.7} 
\]

The costs resulting from investment outlays on the purchase of machines and equipment are often related to their technical and thermodynamic parameters. For example, the compressor price can be defined by the following dependence [30]:

\[
C_c = \alpha_i \cdot \frac{39.5 \cdot m_{1a}}{0.9 - \eta_{ik}} \cdot \beta_k \cdot \ln \beta_k 
\]

And the cost of the expander for the gas turbine installation by [30]:

\[
C_r = \alpha_i \cdot \frac{266.3 \cdot m_{3a}}{0.92 - \eta_{ir}} \cdot \ln \beta_r \cdot [1 + \exp(0.036 \cdot T_{3a} - 54.4 \cdot \alpha_z)] 
\]

where [30]:

- \( \alpha_i \) – constants,
- \( m_{1a} \) – amount of air fed into the compressor, kg/s
- \( \beta_k \) – compressor pressure ratio, -
- \( \beta_r \) – turbine pressure ratio, -
- \( m_{3a} \) – amount of flue gases at the turbine inlet, kg/s
- \( T_{3a} \) – turbine flue gas inlet temperature, K
- \( \eta_i \) – internal efficiency, -

The purchase cost of the entire gas turbine system is about:

\[ C_{GT} = $12,500,000 \]

### 4.2. Gas-air (GT-ABC) system purchase cost

In the case of the gas-air system, in addition to the gas turbine installation, it is necessary to determine the purchase cost of the air heat exchanger AHX, whose price is estimated in the same way as in the case of the heat recovery steam generator of the steam system, i.e. using the information obtained from industry [32]. The materials taken into account here are steels: P235GH, 16Mo3 or 13CrMo4-5 (EN 10216). Chromium steel would only be used at places where the working temperature exceeds the maximum permissible temperature for steel 16Mo3. The cost of purchase of the compressor and the expander was determined in the same way as for individual machines in the gas turbine system. In the case of the gas-air system with an intercooling installation it is necessary to include additionally the purchase cost of the heat exchanger (the interstage cooler), for which material P235GH (EN 10216) is used.
The cost of turbomachinery for simple system depending on the assumed isentropic efficiency is listed in Table 7.

Table 7. Purchase cost of individual air installation turbomachines (simple system)

<table>
<thead>
<tr>
<th>Machine type</th>
<th>Price, USD (at given isentropic efficiency)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>82.5%</td>
</tr>
<tr>
<td>Compressor</td>
<td>260,000</td>
</tr>
<tr>
<td>Expander</td>
<td>380,000</td>
</tr>
</tbody>
</table>

Undoubtedly, the biggest cost is the purchase of the AHX. Preliminary studies show that plate heat exchangers are more effective in minimising the pressure drops in the working media and the heat exchange area. Assuming a counterflow heat exchanger, three steel grades can be used in it at the same time: P235GH, 16Mo3 and 13CrMo4-5 (the following proportion is assumed: 43% of steel P235GH, 36% of steel 16Mo3 and 21% of steel 13CrMo4-5). Dividing costs in this way, the cost of purchase of the heat exchange area only will be about $1,850,000\(^1\). For simple system the purchase cost is bigger (approx. $2,300,000\(^2\)). The shell and tube construction is not longer taken into account because of both bigger heat transfer area and bigger pressure drop of flue gases and air.

In the case of the gas-air system with an interstage cooler, the heat exchanger (the cooler) between the two groups of compressor stages has to be additionally included in the costs. Steel P235GH is assumed for the cooler. The interstage cooler is an air-to-air exchanger. For the assumed heat exchange effectiveness of \(\varepsilon = 0.74\), and keeping the temperature at the inlet into the second compressor at the level of \(t_{3a} = 40^\circ\text{C}\), the cost of purchase for this machine would be about $110,000.

The purchase cost of complex system turbomachinery is shown in Table 8. The presented prices are calculated for the maximum pressure ratio values. Due to the fact that the pressure value before the AHX in the system with an interooler cooler is higher than in the simple gas-air system, the expander price rose.

Table 8. Expander purchase cost for a complex gas-air system

<table>
<thead>
<tr>
<th>Machine type</th>
<th>Price, USD (at given isentropic efficiency)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>82.5%</td>
</tr>
<tr>
<td>Compressor</td>
<td>490,000</td>
</tr>
<tr>
<td>Ekspander</td>
<td>400,000</td>
</tr>
</tbody>
</table>

The price of the generators for the respective gas-air systems is [30,31]:

- a simple system – $1,650,000, (net power output – approx. 29.2 MW; \(\eta_g=0.985\)),
- a complex system – $1,600,000, (net power output – approx. 28.5 MW; \(\eta_g=0.985\)).

According to [33] the purchase cost of other components like ducts, dampers, stack, instrumentation, etc. was estimated on approx $1,200,000 - $1,300,000.

\(^1\) Heat transfer surface area is equal approx. 66500m\(^2\).
\(^2\) Heat transfer surface area is equal approx. 83000m\(^2\).
4.3. Gas-steam (CCPP) system purchase cost

In the case of the steam system, the heat recovery steam generator purchase cost plays an important role. The cost of the facility is estimated based on the information obtained from industry concerning the price of steel of which the individual parts of the heat exchange surfaces are made, on the area of individual heat exchangers, as well as on the coil thickness [32]. The materials used for the individual components of HRSG are shown in Table 9.

Table 9. Materials used to estimate the purchase cost of main HRSG’s heat transfer surfaces

<table>
<thead>
<tr>
<th>HRSG component</th>
<th>Material according to EN 10216</th>
<th>Material according to DIN</th>
</tr>
</thead>
<tbody>
<tr>
<td>Economizer</td>
<td>P235GH</td>
<td>St 35.8 I/III</td>
</tr>
<tr>
<td>Evaporator</td>
<td>16Mo3</td>
<td>15 Mo 3</td>
</tr>
<tr>
<td>Superheater</td>
<td>13CrMo4-5</td>
<td>13 CrMo 44</td>
</tr>
</tbody>
</table>

The steam turbine purchase cost is of significant importance [30,31]:

\[
C_{ST} = 1.36 \left( 3197280 A_{ST}^{0.261} + 823.7 N_{elST}^{1.543} \right) \quad (12)
\]

Cost of generators [30,31]:

\[
C_G = 1.36 \cdot 3082 \cdot \left( \frac{N_{elGT}}{\eta_{ST}} + \frac{N_{elST}}{\eta_{ST}} \right)^{0.58} \quad (13)
\]

Cost of the condenser: [30,31]:

\[
C_{CON} = 1.36 \cdot 275.4 \cdot A_{CON}^{1.01} \quad (14)
\]

where:

- \( A_{ST} \) – exhaust annulus area, \( m^2 \)
- \( \eta_g \) – generator efficiency, \( \% \)

The costs related to the gas-steam installation including the cost of the heat recovery steam generator, the steam turbine, the generator and the condenser are listed in Table 10. It should be noted that the turbine price was estimated for internal efficiency \( \eta_{ST} = 90\% \).

Table 10. Cost of purchase of the steam installation major machines and equipment

<table>
<thead>
<tr>
<th>GT komponent</th>
<th>Price, USD</th>
</tr>
</thead>
<tbody>
<tr>
<td>HRSG</td>
<td>4,000,000</td>
</tr>
<tr>
<td>Steam turbine</td>
<td>6,760,000</td>
</tr>
<tr>
<td>Generator*</td>
<td>1,800,000</td>
</tr>
<tr>
<td>Condenser</td>
<td>1,990,000</td>
</tr>
</tbody>
</table>

* net power output – approx. 34.2 MW

In the case of CCPP with single pressure HRSG the purchase cost of the entire system including other facilities is according to [34] approx. $31,600,000 (includes the price of gas turbine).

4.4. Comparison of the costs of purchase of individual installations

The costs of the three installations under analysis are listed in Table 11. In the case of the gas-air system, the isentropic efficiencies of turbomachines in the air part are assumed at the level of
\[ \eta_c = \eta_{\text{GE}} = 87.5\%. \] It should be noted that the presented costs do not include the purchase cost of the gas turbine.

**Table 11. Cost of purchase of major machines and equipment of individual installations**

<table>
<thead>
<tr>
<th></th>
<th>Simple gas-air system</th>
<th>Complex gas-air system</th>
<th>Gas-steam system</th>
</tr>
</thead>
<tbody>
<tr>
<td>Purchase cost, thousand USD</td>
<td>approx. 6,740</td>
<td>approx. 6,960</td>
<td>approx. 14,550</td>
</tr>
</tbody>
</table>

Based on the electric power which can be obtained from the systems under analysis, it is possible to determine unit investment expenditures for individual installations. The results are presented in Table 12. In order to estimate unit investment expenditures, the power capacities of both the cycles under consideration and of the gas turbine system are totalled.

**Table 12. Unit investment expenditures of the power systems under consideration**

<table>
<thead>
<tr>
<th></th>
<th>Simple gas-air system</th>
<th>Complex gas-air system</th>
<th>Gas-steam system</th>
</tr>
</thead>
<tbody>
<tr>
<td>Unit investment expenditure,  USD/kW</td>
<td>approx. 660</td>
<td>approx. 690</td>
<td>approx. 930</td>
</tr>
</tbody>
</table>

**Conclusions**

Gas-air systems involve smaller investment expenditures than gas-steam cycles. However, they feature a lower efficiency. Both system types include expensive heat exchangers which recover heat energy from flue gases. In either case, the gas turbine is the most expensive component. Obtaining information on the nature of the system operation is an essential element in economic calculations. Is the task of the system to satisfy peak demand for electricity generation, or is the system intended for continuous operation. Is it to power working machines, such as the gas compressor in gas pumping stations etc., or is it to provide the power feed for the electric generator. The detailed data concerning investment expenditures are obtained only after the project has been completed. Studies are the basis for making investment decisions. The price of a gas-air system is predominantly determined by the amount of steel needed to make the AHX. The purchase price of turbomachinery depends mainly on isentropic efficiency. However, the costs related to the compressor and the air expander are not high due to moderate values of the working medium pressure. In the case of the gas-steam system, the principal cost-generating facilities are the heat recovery steam generator and the steam turbine. The total of these components is in fact equal to the price of the gas turbine. Like in the case of the gas-air system, the cost is determined by the amount of the material used for individual areas of heat exchange in the boiler and by the internal efficiency of the turbine. It should also be borne in mind that the gas-steam system is composed of more machinery and equipment, which undoubtedly affects the purchase price of individual installations. As for the heat exchangers, in both installations their price can be lowered at the expense of their effectiveness. Consequently, the purchase costs will be lower but so will be lower the electricity generation efficiency.

The current-generation combined cycles are usually gas-steam cycles. However, the heat contained in the flue gases after the gas turbine can be used differently. If another Brayton cycle with air as the working medium is added after the gas turbine, a combined gas-air system is created whose main advantage is a simple structure of the cycle. The point is that there are no facilities such as the condenser and there is no need to top up the water in the cycle. In the light of the thermodynamic analysis, gas-steam systems appear to be more efficient than gas-air ones. The most efficient
installation of a simple system composed of the ABB GT10 gas turbine and an air part reached 46.51% (Table 2). Using an additional facility such as an intercooler, the efficiency of electricity generation was raised to 47.3% (Table 4). The simple gas-steam system reached a 48.4% efficiency, despite a high exhaust loss. The information on the nature of the future installation operation will be one of the main assumptions for a broad economic analysis. Considering the fact that they are less complex technologically, gas-air systems definitely have an advantage. It should be expected that the assembly of a gas-air system will take less time and, consequently, that the costs related to the commissioning of the new installation will be lower. If a system is placed where there is no access to water (gas pumping stations, for example), gas-air systems have a considerable advantage.

In this paper gas-steam and gas-air cycles are discussed and compared to each other thermodynamically and economically. The efficiency of individual systems and the purchase cost of major machines and equipment are determined.

**Acknowledgments**

The results presented in this paper were obtained from research work co-financed by the National Centre of Research and Development in the framework of Contract SP/E/1/67484/10 – „Strategic Research Programme – Advanced Technologies for obtaining energy: Development of a technology for highly efficient zero-emission coal-fired Power units integrated with CO$_2$ capture”.

**Nomenclature**

- $a$ - Thermal diffusivity, m$^2$/s
- $A$ - Exhaust annulus area, m$^2$
- $ABC$ - Air Bottoming Cycle
- $AHX$ - Air Heat Exchanger
- $AT$ - Air Turbine
- $C$ - cost, USD or compressor
- $CCPP$ - Combined Cycle Power Plant
- $CMB$ - combustion chamber
- $CND$ - condenser
- $c_p$ - specific heat, J/(kg K)
- $CP$ - condensate pump
- $DEA$ - deareator
- $ECO$ - Economizer
- $EVAP$ - Evapurator
- $FWP$ - feed water pump
- $G$ - Generator
- $GT$ - gas turbine
- $HRS$ - Heat Recovery Steam Generator
- $HX$ - heat exchanger
- $IC$ - intercooler
- $LHV$ - low heat value
- $\dot{m}$ - mass flow rate, kg/s
- $N$ - power, MW
- $Nu$ - Nusselt number, -
\( p, P \) - pressure, MPa
\( Pr \) - Prandtl number, -
\( R \) - gas constant, J/(kg K)
\( Re \) - Reynolds number, -
\( SH \) - superheater
\( ST \) - steam turbine
\( T, t \) - temperature, K, °C
\( \dot{Q} \) - heat, W

**Greek symbols**
\( \alpha \) - heat transfer coefficient, W/(m\(^2\)K)
\( \beta \) - pressure ratio
\( \Delta \) - difference, -
\( \varepsilon \) - effectiveness, %
\( \eta \) - efficiency, %, dynamic viscosity, Pa·s
\( \lambda \) - thermal conductivity, W/mK
\( \rho \) - density, kg/m\(^3\)
\( \nu \) - kinematic viscosity, m\(^2\)/s
\( \xi \) - relative pressure drop
\( \zeta \) - temperature ratio, -

**Subscripts and superscripts**
a - Air
amb - Ambient
ABC - Air Bottoming Cycle
AT - Air turbine
c - Cycle
cpt - Cold pinch temperature
C - Compressor
e - Energetic
el - Electric
f - fuel
fg - Flue gas
G - Generator
GT - Gas turbine
i - Internal, isentropic
in - In
ls - Live steam
m - Mechanical
out - Out
pp - Pinch point
sc - Subcooling
ST - Steam turbine
t - temperature, K, °C
T - Turbine

References
[26] Shankar R. Heat transfer to or from a fluid flowing through a tube.
[32] Own information obtained from industry 2011 (Information related to the cost factors).
Application of an Alternative Thermoeconomic Approach to a Two-Stage VaporCompression Refrigeration Cycle with Intercooling

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Abstract:
In any energy system, in the same manner that there are productive components, there also exist dissipative components. Although there has been an advance in the development of thermoeconomic criteria for dissipative components allocation, this problem is still open. The introduction of the negentropy in thermoeconomies represented a great advance, since this magnitude allows quantifying the condenser product. However, negentropy itself does not allow defining the product of values. To overcome such a limitation, in this paper an alternative thermoeconomic approach is applied to a two-stage vapor compression refrigeration cycle with intercooling. The approach is based on disaggregation of the physical exergy into three terms, namely internal energy ($U-\bar{U}$), flow work ($pV-p_0V_0$) and the here called entropic term ($T_0S-TS_0$). The so-called UFS Model allows the valves isolation in the productive structure. The goal is to obtain the exergetic unit cost of each internal flow, considering an ideal cycle and an actual cycle, as application examples. Such an alternative approach yields consistent results, because exergetic unit costs of internal flows are greater than or equal to one and product-fuel ratio of each productive unit are less than or equal to one, depending on the situation, i.e., when considering the actual cycle or the ideal cycle, respectively. Besides the allocation of the external power of the plant, analyses on irreversibilities of each cycle unit are carried out by the exergy balance and by the difference between the defined fuels (resources) and products, according to the systematic of this alternative approach. Thus, it is shown that the UFS Model can be used in order to quantify irreversibilities as well as the conventional exergy analysis. Results show that the product-fuel ratio of each component of the proposed structure varies from zero (for totally irreversible processes) to one (for totally reversible processes).

Keywords: Exergy Disaggregation, Refrigeration Cycle, Thermoeconomics, Valves Isolation.

1. Introduction

Thermoeconomics can be considered a new science which, by connecting Thermodynamics and Economics, provides tools to solve problems in complex energy systems that can hardly or not be solved using conventional energy analysis techniques based on First Law of Thermodynamics (mass and energy balance), as for instance a rational price assessment to the products of a plant based on physical criteria [1].

In any energy system, in the same manner that there are productive components, there also exist dissipative components. An energy system has a defined productive structure, but also dissipative ones and both structures are not independent. Although there has been an advance in the development of criteria for the cost allocation of residues, this problem is still open [2].

One of the thermoecmomic methodologies challenges is to define the productive structure of thermal systems that allows allocating rationally the cost of dissipative components to final products. The way in which we define the productive structure is a key point of the thermoeconomic modeling [3]. Different thermoeconomic methodologies can provide different cost values when they define different productive structures. Cost validation is a key issue in thermoecnomics which has not been properly solved yet. However, one considers that cost
validation can be designed using the physical behavior of the plant together with Thermodynamics, because irreversibility is the physical magnitude generating the cost [4].

According to [5], depending on the type of analysis, different levels of accuracy of the results are required. Sometimes, under a thermoeconomic analysis point of view, it is necessary to consider a component as a group of subsystem (made up of a group of subsystem) or a mass or an energy flow rate consisting of several components, for example thermal, mechanical or chemical exergy, as proposed by [6], or even to include fictitious flow stream (negentropy) without a physical existence in the flow sheet of the plant, as proposed by [7].

The negentropy flow was applied in thermoeconomics joined up with exergy flow [7]. Negentropy was defined as the negative variation of entropy multiplied by the temperature of the environment. This application represented a great advance in the discipline, since it allowed one to quantify the condenser product, which was not possible before because the condenser is a dissipative component, whose product cannot be expressed in terms of exergy. The concept of negentropy was also used in order to define the productive structure of a gas turbine cogeneration system [3,8].

However, when negentropy is applied as a fictitious flow (joined up with exergy flow), it is not possible to obtain an efficiency based on the Second Law of Thermodynamics (product-resource ratio), since the product of dissipative units might be higher than its resource, yielding unit costs lesser than one for some flows [9]. This happens because exergy loss is considered as fuel and negentropy (in this work also called syntropy) as product. To overcome this problem, the H&S Model was developed [9] to allocate wastes of dissipative component (the condenser) in the thermoeconomic analysis of energy systems. The basis of this method is the breakup of exergy into enthalpy and negentropy. Enthalpy flows replace exergy flows and negentropy flows are used as a component of exergy. Therefore, the productive structure is defined using enthalpy and negentropy flows.

The disaggregation of exergy in thermal and mechanical components does not allow defining the product of the condenser. On the other hand, this goal is achieved when negentropy is used both as a fictitious flow [7] and as an exergy component flow [9]. But none of them allows defining the product of valves (for instance, in a refrigeration cycle). Thus, this paper presents an alternative thermoeconomic approach, which is so-called UFS Model, based on disaggregation of physical exergy into internal energy \((U-U_0)\), flow work \((pV-p_0V_0)\) and the here called syntropy \((T_0S-TS_0)\). This disaggregation of the physical exergy allows the isolation of valves in the productive structure, so defining their resources and products, without generating any problems [10].

### 2. Physical Model

A two-stage vapor compression refrigeration cycle with intercooling is used, in this paper, to illustrate the application of the UFS Model. The intercooling of the working fluid is achieved by using both a flash tank and a mixing tank. By using this cycle, this paper shows the capacity of the UFS Model to treat an important point in discussion related to the thermoeconomic methodologies: the dissipative components, as condensers and valves. Figure 1 represents the physical structure of the refrigeration cycle, which is defined as having eight units: the high-pressure compressor \((cmp.h)\), the low-pressure compressor \((cmp.l)\), the high-pressure expansion valve \((vlv.h)\), the low-pressure expansion valve \((vlv.l)\), the evaporator \((evp)\), the condenser \((cnd)\), the flash tank \((flt)\) and the mixing tank \((mxt)\).

Assumptions made for the modeling are:

- Equipments are analyzed as control volumes at steady state and are adiabatic;
- The exergy provided by the evaporator to the cold region is modeled as an exergy flow associated with the heat transfer;
- There is no pressure drop for flow through heat exchangers;
Kinetic and potential energy effects are negligible.

3. Exergy and Thermoeconomic Modelling

In order to carry out a thermoeconomic analysis, the UFS Model defines the productive purpose of the subsystems (resources and products), as well as the distribution of the external resources and internal products throughout the system. The productive structure could be represented by means of a functional diagram. In this section, basic concepts of the cost formation are shown. After that, the UFS Model is applied to generate the productive structure of the studied refrigeration cycle. Finally, the exergy balances are done for each equipment of such a cycle.

3.1. Productive Structure and Costs Formation

The productive structure represents the cost formation process of the cycle. The external resources consumed by the cycle are the mechanical power demanded by the high-pressure compressor \((W_{cmp.h})\) and the low-pressure compressor \((W_{cmp.l})\). The functional product is the exergy, which is associated to the heat transfer, provided by the evaporator \((B_{evp|Q})\). Rectangles are real units that represent the actual equipment of the cycle. Rhombus and circles are fictitious units called junctions and bifurcations, respectively. Each productive unit has inlet and outlet arrows that represent its resources and products, respectively. Each productive flow is defined based on physical flows.

The mathematical model for the mechanical power allocation is obtained by formulating the cost balance equation in each productive unit, or subsystem, of the productive structure, as shown in (1), where \(k\) is the exergetic unit cost of productive flows (unknown variable) and \(Y\) represents the generic productive flow, which can be internal or external. The exergetic unit cost of the mechanical power is equal to one.

\[
\sum (k \cdot Y) = 0. 
\]  

Efficiency is defined as the ratio of the desired result for an event to the input required to accomplish such an event. Therefore, when one defines the fuel and the product during the thermoeconomic modeling, one takes into account that the Second Law efficiency ranges from zero, for a totally irreversible process, to 100 percent, for a totally reversible process [11]. Thus, the exergetic unit cost is the inverse of an efficiency, which is defined as the ratio between the products of a productive unit and the external fuels of the plant.

Fig. 1. Physical Structure of the Refrigeration Cycle.
Since the number of flows is always greater than the number of productive units, some auxiliary equations attribute the same exergetic unit cost to all of productive flows leaving the same bifurcation. The solution of the set of cost balance equations allows the attainment of the exergetic unit cost of each internal flow and final product.

### 3.2. The UFS Model

The physical exergy \((B_i)\) of a refrigerant stream is written as shown in (2), where \(h_0\) and \(s_0\) are the specific enthalpy and the specific entropy, both at the dead state, respectively. Dead state is at \(T_0\), the reference temperature, and at \(p_0\), the reference pressure.

\[
B_i = m_i \cdot \left[ (h_i - h_0) - T_0 \cdot (s_i - s_0) \right].
\]  

To generate the productive structure, the so-called UFS Model considers that the physical exergy must be disaggregated into three components, which are the internal energy \((U_i)\), the flow work \((F_i)\), and the syntropy \((S_i)\). The productive structure is shown in Fig. 2.

The products and the fuels of each subsystem, in terms of internal energy, flow work and chemical exergy component, are defined based on the quantity of these magnitudes added to and removed from the working fluid, respectively. On the other hand, the entropic component flows are the products of the subsystems that decrease the working fluid entropy, and subsystems that increase the working fluid entropy are entropic components consumers.

The physical flows are calculated as shown in (3-5), where \(u_0\) and \(v_0\) are the specific internal energy and the specific volume, both at the dead state, respectively. One must remind that the enthalpy is defined as the sum between the internal energy and the flow work.

\[
U_i = m_i \cdot (u_i - u_0), \quad (3)
\]
\[
F_i = m_i \cdot (p_i v_i - p_0 v_0), \quad (4)
\]
\[
S_i = m_i \cdot T_0 \cdot (s_i - s_0). \quad (5)
\]

Chemical exergy is not considered because there is no changing in the composition of the fluid throughout the processes of the cycle.

The disaggregation of the enthalpy is done because when one evaluates an adiabatic valve at steady state, the enthalpy variation between its upstream and downstream flows is equal to zero, but the increasing of its flow work is equal to the decreasing of its internal energy. In light of this, it is now possible to define a product for a valve.

Internal flows of the productive structure are calculated using (6-11).

\[
U_{i,j} = m_i \cdot (u_i - u_j), \quad (6)
\]
\[
F_{i,j} = m_i \cdot (p_i v_i - p_j v_j), \quad (7)
\]
\[
S_{i,j} = m_i \cdot T_0 \cdot (s_i - s_j), \quad (8)
\]
\[
U_{i,j} = m_j \cdot (u_i - u_j), \quad (9)
\]
\[
F_{i,j} = m_j \cdot (p_i v_i - p_j v_j), \quad (10)
\]
\[ S_{i,j} = m_j \cdot T_0 \cdot (s_i - s_j). \]  

(11)

The set of cost balance equations is given by (12-22).

\[
k_{\text{emp}, i} \cdot (U_{23} + F_{23}) - k_S \cdot S_{23} = W_{\text{emp}, i},
\]

(12)

\[
k_{\text{emp}, h} \cdot (U_{43} + F_{43}) - k_S \cdot S_{43} = W_{\text{emp}, h},
\]

(13)

\[
k_{\text{vl}, i} \cdot F_{87} - (k_{U_i} \cdot U_{78} + k_S \cdot S_{87}) = 0,
\]

(14)

\[
k_{\text{vl}, h} \cdot F_{65} - (k_{U_i} \cdot U_{56} + k_S \cdot S_{65}) = 0,
\]

(15)

\[
k_{\text{vnd}, i} \cdot S_{45} - (k_{U_i} \cdot U_{45} + k_F \cdot F_{45}) = 0,
\]

(16)

\[
k_{\text{ep}} \cdot \left( B_{\text{ep}, Q} + U_{18} + F_{18} \right) - k_S \cdot S_{18} = 0,
\]

(17)

\[
k_{\text{fl}, i} \cdot (U_{96} + F_{96} + S_{67}) - (k_{U_i} \cdot U_{67} + k_F \cdot F_{67} + k_S \cdot S_{96}) = 0,
\]

(18)

\[
k_{\text{mas}, i} \cdot (U_{39} + F_{39} + S_{23}) - (k_{U_i} \cdot U_{23} + k_F \cdot F_{23} + k_S \cdot S_{39}) = 0,
\]

(19)

\[
k_{\text{U}, i} \cdot (U_{78} + U_{56} + U_{45} + U_{67} + U_{23}) - (k_{\text{emp}, i} \cdot U_{23} + k_{\text{emp}, h} \cdot U_{43} + k_{\text{ep}} \cdot U_{18} + k_{\text{fl}, i} \cdot U_{96} + k_{\text{mas}, i} \cdot U_{39}) = 0,
\]

(20)

\[
k_{F, i} \cdot (F_{45} + F_{67} + F_{23}) - (k_{\text{emp}, i} \cdot F_{23} + k_{\text{emp}, h} \cdot F_{43} + k_{\text{vl}, i} \cdot F_{87} + k_{\text{vl}, h} \cdot F_{65} + k_{\text{ep}} \cdot F_{18} + k_{\text{fl}, i} \cdot F_{96} + k_{\text{mas}, i} \cdot F_{39}) = 0,
\]

(21)

\[
k_S \cdot (S_{39} + S_{43} + S_{87} + S_{65} + S_{18} + S_{96} + S_{39}) - (k_{\text{vnd}, i} \cdot S_{45} + k_{\text{fl}, i} \cdot S_{67} + k_{\text{mas}, i} \cdot S_{23}) = 0.
\]

(22)

In Fig. 2, orange lines represent \( U_{ij} \) and \( U_{ij} \), blue lines represent \( F_{ij} \) and \( F_{ij} \), red lines represent \( S_{ij} \) and \( S_{ij} \), gray lines represent mechanical power, black lines represent the resource and the product of productive units and the green line represents the product of the refrigeration cycle.

One should note that Fig. 2, i.e., the productive structure of the refrigeration cycle, is the graphical representation of (12-22).
Fig. 2. Productive Structure of the Refrigeration Cycle
Equations (23-33) are generated by taking the difference between resources and products for all units, both real and fictitious ones, of the productive structure.

\[(R - P)_{cmp,l} = (W_{cmp,l} + S_{21}) - (U_{21} + F_{21}),\]  
\[(R - P)_{cmp,h} = (W_{cmp,h} + S_{43}) - (U_{43} + F_{43}),\]  
\[(R - P)_{vl,l} = (U_{78} + S_{87}) - F_{87},\]  
\[(R - P)_{vl,h} = (U_{56} + S_{65}) - F_{65},\]  
\[(R - P)_{emd} = (U_{45} + F_{45}) - S_{45},\]  
\[(R - P)_{exp} = S_{18} - (B_{exp} + U_{18} + F_{18}),\]  
\[(R - P)_{flh} = (U_{67} + F_{67} + S_{96}) - (U_{96} + F_{96} + S_{67}),\]  
\[(R - P)_{med} = (U_{23} + F_{23} + S_{39}) - (U_{39} + F_{39} + S_{23}),\]  
\[(R - P)_{u} = (U_{21} + U_{43} + U_{18} + U_{96} + U_{96} + U_{39}) - (U_{78} + U_{56} + U_{45} + U_{67} + U_{23}),\]  
\[(R - P)_{f} = (F_{23} + F_{43} + F_{67} + F_{65} + F_{18} + F_{96} + F_{39}) - (F_{45} + F_{67} + F_{23}),\]  
\[(R - P)_{s} = (S_{45} + S_{67} + S_{23}) - (S_{21} + S_{43} + S_{67} + S_{65} + S_{18} + S_{96} + S_{39}).\]  

3.3. Exergy Balances
The exergy balance is done for each control volume in the physical structure. The sum between the exergy destruction \( (B_D) \) and the exergy loss \( (B_L) \) is evaluated for each equipment that lies in such a structure. Applying the exergy balance for control volumes at steady state, (34), it generates (35-42).

\[
\left(1 - \frac{T_0}{T}\right)Q - W - B_D = \sum_{ou} B_{ou} - \sum_{in} B_{in},
\]  
\[B_{D,cmp,l} = W_{cmp,l} + B_1 - B_2,\]  
\[B_{D,cmp,h} = W_{cmp,h} + B_3 - B_4,\]  
\[B_{D,vl,l} = B_7 - B_8,\]  
\[B_{D,vl,h} = B_5 - B_6,\]  
\[B_{D,emd} + B_{L,emd} = B_4 - B_5,\]  
\[B_{D,exp} = B_8 - B_1 - B_{exp},\]  
\[B_{D,flh} = B_6 - B_7 - B_9,\]  
\[B_{D,med} = B_9 + B_2 - B_3.\]
On the condenser, it is considered that the exergy loss is equal to the net exergy associated to the cooling water flow, i.e., the difference between the exit and inlet water flows exergies. Nevertheless, for purposes of this paper, it is not necessary to know the value of that exergy loss, because such a loss is internalized in the productive structure.

### 3.4. Applying Examples

In order to exemplify the applying of the UFS Model, such an approach is going to be used to evaluate a defined actual cycle and a defined non-actual cycle, according to the data given in the next paragraph. This is done to check the consistency of the proposed thermoeconomic approach.

Considering the actual cycle, let the isentropic efficiencies of both high-pressure compressor ($\eta_{C_{\text{high}}}$) and low-pressure compressor ($\eta_{C_{\text{low}}}$) be equal to 0.90, the temperature difference between the cold region that receives the exergy provided by the evaporator ($\Delta T_{\text{exp}}$) and the state 8 be equal to 5 K and the temperature difference between the state 5 of the physical structure and the environment ($\Delta T_{\text{end}}$) be equal to 5 K. Considering the non-actual cycle, the compression processes are isentropic ($\eta_{C_{\text{high}}}=\eta_{C_{\text{low}}}=1$) and there are no temperature differences ($\Delta T_{\text{end}}=\Delta T_{\text{exp}}=0$ K).

Streams parameters of the refrigeration cycle for both the actual and non-actual situations are presented in Table 1. Streams 2, 3 and 4 have different thermodynamic properties in each situation (subscripts $a$ and $n-a$, respectively). Values are based on [12]. The refrigerant is R-134a. Thermodynamic properties, in this case specific internal energy, specific volume and specific entropy, of the fluid are evaluated from the database of the software Engineering Equation Solver (EES).

Considering the actual cycle, the reference temperature is equal to 314.44 K. Considering the non-actual cycle, the reference temperature is equal to 319.44 K. The exergy provided by the evaporator is calculated by multiplying the exergetic temperature factor, considering the evaporation temperature, and the value of the heat transfer associated to such equipment.

<table>
<thead>
<tr>
<th>Physical Flow</th>
<th>$m$ [kg/s]</th>
<th>$p$ [kPa]</th>
<th>$T$ [K]</th>
<th>$u$ [kJ/kg]</th>
<th>$v$ [m³/kg]</th>
<th>$s$ [kJ/kg-K]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 Vapor ($x=1$)</td>
<td>2.091</td>
<td>84.43</td>
<td>243.15</td>
<td>213.11</td>
<td>0.22580</td>
<td>0.9558</td>
</tr>
<tr>
<td>$2_a$ Vapor</td>
<td>2.091</td>
<td>400.00</td>
<td>294.76</td>
<td>245.49</td>
<td>0.05464</td>
<td>0.9678</td>
</tr>
<tr>
<td>$2_{n-a}$ Vapor</td>
<td>2.091</td>
<td>400.00</td>
<td>290.15</td>
<td>242.38</td>
<td>0.05363</td>
<td>0.9558</td>
</tr>
<tr>
<td>$3_a$ Vapor</td>
<td>2.907</td>
<td>400.00</td>
<td>291.17</td>
<td>242.56</td>
<td>0.05369</td>
<td>0.9565</td>
</tr>
<tr>
<td>$3_{n-a}$ Vapor</td>
<td>2.907</td>
<td>1200.00</td>
<td>334.08</td>
<td>268.47</td>
<td>0.01851</td>
<td>0.9645</td>
</tr>
<tr>
<td>$4_a$ Vapor</td>
<td>2.907</td>
<td>1200.00</td>
<td>329.13</td>
<td>263.61</td>
<td>0.01794</td>
<td>0.9478</td>
</tr>
<tr>
<td>$4_{n-a}$ Vapor</td>
<td>2.907</td>
<td>1200.00</td>
<td>319.44</td>
<td>116.70</td>
<td>0.00089</td>
<td>0.4244</td>
</tr>
<tr>
<td>5 Liquid ($x=0$)</td>
<td>2.091</td>
<td>400.00</td>
<td>282.06</td>
<td>111.79</td>
<td>0.01495</td>
<td>0.4385</td>
</tr>
<tr>
<td>6 Mixture ($x=0.2809$)</td>
<td>2.907</td>
<td>400.00</td>
<td>282.06</td>
<td>111.79</td>
<td>0.01495</td>
<td>0.4385</td>
</tr>
<tr>
<td>7 Liquid ($x=0$)</td>
<td>2.091</td>
<td>400.00</td>
<td>282.06</td>
<td>111.79</td>
<td>0.01495</td>
<td>0.4385</td>
</tr>
<tr>
<td>8 Mixture ($x=0.2336$)</td>
<td>2.091</td>
<td>40.00</td>
<td>282.06</td>
<td>59.44</td>
<td>0.05331</td>
<td>0.2639</td>
</tr>
<tr>
<td>9 Vapor ($x=1$)</td>
<td>0.817</td>
<td>400.00</td>
<td>282.06</td>
<td>235.07</td>
<td>0.05120</td>
<td>0.9269</td>
</tr>
</tbody>
</table>

### 4. Results and Discussion

Table 2 shows the exergy balance sheet for the given numerical data. Percentages are based on total exergy, which is defined by the sum between the mechanical power inlets. Such a value is also equal to the sum between the exergy provided by the refrigeration cycle and the exergy destructions and losses of all equipments.
Table 2. Exergy Balance Sheet

<table>
<thead>
<tr>
<th>Description</th>
<th>Actual Cycle</th>
<th></th>
<th>Non-Actual Cycle</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Value [kW]</td>
<td>Percentage</td>
<td>Value [kW]</td>
<td>Percentage</td>
</tr>
<tr>
<td>Total Exergy</td>
<td>151.01</td>
<td>100%</td>
<td>134.88</td>
<td>100%</td>
</tr>
<tr>
<td>Power in:</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>HP Compressor</td>
<td>77.47</td>
<td>51.30%</td>
<td>68.69</td>
<td>50.93%</td>
</tr>
<tr>
<td>LP Compressor</td>
<td>73.54</td>
<td>48.70%</td>
<td>66.19</td>
<td>49.07%</td>
</tr>
<tr>
<td>Product Exergy:</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Evaporator</td>
<td>93.96</td>
<td>62.22%</td>
<td>110.35</td>
<td>81.82%</td>
</tr>
<tr>
<td>Exergy Destructions and Losses:</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>LP Compressor</td>
<td>7.90</td>
<td>5.23%</td>
<td>0.00</td>
<td>0.00%</td>
</tr>
<tr>
<td>HP Compressor</td>
<td>7.32</td>
<td>4.85%</td>
<td>0.00</td>
<td>0.00%</td>
</tr>
<tr>
<td>LP Valve</td>
<td>10.74</td>
<td>7.11%</td>
<td>10.91</td>
<td>8.09%</td>
</tr>
<tr>
<td>HP Valve</td>
<td>12.84</td>
<td>8.50%</td>
<td>13.05</td>
<td>9.67%</td>
</tr>
<tr>
<td>Condenser</td>
<td>8.95</td>
<td>5.93%</td>
<td>0.51</td>
<td>0.38%</td>
</tr>
<tr>
<td>Evaporator</td>
<td>9.14</td>
<td>6.05%</td>
<td>0.00</td>
<td>0.00%</td>
</tr>
<tr>
<td>Flash Tank</td>
<td>0.00</td>
<td>0.00%</td>
<td>0.00</td>
<td>0.00%</td>
</tr>
<tr>
<td>Mixing Tank</td>
<td>0.17</td>
<td>0.11%</td>
<td>0.08</td>
<td>0.06%</td>
</tr>
</tbody>
</table>

The exergetic efficiency of the refrigeration cycle, (43), is equal to the ratio between the product exergy of the cycle and its mechanical power inputs. As seen in Table 2, such a value is equal to 62.22% for the actual cycle and equal to 81.82% for the non-actual cycle.

\[ \varepsilon = \frac{B_{cyl Q}}{W_{cyl H} + W_{cyl L}}. \]  

(43)

Table 3 shows productive flows, its values and its respective exergetic unit costs for the given numerical data. One should note that there is no exergetic unit cost value less than one.

Taking the inverse of the exergetic unit cost of the product exergy, (44), one obtains the same value of the exergetic efficiency of the refrigeration cycle, i.e., 62.22% for the actual cycle and 81.82% for the non-actual cycle.

\[ \varepsilon = \frac{1}{k_{exp}}. \]  

(44)

Table 4 shows the values of the difference between resources and products of both real and fictitious units of the productive structure for the given numerical data.

For the real productive units, the differences between resources and products (Table 4) are equal to the sums of both exergy destruction and loss of the same units of the physical structure (Table 2). Actually, one can do some algebra from (23-30) and (6-11) and obtain (35-42). For the fictitious productive units, the differences between resources and products (Table 4 and 31-33) are equal to zero. Therefore, exergetic costs are fairly distributed among the productive structure.

Considering the non-actual cycle and its values from Table 4, the difference of the condenser is not equal to zero because the desuperheating and the difference of the mixing tank is not equal to zero because its inlet streams are at different thermodynamic states.
### Table 3. Exergetic Unit Costs

<table>
<thead>
<tr>
<th>Productive Flow</th>
<th>Value [kW]</th>
<th>Exergetic Unit Cost [kW/kW]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Actual Cycle</td>
<td>Non-Actual Cycle</td>
</tr>
<tr>
<td>U (_{1:8})</td>
<td>321.25</td>
<td>321.25</td>
</tr>
<tr>
<td>U (_{2:1})</td>
<td>67.71</td>
<td>61.19</td>
</tr>
<tr>
<td>U (_{2:3})</td>
<td>6.13</td>
<td>4.30</td>
</tr>
<tr>
<td>U (_{3:9'})</td>
<td>6.11</td>
<td>4.29</td>
</tr>
<tr>
<td>U (_{4:3})</td>
<td>75.33</td>
<td>67.70</td>
</tr>
<tr>
<td>U (_{4:5})</td>
<td>441.25</td>
<td>427.12</td>
</tr>
<tr>
<td>U (_{5:6})</td>
<td>14.27</td>
<td>14.27</td>
</tr>
<tr>
<td>U (_{6:7'})</td>
<td>100.70</td>
<td>100.70</td>
</tr>
<tr>
<td>U (_{7:8})</td>
<td>8.75</td>
<td>8.75</td>
</tr>
<tr>
<td>U (_{9:6})</td>
<td>100.70</td>
<td>100.70</td>
</tr>
<tr>
<td>F (_{1:8})</td>
<td>30.45</td>
<td>30.45</td>
</tr>
<tr>
<td>F (_{2:1})</td>
<td>5.83</td>
<td>4.99</td>
</tr>
<tr>
<td>F (_{2:3})</td>
<td>0.79</td>
<td>0.56</td>
</tr>
<tr>
<td>F (_{3:9'})</td>
<td>0.81</td>
<td>0.57</td>
</tr>
<tr>
<td>F (_{4:3})</td>
<td>2.14</td>
<td>0.99</td>
</tr>
<tr>
<td>F (_{4:5})</td>
<td>61.46</td>
<td>59.46</td>
</tr>
<tr>
<td>F (_{6:5})</td>
<td>14.27</td>
<td>14.27</td>
</tr>
<tr>
<td>F (_{6:7'})</td>
<td>11.84</td>
<td>11.84</td>
</tr>
<tr>
<td>F (_{8:7})</td>
<td>8.75</td>
<td>8.75</td>
</tr>
<tr>
<td>F (_{9:6})</td>
<td>11.84</td>
<td>11.84</td>
</tr>
<tr>
<td>S (_{1:8})</td>
<td>454.80</td>
<td>462.03</td>
</tr>
<tr>
<td>S (_{2:1})</td>
<td>7.90</td>
<td>-----</td>
</tr>
<tr>
<td>S (_{2:3})</td>
<td>7.43</td>
<td>5.36</td>
</tr>
<tr>
<td>S (_{3:9'})</td>
<td>7.60</td>
<td>5.44</td>
</tr>
<tr>
<td>S (_{4:3})</td>
<td>7.32</td>
<td>-----</td>
</tr>
<tr>
<td>S (_{4:5})</td>
<td>493.75</td>
<td>486.07</td>
</tr>
<tr>
<td>S (_{6:5})</td>
<td>12.84</td>
<td>13.05</td>
</tr>
<tr>
<td>S (_{6:7'})</td>
<td>125.45</td>
<td>127.45</td>
</tr>
<tr>
<td>S (_{8:7})</td>
<td>10.74</td>
<td>10.91</td>
</tr>
<tr>
<td>S (_{9:6})</td>
<td>125.45</td>
<td>127.45</td>
</tr>
<tr>
<td>B (_{evp</td>
<td>Q})</td>
<td>93.96</td>
</tr>
<tr>
<td>W (_{cmp.l})</td>
<td>73.54</td>
<td>66.19</td>
</tr>
<tr>
<td>W (_{cmp.h})</td>
<td>77.47</td>
<td>68.69</td>
</tr>
</tbody>
</table>

### Table 4. Differences between Resources and Products of Productive Units

<table>
<thead>
<tr>
<th>Productive Unit</th>
<th>Difference Value [kW]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Actual Cycle</td>
</tr>
<tr>
<td>LP Compressor (cmp.l)</td>
<td>7.90</td>
</tr>
<tr>
<td>HP Compressor (cmp.h)</td>
<td>7.32</td>
</tr>
<tr>
<td>LP Valve (vlv.l)</td>
<td>10.74</td>
</tr>
<tr>
<td>HP Valve (vlv.h)</td>
<td>12.84</td>
</tr>
<tr>
<td>Condenser (cnd)</td>
<td>8.95</td>
</tr>
<tr>
<td>Evaporator (evp)</td>
<td>9.14</td>
</tr>
<tr>
<td>Flash Tank (flt)</td>
<td>0.00</td>
</tr>
<tr>
<td>Mixing Tank (mxt)</td>
<td>0.17</td>
</tr>
<tr>
<td>Internal Energy (U)</td>
<td>0.00</td>
</tr>
<tr>
<td>Flow Work (F)</td>
<td>0.00</td>
</tr>
<tr>
<td>Syntropy (S)</td>
<td>0.00</td>
</tr>
</tbody>
</table>
Taking the ratios between products and resources of the productive units, such ratios are defined based on the Second Law of Thermodynamics, the set (45-55) is generated.

\[
\eta_{\text{cmp}.l} = \frac{U_{21} + F_{21}}{W_{\text{cmp}.l} + S_{21}}, \quad (45)
\]

\[
\eta_{\text{cmp}.h} = \frac{U_{43} + F_{43}}{W_{\text{cmp}.h} + S_{43}}, \quad (46)
\]

\[
\eta_{\text{vlv}.l} = \frac{F_{87}}{U_{78} + S_{87}}, \quad (47)
\]

\[
\eta_{\text{vlv}.h} = \frac{F_{65}}{U_{56} + S_{65}}, \quad (48)
\]

\[
\eta_{\text{cnd}} = \frac{S_{45}}{U_{45} + F_{45}}, \quad (49)
\]

\[
\eta_{\text{evp}} = \frac{B_{\text{evp}} Q + U_{18} + F_{18}}{S_{18}}, \quad (50)
\]

\[
\eta_{\text{flt}} = \frac{U_{96} + F_{96} + S_{67}}{U_{67} + F_{67} + S_{96}}, \quad (51)
\]

\[
\eta_{\text{mxt}} = \frac{U_{39} + F_{39} + S_{23}}{U_{23} + F_{23} + S_{39}}, \quad (52)
\]

\[
\eta_{U} = \frac{U_{78} + U_{56} + U_{45} + U_{67} + U_{23}}{U_{21} + U_{43} + U_{18} + U_{96} + U_{39}}, \quad (53)
\]

\[
\eta_{F} = \frac{F_{45} + F_{67} + F_{23}}{F_{21} + F_{43} + F_{87} + F_{65} + F_{18} + F_{96} + F_{39}}, \quad (54)
\]

\[
\eta_{S} = \frac{S_{21} + S_{43} + S_{87} + S_{65} + S_{18} + S_{96} + S_{39}}{S_{45} + S_{67} + S_{23}}, \quad (55)
\]

Table 5 shows the values of the product-resource ratios of the productive units for the given numerical data. One should note that there is no product-resource ratio value greater than 100%.

**Table 5. Product-Resource Ratios of Productive Units**

<table>
<thead>
<tr>
<th>Productive Unit</th>
<th>Ratio Value</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Actual Cycle</td>
</tr>
<tr>
<td>LP Compressor (cmp.l)</td>
<td>90.30%</td>
</tr>
<tr>
<td>HP Compressor (cmp.h)</td>
<td>91.37%</td>
</tr>
<tr>
<td>LP Valve (vlv.l)</td>
<td>44.90%</td>
</tr>
<tr>
<td>HP Valve (vlv.h)</td>
<td>52.64%</td>
</tr>
<tr>
<td>Condenser (cnd)</td>
<td>98.22%</td>
</tr>
<tr>
<td>Evaporator (evp)</td>
<td>97.99%</td>
</tr>
<tr>
<td>Flash Tank (flt)</td>
<td>100.00%</td>
</tr>
<tr>
<td>Mixing Tank (mxt)</td>
<td>98.86%</td>
</tr>
<tr>
<td>Internal Energy (U)</td>
<td>100.00%</td>
</tr>
<tr>
<td>Flow Work (F)</td>
<td>100.00%</td>
</tr>
<tr>
<td>Syntropy (S)</td>
<td>100.00%</td>
</tr>
</tbody>
</table>
The UFS Model was proposed in order to define a product for valves and, in general, for units or processes that are modelled as isenthalpic, because the H&S Model is not able to do it. Nevertheless, the UFS Model keeps all features of the H&S Model, i.e., one can say that the UFS Model is an extension of the H&S Model and the application of the former only could be justified, e.g., whether there is a valve in the structure, because of the increasing of the modeling complexity.

5. Closing Remarks

In this paper, an exergy and a thermoeconomic analyses were carried out to a two-stage vapor compression refrigeration cycle with intercooling. The thermoeconomic approach used was the UFS Model, which is based on the disaggregation of physical exergy into internal energy ($U_i$), flow work ($F_i$) and syntropy ($S_i$).

Results of both exergy and thermoeconomic evaluations were compared. It was seen that the sum between the exergy destructions and losses of each physical structure equipment were equal to the difference between the fuels and the products of each real equipment of the productive structure. Besides that, the exergetic efficiency of the refrigeration cycle evaluated by the product-fuel ratio was equal to the one evaluated by the inverse of exergetic unit cost value of the productive unit associated to the product of the cycle. Thus, one can conclude that this thermoeconomic modeling can be used as well as the exergy balance in order to quantify both internal and external irreversibilities of each equipment that lies in the cycle. One can conclude as well that the exergetic costs were fairly allocated throughout the cycle. It was seen as well that exergetic unit costs of internal flows and products are greater than or equal to one and product-resource ratio of each productive unit are less than or equal to one, depending on the situation, i.e., when considering the actual cycle or the non-actual cycle, respectively. Thus, one can say that the UFS Model is in accordance with the Second Law of Thermodynamics. Results also show that the product-resource ratio of each component of the productive structure varies from zero (for totally irreversible processes) to one (for totally reversible processes), whether one considers the trend of such ratios.

Acknowledgments

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References


Comparative Performance of Advanced Power Cycles for Low-Temperature Heat Sources

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Abstract:
In this study the performance benefit of numerous advanced power cycles including supercritical, dual-pressure and several cycles using an ammonia/water mixture is assessed against a baseline of a state-of-the-art single pressure organic Rankine cycle with heat source temperatures ranging from 80°C to 200°C. Different sets of boundary conditions (heat exchanger minimum approach temperature and pressure loss) are investigated in order to evaluate their influence on each of the cycles. Both the relative power output of these cycles and the relative increase in the size of the heat exchangers are reported in order to allow estimating their economic benefits. Each of the investigated cycles is optimized to maximize the net power output over the whole range of heat source temperatures. Several heat exchanger designs are laid out for the boilers and condensers of selected cycles in order to assess their relative size from one cycle to another, taking into account the properties of the different working fluids and the operating conditions. The pressure loss and the overall heat transfer coefficient for the different working fluids are showing no significant benefit for ammonia/water mixtures or hydrocarbons over refrigerants that would justify applying lower temperature differences and losses as design boundary conditions for cycle comparison. To meet given approach and pinch temperatures, however, different cycles require very different sizes of heat exchangers and the total amount of required UA (heat transfer coefficient (U) multiplied by the transfer area (A)) is shown. The study shows that advanced cycles can significantly increase the power output over the range of temperatures considered. With similar temperature boundary conditions, the power output can be increased by up to 35% compared to the baseline at the lowest heat source temperatures, but little benefit was found at the higher end of the temperature range. Ammonia/water mixture cycles, dual-pressure organic Rankine cycles and supercritical cycles can each offer a significant increase in power production for different heat source temperatures, generally at the cost of an increased system complexity and heat exchanger size.

Keywords:
Low-temperature cycles, advanced organic Rankine cycles, ORC comparison, ORC heat exchanger.

1. Introduction

The industrial sector energy consumption in the U.S. accounted in 2002 for roughly 22 quadrillions Btu/year (or 6.5 PWh/year), 25% to 55% of this energy being lost to the surroundings in the form of heat [1]. The IEA Blue Map Hi-REN scenario [2] aiming to reduce CO\textsubscript{2} emissions by 50% in 2050 predicts an installed geothermal power capacity of 200 GWe by that date. While the potential for industrial waste heat recovery and geothermal power is great, a large fraction of these resources is dominated by low grade heat sources. The organic Rankine cycle (ORC) is a proven technology which allows generating electricity from low-temperature heat sources in a far more efficient way than conventional steam cycles. However, for heat source temperatures below 200°C, the performance of ORCs is hampered by the large fraction of exergy being destroyed during the heat exchange process, mainly due to the constant temperature boiling phase of the working fluid. This evaporation phase compromises the thermal efficiency of the system to maintain good heat source utilization in order to maximize the power generation. Advanced solutions exist in order to minimize this exergy destruction at low heat source temperature and increase the overall system
efficiency. These solutions enhance the match between the heat source and the working fluid heating curves by working in the supercritical regime (removing the evaporation phase), increasing the number of evaporating pressure levels or using a fluid mixture as working fluid (thereby taking advantage of a temperature glide during the evaporation). The enhanced performance results from the better match between the heat source and the working fluid but comes at the cost of a (desired) lower overall temperature difference over the heat exchangers, leading to a higher cost for this equipment. This study aims at evaluating the relative power generation advantage and the penalty in required heat exchanger area of advanced low-temperature power cycles. Seven cycles, including a supercritical cycle, a dual-pressure organic Rankine cycle and several ammonia/water mixture cycles (“Kalina” cycles) are optimized for net power output on heat source temperatures ranging from 80°C to 200°C and compared to a single-pressure ORC baseline. These cycles are investigated using two sets of boundary conditions, mainly affecting the heat exchangers. The impact of these boundary conditions on the cycle performance and the relative size increase of the heat exchangers will be evaluated to assess the potential economic benefits of each cycle. As absolute power scales with the mass flow of the heat source for an optimized cycle and the cycles are calculated for comparison to one another with the same heat source, only relative power, normalized by a baseline cycle, is used throughout this paper. Finally, heat exchangers designs will be laid out for a selection of heat source temperatures and cycles in order to assess the impact of the different working fluids and operating conditions on the predicted cost of the heat exchangers, and to quantify possible advantages for some fluid/cycle combinations which might motivate the use of a different set of boundary conditions to make a fair comparison with the ORC baseline.

2. Investigated Cycles

Several cycles offering a better match between the heat source and working fluid heating curve have been investigated. Their relative net power output and relative heat exchanger size are compared to a state of the art subcritical, single-pressure organic Rankine cycle, referred to as “subcritical ORC” in the following.

2.1 - Subcritical Organic Rankine cycle

![Fig. 1. a) Subcritical, single-pressure ORC, b) Supercritical ORC](image)

Subcritical ORCs are a proven technology based on the steam Rankine cycle. Water is replaced by a working fluid more suitable for low-temperature heat sources such as hydrocarbons or refrigerants. The layout of the subcritical ORC cycle is given in Fig. 1-a. The lower boiling temperature of these fluids, combined with their “dry fluid” properties allows increasing the efficiency of these cycles while reducing the need for superheating even at low temperatures. However, subcritical ORCs
suffer from the constant temperature boiling phase for low heat source temperature. The pinch point occurring at the beginning of the working fluid evaporation indeed limits the extent to which the heat source can be cooled, affecting the system’s second law efficiency. Considering the low heat source temperatures investigated, a recuperator is usually not beneficial from a net power output point of view [3] (the minimum heat source temperature is set to 40°C, which is low enough for it not to become a limiting factor for the net power output optimization). These systems can be either water- or air-cooled, however in this study all the cycles investigated will be assumed to be air-cooled.

### 2.2 - Supercritical Organic Rankine Cycle

One way to reduce the exergy destruction caused by the imperfect heat transfer is to simply remove the constant temperature evaporation by working in the supercritical regime. The layout is similar to the subcritical ORCs given in Figure 1-b.

### 2.3 - Dual-Pressure, Subcritical ORC

Dual-pressure cycles introduce an additional pressure level, leading to an evaporation occurring at two different temperatures. The power cycle is split into a high-pressure and a low-pressure loop. The high temperature part of the heat source is used to evaporate the high-pressure loop. Where a single-pressure ORC would keep the same mass flow to preheat the high-pressure loop, the dual-pressure cycle splits the heat source around the beginning of the high-pressure loop evaporation. The minimum amount required for the high-pressure loop preheating is sent to the high-pressure preheater, while the rest of the flow is used to evaporate a low-pressure loop, as seen in Figure 2. The layout given in Figure 2, describing the cycle used in this study, is just one solution for the implementation of a dual-pressure ORC among others.

![Fig. 2. Subcritical, dual-pressure organic Rankine cycle (dual-pressure ORC)](image)

The dual-pressure cycle allows working with high pressure while still offering high heat source utilization. This concept can be generalized with more pressure levels, approaching the theoretical Lorenz cycle with an infinite number of pressure levels. In this study, however, the analysis is limited to the assessment of cycles with two pressure levels. It should be noted that on top of the
overall increase in system complexity, working with two pressure levels may require to use two turbines, potentially leading to a further increase of the system cost.

2.4 - Ammonia/Water Mixture (Kalina) Cycles

Power cycles designed with fluid mixtures benefit from a temperature glide both during the evaporation and the condensation process, as those will occur at non-constant temperature and reduce the losses associated with imperfect heat transfer between the heat source or the heat sink and the working fluid. Examples of these cycles are the ammonia/water mixture cycles, commonly referred to as Kalina cycles [4]. This study investigates several configurations with variable complexity using an ammonia/water mixture as a working fluid. These configurations can be classified into first and second generation’s cycles (the second generation cycles are more complex than the first generation). Each of this generation is declined into a low and a high temperature version. In total, five ammonia/water cycle (AW) configurations are investigated: three first generation (AWa, AWb, AWc), and two second generation cycles (AWd, AWe). Configuration (AWa), shown in Figure 3, is the low-temperature version of the first generation ammonia/water mixture cycle, commonly referred to as KCS34 Kalina cycle [3].

In this cycle a basic ammonia/water mixture is partially evaporated using the heat source and internal recuperation. The stream is then separated, where the ammonia-rich vapour is sent to the turbine and the ammonia-lean liquid is used for recuperation before being absorbed at the turbine exhaust. This recuperated cycle maximizes the efficiency cooling the source down to low temperatures thanks to the temperature glide during the evaporation. The effects of the operating pressure and the ammonia concentration on the performance of the cycle have already been investigated in previous research work [5]. This cycle is now proven and operating in several geothermal (e.g. in the Húsavik power plant [4]) and industrial heat sources.

It is possible that the pinch point at the beginning of the evaporation occurs during the recuperation step. In such cases it is possible to further increase the heat source utilization and therefore the power output of the system by slightly modifying the (AWa) layout with the introduction of a heat source heat exchanger in parallel to the low-temperature recuperator. This configuration (AWb),
shown in Figure 4 allows a better management of the heat source utilization at a moderate cost of complexity.

Fig. 4 and Fig. 5. Ammonia/water mixture cycle (AWb), left and (AWc), right.

For high-temperature heat sources, the optimum design fully evaporates the ammonia/water mixture at the heat exchanger outlet. In these cases, a separator is not required anymore, and the first generation ammonia/water mixture cycles degenerates into configuration (AWc), shown in Figure 5. This cycle, often referred to as KCS11 [7], is a simplified version of the previous cycle. The main difference here is that a constant concentration of ammonia is used on the whole cycle as opposed to the two previous cycles where three different ammonia concentrations occur in the cycle. More complex configurations designed for higher temperature heat source using two condensers exist [8, 9, 10], but are not investigated in this study. The main advantage of variable concentration cycles is their off-design capability; it is indeed possible to change the ammonia concentrations in order to optimize for different ambient temperatures [6].

The next two cycles investigated belong to the second generation of ammonia/water mixture cycles, developed by Kalex LLC [6]. The low-temperature version of these cycles, configuration (AWd), is shown in Figure 6-a. The main feature of these second generation cycles (often referred to as SG2 cycles [6]) is an internal recirculation loop which allows decreasing the condenser load. This has a positive impact on the condenser parasitic load since it decreases the working fluid flow rate through the condenser while maintaining a higher flow rate through the turbine. The liquid resulting from the first separation process is flashed to an intermediate pressure, and undergoes a second separation. The vapour stream is then sent back to the heat exchangers without being condensed, thereby reducing the condensation heat load. In order to pump this vapour stream back to the cycle’s top pressure, it is absorbed by a liquid stream resulting from the third separator placed downstream of the turbine.

If the heat source temperature is high enough, the optimum design for the (AWd) cycle leads to full evaporation of the ammonia/water mixture after the second heat source heat exchanger. In that case, the cycle can be simplified and degenerates into the (AWe) cycle shown in Figure 6-b.
3. Boundary Conditions

The previously described cycles are optimized for net power output on a set of eight heat source temperatures ranging from 80°C to 200°C with steps of 100°C, 110°C, 120°C, 130°C, 150°C and 175°C. Two sets of boundary conditions have been used for this study in order to evaluate the sensitivity of the investigated cycles relative to the heat exchangers design. “Conservative” and “aggressive” boundary conditions are defined, with different allowable minimum approach and pressure loss for the heat exchangers. The rest of the boundary conditions (i.e. isentropic and electrical efficiencies, ambient temperature, allowable air temperature rise over the air-cooled condenser, minimum heat source temperature) remain essentially unchanged. Table 1 shows the conservative boundary conditions in use for the heat exchangers and heat sink.

The set of boundary conditions described in Table 1 is slightly modified when simulating ammonia/water mixture cycles. Given the lower mass and volume flow rate at the expander outlet in ammonia/water cycles, the expected pressure loss in the condenser and the recuperator can be expected to be lower with similarly sized equipment when compared to organic Rankine cycles. For this reason, the pressure loss for the condenser and the recuperators in cycles running with ammonia/water mixture as a working fluid has been decreased to 1% of the absolute inlet pressure, while the other boundary conditions remain unchanged.
**Table 1. Conservative boundary conditions**

<table>
<thead>
<tr>
<th>Heat exchangers</th>
<th>Heat sink</th>
</tr>
</thead>
<tbody>
<tr>
<td>Minimum approach (heat source heat exchanger)</td>
<td>10 K Ambient temperature</td>
</tr>
<tr>
<td>Minimum approach (recuperators)</td>
<td>5 K Minimum air temperature rise over condenser</td>
</tr>
<tr>
<td>Minimum approach (condenser)</td>
<td>10 K Cooling medium</td>
</tr>
<tr>
<td>Pressure loss (heat source heat exchanger)</td>
<td>7% of absolute inlet pressure</td>
</tr>
<tr>
<td>Pressure loss (recuperators)</td>
<td>3% of absolute inlet pressure</td>
</tr>
<tr>
<td>Pressure loss (condenser)</td>
<td>5% of absolute inlet pressure</td>
</tr>
<tr>
<td>Minimum heat source temperature</td>
<td>40 °C</td>
</tr>
</tbody>
</table>

The aggressive boundary conditions are derived from a paper published by Dr. Kalina, describing the second generation Kalina cycles [7]. The boundary conditions for the heat exchangers calculated from the state points of an SG-2a cycle given in this paper are given in Table 2. It can be seen that the minimum approach and the pressure losses on the heat exchangers are much lower than for the previous “conservative” set of boundary conditions. The rest of the boundary conditions and the ambient temperature remain unchanged.

**Table 2. Aggressive boundary conditions**

<table>
<thead>
<tr>
<th>Heat exchangers</th>
<th>Optimization variables</th>
</tr>
</thead>
<tbody>
<tr>
<td>Minimum approach (heat source heat exchanger)</td>
<td>2.8 K</td>
</tr>
<tr>
<td>Minimum approach (recuperators)</td>
<td>2.8 K</td>
</tr>
<tr>
<td>Minimum approach (condenser)</td>
<td>5 K</td>
</tr>
<tr>
<td>Pressure loss (heat source heat exchanger)</td>
<td>2 psia / 0.14 bar</td>
</tr>
<tr>
<td>Pressure loss (recuperators)</td>
<td>2 psia / 0.14 bar</td>
</tr>
<tr>
<td>Pressure loss (condenser)</td>
<td>2 psia / 0.14 bar</td>
</tr>
<tr>
<td>Minimum heat source temperature</td>
<td>40 °C</td>
</tr>
</tbody>
</table>

The list of the optimization variables for each of the cycles investigated is given in table 3. The rest of the cycle parameters are imposed by fixed conditions (e.g. degree of superheating) or their optimum value can be calculated directly without optimization (e.g. condensing pressure).

**Table 3. Optimization variables for each cycle**

<table>
<thead>
<tr>
<th>Cycle</th>
<th>Optimization variables</th>
</tr>
</thead>
<tbody>
<tr>
<td>Subcritical, single pressure ORC</td>
<td>Top cycle pressure</td>
</tr>
<tr>
<td>Supercritical ORC</td>
<td>Top cycle pressure</td>
</tr>
<tr>
<td></td>
<td>Top cycle temperature</td>
</tr>
<tr>
<td>Dual-pressure ORC</td>
<td>Top cycle pressure</td>
</tr>
<tr>
<td></td>
<td>Intermediate cycle pressure</td>
</tr>
<tr>
<td>(AWa), (AWb), (AWc)</td>
<td>Top cycle pressure</td>
</tr>
<tr>
<td></td>
<td>Ammonia concentration</td>
</tr>
<tr>
<td>(AWd), (AWe)</td>
<td>Top cycle pressure</td>
</tr>
<tr>
<td></td>
<td>Ammonia concentration</td>
</tr>
<tr>
<td></td>
<td>Recirculation loop flow rate</td>
</tr>
</tbody>
</table>
The cycles are modeled using the commercial chemical process modeling tool Aspen HYSYS. For organic Rankine cycles, a set of common refrigerants and hydrocarbons are investigated (including, but not limited to R125, R134a, R245fa, R152a, isobutane, n-butane, propane, isopentane, n-pentane), and the fluid properties are calculated using the Zudkevitch-Joffee [11] equation of state available in HYSYS. For ammonia/water mixture cycles, the SRK [12] equation of state is used in order to calculate the fluid properties.

4. Cycle Performance Comparison

The previously introduced cycles are optimized for net power output for each of the eight heat source temperatures, for both sets of boundary conditions.

4.1. Conservative Boundary Conditions

Figure 7 shows the relative net power output of the investigated cycles against the single-pressure, subcritical ORC using the conservative set of boundary conditions. The working fluid used for the reference subcritical ORC is given at the bottom of the graph. As a general trend, the advanced cycles tend to offer larger power output benefits for lower heat source temperatures: the power output increase can reach up to 35% with an 80°C heat source, while benefits up to only 10% can be expected for a 200°C heat source. This can be explained by the strong increase in performance of the subcritical ORC as we leave the very low heat source temperature range.

![Fig. 7. Relative performance comparison, conservative boundary conditions](image_url)

When looking closer into the power output of the different cycles, it is apparent that ammonia/water cycles perform significantly better than the ORC baseline for heat source temperatures below about 150°C. Among these cycles, the second generation cycles offer greater benefits than the first generation cycles over the same temperature range. The modified version of KCS34, the AWb cycle, significantly increases the performance of the first generation cycles to bring them close to the second generation configuration. For heat source temperatures below 110°C, the second
generation ammonia/water cycles offer the highest performance. It should be noted that with the same set of conservative boundary conditions, the ammonia/water mixture cycles investigated cannot compete with the organic Rankine cycles for temperatures above 175°C. Dual-pressure cycles are the only advanced solution to offer a power output increase over the whole range of heat source temperature considered, even though this advantage decreases with the heat source temperature. Above 120°C, supercritical cycles become the most efficient cycles. It is worth noting that for very low heat source temperature, the choice of working fluids that can operate supercritical is very limited, which partly explains the strong decrease in performance for this temperature range. If an ideal fluid could be tailored for any temperature with an adjusted critical point, one could expect that the power output benefit would be maintained even for very low heat source temperatures.

Considering the net power output by itself may not be enough for a comprehensive cycle assessment. The advanced cycles aim at improving the match between the cooling curve of the heat source and the heating curve of the working fluid to reduce the second-law losses during the heat transfer. The direct consequence in a cycle where the input and approach temperature boundary conditions are fixed is a reduction of the average temperature difference across the heat exchangers. A reduced mean temperature difference typically will result in a larger heat exchanger area to transfer the same amount of heat, hence an increase in size and cost. In order to capture this effect, the total UA value of the heat exchangers in the cycle is recorded for each of the optimized cycles (the UA of a heat exchanger is the overall heat transfer coefficient multiplied by the heat transfer area and equals the quotient of the heat duty divided by the mean temperature difference. The mean temperature difference takes into account the non-linear temperature-heat load profile in the heat exchangers for the supercritical and ammonia/water based cycles). The total UA value (summed over the heat source heat exchangers, recuperators and condenser) required per unit of power produced is calculated to represent the size of the heat exchangers relative to the power output increase of the advanced cycles. This value is shown in Figure 8 for each cycle and heat source temperature.

![Fig. 8. Relative total heat exchanger UA per unit of power produced, conservative boundary conditions](image-url)
Figure 8 shows that even if the AWb configuration helped to bring the performance of the first generation of ammonia/water mixtures closer to the second generation, it requires far more heat exchanger area under the same boundary conditions. The second generation cycles indeed achieve a higher power output increase while being relatively less demanding from a heat exchanger size point of view. This can be explained by the use of more separators and mixers in the second generation configurations which achieve internal recuperation via direct mixing, therefore saving on heat exchanger UA.

The required total UA per unit of power produced is remarkably constant for the first generation AWa and AWc configurations and for the supercritical cycle. This value oscillates between 115% and 135% for the range of heat source temperatures where these cycles offer a performance advantage. For the dual-pressure cycle the required UA per unit of power produced is barely higher than for the subcritical, single-pressure ORC, or even lower for temperatures around 150°C. This means that the dual-pressure cycle will offer a strong power output increase, while the relative size of the heat exchanger is expected to stay almost the same. The dual-pressure and ammonia/water based cycles however have an additional cost associated with the increase in cycle complexity, additional components (mixers, separators) and controls, and with the possible need for two expanders instead of one for the dual-pressure cycle.

### 4.2. Aggressive Boundary Conditions

The same exercise as above is repeated with the second set of boundary conditions; the results are shown in Figure 9. The general trend is the same: under comparable boundary conditions, the performance advantage offered by the advanced cycles decreases with increasing heat source temperature, with a maximum benefit of about 35% at a heat source temperature of 80°C, down to about 5% at 200°C.

![Fig. 9. Relative performance comparison, aggressive boundary conditions](image)

Similar to the previous conservative boundary conditions case, the ammonia/water cycles perform significantly better than the organic Rankine cycles for heat source temperatures of 150°C and...
below. For heat source temperatures below 120°C, the second generation ammonia/water cycle is the best cycle from a pure performance point of view. In the region where they offer benefits over the baseline, the second generation ammonia/water cycles perform significantly better than the first generation at base configuration (AWa). The modified version, AWb, brings the performance of the first generation closer to the second generation configurations, especially for heat source temperatures of 120°C and above.

The dual-pressure cycle offers a strong performance increase for heat source temperatures between 80°C and 150°C, with a gain of roughly 10% to 20% in net power in that range. The power increase offered by dual-pressure cycles increases as the heat source temperature gets lower, but reaches a plateau of 20% net power increase for heat source temperatures of 120°C and below. The supercritical cycle offers the best performance increase for heat source temperatures of 130°C and above. It can be noted again that for very low heat source temperatures the benefit of the supercritical cycle decreases dramatically, and even drops below the subcritical ORC performance for an 80°C heat source temperature. This is because the choice of working fluids with a critical temperature low enough to be used in these cycles is limited as the heat source temperature goes below 100°C.

The relative increase in size of the heat exchangers for each cycle is calculated for the whole temperature range. This increase is measured by considering the amount of heat exchanger UA required in order to generate a unit of power. Figure 10 shows how this value evolves for each cycle and heat source temperature relatively to the subcritical, single pressure ORC baseline.

![Figure 10. Relative total heat exchanger UA per unit of power produced, aggressive boundary conditions](image-url)

The trend for the total UA required per unit of power produced is very similar to the trends identified for the conservative boundary conditions case. Even though the AWb configuration brings the performance of the first generation ammonia/water cycles closer to their second generation counterpart, they are likely to require far more heat exchanger area relatively to the power they produce, and might not represent an economical solution. The second generation cycles offer equal or superior performance at a relatively lower required heat exchanger area per unit of power produced. It is interesting to note that for a 150°C heat source, the required heat exchanger UA per unit of power produced for the AWd configuration is lower than for the subcritical ORC
baseline. This means that this cycle not only produces more power than the baseline on a 150°C heat source, but it is also likely to have smaller heat exchangers relatively to the power produced by the cycle. The AWa configuration and the dual-pressure cycle are for the same reasons particularly interesting for the lowest heat source temperatures considered, where they offer strong performance benefits at only a moderate increase in the relative UA of the heat exchangers. It has been shown that the supercritical cycle offers the highest performance benefits for heat source temperatures of 130°C and above. In this same range of heat source temperatures, its relative required UA per unit of power produced remains between 5% and 20% higher than the subcritical ORC baseline.

4.3. Mixed Boundary Conditions

While the previous results help to assess the attractiveness of one cycle to another with the same set of boundary conditions being applied to all cycles, they do not give any indication on the best choice for boundary conditions or specifications for the heat exchangers. The following figures 11 and 12 show the net power output increase and the impact on the required total heat exchanger UA per unit of power produced for both sets of boundary conditions relative to the subcritical ORC baseline with conservative boundary conditions.

![Fig. 11. Mixed boundary condition cycle performance comparison](image)

A first observation is that within the same set of boundary conditions, the difference between the different cycles seems to decrease, especially for the relative net power output. The important conclusion is that the change from one set of boundary conditions to another has at least as much impact as the change from one cycle to another assuming the same set of boundary conditions. Another observation is that the advantage of using an aggressive set of boundary conditions is the highest for the lowest heat source temperature range. At the same time, the relative required total heat exchanger UA per unit of power produced tends to get much higher with the aggressive set of boundary conditions than with the conservative set at higher heat source temperatures. This is an indication that the aggressive set of boundary conditions may provide high power production.
benefits at a moderate increase of the relative size of the heat exchangers for the lowest heat source temperature range. This is particularly interesting when working with very expensive heat sources such as geothermal brines. On the other hand, the conservative set of boundary conditions seems most appropriate when working with higher heat source temperatures, as the power output benefits relative to the baseline tends to decrease when switching to more aggressive set of boundary conditions while the relative size of the heat exchangers increases dramatically. The previous observations are made considering only the heat exchanger UA required to meet certain design temperatures and pressure losses given in the boundary condition sets. The required UA is a good indication of the variations of the heat exchanger size when working with the same fluid under similar operating conditions and heat exchanger geometries. However, variations of the overall heat transfer coefficient U can be expected between the subcritical ORC, supercritical ORC and the ammonia/water cycles owing to different fluid properties and flow conditions even between geometrically similar heat exchangers. These variations influence the required area and ultimately the size and cost of the heat exchangers. For instance, an increase of the U value of one cycle would decrease the area for the required UA; despite a higher UA requirement, one cycle with higher U may not require a larger area than another. The UA required for a certain cycle is therefore not fully sufficient to fairly compare different cycles in order to estimate relative equipment cost. In order to assess the impact of the fluid and the flow conditions on the heat exchanger size and cost for a given UA resulting from the boundary conditions set in this study, detailed heat exchanger designs are laid out for a selection of cycles and common heat source temperatures.

**Fig. 12. Mixed boundary conditions, relative total heat exchanger UA per unit of power produced comparison**
5. Detailed Heat Exchanger Design Comparison

Different cycles under given fluid temperature and pressure boundary conditions and a fixed heat source mass flow and inlet temperature require not only different heat flows but also different approach and pinch point temperatures because of the different heating (boiling or condensing) curves, resulting in different overall heat transfer and surface area (UA) requirements for each heat exchanger. To answer questions about how the UA requirement for different cycles affects the size and cost of the heat exchangers, shell-and-tube (S&T), plate and fin-tube boilers (PHE) and fin-tube (F-T) condensers have been designed using Aspen Exchanger Design and Rating software (Aspen v7.2). The cycles selected for this investigation are the subcritical ORC with isobutane and with a refrigerant, the supercritical ORC and the recuperated simple KCS34 Kalina cycle (AWa) with a heat source temperature of 120°C and “conservative” boundary conditions. The heat exchangers are designed with consistent features and design guidelines across the range of cycles to ensure these are comparable one to another and the area and cost implications due to fluid and cycle specifics are expressed clearly.

![Graph showing specific primary area and cost per unit for different cycles](image1)

For the three types of boilers the required surface areas for the different cycles are shown in Figure 13. “Specific” here refers to normalization by the cycle net power output. For the ammonia/water cycles with a boiling two-phase ammonia-water mixture, the average fluid-side heat transfer coefficients in a heat exchanger can be more than twice as high as hydrocarbons or refrigerants, particularly when allowing for higher pressure losses on the grounds of smaller volume mass flow rate in the feed pump. However, when the heat transfer coefficients on the other side of the boiler or condenser, single-phase water or air, enter the equation, the overall heat transfer coefficient (U) that determines the heat exchanger’s area and cost, does not increase dramatically over that seen for hydrocarbons or refrigerants, by about 10% for shell-and-tube and fin-tube heat exchangers, hence the modest decrease in required area. Still, the relatively low mass flow and a mostly lower pressure of the ammonia/water cycles compared to ORCs allow designing a shell-and-tube or fin-tube boiler somewhat more compact and inexpensive, with the cost per unit tube surface area about 10% lower than for an ORC in the cases studied. Together with the higher U for the ammonia/water heat exchangers, the cost per unit UA of shell-and-tube and fin-tube boiler can be more than 10% lower for the ammonia/water cycle than for refrigerants and hydrocarbons; this cost is shown in Figure 14.

The boilers designed in this study showed that for a shell-and-tube heat exchanger the pressure losses on both sides are significantly below the limits set in the boundary conditions, size and cost are driven primarily by the desired UA. For plate heat exchangers, however, for cycles with
relatively large volume flow, such as for an ammonia water mixture or an evaporating refrigerant, the exchanger size and cost is primarily driven by acceptable pressure loss rather than by the desired UA alone, hence a larger UA than required came out of the design and the cost per unit UA (Figure 14) is very low. Thus, for a plate heat exchanger in ammonia/water cycles further opportunity for reduced surface area due to increased U compared to ORCs exist if pressure losses are allowed to rise as pointed out in the following considerations on design boundary conditions based on pumping power. A plate heat exchanger is not a likely choice for a supercritical ORC because of the high pressure. For a fin-tube boiler, the design, size and cost is mostly driven by the air-side pressure loss limit and associated limited velocity and heat transfer coefficient; achieving high values for U and reaching small approach temperatures to meet more “aggressive” boundary conditions is very difficult and costly.

Figure 15: Specific cost of boiler

Figure 16: Condenser cost specific to UA and to cycle net power

Generally speaking, a boiler with a liquid heat source does not present a significant cost challenge under either set of boundary conditions considered here, whereas a fin-tube boiler for a gaseous heat source does come at a significant cost, about five times that of a shell-and-tube boiler for “conservative” conditions. The specific cost relative to the net power output of the boilers is shown in Figure 15. Attempting to meet more “aggressive” design boundary conditions by decreasing the desired approach temperatures may lead to size and cost increases beyond what is economical.

Customarily the boundary conditions for heat exchanger design are given as minimum approach temperatures, pinch-points and relative pressure losses, such as in the sets of boundary conditions used in this paper. An alternative to setting the desired temperatures is to design for matching a desired UA that is deemed economical, as the UA is more closely related to the cost than minimum temperature differences as this investigation has shown. An alternative way to set the fluid-side pressure loss limit in the boiler to a fixed fraction of absolute pressure is instead setting a required pumping power to overcome the losses as a fraction of the cycle’s net power. This way cycles with low mass flow, such as hydrocarbons or ammonia-water mixture, allow for higher absolute pressure losses in the boiler (smaller or longer pipes), which may further reduce the boiler size and the approach temperatures depending on the design and the heat source-side pressure loss limit relative to ORCs, specifically for a pressure loss limited plate heat exchanger.

An air-cooled condenser is arguably the largest and most costly heat exchanger in a Rankine cycle. To save cost at the cold end, a water-cooled condenser (shell-and-tube or plate heat exchanger) and a source of cooling water, such as a river or a cooling tower could be applied. In this study, only an air-cooled fin-tube condenser with forced convection electric fans is considered. To trade the cost of fan power requirements against the area of the condenser, the design fan power has been kept at or
below about 1% of the heat duty. For the boiler and the condenser the performance impact in terms of relative change in power output versus relative change in required UA are basically similar (roughly 1% net power for 4% UA, depending on the design point, resulting ultimately from Carnot’s efficiency for low source temperature). However, the cost impact of an air-cooled condenser on a cycle is almost an order of magnitude larger than that of a shell-and-tube or plate boiler. The specific cost of the air-cooled condenser is shown in Figure 16, along with the cost per unit UA in $-K/kW.

The ammonia/water cycles have a rather small required UA as they are usually recuperated and less heat is discharged in the condenser although the temperature differences are also smaller than for ORCs because of the temperature glide when condensing a mixture, overall resulting in a lower specific cost, shown in Figure 16. Despite the higher heat transfer coefficients on the inside compared to refrigerants and hydrocarbons, the cost per unit UA is only insignificantly lower for ammonia/water cycles condensers, shown also in Figure 16. The supercritical ORC with its high mass flow rate has a somewhat higher cost per UA to meet pressure loss limits but the specific cost is not higher than for other ORCs. The fluid-to-air condenser heat transfer is mainly limited by the air-side, as the inside heat transfer coefficients are about two times higher than on the outside relative to the bare tube area. Since the outside heat transfer resistance is independent of the fluid conditions and similar air velocities result from the fan power limits, the differences between the cycles are rather small.

When designing the condensers to meet either the “conservative” or the “aggressive” boundary conditions, it emerges that the differences in required area and estimated cost are rather small, but the fan power requirement increases to allow a higher mass flow of air and the arrangement of fin tubes and headers can be different to meet the fluid pressure loss limitations. Only for supercritical ORC and the ammonia/water cycles, the required UA to meet the “aggressive” boundary conditions increased by about 20%.

6. Conclusions

The performance of advanced power systems, including dual-pressure ORCs, supercritical cycles and ammonia/water cycles for low temperature heat sources have been evaluated relatively to an organic Rankine cycle baseline. These cycles have been compared using two different sets of boundary conditions in order to evaluate their sensitivity to the heat exchanger design. Under the same set of boundary conditions, ammonia/water mixture cycles outperform all other cycles for heat source temperatures below 120°C. For this same range of heat source temperatures, the second generation cycles offer better performance than the first generation of ammonia/water cycles. These performance benefits are achieved at the price of a larger required total heat exchanger UA for these cycles compared to the subcritical ORC baseline. For heat source temperatures between 120°C and 150°C, the dual pressure cycle offers a solid performance benefit over the subcritical ORC, at a moderate increase of the relative size of the heat exchangers. For heat source temperatures above 150°C, the supercritical cycle is the best advanced cycle solution. However, the general performance benefit of advanced cycles over subcritical organic Rankine cycles tends to decrease as the heat source temperature increases, resulting from an increase of the baseline performance as the heat source temperature approaches 200°C.

Ammonia/water mixture based cycles need somewhat less primary area for the boilers per unit UA because of a higher U, but at the same time require higher UA to meet the same approach temperatures compared to ORCs. The result is that any effect on specific area or cost reduction is small and more “aggressive” temperature design boundary conditions than in an ORC are not justified. Supercritical ORCs entail higher costs for the boiler than other cycles because of high UA requirements and higher mass flow. Thermally efficient cycles that reject relatively less heat for the same input have an advantage of a smaller condenser requirement. The specific cost for air-cooled condensers is about one order of magnitude higher than that of shell-and-tube or plate boilers and
several times that of fin-tube boilers. Although having a similar impact on the efficiency of low-temperature cycles, an economic condenser will require more generous approach temperatures than a shell-and-tube or plate boiler to optimize the cycle. Generally, meeting more “aggressive” boundary conditions than the “conservative” set seems practicable for air-cooled condensers and boilers, specifically for plate boilers, while larger approach temperatures are to be expected in fin-tube boilers for gaseous heat sources. Reasonable boundary conditions for design need to account for the type of heat exchanger and the media on both sides and cost considerations; an apple-to-apple comparison of cycles should include comparable heat exchanger areas or costs.

In any case, there is no single best technology for low temperature waste heat recovery. The optimum design will indeed vary strongly from one application to another and the author’s company supports specific studies to offer the appropriate solution fitting each customer’s needs.

References
Comparison of Nuclear Steam Power Plant and Conventional Steam Power Plant through Energy Level and Thermoeconomic Analysis

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Abstract:
Exergetic and thermoeconomic analyses were performed for a 315 MW conventional steam and 1076 MW VVER nuclear power plants. In these analyses, mass and energy conservation laws were applied to each component of the system. Quantitative balances of the exergy and exergetic cost for each component, and for the whole system was carefully considered. The exergoeconomic model, which represented the productive structure of the system considered, was used to visualize the cost formation process and the productive interaction between components. The computer program developed in this study can determine the production costs of each component of steam and combined cycle power plant such as electricity production in steam turbines or gas turbine. The code can be also be used to study plant characteristics, namely, thermodynamic performance and sensitivity to changes in process and/or component design variables.

Keywords:
Exergy, Exergoeconomic, Steam power plant, Nuclear power plant.

1. Introduction

Energy systems involve a large number and various types of interactions with the world outside their physical boundaries. The designer must, therefore, face many issues, which deal primarily with the energetic and economic aspects of the system. Thermodynamic laws govern energy conversion processes, costs are involved in obtaining the final products (expenses for the purchase of equipment and input energy resources, operation and maintenance costs), and the effects of undesired fluxes to the ambient must be evaluated in order to answer environmental concerns. Second law analysis has been widely used in the last several decades by many researchers. Exergy analysis usually predicts the thermodynamic performance of an energy system and the efficiency of the system components by accurately quantifying the entropy-generation of the components [1]. Furthermore, exergoeconomic analysis estimates the unit cost of products such as electricity, steam and quantifies monetary loss due to irreversibility. Also, this analysis provides a tool for the optimum design and operation of complex thermal systems [1], [2], [3]. At present, such analysis is in great demand because proper estimation of the production costs is essential for companies to operate profitably. In addition, it is vital to display the system information graphically for one to visualize the performance of system in different cases by applying improved combined pinch and exergy analysis. In contrast, the power of exergy analysis is that it can identify the major causes of thermodynamic imperfection of thermal and chemical processes and thus promising modifications can be determined [4]. By combining the strengths of pinch and exergy methods, the
The proposed method can represent a whole system, including individual units on one diagram, which helps to screen the promising modifications quickly for improving a base case design [5].

In this study, exergetic, thermoeconomic and combined pinch and exergy analyses have been performed for 1076MW nuclear steam cycle and 315MW gas fired steam power plants. In these analyses, mass and energy conservation laws were applied to each component. Quantitative balance of the exergies and exergy costs for each component and for the whole system was carefully considered. The exergy-balance equation developed by Oh et al. [6] and the corresponding exergy cost-balance equations developed by Kim et al. [7] were used in these analyses.

In this regard, computer program has been developed for energy, exergy, exergoeconomic and exergy analysis of both of cases in different load conditions. Furthermore, it can also use to study plant characteristics, namely, thermodynamic performance and sensitivity to changes in process and/or component design variables.

In this paper, the authors evaluate and compare gas fired steam and nuclear power plants in view of exergy and thermoeconomic analysis at different load conditions. The main objective of this study is to provide insights into the performance of system components in particular by applying methods.

2. Process Description

In this paper, two cases have been considered at different load conditions. The first case is 315 MW gas fired steam power plant such as RAMIN power plant that is located in southwest of Iran in Ahvaz city. The scheme of 315 MW power plant and its steam turbines have been shown in Figure 1. The type of fuel is natural gas that its Low Heating Value (LHV) is 48748 kJ/kg and the net plant efficiency based on LHV is about 38.5.

The second case is 1076 MW nuclear steam power plant such as BUSHEHR power plant that is located in south of Iran in BUSHEHR city. Figure 2 represents the flow diagram of this plant. The net plant electricity efficiency in BUSHEHR power plant is 34.98.

![Fig 1. The scheme of a 315 MW Steam Power Plant](image)
3. Exergoeconomic Analysis

3.1. Exergy Analysis

Exergy is the maximum theoretical useful work attainable from an energy carrier under the conditions imposed by an environment at given pressure $p_0$ and temperature $T_0$, and with given amounts of chemical elements [8]. The purpose of an exergy analysis is generally to identify the location, the source, and the magnitude of true thermodynamic inefficiencies in thermal systems. Disregarding kinetic and potential energy changes the specific flow exergy of a fluid at any cycle state is given by:

$$e = h - h_0 - T_0(s - s_0)$$

The reversible work as a fluid goes from an inlet state to an exit state is given by the exergy change between these two states. That is:

$$e_2 - e_1 = h_2 - h_1 - T_0(s_2 - s_1)$$

Where the subscripts 1 and 2 represent the inlet and the exit state for a flowing fluid. Now, we present the exergy destruction and exergy efficiency relations for various cycle components in plant [9].

3.2. Cost equation for plant component

All costs due to owning and operating a plant depend on the type of financing, the required capital, the expected life of a component, and so on. The annualized (levelized) cost method of Moran was used to estimate the capital cost of system components in this study. The amortization cost for a particular plant component may be written as:

$$PW = C_i - S_n PWF(0, n)$$

Fig 2. The scheme of 1076 MW Steam Power Plant
The present worth of the component is converted to annualized cost by using the capital recovery factor $CRF(i, n)$, i.e. [2]. Dividing the levelized cost by 8000 annual operating hours, we obtain the following capital cost for the $k$th component of the plant.

$$Z_k = \frac{\Phi_k t_k}{3000 \times 8000}$$

(5)

The maintenance cost is taken into consideration through the factor $\Phi_k = 1.06$ for each plant component whose expected life is assumed to be 15 years [2].

### 3.3. Exergoeconomic Modelling

The results from an exergy analysis constitute a unique base for exergoeconomics, an exergy-aided cost reduction method. A general exergy-balance equation, applicable to any component of a thermal system may be formulated by utilizing the first and second law of thermodynamics [1], [2], [11], [12].

The cost balance expresses that the cost rate associated with the product of the system ($C_P$), the cost rates equals the total rate of expenditure made to generate the product, namely the fuel cost rate ($C_F$), the cost rates associated with capital investment ($Z_{CI}$), operating and maintenance ($Z_{OM}$) [13].

In a conventional economic analysis, a cost balance is usually formulated for the overall system (subscript tot) operating at steady state [14]:

$$C_{P,tot} = C_{F,tot} + C_{L}Z_{tot}$$

(6)

Accordingly, for a component receiving a heat transfer and generating power, we would write [2]:

$$\sum_k \text{EMBED Equation.3 Eq. k} + \sum_k \text{EMBED Equation.3 Eq.k} = \text{EMBED Equation.3 Eq. k}$$

(7)

To solve for the unknown variables, it is necessary to develop a system of equations applying Eq. (6) to each component, and it some cases we need to apply some additional equations, to fit the number of unknown variables with the number of equations [15],[16],[17].

To derive the cost balance equation for each component, we assigned a unit cost to the principal product for each component. Depending on the type of fuel consumed in the production process different unit cost of product should be assigned [18], [19], [20],[21].

### 3.4. Energy Level Analysis

Pinch analysis has become a general methodology for targeting and design of thermal and chemical processes and associated utilities [4]. The composite curves (CC) and the grand composite curves (GCC) are two basic tools in pinch analysis, and they are constructed using temperature versus enthalpy axes. The energy targets set by the CC and GCC are mainly in terms of heat loads. To deal with systems involving heat and power, the concepts of both the composite curves and the grand composite curves have been extended. As a result, the exergy composite curves (ECC) and the exergy grand composite curves (EGCC) were proposed which are based on Carnot factor ($\eta$) versus enthalpy [4].

The composite curves (T-H diagram) for a heat transfer system can be converted into the exergy composite curves and the grand composite curves. The shaded areas indicate the exergy loss associated with the heat transfer process. By combining pinch analysis and exergy analysis in such a
manner, it is possible to predict the shaft work requirement or generation for both power systems and refrigeration systems with certain accuracy [4]. It must be noted that this combined pinch and exergy analysis was developed mainly for the purpose of shaft work targeting. When dealing with process modifications, it has severe limitations. Particularly, only processes related to heat transfer can be represented on the ηc-H diagram but not the processes associated with pressure and composition changes, since the diagram is constructed based on temperatures [4]:

\[
\eta = 1 - \frac{T_0}{T}
\]

(8)

To overcome limitation of ηc-H diagram, a generic diagram has been introduced which is the so called Ω-H by Feng in 1996 [4], where Ω indicates the energy level and H states the amount of energy. Both energy and exergy balances for a whole system can therefore be represented simultaneously on this diagram. Using this diagram, the major advantages of both pinch and exergy analysis are combined since the diagram enables one to view the performance of a system and set targets for improvement.

The Energy Level Representation (ELR) based on pinch and exergy analysis draw on the earlier strategies of the thermodynamic approach to process integration, namely the concept of composite curves and the methodology of combined pinch and exergy analysis.

The graphical representation of process units involving energy in terms of heat and power has been made possible with the introduction of a variable referred to as energy level (Ω) defined [23]:

\[
\Omega = \frac{\text{Energy}}{\text{Exergy}}
\]

(9)

Thus, for work

\[
\Omega = 1
\]

(10)

and for heat

\[
\Omega = 1 - \frac{T_0}{T}
\]

(11)

and for a steady-state-flow system

\[
\Omega = \frac{\Delta E}{\Delta H}
\]

(12)

4. Computer Program

A computer program for exergy, exergoeconomic, exergy destruction and exergy destruction level analyses of the 1076-MW nuclear and 315-MW steam gas fired power plants have been developed. The program uses the following input data:

(a) Standard pressure (P₀) and temperature (T₀);

(b) Fuel compositions and costs

(c) Air composition and relative humidity of air;

(d) Different load conditions;

(d) Gross shaft power of gas turbine, steam turbine;

(e) Shaft work consumption in compressor and pumps;

(f) Mass flow rate (kg/s), pressure (MPa) and temperature (°C) for fluid streams at the inlet and outlet of each component;

(g) Initial investment of capital cost, interest rate, salvage value factor;
Using these input data; one can calculate the number of moles of combustion products, adiabatic flame temperature and enthalpy (MW) and entropy (MW/K) for fluid streams at various states. Using the values of these properties, we calculated the net flow rate of various exergies and entropies, the exergy efficiencies of the components and the lost exergy occurred in each component. The heat transfer rate from a component was calculated to satisfy the exergy balance for the component. Once exergy balances for the components, junctions and the plant boundary were established, the unit cost of various exergies and products were calculated by solving the cost balance equations simultaneously. Also, this program can generate Energy Level representations that can be used for combined pinch and exergy analysis.

5. Results and discussion

In this paper, computer program have been developed for thermodynamic simulation and analysis of 315-MW gas fired steam and 1076-MW nuclear power plants in different load conditions. The enthalpy and entropy of non-interacting gas species were calculated by using appropriated polynomials fitted to the thermophysical data in the JANAF Tables [22]. Also the values of physical properties such as enthalpy and entropy for water and steam were evaluated by using equations suggested by the International Association for the Properties of Water and Steam (IAPWS-IF97) [23]. Exergy and exergoeconomic rates of each streams in both thermal power plants calculated by computer program have been provided in this study. Also, exergy and exergoeconomic analysis have been performed for each component to calculation of exergy and exergy cost destruction with and without considering capital investment at various load conditions in steam gas fired and VVER nuclear steam power plants as shown in Table 1 and Table 2.

Table 1. Exergy flow and cost flow rates of exergy destruction with and without considering capital investment for each streams in a 315 MW steam power plant at various load condition.

<table>
<thead>
<tr>
<th>Load</th>
<th>Equip Unit</th>
<th>$/hr</th>
<th>$/hr</th>
<th>$/hr</th>
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<th>$/hr</th>
<th>$/hr</th>
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<tbody>
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<td>100</td>
<td>FWH1</td>
<td>1.74</td>
<td>19.40</td>
<td>20.02</td>
<td>0.59</td>
<td>6.98</td>
<td>7.19</td>
<td>0.60</td>
<td>6.94</td>
<td>7.37</td>
<td>0.12</td>
<td>1.46</td>
<td>1.67</td>
</tr>
<tr>
<td>75</td>
<td>FWH2</td>
<td>0.76</td>
<td>8.46</td>
<td>7.94</td>
<td>0.39</td>
<td>4.58</td>
<td>3.69</td>
<td>0.28</td>
<td>3.26</td>
<td>1.73</td>
<td>0.14</td>
<td>1.83</td>
<td>2.09</td>
</tr>
<tr>
<td>50</td>
<td>FWH3</td>
<td>0.89</td>
<td>9.96</td>
<td>10.28</td>
<td>0.86</td>
<td>10.17</td>
<td>10.48</td>
<td>0.33</td>
<td>3.85</td>
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<td>0.17</td>
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<td>2.40</td>
</tr>
<tr>
<td>25</td>
<td>FWH4</td>
<td>0.79</td>
<td>8.80</td>
<td>9.08</td>
<td>0.42</td>
<td>4.96</td>
<td>5.11</td>
<td>0.36</td>
<td>4.18</td>
<td>4.44</td>
<td>0.21</td>
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</tr>
<tr>
<td></td>
<td>FWH5</td>
<td>0.99</td>
<td>12.55</td>
<td>13.33</td>
<td>0.61</td>
<td>8.08</td>
<td>8.70</td>
<td>0.05</td>
<td>0.63</td>
<td>0.71</td>
<td>0.23</td>
<td>3.68</td>
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<tr>
<td></td>
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<td>0.75</td>
<td>8.97</td>
<td>9.24</td>
<td>0.35</td>
<td>4.09</td>
<td>4.34</td>
<td>0.21</td>
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<td></td>
<td>FWH7</td>
<td>1.24</td>
<td>13.75</td>
<td>14.20</td>
<td>0.36</td>
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Table 2. Exergy flow and cost flow rates of exergy destruction with and without considering capital investment for each streams in a 1076 MW nuclear power plant at various load conditions.

<table>
<thead>
<tr>
<th>Load</th>
<th>Equipment/Unit</th>
<th>Parameter</th>
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<td>C_D0</td>
<td>C_D</td>
<td>E_D</td>
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<tr>
<td></td>
<td></td>
<td>MW/SHR</td>
<td>MW/SHR</td>
<td>MW/SHR</td>
<td>MW/SHR</td>
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<td>HPT</td>
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<td>426.89</td>
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<td>43.51</td>
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<td>96.15</td>
<td>386.60</td>
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</table>

These results represented that boiler in gas fired steam and reactor in nuclear power plant have most exergy and exergy cost destruction due to nature of combustion and fission; however boiler in steam plant shares about 86 % TED, 88 % TCD0 and 87% TCD and reactor in nuclear plant shares about 76% TED, 71% TCD0 and 70% TCD. In next steps, condenser and steam turbines for gas fired steam and steam generator, condenser and steam turbine for nuclear case have most exergy and exergy cost destruction. As results shown, TCD0 and TCD reduce when load condition decreases and vice versa because the fuel consumption decreases when load condition reduces and vice versa, so TCD0 and TCD have direct relation with load conditions.

In this study, exergy and energy efficiency comparison of both power plants at different load conditions have been considered as represented in Fig.3. The energy and exergy efficiency based on first and second laws of thermodynamics for RAMIN steam power is more than BUSHEHR nuclear plant in different load conditions; however, we can divide two sections for analysis of energy and exergy comparison of both power plants. In section 1, that refer to 25-75 % load, both efficiency in both plants approach together when load condition increase to 75% (efficiency will be about 35% for BUSHEHR and 38% for RAMIN). Section 2 begins from 75% and be finished in full load condition that in this domain with increasing load condition form 75%, both efficiency in both plants almost be fixed.
Fig. 3. Exergy and energy efficiency comparison of BUSHEHR and RAMIN power plants at different load.

The details comparison of main parameters such as TED (total exergy destruction), EF (rate of exergy related to fuel) and Wnet (net power generation) in RAMIN and BUSHEHR plants have been illustrated in Fig. 4.

Fig. 4. EF, TED and net power comparison of BUSHEHR and RAMIN power plants at different loads.

As shown in Fig. 4, rate of exergy related to fuel in BUSHEHR is more than in RAMIN but with increasing load condition the difference between them is increased subsequently. In addition, net power generation in BUSHEHR is more than RAMIN in different loads; however load difference between net power in both plants is increased. Also, total exergy destruction (TED) in both plants is increased and difference between TED of BUSHEHR and RAMIN is increased.
The comparison of cost per unit exergy of product for turbine systems of both plants at different load conditions with and without considering capital cost have been demonstrated in Fig. 5 and Fig. 6 consecutively. As shown in these figures, $c_{P0}$ of HP steam turbine in BUSHEHR in full load is more than other turbines significantly; however between 25-60% loads, $c_{P0}$ and $c_p$ of BUSHEHR HP steam turbine are decreased, between 60-100% loads these parameters are increased. After $c_{P0}$ and $c_p$ of BUSHEHR HP steam turbine, high pressure steam turbine (STHP) in RAMIN has most $c_{P0}$ and $c_p$ than other turbines. Also, in domain 25-50% loads $c_{P0}$ and $c_p$ are decreased quickly, between 50-75% loads they are fixed and both parameters between 50-100% load are decreased slowly. RAMIN low pressure steam turbine (STLP) locates in next level and it behavior similar to STHP but sloop between 25-50% is gentler than STHP. RAMIN intermediate pressure steam turbine (STIP) locates in next level Also, between 25-50% these parameters are decreased, between 50-100% are almost fixed and between 75-100% are increased with very gentle sloop. Finally, BUSHEHR low pressure steam turbine (STLP) has lowest $c_{P0}$ and $c_p$.

**Fig. 5.** $c_{P0}$ comparison of BUSHEHR and RAMIN power plants at different loads.

**Fig. 6.** $c_p$ comparison of BUSHEHR and RAMIN power plants at different loads.
Comparison of cost per hour of exergy destruction of both plants at different load conditions with and without considering capital cost has been demonstrated in Fig. 7. As shown in these figure, cost exergy destruction (CD and CD₀) in BUSHEHR is more than cost exergy destruction in RAMIN. Both parameters are increased when load condition increase to full load. However but RAMIN sloop is gentler than BUSHEHR. Also, the energy level representation of BUSHEHR and RAMIN power plants at full load condition have been shown in Fig 8 and Fig 9 consequently. The interaction between different components in each power plant have been demonstrated.

Fig. 7. CD, CD₀ and net power comparison of BUSHEHR and RAMIN power plants at different loads.

Fig. 8. Energy Level Representation of BUSHEHR power plant
6. Conclusions

In this paper, exergy and exergoeconomic methods have been applied for analysis and comparison of a 315-MW steam gas fired and a 1076-MW nuclear steam cycle power plants in different load conditions.

An exergy-costing method has been applied to both cases to estimate the unit costs of electricity produced from steam turbines. The computer program that was developed which shows that the exergy and the thermoeconomic analysis presented here can be applied to any energy system systematically and elegantly. If correct information on the initial investments, salvage values and maintenance costs for each component can be supplied, the unit cost of products can be evaluated.

Although the overall picture of a system can be shown and major directions for improving the system performance can be identified from the above two levels of analysis, the maximum potential or the limit of improvement for individual units and processes are still uncertain, since the exergy loss analysis so far is based on the concept of total exergy loss. In some cases, the suggestions for promising modifications based on the total exergy loss may be misleading, since they do not consider the minimum exergy loss which is required to operate a process.

The combined pinch and exergy analysis or Energy Level Representation shows the maximum potential for improvement or limitations. There are three significant advantages of knowing the practical maximum potential and limit for improvement. First, the performance of a process and equipment can be evaluated based on the maximum potential which is achievable in current technical and economical conditions. The practical maximum potential for improvement defined as such distinguishes itself from the theoretical maximum potential, which cannot be realized either technically or economically. Therefore, the practical maximum potential indicates what can be done and what cannot be done in current conditions. Secondly, by knowing the practical maximum potential for improvement, a designer sets the target for improvement by making modifications. Different modifications can then be compared in terms of how much benefit can be achieved and what is the capital cost involved. Thirdly, any processes or units with very small potential for improvement can be immediately ruled out from consideration.
### Nomenclature

- \(c\) cost per unit exergy ($/MW)
- \(c_p\) molar specific heat capacity (J/kmol.K)
- \(C\) cost flow rate ($/hr)
- \(CP\) heat capacity (J)
- \(CRF\) capital recovery factor
- \(e\) exergy rate per mass (MW/kg)
- \(E\) specific exergy (MW)
- \(h\) specific enthalpy (kJ/kg)
- \(i\) interest rate
- \(m\) mass flow rate (kg/s)
- \(N\) Molar mass flow rate of saline water
- \(PWF\) Present forth factor
- \(PW\) Present worth
- \(p\) pressure (bar)
- \(R\) universal gas constant (bar.m³K⁻¹)
- \(s\) specific entropy (MW.K⁻¹)
- \(T\) temperature (°C)
- \(T_0\) ambient temperature (°C)
- \(W\) shaft work, electricity (MW)
- \(Z\) capital cost rate of unit ($/hr)
- \(TED\) Total exergy destruction
- \(TCD\) Total exergy cost of destruction
- \(STHP\) steam turbine high pressure
- \(STIP\) Steam turbine Intermediate pressure
- \(STLP\) Steam turbine low pressure
- \(GT\) gas turbine
- \(ST\) steam turbine

### Greek symbols

- \(\rho\) density (kg/m³)

### Superscript

- \(CI\) capital investment
- \(OM\) operating and maintenance cost
- \(ph\) Physical
- \(ch\) Chemical

### Subscript

- \(P\) Product
- \(f\) Fuel
- \(tot\) Total
- \(D\) Destruction
References


Economic and exergoeconomic analysis of micro GT and ORC cogeneration systems

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Abstract:
Exergoeconomic analysis is a powerful tool which can be used to analyse energy generation systems. However, due to its complexity it is rarely used, especially for smaller systems. These systems are usually assessed using traditional energy and economic analysis. The aim of this study is to compare the differences between the two approaches, i.e. economic and exergoeconomic. For that purpose a commercial micro gas turbine (GT) cogeneration system and a cogeneration system consisting of a gas turbine and micro organic Rankine cycle (GT-ORC) have been investigated. Results show that exergy costs of products calculated using exergoeconomic analysis depend on ambient reference temperature and these costs reduce when the temperature decreases. Heat energy costs, calculated using the exergy factor from exergy costs, increase slightly when the reference temperature decreases. It is also noted that exergoeconomic cost analysis may provide an attribution of costs which gives a better guide to the relative value of electricity and heat than economic cost analysis.

Keywords:
CHP; cogeneration; economic analysis; exergy analysis; exergoeconomic analysis; gas turbine; ORC

1. Introduction

Traditionally energy efficiency and economic analyses are used when energy conversion systems have to be assessed, particularly for investment purposes. Sometimes energy analysis is not sufficient, especially when different types of cogeneration systems are compared. Therefore, economic evaluation plays an important role in deciding which cogeneration system should be chosen. However, economic analysis cannot be relied upon when attributing costs to electricity and heat as it may not reflect their quality or value and alternative methods such as thermoeconomic analysis need to be used.

Thermoeconomic analysis combines economic and thermodynamic analysis by applying the concept of cost [1]. First attempts to combine exergy and costs were undertaken in early 1930’s [2]. Later many more thermoeconomic methods were developed. During the last decade several attempts to generalise all thermoeconomic methods were made. It led to the evolution of two thermoeconomic methods. One of them, Structural Theory of Thermoeconomics, is a standard mathematical formulation for all methodologies [3]. Another, Specific Exergy Costing (SPECO) is a methodology for defining and calculating exergy efficiencies and exergy related costs in thermal systems [4].

The SPECO method provides an unambiguous and systematic procedure for exergy cost calculations. This method has been used to study community energy supply systems [5] and micro cogeneration plants [6]. In this paper the investigation of micro cogeneration systems is extended. As in the previous papers the SPECO method has been used but the micro gas turbine cogeneration system (GT) has been modified by connecting an organic Rankine cycle turbine (GT-ORC). Then exergoeconomic analysis has been performed and the effect of ambient reference temperature on
the product costs has been investigated. Finally an economic assessment of both systems has been conducted from a cost and investment perspective.

2. System description

The GT and GT-ORC systems were modelled using the Cycle-Tempo modelling tool [7] with an ambient reference temperature of 288K ($T_0$) and a pressure of 101.3kPa ($P_0$) used for exergy calculations.

2.1. Micro GT cogeneration system

A micro gas turbine cogeneration system available on the market was chosen (Fig. 1). Air for combustion is compressed in the compressor (1) and passes through the recuperator (4). The temperature of the air is increased in the recuperator (4) and the hot air supplied to the combustion chamber (2), where air and gas mixture is burned. The gas expands in the gas turbine (3) and is supplied to the heat exchanger (5), passing by the recuperator (4), where it is cooled. In the heat exchanger (5), high temperature exhaust gas cools and heats water in the heating system (6). Pump (7) is used to circulate water in the heating system (6). Finally the cooled exhaust gas is delivered to the flue stack (8). Electrical energy generated in the gas turbine and heat energy generated in the heating system (6) are shown using arrows (13) and (15). Simulation data are presented in table 1.

![Fig. 1. Micro GT cogeneration system model [6]](image)

2.2. Micro GT-ORC cogeneration system

This consists of the micro GT-ORC system described in 2.1 and a commercially available ORC turbine [8] (Fig. 2).

The main difference with this cogeneration system is that in the heat exchanger (5) high temperature exhaust gas is used to heat water to +140°C under elevated pressure conditions. Pump (7) is used to circulate water between heat exchangers (5) and (8). In the heat exchanger (8) water heats the refrigerant pentafluoropropane (R245fa), which is used as working fluid in the ORC system. Cooled water is then supplied back to the heat exchanger (5). High pressure and temperature R245fa vapour is delivered to the turbine (9), where it expands. Exhaust vapour from
turbine (9) is condensed in the heat exchanger (10). Then the liquid is pumped to the heat exchanger (8) and the cycle repeats. Water in the heating system is heated from 50°C to 70°C in the heat exchanger (10) and is supplied to the heating system (15) using the circulation pump (14).

![Diagram of Micro GT-ORC cogeneration system model]

**Fig. 2. Micro GT-ORC cogeneration system model**

**Table 1. Micro GT and micro GT-ORC simulation data**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Micro GT</th>
<th>Micro GT-ORC</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fuel energy (HHV), kW</td>
<td>369.7</td>
<td>369.7</td>
</tr>
<tr>
<td>Electrical output, kW</td>
<td>102.1</td>
<td>112.1</td>
</tr>
<tr>
<td>Thermal output, kW</td>
<td>157.6</td>
<td>148.0</td>
</tr>
<tr>
<td>Exhaust gas temperature after GT recuperator, ºC</td>
<td>270</td>
<td>270</td>
</tr>
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<td>Gas temperature in chimney, ºC</td>
<td>70</td>
<td>70</td>
</tr>
<tr>
<td>Supply/return heating system temperature, ºC</td>
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<td>70°/50°</td>
</tr>
<tr>
<td>Supply/return water temperature for ORC cycle, ºC</td>
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<td>140°/115°</td>
</tr>
<tr>
<td>Maximum refrigerant vapour temperature, ºC</td>
<td>-</td>
<td>120°</td>
</tr>
</tbody>
</table>

In the GT-ORC system, electricity is generated in gas turbine (3) and ORC turbine (9) (electrical energy streams (24) and (31)), and heat is generated in heating systems (13) and (15) (heat energy streams (29) and (30)). Simulation data are presented in table 1.
3. Energy and exergy efficiency analysis

Energy and exergy efficiencies of the micro GT and micro GT-ORC systems including auxiliary pumps are shown in Table 2. Efficiencies are calculated using the higher heating value (HHV) of fuel.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Micro GT</th>
<th>Micro GT-ORC</th>
</tr>
</thead>
<tbody>
<tr>
<td>Electric energy efficiency, %</td>
<td>27.5</td>
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<tr>
<td>Heat energy efficiency, %</td>
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<tr>
<td><strong>Total exergy efficiency, %</strong></td>
<td><strong>35.5</strong></td>
<td><strong>37.6</strong></td>
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</tbody>
</table>

Notes: 1 - Including auxiliary power consumption

The total energy efficiency of the GT-ORC system is slightly lower than that of the GT system. However, more electrical energy is generated in the GT-ORC system. The reduction of heat exergy efficiency in the GT-ORC system by 0.4% is accompanied by a 2.5% increase in electrical exergy efficiency. Consequently, the total exergy efficiency of the GT-ORC system is higher than that of the GT system.

4. Exergoeconomic analysis

In order to perform exergoeconomic and economic analysis, capital costs were required. The total cost of the micro GT system was provided by the manufacture and component costs were then estimated based on the cost functions provided by Galanti and Massardo [9] supplemented by manufacturers’ quotes. The total cost of the micro GT system is €130.2 k.

The costs of components: heat exchangers, ORC turbine and pumps for the GT-ORC system, were obtained from the manufacturers’ pricelists. The total cost of the micro GT-ORC system is €156.7 k.

The following assumptions were made for both cogeneration systems:
- 10% return on investment,
- 15 years investment repayment period,
- 7000 hours annual operating hours,
- operation and maintenance costs are not included,
- 2.31 c/kWh fuel price (HHV) based on natural gas for industrial consumers in UK in 2011 (taken from EU Energy Portal (www.energy.eu)).

4.1. Exergoeconomic calculations

The Specific Exergy Costing (SPECO) method [4] was used in this study. At first, identification of the exergy streams was carried out using Cycle-Tempo software. Total exergy was used in this study because the use of separate forms of exergy, such as: thermal, mechanical or chemical, only marginally improves calculation accuracy [4]. The next step, definition of products and fuels, was carried out using the methodology described by Tsatsaronis [10, 11]. Finally the last step, construction of cost equations, was carried out by constructing cost equations for each component in the model and by formulating auxiliary cost equations. More detailed description of application of SPECO method can be found in [5, 6].
Using the exergoeconomic approach cost rates of exergy destruction, capital cost rates and exergoeconomic factors of the micro GT and micro GT-ORC systems were calculated. The exergoeconomic factor was calculated using equation [11]:

$$f_k = \frac{\hat{f}_k}{(\hat{Z}_k + C_{D,k})}$$  \hspace{1cm} (1)

Here $f_k$ – exergoeconomic factor of k-th component (%), $\hat{Z}_k$ – capital cost rate (€/h), $C_{D,k}$ – cost rate of exergy destruction based on fuel (€/h). The exergoeconomic factor shows the contribution of non-exergy related cost (investment cost) to the total cost increase. A low value indicates that a higher cost of the component would be acceptable if the exergy destruction were reduced. A high value of $f_k$ indicates the cost of the component should be reduced, even if the exergy efficiency of the component decreases.

Results of calculation of the GT system are presented in Fig. 3. It is seen that the cost rate of exergy destruction in the combustor is the highest. The exergoeconomic factor of the combustor is low, which suggests further investment in this component may be justifiable if higher exergy efficiency were achieved.

![Fig. 3. Cost rate and exergoeconomic factors of micro GT system components](image)

The cost rate of exergy destruction of turbine, heat exchanger (5) (HEX5) and compressor are similar. The exergoeconomic factor of turbine is around 34%. This suggests that a less efficient gas turbine produced at lower cost can be chosen, if its exergy efficiency were not significantly reduced.

The cost rate of exergy destruction (0.81 €/kWh) of the recuperator (4) (HEX4) is low compared with other system components. However its capital cost rate (0.89 €/kWh) is significantly higher than that of other components. The exergoeconomic factor of the recuperator (4) is about 52%, which indicates that its capital cost should be reduced even if the exergy efficiency decreased.

Results of calculation of the GT-ORC system are presented in Fig. 4. It is seen that the cost rate of exergy destruction in the combustor and turbine are the highest, as in the GT system.

A similar cost rate of exergy destruction of the recuperator (4) (HEX4) is observed in both the GT (Fig. 3) and GT-ORC systems (Fig. 4). However the cost rate of exergy destruction of the heat exchanger (5) (HEX5) is significantly lower in the GT-ORC system (Fig. 4) compared with the GT system (Fig. 3).

A single heat exchanger is used to reduce the temperature of the flue gas in the GT system (heat exchanger (5), Fig. 1) whereas two heat exchangers (5) and (6) are used in the GT-ORC system (Fig. 2). The cost rate of exergy destruction of the heat exchanger (5) in the GT system (1.94 €/h, Fig. 3) is higher than the sum of two cost rates of exergy destruction of the heat exchangers (5) and
It shows that gradual reduction of the exhaust gas temperature is more beneficial than sudden reduction of temperature using one heat exchanger.

There are more components, such as: heat exchangers (6), (8) and (10), turbine (9) and pump (11) and (14) (Fig. 2), which contribute to the increase of the total cost rate of exergy destruction in the GT-ORC system compared with the GT system. The contribution of pumps (7) and (12) (Fig. 2) to the cost of exergy destruction formation is negligibly small. However, the total cost rate of exergy destruction of the GT system (11.46 €/h) is higher than that of the GT-ORC system (11.30 €/h). The decrease exergy destruction cost is obtained because the additional ORC system is used. However, the use of additional system increases the total cost rate, which is the sum of cost rates of exergy destruction and capital costs. The total cost rate is 13.61 €/h in the GT system and 13.93 €/h in the GT-ORC system.

One of the main objectives of exergoeconomic analysis is to evaluate the production costs of products in an energy conversion system [2]. The exergy costs of electricity and heat generated in the micro GT and micro GT-ORC systems are shown in Figure 5. It is seen that electricity exergy costs of the GT system is lower than the average cost of the GT-ORC system. However, the heat exergy cost of the GT is higher than the average heat exergy cost of the GT-ORC system. The exergy cost of electricity generated in the ORC turbine of GT-ORC system is significantly higher than the exergy cost of electricity generate in the gas turbine. The reason for that is low electrical efficiency of the ORC.

Fig. 4. Cost rate and exergoeconomic factors of micro GT-ORC system components

Fig. 5. Exergy costs
4.2. Calculation of energy cost based on exergoeconomic analysis

Exergoeconomic analysis is a powerful tool which allows an understanding of the cost formation process and the calculation of the costs of each product. However, exergy is not a commodity. Therefore, exergy cost calculated using exergoeconomic analysis, has little practical value. In order to use the exergoeconomic method for practical calculations exergy costs must be converted to energy costs.

When conducting exergy and exergoeconomic analysis the reference ambient temperature $T_0$ must be carefully selected as any variation affects the exergy efficiency of the system. In this study exergy and exergoeconomic analysis were conducted using an ambient reference temperature $+15^\circ C$. Other ambient reference temperatures of $+9^\circ C$ and $+4^\circ C$ were explored to understand how exergy costs are affected for the GT system only.

In order to calculate the heat energy costs from heat exergy costs the exergy factor $\tau$ is used determined from:

$$\tau = \frac{E}{Q} = 1 - \frac{T_0 \ln \frac{T_S}{T_R}}{T_S - T_R}$$

Here $E$ is heat exergy; $Q$ is heat energy, $T_0$ (K) is ambient reference temperature, $T_S$ (K) is heating system supply water temperature and $T_R$ (K) is heating system return water temperature. Calculated energy and exergy costs at different reference temperature for micro GT system are shown in Fig.6.

It is seen that exergy costs of electricity and heat reduces when the lower ambient reference temperature is used with a larger reduction in heat cost than electricity. This is because there is an increased quantity of exergy when the lower reference temperature is used. Electricity energy cost is equal to its exergy cost whereas heat energy cost increases with temperature due to the augmentation of exergy factor. In general the change of reference temperature has little effect on electricity and heat energy costs.
The energy costs, calculated using exergoeconomic analysis at ambient reference temperature +15°C for the GT and GT-ORC systems are presented in Fig. 7. These costs ensure that all fuel and capital costs are recovered. As expected, electricity energy costs are the same as exergy costs but it can be seen that the heat energy costs, calculated using exergoeconomic analysis, are significantly lower than heat exergy costs.

5. Economic analysis

Exergy analysis can identify elements of the system where the exergy destruction is the greatest and exergoeconomics enables the costing of that destruction to be performed. From this the scope for improvements can be identified. However, this costing may bear little relationship to traditional economic analysis which will ultimately form the basis of any investment case. The economic analysis of both co-generation systems is presented here and then compared with those from the exergoeconomic analysis.

The following assumptions have been made:

1. The main objective of the investment is for electricity production with heat as a by-product. Hence heat production is marginal to electricity production. So only the capital and variable costs which can be solely attributed to heat are allocated to heat.
2. The asset capital costs which can be attributed to heat production are:
   a. GT system - heat exchanger 5 and pump 7 with a total cost of €2200.
   b. GT-ORC system – heat exchangers 6 and 10, pumps 12 and 14 with a total cost of €5500.
3. Overheads, operating and maintenance costs have not been included. Otherwise all other assumptions are the same as those for the exergoeconomic analysis.

5.1. Cost analysis

This comprises the fixed and variable costs. Fixed costs are the capital costs associated with the investment. In practice there will be other fixed costs such as overheads but for the purpose of this analysis they have not been included. The capital repayment is based on 10% return on investment with a 15 year repayment period and 7000 hours/years of operation. Cost attribution of heat is determined from those costs that can be directly attributed to heat only with the remainder to electricity.
Table 4. Costs of micro GT and GT-ORC cogeneration systems based on economic analysis

<table>
<thead>
<tr>
<th></th>
<th>Micro GT</th>
<th>Micro GT-ORC</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total capital cost, €</td>
<td>130200</td>
<td>156700</td>
</tr>
<tr>
<td>Heat asset costs, €</td>
<td>2200</td>
<td>5500</td>
</tr>
<tr>
<td>Fixed electricity cost, c/kWh</td>
<td>2.35</td>
<td>2.53</td>
</tr>
<tr>
<td>Fixed heat cost, c/kWh</td>
<td>0.03</td>
<td>0.07</td>
</tr>
<tr>
<td>Total fuel cost, €/a</td>
<td>59780</td>
<td>59780</td>
</tr>
<tr>
<td>Variable electricity cost, c/kWh</td>
<td>8.36</td>
<td>7.62</td>
</tr>
<tr>
<td>Variable heat cost, c/kWh</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td><strong>Total electricity cost, c/kWh</strong></td>
<td><strong>10.72</strong></td>
<td><strong>10.15</strong></td>
</tr>
<tr>
<td><strong>Total heat cost, c/kWh</strong></td>
<td><strong>0.03</strong></td>
<td><strong>0.07</strong></td>
</tr>
</tbody>
</table>

Variable costs are the fuel costs associated with plant operation. In practice there will be other running costs such as plant maintenance but for the purpose of this analysis they have not been included. Fuel is converted to produce both electricity and heat and so the methodology for attributing fuel needs to be considered. For both of these cogeneration systems heat production has no impact on fuel input or electricity output and as a consequence the variable cost of heat production is zero. Thus the fuel cost must be fully attributed to electricity production. The fixed, variable and total costs for heat and electricity are shown in Table 4.

5.2. Investment analysis

The results of the cost analysis do not permit a decision to be made in terms of choice of cogeneration system or whether or not to proceed with either investment. This is because the costing of the electricity and heat products may not reflect their market value. For example, the heat cost shown in table 4 is less than 0.1 c/kWh, which places a very low value on heat.

For an investment to be justified the economic analysis needs to include an assessment of the market value of the products in order to determine the potential sales revenue. For the purposes of this analysis the electricity price 10.9 c/kWh has been obtained from the EU Energy Portal for industrial customers. Heat prices are not published and so an estimate has been made from the production cost of heat using a gas boiler based on an efficiency of 80% and using the same gas price as used for the cogeneration systems to give 2.89 c/kWh. Table 5 shows the results of the investment analysis.

Table 5. Investment analysis of micro GT and GT-ORC cogeneration systems

<table>
<thead>
<tr>
<th></th>
<th>Micro GT</th>
<th>Micro GT-ORC</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total revenue, €k pa</td>
<td>110</td>
<td>115</td>
</tr>
<tr>
<td>Capital repayment, €k pa</td>
<td>17</td>
<td>21</td>
</tr>
<tr>
<td>Fuel cost, €k pa</td>
<td>61</td>
<td>60</td>
</tr>
<tr>
<td><strong>Total cost, €k pa</strong></td>
<td><strong>77</strong>*</td>
<td><strong>81</strong>*</td>
</tr>
<tr>
<td>Gross margin (Total revenue – Total costs), €k pa</td>
<td>33</td>
<td><strong>35</strong>*</td>
</tr>
<tr>
<td>Project Net Present Value (NPV), €k</td>
<td><strong>250</strong></td>
<td><strong>266</strong></td>
</tr>
</tbody>
</table>

* Rounding applied.

The investment analysis shows that both projects have a positive NPV and has also identified that the GT-ORC generation system has a higher NPV and on this criterion only should be preferred.
5.3. Comparison of exergoeconomics with economic analysis

Both exergoeconomic and economic energy costing analysis ensure all cogeneration costs (i.e. capital and fuel) are fully recovered in their attribution to the electricity and heat produced. However, it is can be seen from table 6 that the cost attribution is very different.

<table>
<thead>
<tr>
<th>Table 6. Electricity and heat costs from exergoeconomic and economic analysis of micro GT and GT-ORC cogeneration systems.</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Exergoeconomic cost</strong></td>
</tr>
<tr>
<td>Electricity, c/kWh</td>
</tr>
<tr>
<td>Heat, c/kWh</td>
</tr>
<tr>
<td><strong>Economic cost</strong></td>
</tr>
<tr>
<td>Electricity, c/kWh</td>
</tr>
<tr>
<td>Heat, c/kWh</td>
</tr>
<tr>
<td><strong>Market price (based on prices for UK industry)</strong></td>
</tr>
<tr>
<td>Electricity, c/kWh</td>
</tr>
<tr>
<td>Heat, c/kWh (estimated)</td>
</tr>
</tbody>
</table>

In a commercial environment an investment decision is more likely to be dependent on the market value of the product streams but it is important to note that these may bear no relationship to the costs derived from either methodology. This in itself can be problematic as energy prices have been very volatile [12] and substantial changes in market prices could continue to be seen over the economic life of the cogeneration system.

It is worthwhile noting that the exergoeconomic cost of electricity relative to heat is comparable to that of market price of electricity to heat and thus seems to provide a better estimate of heat value than that from economic cost analysis.

6. Conclusion

Exergoeconomic and economic analysis of micro GT and GT-ORC systems was conducted and the effect of ambient reference temperature on the exergy costs of products was investigated. The results of this study show:

- Exergoeconomic analysis can assist in identifying improvements in performance and capital costs.
- Exergy costs fall with a reduction in reference temperature.
- Exergoeconomic analysis can be used to ensure all cogeneration costs are fully recovered through their attribution to electricity and heat.
- The change of reference temperature has little effect on electricity and heat energy costs when calculated using exergoeconomic analysis and exergy factor.
- Exergoeconomic cost analysis may provide an attribution of costs which gives a better guide to the relative value of electricity and heat than economic cost analysis.
- Economic cost analysis is fundamental to any investment case but it is not affected by the quality or value of the product streams. Hence it must be supplemented by the market value of the energy products in order to enable an investment decision to be made.
Acknowledgments

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References

Exergoeconomic comparison of wet and dry cooling technologies for the Rankine cycle of a solar thermal power plant

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Abstract:
This work describes and analyses the Rankine cycle of the 50 MW parabolic trough power plant Andasol 1 in the Spanish region of Andalusia, a prime example of the use of solar energy for electricity generation. By means of an exergoeconomic analysis, all components of the thermal plant are considered individually. Thermodynamic inefficiencies within the system are located, quantified and economically evaluated. Generally, various wet and dry cooling methods come into consideration for the re-cooling of the steam. Andasol 1 uses the more effective wet cooling method, because in the vicinity sufficient water reserves are available. In this work, the water-cooled system is replaced by an air-cooled condenser and the two variants are compared under exergoeconomics aspects. Also the impact of a construction of Andasol 1 in a hotter and drier climate than the northern Sahara is simulated.

The analysis shows that within the air-cooled condenser much more exergy is destroyed than within the wet cooling system. Thereby the achievable condenser pressure is one of the critical process parameters. The electricity production cost in the zone around the Andalucian Granada are 15.27 ct/kWh when using wet cooling and 16.08 ct/kWh in case of dry cooling. In the Sahara, these costs increase to 15.52 ct/kWh for wet cooling and 17.52 ct/kWh for dry cooling technology.

Keywords:
Exergy analysis, Exergoeconomic analysis, Thermal power plant, Rankine cycle, Cooling tower, Air cooled condenser.

1. Introduction
For the last few years, the energy industry finds itself in a noticeable change. Sustainable, safe and resource-saving energy will play a key role in the coming years to keep nature and environment safeguarded for future generations and to strengthen and expand the current prosperity. This requires that the renewable energy conversion methods have to be optimized and distributed and further practical experiences have to be collected.

This paper describes and analyses the parabolic trough power plant Andasol 1, a prime example of the use of solar energy for electricity generation. In a steam cycle (Rankine cycle) heat is converted into mechanical energy. This is done in a steam turbine which is coupled to a generator to produce electricity. Once the steam has passed through the turbine, it is condensed by heat transfer to the ambient. Generally, various wet and dry cooling methods come into consideration for the re-cooling of the steam. For wet cooling a cooling water cycle including cooling tower, cooling pump and water-cooled condenser is the most common method. For dry cooling an air-cooled condenser is the most used variant.

In this paper the Rankine cycle of the power plant is simulated and analyzed by using the exergoeconomic method. Furthermore, the water-cooled system which is used in the real Andasol 1 plant (Configuration 1) is replaced by an air-cooled condenser (Configuration 2) and the two variants are compared under exergoeconomics aspects. Also the impact of a construction of Andasol 1 in a hotter and drier climate as the northern Sahara is simulated.
In an exergoeconomic analysis, all components of the thermal plant are considered individually. Thermodynamic inefficiencies within the system can be located, quantified and economically evaluated. Thus, the process variants presented in this work can be compared to each other objectively and suggestions for improvements to reduce electricity generation costs can be made.

One exergy analysis of Andasol 1 power plant with wet and dry cooling at two different condenser pressures has already been made in a previous paper Blanco-Marigorta et al. [1]. This analysis is optimized and economic and exergoeconomic analysis are added.

Related to solar-thermal power plants, Singh et al [2] presented a second law analysis based on an exergy concept for a solar thermal power system. Singh et al evaluated the respective losses as well as exergetic efficiency for typical solar thermal power systems under given operating conditions. They found that the main energy loss takes place in the condenser. Their exergy analysis shows that the collector–receiver assembly is the part where the losses are maximal. Gupta and Kaushik [3] carried out the energy and exergy analysis for the different components of a proposed conceptual direct steam generation solar-thermal power plant. In Kaushik et al [4], a 35 MW solar thermal power plant was analyzed with the aid of exergoeconomics.

2. Description of the plant

The parabolic trough power plant Andasol 1 has a net power capacity of 50 MWe. The power cycle is a conventional reheat design with 5 closed and 1 open extraction feedwater heaters. A schematic of the Rankine Cycle of the plant is shown in Fig. 1. The streams S50 and S52 come from the solar part and the streams S51 and S55 go back to the solar part.

GateCycle [5] software has been used in this work for simulation purposes. Real operation parameters have been introduced in the simulation.

Thermo oil VP1 is used as heat transfer fluid: it is heated up in the solar collectors and cooled down while producing steam in the steam generator. A part of this heat transfer fluid is also used to heat up the molten salt-storage tanks. The hours at which the plant runs with the stored heat in the heat tanks is considered in the total annual time of system operation at full load. The steam generation system consists of two parallel heat exchanger trains (preheater (ECON1)/ evaporator (EVAP1)/ superheater (SPHT1)) and two reheaters (SPHT2), again connected in parallel. The produced superheated steam enters the high-pressure turbine with a pressure of 100 bar, a mass flow of 60.34
kg/s and a temperature of 373 °C. A conventional Rankine cycle begins. After expanding in the high-pressure turbine, the steam is reheated and enters the low-pressure turbine with a pressure of 16.5 bar and a temperature of 373 °C. To preheat feedwater in one of the five feedwater heaters, one extraction is taken from the high-pressure turbine. After expanding in the low-pressure turbine, the steam reaches the condenser at a pressure of 0.063 bar. In the low-pressure turbine, five steam extractions take place. One goes to the deaerator, the other four preheat the feedwater in the remaining four feedwater heaters. In the condenser the steam is cooled down by means of wet cooling technology. The feedwater pump leads the deaerated water through two more consecutive feedwater heaters and to the heat recovery steam generator, where the cycle closes. Thermodynamic parameters of the most important streams can be seen in Table 1.

In contrast to other solar thermal trough projects, the electricity production of Andasol 1 can be adjusted to the demand thanks to a molten salt thermal storage system. A part of the thermal energy received by the solar field during the day is stored in one of two liquid salt-tanks. A full storage tank, which holds 1 010 MWh of heat, can generate power at full load for 7.5 hours [6].

<table>
<thead>
<tr>
<th>Stream</th>
<th>( \dot{m} ), kg/s</th>
<th>( p ), bar</th>
<th>( T ), °C</th>
<th>( \dot{E}^{PH} ), kW</th>
<th>( \dot{E}^{CH} ), kW</th>
<th>( \dot{E}^{TOT} ), kW</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>60.08</td>
<td>100.0</td>
<td>373.0</td>
<td>72 158</td>
<td>150</td>
<td>72 308</td>
</tr>
<tr>
<td>2</td>
<td>53.87</td>
<td>18.5</td>
<td>208.5</td>
<td>47 053</td>
<td>135</td>
<td>47 187</td>
</tr>
<tr>
<td>3</td>
<td>53.87</td>
<td>16.5</td>
<td>373.4</td>
<td>57 789</td>
<td>135</td>
<td>57 923</td>
</tr>
<tr>
<td>4 (Conf. 1)</td>
<td>40.98</td>
<td>0.063</td>
<td>37.1</td>
<td>3 495</td>
<td>102</td>
<td>3 598</td>
</tr>
<tr>
<td>4 (Conf. 2)</td>
<td>41.18</td>
<td>0.075</td>
<td>40.1</td>
<td>4 404</td>
<td>103</td>
<td>4 507</td>
</tr>
<tr>
<td>5 (Conf. 1)</td>
<td>48.47</td>
<td>0.063</td>
<td>37.1</td>
<td>43</td>
<td>121</td>
<td>165</td>
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<tr>
<td>5 (Conf. 2)</td>
<td>48.47</td>
<td>0.073</td>
<td>39.7</td>
<td>66</td>
<td>121</td>
<td>188</td>
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<tr>
<td>13</td>
<td>60.08</td>
<td>103.0</td>
<td>235.0</td>
<td>13 945</td>
<td>150</td>
<td>14 096</td>
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<tr>
<td>26</td>
<td>7.48</td>
<td>0.370</td>
<td>42.2</td>
<td>14</td>
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<td>33</td>
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<tr>
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<td>2 814</td>
<td>2 814</td>
<td>36.3</td>
<td>2663</td>
<td>7 035</td>
<td>9 698</td>
</tr>
<tr>
<td>31 (Conf. 1)</td>
<td>2 814</td>
<td>2 814</td>
<td>28.7</td>
<td>237</td>
<td>7 035</td>
<td>7 272</td>
</tr>
<tr>
<td>37 (Conf. 1)</td>
<td>2 814</td>
<td>2 814</td>
<td>28.7</td>
<td>500</td>
<td>7 036</td>
<td>7 535</td>
</tr>
<tr>
<td>40 (Conf. 1)</td>
<td>1 645</td>
<td>0.948</td>
<td>28.1</td>
<td>72 304</td>
<td>12 691</td>
<td>84 995</td>
</tr>
<tr>
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<tr>
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<td>471 955</td>
<td>82 839</td>
<td>554 794</td>
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<tr>
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<td>10 740</td>
<td>0.948</td>
<td>309.7</td>
<td>474 166</td>
<td>82 839</td>
<td>557 004</td>
</tr>
</tbody>
</table>

*neglected

The wet cooling system of the solar trough power plant Andasol 1 consists of five induced draft cooling tower cells, three cooling pumps and a surface condenser (Conf. 1). To cool down the steam of the Rankine cycle and for other water-consuming equipment, 870 000 m³ of water of the nearby Sierra Nevada mountains are required a year. In water scarce regions, a hypothetic alternative is dry cooling technology by means of air cooled condensers (ACC) (Fig. 2). In the process with air cooled condenser no further cooling tower and no cooling water pump is needed (Conf. 2). However, heat transfer by forced air convention is less effective than evaporative heat transfer; therefore, larger heat exchanger areas and greater fan power are required to achieve the same heat rejection.
3. Thermodynamic evaluation

3.1. Basic assumptions for the thermodynamic analysis

A representative steady flow process at design conditions is analyzed. Start-up, shutdown and part load procedures and processes are not subject of this work. A constant solar radiation (900 W/m²) which allows running the plant at design conditions is assumed [7].

In the Rankine cycle only the components shown in figure 1 and 2 are taken into consideration. All other components such as valves and auxiliary systems are neglected. The same applies to the solar part. It is treated as one component including the solar field, the expansion and overflow tank, the main HTF-pumps, the HTF anti-freeze system, the HTF piping system, the salt tanks, the heat exchangers between oil and salt and the salt circulation pumps. It is assumed that in the pipes connecting the main components no losses take place.

The cooling tower cells are designed in GateCycle as close as possible to the ones used in the real power plant. Main plant operation data [8] were fed to the software as input variables. The theoretical cooling alternative of an air cooled condenser is designed with an ACC calculator [9].

The net work output is the net electricity fed into the grid. The plant’s consumption due to solar field tracking, auxiliary components, HTF- and liquid salt-pump are estimated to 4 250 kW [10].

3.2. Energy assessment

The pressure and the temperature of the steam both entering and leaving the turbine define, in large part, the efficiency of the Rankine Cycle. The steam conditions at the turbine outlet are defined by the temperature at which the steam is condensed and the latent heat of vaporization can be transferred to the environment. The lower the ambient temperature, the lower the feasible operation pressure of the condenser ($p_{CND}$) and the higher the energetic efficiency of the cycle. For wet cooling technologies, the lowest ambient temperature available is the wet bulb temperature since an evaporation process is used to provide the cooling water source for the condenser. Taken into account the ambient conditions of the region where Andasol 1 is located, $p_{CND}$ was set to 0.063 bar.

Dry cooling technologies reject heat to the environment at the dry bulb temperature. Therefore, first of all, an optimal condenser pressure has to be found. Fig. 3 shows the condenser pressure as a function of the net work output of the process using either a wet cooling or a dry cooling technology in the Rankine Cycle of Andasol 1 at design conditions. The ambient temperature in Granada is assumed to be 28.05 °C and the relative air humidity is assumed to be 60 %.

For this purpose, in calculations, air mass flow streaming through ACC and water flow streaming through the wet condenser have been adjusted. In both cases all other parameters such as steam...
mass flow or ambient conditions are kept constant. It is assumed that the design condenser pressure of 0.063 bar used in the real power plant is the lowest reachable condenser pressure for this plant.

Rankine cycle with ACC shows a maximum net power value of 47.37 MW corresponding to an optimum pressure value of 0.073 bar (at lower pressures, additional fan power outweighs additional electricity production by the steam turbine). This output is 2.54 MW lower compared to the possible maximum in configuration with wet cooling. From this pressure operation value on, the higher the condenser pressure, the more both curves converge and the lower gets the efficiency of the cycle.

According to GEA’s ACC calculator [9], 15 fan modules divided in 3 A-frames are required for this optimum configuration. ACC’s plot area is 2 100 m² and inlet height is 18 m; the diameter of each fan is 9.75 m.

To reach a worse condenser pressure, dry cooling system consumes over three times more electricity (2 660 kW) in confront to the wet cooling system (810 kW). This shows that from an energetic point of view a wet cooling system would be much more efficient and using a dry cooling system could only be justified with a lack of a water source close to the plant.

The dry ambient temperature, the wet ambient temperature and the relative humidity of the ambient change within a year. Therefore, also condenser pressure may change to avoid a too high power consumption of the fans in the cooling tower cells as well as in the ACC.
The diagram in Fig. 4 (a) shows the net electrical power output of both configurations in a year. Fig. 4 (b) shows the consumption of the fans in the same period; a polynomial trend line of the development of ACC’s fan-consumption is added.

Fig. 4 shows that the ambient temperature during the year has a high impact on the effectiveness of ACC fans and electric power output. On the contrary, net electric output of configuration with wet cooling technology does not change significantly over the year. For dry cooling technology, condenser pressure has to be increased in summer to avoid a high consumption of ACC-fans.

At outside temperatures over about 20 ºC the difference in net power output between the two configurations gets higher than 1 MW. In the simulated extreme case of the Sahara (45 ºC, 20 % relative air humidity) there is even a difference of 5.6 MW of net power output between the two configurations.

4. Exergoeconomic Analysis

An exergoeconomic analysis, also called thermoeconomic analysis, is a combination of exergy and economic analysis. Therefore, a complete exergoeconomic investigation consists of an exergy analysis, an economic analysis and a thermoeconomic analysis [11]. The questions how much thermodynamic inefficiencies in the process cost and which variables have to be changed in order to obtain a cost-effective system are answered [12].

4.1. Exergy Analysis

An exergy analysis is the first step for the exergoeconomic analysis. With an exergy analysis it is possible to get to know the thermodynamic inefficiencies in a thermal system. The location, the size and the sources of the inefficiencies can be discovered. The exergy of each stream, the exergy losses, the exergy destruction and the exergetic efficiency can be identified.

The exergy of the flow streams was calculated according to the definitions given in [11]. In this work, the conditions of thermodynamic environment are set to $T_0 = 298.15$ K and $p_0 = 1.013$ bar [15]. This implicates that thermodynamic environment and ambient conditions are not equal. Kinetic and potential exergy are neglected and chemical exergy is calculated with Ahrendts [13].

Thermodynamic properties of Water are calculated from IAPWS 97 [14] Air is presumed to be a mixture of $N_2$, $O_2$ and $H_2O$ (60 % relative air humidity) treated as ideal gases and neglecting all other substances. Enthalpy and entropy of air is taken from [15]. Thermo oil VP1 is a mixture of 73.5 % biphenyl oxide and 26.5 % diphenyl [16]. The physical exergy of this fluid has already been calculated [1]; the chemical exergy of the HTF has been neglected because it does not play a role.

Table 2 shows the values of exergy rates associated with fuel and product, exergy destruction, $\dot{E}_{D,k}$, exergetic efficiencies, $\epsilon_k$, as well as the exergy destruction ratio, $y_{D,k}$, for the Rankine Cycle of the system. Solar part is excluded.

It is not possible to define exergy rates associated with fuel and product of the dissipative components CT1, CND1 and ACC1. To be able to define an exergetic product and an exergetic fuel of the two cooling systems and to make them more comparable, both cooling variants are packed together in a system with the low pressure steam turbine ST2&3 [17, 18]. Exergy rates associated with fuel and product as well as exergy destruction for a system with LP turbine and wet cooling system are defined as follows:

\[
\dot{E}_{F, ST2&3+WCS} = \dot{E}_{3} + \dot{E}_{26} - \dot{E}_{5} - \dot{E}_{18} - \dot{E}_{20} - \dot{E}_{21} - \dot{E}_{23} - \dot{E}_{25} \quad (1)
\]

\[
\dot{E}_{P, ST2&3+WCS} = \dot{W}_{ST2&3} + \dot{W}_{PUMP3} - \dot{W}_{FAN, CT} \quad (2)
\]

\[
\dot{E}_{L, ST2&3+WCS} = \dot{E}_{32} + \dot{E}_{33} + \dot{E}_{36} + \dot{E}_{41} - \dot{E}_{35} - \dot{E}_{40} \quad (3)
\]
For a system with LP turbine and dry cooling system, instead:

\[
\dot{E}_{F,ST2&3-DCS} = \dot{E}_s + \dot{E}_{26} - \dot{E}_5 - \dot{E}_{18} - \dot{E}_{20} - \dot{E}_{21} - \dot{E}_{23} - \dot{E}_{25} \tag{4}
\]
\[
\dot{E}_{P,ST2&3-DCS} = \dot{W}_{ST2&3} - \dot{W}_{FAN,ACC} \tag{5}
\]
\[
\dot{E}_{L,ST2&3-DCS} = \dot{E}_{41} - \dot{E}_{40} \tag{6}
\]

The exergetic efficiency of the system which includes low pressure steam turbine and wet cooling system \( \varepsilon_{ST2&3-WCS} \) is 79.8 % while with dry cooling \( \varepsilon_{ST2&3-DCS} \) is 74.5 %. This difference and the fact that \( \dot{E}_{D,ACC1} \) (4 806 kW) is more than twice as high as \( \dot{E}_{D,CND1} \), \( \dot{E}_{D,CT1} \) and \( \dot{E}_{D,PUMP3} \) together (1 865 kW), shows among others that from an exergetic point of view, the air cooled condenser is much more ineffective than the water-cooled system.

The exergy destruction ratio \( \dot{y}_{D,CT1+CND1+PUMP3}^* \), which is the quotient of the exergy destruction in CND1, CT1 as well as PUMP3 and the total exergy destruction in configuration 1 is 9.7 %. In configuration 2, \( \dot{y}_{D,ACC1}^* \) is 21.9 %.

It should be noted that the biggest exergy destruction in the whole process of the parabolic trough power plant takes place in the solar part. If the exergy destruction within the solar part (387 021 kW) is included in the calculation of total exergy destruction, \( \dot{y}_{D,SP}^* \) is 95.3 % in configuration 1 and 94.6 % in configuration 2. Therefore exergy destruction of the Rankine cycle is only about 5 % in both configurations. The exergetic efficiency of the solar part is around 16.5 % in both configurations. Thereby it is assumed that the only exergetic product of the solar part is to heat the streams 2, 13, 14 and 15.
4.2. Economic Analysis

The aim of an economic analysis as a part of a thermoeconomic analysis is to relate entire costing on the total annual time of system operation at full load. Levelized carrying charges, \( CC_l \), and operating and maintenance costs, \( OMC_l \), can be assigned to components according to the relative fraction of the \( k \) th component associated to the purchased-equipment costs. In this paper is used the total revenue requirement method, TRR [17].

Therefore the cost rates linked with capital investment \( Z_{CI} \) as well as the cost rate coupled with operating and maintenance cost \( Z_{OMC} \) each related to the \( k \) th component have been calculated. The sum of these terms is defined as \( Z_k \):

\[
Z_k = Z_{CI} + Z_{OMC}
\]

Parameters for the economic calculation are given in Table 3. Depreciation is linear. Income taxes and costs of licensing, research and development are neglected. Overnight construction is assumed. Calculations will be made in constant 2011 € (without inflation).

Table 3. Basic conditions for economic calculations

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Average annual inflation rate, ( r_i )</td>
<td>0 %/a</td>
</tr>
<tr>
<td>Average annual nominal discount rate, ( i_{eff,n} )</td>
<td>7 %/a</td>
</tr>
<tr>
<td>Nominal average annual escalation rate for fuel, ( r_{n,FC} )</td>
<td>4 %/a</td>
</tr>
<tr>
<td>Nominal average annual escalation rate for other costs, ( r_n )</td>
<td>3 %/a</td>
</tr>
<tr>
<td>Average annual real discount rate, ( i_{eff,r} )</td>
<td>3.92 %/a</td>
</tr>
<tr>
<td>Real average annual escalation rate for fuel, ( r_{r,FC} )</td>
<td>0.98 %/a</td>
</tr>
<tr>
<td>Real average annual escalation rate for other costs, ( r_r )</td>
<td>1.96 %/a</td>
</tr>
<tr>
<td>Full load hours</td>
<td>3600 h/a</td>
</tr>
<tr>
<td>Economic life time</td>
<td>40 years</td>
</tr>
</tbody>
</table>

The costs of the generator which converts mechanical into electrical energy are neglected. In contrast to conventional power plants, fuel costs of solar radiation are equal to zero. For part load situations, an auxiliary fossil fuel is necessary. In case of Andasol 1, liquefied natural gas or liquefied petroleum gas is used [8]. But because at base load there is not used a co-firing gas stream, fuel costs have to be treated in a special manner. In this analysis they are apportioned to the components, a similar procedure as for the operating and maintenance costs.

The total capital investment of both configurations is the same (314.2 million €), because the package consisting of cooling tower, cooling pump and wet condenser (configuration 1) and ACC (configuration 2) are assumed to cost the same.

In this plant the levelized main product unit costs are equal to the electricity generation costs. In case of configuration 1 with the ambient conditions of Granada, \( MPUC_{l,Conf1} \) are 15.27 ct/kWh of generated electricity. With the same ambient conditions, levelized main product unit costs in configuration 2 are 16.08 ct/kWh. This shows that using an air cooled condenser is 0.81 ct/kWh more expensive than using wet cooling tower cells.

\( MPUC_{l,Conf2,Sahara} \) which are the levelized main product unit costs in configuration 1 when the power plant is built in the Sahara, are 15.52 ct/kWh. This is still more economic than configuration 2 in Granada. When air cooling is used in the northern Sahara, the electricity generation costs \( MPUC_{l,Conf2,Sahara} \) are 17.52 ct/kWh. From the economic analysis follows that water cooled systems are much more economic than air cooled systems.
4.3. Exergoeconomic analysis

For the exergoeconomic analysis, cost flows are assigned to exergy flows [17]. This method is called "Exergy Costing". Combining the results of the exergetic and the economic analyses, the cost rates of all exergy streams within the process were calculated according to the definitions given in [11]. To calculate the costs of exiting streams auxiliary relations based on the P-rule and the F-rule [17] are used.

Following variables are used in a exergoeconomic analysis for improving the cost effectiveness of the k-th component:

The average costs per exergy unit of fuel:

\[ c_{F,k} = \frac{\dot{C}_{F,k}}{\dot{E}_{F,k}} \]  \hspace{1cm} (8)

The average costs per exergy unit of product:

\[ c_{P,k} = \frac{\dot{C}_{P,k}}{\dot{E}_{P,k}} \]  \hspace{1cm} (9)

By the use of the average costs per exergy unit of fuel the current costs of exergy destruction can be calculated:

\[ \dot{C}_{D,k} = c_{F,k} \dot{E}_{D,k} \]  \hspace{1cm} (10)

It is assumed that the average costs per supplied exergy unit of product and the exergetic product stay constant even if the exergy destruction of the component changes [17].

The relative cost difference between the average costs per exergy unit of product \( c_{P,k} \) and the average costs per exergy unit of fuel \( c_{F,k} \) can be determined as follows:

\[ r_k = \frac{c_{P,k} - c_{F,k}}{c_{F,k}} \]  \hspace{1cm} (11)

Exergy destruction and investment costs have the biggest impact on \( r_k \) value. By exergoeconomic factor the values of these two main variables are compared:

\[ f_k = \frac{\dot{Z}_k}{Z_k + \dot{C}_{D,k}} \]  \hspace{1cm} (12)

To interpret the system exergoeconomically, the exergoeconomic key figures of both configurations have to be determined. Primarily the exergoeconomic interpretation gives information about whether the investment costs in single components should be increased or reduced. Increasing the investment costs would also increase the exergetic efficiency and vice versa.

The values of the exergoeconomic key figures average costs per exergy unit of product and fuel, sums of investment cost rates of the components plus cost streams of exergy destruction, relative cost differences as well as exergoeconomic factors of both configurations are shown in Table 4.

In accordance with exergoeconomic analysis, the most significant component in the Rankine cycle in both configurations is the low pressure steam turbine, which has at the same time the highest values of exergy destruction and the highest investment costs. Although in the evaporator takes place the second highest exergy destruction, the high-pressure turbine has the second highest exergoeconomic significance in the process due to its high investment cost. However, the \( f_k \) - factors of the turbines are within the acceptable range. Nevertheless a reduction of exergy destruction and an increase of exergetic efficiencies are recommended.
Table 4. Exergoeconomic data of main components of the Rankine cycle $\dot{Z}_k + \dot{C}_{D,k}$

<table>
<thead>
<tr>
<th>Component</th>
<th>$c_{F,k}$, €/MWh</th>
<th>$c_{P,k}$, €/MWh</th>
<th>$\dot{Z}<em>k + \dot{C}</em>{D,k}$, €/h</th>
<th>$r_k$</th>
<th>$f_k$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Low-pressure turbine ST2&amp;3</td>
<td>103.4</td>
<td>136.9</td>
<td>1333.0</td>
<td>0.324</td>
<td>0.565</td>
</tr>
<tr>
<td>High-pressure turbine ST1</td>
<td>101.1</td>
<td>139.9</td>
<td>649.3</td>
<td>0.384</td>
<td>0.645</td>
</tr>
<tr>
<td>Evaporator EVAP1</td>
<td>84.5</td>
<td>94.7</td>
<td>397.1</td>
<td>0.120</td>
<td>0.316</td>
</tr>
<tr>
<td>Reheater SPHT2</td>
<td>84.5</td>
<td>113.7</td>
<td>313.4</td>
<td>0.345</td>
<td>0.338</td>
</tr>
<tr>
<td>Economizer ECO1</td>
<td>84.5</td>
<td>96.5</td>
<td>124.4</td>
<td>0.142</td>
<td>0.098</td>
</tr>
<tr>
<td>Superheaters SPHT1</td>
<td>84.5</td>
<td>97.3</td>
<td>111.8</td>
<td>0.151</td>
<td>0.599</td>
</tr>
<tr>
<td>Feedwater pump PUMP2</td>
<td>137.8</td>
<td>216.0</td>
<td>53.7</td>
<td>0.568</td>
<td>0.732</td>
</tr>
<tr>
<td>Feedwater heaters 5 FWH5</td>
<td>101.1</td>
<td>381.0</td>
<td>46.5</td>
<td>0.098</td>
<td>0.094</td>
</tr>
<tr>
<td>Deaerator DA1</td>
<td>89.2</td>
<td>109.3</td>
<td>37.0</td>
<td>0.225</td>
<td>0.453</td>
</tr>
<tr>
<td>Feedwater heater 1 FWH1(Conf. 1)</td>
<td>103.4</td>
<td>170.4</td>
<td>36.9</td>
<td>0.648</td>
<td>0.095</td>
</tr>
<tr>
<td>Feedwater heaters 4 FWH4</td>
<td>102.8</td>
<td>116.9</td>
<td>35.1</td>
<td>0.137</td>
<td>0.112</td>
</tr>
<tr>
<td>Feedwater heater 1 FWH1(Conf. 2)</td>
<td>103.4</td>
<td>165.0</td>
<td>32.6</td>
<td>0.596</td>
<td>0.108</td>
</tr>
<tr>
<td>Feedwater heater 2 FWH2</td>
<td>103.4</td>
<td>133.9</td>
<td>30.8</td>
<td>0.295</td>
<td>0.109</td>
</tr>
<tr>
<td>Feedwater heater 3 FWH3</td>
<td>103.4</td>
<td>124.6</td>
<td>29.9</td>
<td>0.205</td>
<td>0.109</td>
</tr>
<tr>
<td>Condensate pump PUMP1(Conf. 2)</td>
<td>137.9</td>
<td>389.6</td>
<td>10.1</td>
<td>1.826</td>
<td>0.792</td>
</tr>
<tr>
<td>Condensate pump PUMP1(Conf. 1)</td>
<td>137.8</td>
<td>110.9</td>
<td>10.0</td>
<td>1.765</td>
<td>0.802</td>
</tr>
<tr>
<td>ST2&amp;3, CND1, CT1, PUMP3</td>
<td>103.4</td>
<td>136.9</td>
<td>1720.0</td>
<td>0.324</td>
<td>0.551</td>
</tr>
<tr>
<td>ST2&amp;3 and ACC1</td>
<td>103.4</td>
<td>137.0</td>
<td>2009.0</td>
<td>0.326</td>
<td>0.472</td>
</tr>
<tr>
<td>Total (Conf. 1)</td>
<td>84.6</td>
<td>137.8</td>
<td>3382.0</td>
<td>0.623</td>
<td>0.520</td>
</tr>
<tr>
<td>Total (Conf. 2)</td>
<td>84.5</td>
<td>137.9</td>
<td>3614.0</td>
<td>0.631</td>
<td>0.486</td>
</tr>
</tbody>
</table>

Low-pressure turbine, high-pressure turbine, evaporator, re heater and economizer have in descending order the highest $\dot{Z}_k + \dot{C}_{D,k}$ - values. With these components is a need of action within the framework of an exergoeconomic optimization.

The SPHT1 is the only component with a $f_k$ - factor outside of the desired range. The high $f_k$ - factor (59.9 %) of the SPHT1 recommends a reduction of the investment costs of the component that can be accomplished by accepting the increase of the exergy destruction within the component. FWH1 should be redesigned for both configurations, not least due to its low exergetic efficiencies of 63.1 % (configuration 1) and 65.3 % (configuration 2) respectively. The simplifying assumption in this work that FWH1 can be designed with the same size in both configurations is not recommended and has to be reconsidered.

The system with ST2&3, CND1, CT1 and PUMP3 has a much lower $\dot{Z}_k + \dot{C}_{D,k}$ - value (1 720 €/h) in comparison to the system with ST2&3 and ACC1 (2 009 €/h), thus a larger $f_k$ - factor (55.1 % in comparison to 47.2 %). This shows that the exergy destruction in the system with ST2&3 and ACC1 is much higher.

In the solar part $c_{F,SP}$ is very low, because the sun as a fuel is free of charge. This implicates that costs of exergy destruction are therefore very low as well. The fact that about 95 % of the exergy destruction in both configurations takes place in the solar part is not expressed by $r_{SP}$ and $f_{SP}$ (63.6 % and 0.92). However, the investment costs are still very high, because of the rare usage of this solar technology. Research and development work in order to reduce the investment costs are therefore necessary. As proposed in [19], investment costs can be reduced as well by improving the efficiency of the turbine and reducing the efficiency of the solar part.
5. Conclusions

Two configurations with different cooling systems for the Rankine cycle of Andasol 1 power plant are simulated. Configuration 1 includes a wet cooling system (cooling tower cells, wet condenser and cooling pump) and is used in the real Andasol 1 plant. In configuration 2, the wet cooling system is replaced by an air cooled condenser.

When using an air cooled condenser in Granada (28.05 °C, 60 % relative air humidity), the optimal condenser pressure increases to 0.073 bar in comparison to 0.063 bar with a wet cooling technology. For the dry cooling system 2 660 kW have to be used on electrical energy. For the wet cooling system have to be spent only 810 kW. These facts result in a net power output of 47.34 MW for configuration 2 and 49.91 MW in configuration 1.

In case of the construction of an Andasol-type plant in the northern Sahara (45 °C, 20 % relative air humidity), the optimal condenser pressure increases drastically and only 43.50 MW net power output can be obtained. With wet cooling technology it is still possible to provide 49.10 MW net power output in the Sahara.

The main unit product costs of configuration 1 with a wet cooling system in Granada are 15.27 ct/kWh and 16.08 ct/kWh for configuration 2 with a dry cooling system. In the Sahara they increase to 15.52 ct/kWh in configuration 1 and 17.52 ct/kWh in configuration 2.

The difference in electricity generation costs of both configurations (0.81 ct/kWh in Granada and 2 ct/kWh in the Sahara) compared to the costs, which make water supply possible, are crucial to be able to judge which cooling system to take.

The values of exergoeconomic factors are all in the frame of the values determined in conventional power plant technology, except the one of the SPHT1.

The $\tilde{Z}_t + \tilde{C}_{D,t}$ - value of the system with ST2&3 and AC1 is much higher than the one of the system with ST2&3, CND1, CT1 and PUMP3 (2 009 €/h in comparison to 1 720 €/h).

This implies that the exergy destruction in the system with ST2&3 and AC1 is much higher, for the reason that the investment costs of both systems are the same.

The trade-off between investment cost and fuel cost which is found in fossil-fuel fired plants does not exist to the same extend in solar plants, because the main fuel, solar energy, is for free. Therefore, the meaningfulness and the validity of empirical exergoeconomical factors from conventional power plant which are used to judge solar power plant technology are questionable. Nevertheless it is evident that increasing the effectiveness of the turbine and decreasing the investment costs of the solar part are most important potentials to theoretically improve the cost effectiveness of both configurations.

Nomenclature

- $c$: cost per unit of exergy, €/MWh
- $\dot{C}$: cost rate associated with exergy, €/h
- $e$: specific exergy, kJ/kg
- $\dot{E}$: exergy flow rate, kW
- $f$: exergoeconomic factor, %
- $h$: specific enthalpy, kJ/kg
- $m$: massflow rate, kg/s
- $p$: pressure, bar
- $\dot{Q}$: heat rate
- $r$: relative cost difference
specific entropy, kJ/kg

entropy flow rate, kW

stream

temperature, K

power

exergy destruction ratio, %

cost rate associated with capital investment, €/h

exergetic efficiency, %
total annual time of system operation at full load, h

destruction
fuelp

generated

j

jth stream

k

kth component

product

associated with transfer of heat

loss

levelized

total

associated with transfer of work

thermodynamic environment (reference state)

chemical

kinetic

physical

potential

air cooled condenser

condenser

carrying charges
capital investment
configuration

cooling tower
daerator
dry cooling system

economizer

evaporator

fuel costs

feedwater heater

heat transfer fluid
LP  low pressure
MPUC  main product unit cost
PEC  purchased-equipment cost
OMC  operating and maintenance costs
SPHT  superheater
ST  steam turbine
SP solar part
TRR total revenue requirement
WCS  wet cooling system

References


Influence of renewable generators on the thermo-economic multi-level optimization of a poly-generation smart grid

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Abstract:
In this paper, the impact of not controllable renewable energy generators (wind turbines or solar photovoltaic panels) on the thermo-economic optimum performance of poly-generation smart grids is investigated using an original time dependent hierarchical approach. The grid used for the analysis is the one installed at the University of Genoa for research activities. It is based on different prime movers: (i) 100 kWe micro gas turbine, (ii) 20 kWe internal combustion engine powered by gases to produce both electrical and thermal (hot water) energy and (iii) a 100 kWth adsorption chiller to produce cooling (cold water) energy. The grid includes thermal storage tanks to manage the thermal demand load during the year and appropriate systems to model any kind of thermal and electrical load profiles. The system is also equipped with two renewable non-controllable generators: a small size wind turbine and photovoltaic solar panels. The system optimization (size and management) including the renewable generators has been carried out using a time-dependent thermo-economic hierarchical approach developed by the authors, considering the time-dependent electrical, thermal and cooling load demands during the year as constraints, in order to minimize both grid capital and variable costs. The results are presented and discussed in depth and show the strong interaction between fossil and renewable resources, and the importance of an appropriate storage system to optimize the RES impact taking into account the multiproduct character of the grid under investigation.

Keywords:
Poly-generative smart grid, renewable sources, thermo-economic time-dependent optimization, optimal size.

1. Introduction

The development of poly-generation smart grids represents an interesting solution to satisfy electricity and heat demand and emission reduction[1][2]: poly-generation smart grids generate electricity and heating and cooling thermal power close to end users, solving the main disadvantages of the centralized generation approach, due to energy transmission [3]. In fact, the distributed generation approach has several benefits over the others, such as (i) reduction of transmission and distribution costs: about the 30% of the costs related to electricity supply relates to these costs. Local connections do not generate high capital costs and energy losses for long distances to be wired with overhead facilities; (ii) decrease of energy dissipation: piping and conversion devices dissipate almost 6% of produced energy,[4] increasing costs and emissions; in a smart grid these kinds of losses are avoided; (iii) increase of energy efficiency: the simultaneous supply of electrical and thermal demand allows to reduce energy waste, improving system global efficiency; since thermal energy is less easily transported than electricity, distributed generation approach (production close to users) is essential; (iv) integration of renewable generators: traditional prime movers could be easily integrated at local site with renewable generators decreasing emissions.

Therefore, the main goal is to analyze and optimize a poly-generation smart grid, analyzing the effect of renewable generators and correlation with the storage tank for hot thermal energy. To understand the importance of renewable generators in influencing the whole system, it is worth...
remembering that an optimal poly-generation smart grid must assure the end users’ demands (multi product systems) simultaneously (heating, cooling and electrical demand). Increasing the size of renewable generators involves several advantages such as low environmental impact and fossil fuel consumption reduction; on the other hand, renewable generation is intrinsically random over time, thus their “unpredictable behaviour” could affect the operating conditions of prime movers over time. Since prime movers are co-generative, changes to operating conditions influence simultaneously electrical and thermal production. Thus, by varying the optimal size of renewable electrical generators, thermal storage size and behaviour could be influenced too. Therefore, the most important point of the analysis is represented by the integration between the prime mover constraints (continuous operation, high performance, etc.) and the uncontrolled renewable production (whose performance are random according to wind and solar irradiation condition).

2. Prime movers and device test rig

The poly-generation smart grid analyzed here is based on the one installed in the laboratory of the University of Genoa. The facilities installed are based on different kinds of technology with the aim to produce both electrical and thermal energy [5] and designed using different tools developed at TPG lab.[6-9]

Referring to Fig.1 the poly-generative smart grid main components are [10]:

- **100 kWe Recuperated Micro Gas Turbine**: It is the largest prime mover installed and produces the highest amount of both thermal and electrical energy; the machine is a Turbec T100 PHS Series [11].
consists of a power generation module (100 kW of electrical power at nominal conditions), a cogeneration heat exchanger located downstream of the recuperator outlet (hot side).

- **20 kWe Internal Combustion Engine**: the internal combustion engine installed is a FIAT TANDEM-T20-A, powered by natural gas; this commercial unit generates 20 kW of electrical power at nominal conditions; a cogenerative heat exchanger (named recuperator) is placed downstream of the ICE outlet. It produces 47.5 kW of thermal power at nominal conditions.

- **100 kWe Adsorption Chiller**: in order to generate cooling thermal power, mGT T100 has been connected to an adsorption chiller LWM-W300 produced by LS. It is powered directly by the mGT heat, assuring 100kW cooling power generation.

- **Renewable generators**: the grid also includes renewable generators; photovoltaic solar panels, model eco line 72/180-185 W produced by Luxor; wind energy produced by wind turbines of MAGLEV series.

- **Storage tank**: the grid analyzed is equipped with several storage tanks (i.e., for hot and cold water of the whole grid, and for each main engines). However in this analysis only the storage tank (Pufferspeicher series. [5]) for thermal energy is considered.

The grid is also integrated with a 450 kW SOFC Hybrid System Emulator developed by the authors in collaboration with Rolls-Royce FCS [12-15] for cogeneration applications, but it is not considered in the analysis carried out this paper.

### 3. EPoMP multilevel thermo-economic optimization approach

To optimize a poly-generative smart grid optimal size and management including RES, a time-dependent thermo-economic analysis is mandatory. The approach used here represents the further development of the one for co-generative power plants [16][17]. The model has been developed by the authors and a modular program named EPoMP (Economic Poly-generation Modular Program) has been created [18].

Fig. 2 shows the model structure which is based on a hierarchical optimization structure. There are two different optimization levels: the low and the high level respectively. At the low level, lay-out and plant size are fixed (therefore capital costs are fixed) and the code finds the best operational strategy, in order to minimize the function that represents the hourly (or less time period) variable cost.

\[
C_{\text{var}} = F_f \cdot \sum_{i=1}^{N} c_{\text{fuel}_i} + c_{\text{el}} \cdot E_{\text{acq}} + c_{\text{virt}} \cdot \left( F_{\text{virt}} + E_{\text{virt}} + Q_{\text{virt}} \right) \tag{1}
\]

Variable costs are made up of the following terms: (i) a term related to fuel consumption costs, (ii) a term related to electrical energy costs, and (iii) a term that represents “virtual costs”.

The electrical energy costs term represents the product of the electrical energy purchased from external grid and the electricity specific cost: when the electricity produced by the plant is not sufficient to satisfy the electrical load (which is one of the problem constraints), electricity is purchased from the external grid. About the term “virtual flows”, added to the cost function, it is important to underline that it is not a real cost, but rather represents exchanges between system and environment (electrical grid, fuel grid, storage system, etc.) necessary only to meet during the calculation the optimization procedure constraints. Since the virtual cost is considered very high, to find the optimum conditions without any virtual energy request (constraint violation such as the gas flow rate or available electricity from the grid), the code is forced to find a plant configuration which minimizes virtual flows [16] (i.e. zero virtual costs).

The main inputs of the model are: (i) electricity, heating, cooling, etc. load curves versus time; (ii) economic scenario where the grid operates, including trade prices (fuel cost, energy cost, plant lifetime etc.); (iii) component capital costs vs. size; (iv) operating and maintenance costs vs. time;
(v) prime movers off design performance curves; (vi) technology constraints for grid devices (i.e. starting time, flexibility, etc.).

Constraints of the problem are the balance equation between supply and demand of the components. For example, in the energy balance, the energy produced by the prime movers (ICE, gas turbines, etc.) in the system, the energy sold to the user and the energy consumed by system components (i.e. electrolytic cells, etc.) are included:

\[ E_{\text{req}} = \sum_{i=1}^{N_i} E_{i,\text{prod}} + E_{\text{acq}} + E_{\text{virt}} - \sum_{i=1}^{N_i} E_{i,\text{cons}} \]  

(2)

The same concept is the basis of the thermal energy balance:

\[ Q_{\text{req}} = \sum_{i=1}^{N_i} Q_{\text{prod}} + Q_{\text{virt}} \]  

(3)

At the high level, the plant component optimal size, minimizing total annual cost, is evaluated as the sum of variable costs \( C_{\text{var}} \) calculated at low level analysis and capital plant costs \( C_{\text{cap}} \).

\[ C_{\text{tot}} = C_{\text{var}} + C_{\text{cap}} \]  

(4)

The plant total capital cost is the sum of the total capital cost of any plant component.

\[ C_{\text{cap}} = \sum_{i=1}^{N_i} C_{\text{cap},i} \]  

(5)

Therefore, the approach optimizes simultaneously the plant and the component sizes, together with plant operation management [16][17].

EPoMP, using two different optimization routines (one for each level of the model at every iterative cycle), calculates the total annual cost changing the value of the component nominal size: in this step virtual flows are very important to find the optimal component size, in order to determine the global minimum value of the objective function. At low levels, only variable costs are optimized, reducing to zero the penalty costs associated with the virtual flows that would represent not satisfied constraints. At the high optimization levels, the model also takes into account fixed costs, finding the optimal size for plant components. At every optimization level, different constraints must be satisfied; if these conditions are not verified, strong penalties are applied and objective functions are raised because of costs \( C_{\text{virt}} \) associated with virtual flows [18].

![Fig. 2: EPoMP code hierarchical structure](image)

To build in an easy way complex plant configurations, a modular visual approach has been developed by the authors (37 different modules are available at the moment), as already presented...
For each component, four subroutines are developed to calculate (i) design and off-design performance; (ii) capital costs; (iii) variable costs, and (iv) operating and maintenance costs. The calculation is then carried out by dividing the operational time (usually a year) into a sufficient number of representative periods, one hour or less depending by the particular application.

Due to their high importance in the poly-generative smart grid under analysis in this paper, a short description of renewable generators and thermal storage models is presented.

- **Renewable generator module:** the innovative aspect of this module is its ability to simulate the functional behaviour and the energy production of any kind of renewable generator, including random aspects, such as wind turbines, hydraulic plants, solar panels, etc. Thanks to its generic nature, it is possible to simulate several plant configurations. The module needs, as inputs: (i) solar radiation and wind curves [20] in order to take properly into account the random character of these energy resources; (ii) nominal size of the installed renewable generator; (iii) number of the modules installed in the plant; (iv) kind of the installed renewable generators. Energy production is computed as the product of renewable generator installed power and the availability curves, varying in each period, as reported in Eq. (6):

\[
P_{\text{prod},i} = P_{\text{inst}} \cdot \alpha_i
\]

\(\alpha_i\) is strictly related to renewable sources availability curves, thus it can assume all the values between 0 and 1. As far as the economical analysis is concerned, for any kind of renewable generator, specific cost functions are implemented, in order to calculate both fixed and variable costs.

It is worth noting the innovative feature of this module; in a standard prime mover (ICE, mGT, etc.) powered by fuel, working conditions are optimized at hierarchical low level taking into account off-design curves, implemented in the model, and satisfying load demands (constraints of the problem) in each period; on the contrary, renewable generators behaviour cannot be regulated by the users, since energy production amounts depend only on random renewable sources (water, solar, wind) availability curves.

- **Thermal storage module:** the optimization of this component is realized by the introduction of appropriate virtual flows and virtual costs. The calculation is carried out over a selected period of time, in order to predict the weighted average performance of the plants working under variable loads. The new system of virtual costs has been introduced in order to optimize storage management, associating virtual costs with the filling and emptying operations [21].

3.1 Thermo-economic simulation

To achieve thermo-economic optimization, EPoMP needs a large number of input, most of them related to the economic scenario where the plant is installed. The main plant data considered here are as follow:

- **Electrical and thermal load curves:** The simulation was carried out on an entire year time period, considering average monthly values for electrical, thermal and cooling demand values. Load curves versus time represent the problem constraints to avoid penalties; they are shown in Fig. 3 and represent typical university campus energy demands, as reported in [20].
Energy cost: electricity cost has been assumed to be equal to 0.20 €/kWh; for both hot and cold thermal energy a sale rate of 0.10 €/kWh has been assumed [20].

Capital costs of each installed device have been assumed based on laboratory plant data [5][11][5]. In particular, for renewable generators a value of 2,500 €/kW peak for photovoltaic panels and of 2,000 €/kW for wind generator have been considered.

Fig. 4 shows the constraints of the problem, in particular: (i) cooling demand, satisfied by adsorption chiller, which is fuelled by mGT heat; (ii) thermal demand, satisfied by a storage tank, whose level depends on operation of the prime movers (Q_mGT and Q_ICE); (iii) electrical demand,
satisfied by the prime movers (E_{mGT} and E_{ICE}) and renewable generators (E_{SP} and E_{WT}). As shown in Fig. 4, electrical demand may also be satisfied by taking electricity from the external electrical network (i.e. when a peak demand is higher than maximum installed plant production capacity). In this case, a limit to the electricity from the external grid is considered for contract cost reasons, and therefore a virtual cost may be applied also for this connection.

The plant analysis was performed on an entire year time period (8760 hourly periods) considering the percentage load of the prime movers (mGT, ICE, adsorption chiller) as decision variables at model low level; renewable generators (solar panels and wind turbines) and thermal storage size have been assumed as decision variables at high level optimization. The analysis was performed at high levels considering storage volume and renewable generators size as decision variables. Renewable generators have been considered in the range between 1 and 25 kW for both photovoltaic panels and wind turbines.

4. Results

Through the use of the EPoMP code, an optimum value of 17 kW for photovoltaic panels peak power, 3 kW for wind turbine size, and 34 m$^3$ for the storage tank volume has been obtained.

To carry out a detailed analysis of the results for the different aspects of the grid along the year, they have been properly organized in “power or size vs. time plots” showing the entire year behaviour.

In Fig. 5, the mGT, the internal combustion engine, the renewable generators productions and the electricity taken from the grid are shown; the electrical load demand is represented by a black line. Specifically, mGT production satisfies the base load, ICE supplies electricity during peak periods and at night hours as well, when the demand is lower and mGT would work at strong off design conditions, where efficiency is particularly low and emissions high. In some periods, electrical demand could be higher than maximum prime movers production, thus the system has to buy electricity from the external network to satisfy the load demand, previously shown in Fig. 4. The electrical demand trend is nearly the same throughout the entire year, no significant differences between the seasons are evident. It is worth noting that the optimization minimizes the number of mGT start-up and shut-down, which affects the operating life of the engine (similarly for ICE): the cost related to these operations is considered in the fuel consumption term, included in Eq. (1).

As far as renewable generators are concerned, their behaviour is shown in Fig. 6. It is worth remembering that their production follows not only night/day variation, but it also depends on the season of the year and is substantially “random”, therefore it is unpredictable and not controllable. It is also necessary to highlight the great difference between photovoltaic panels (17 kW) and wind turbine (3 kW) optimal size. Although both of them are uncontrollable generators, photovoltaic panels follow the night/day insulation (i.e. easily predictable) behaviour: this feature allows a less complex matching for PV panels with mGT and ICE, compared to wind turbine. In particular, since wind turbine works even during night periods, a further size increase would result in the shut-down of the ICE during night periods, a thermal power reduction and greater difficulty in satisfying thermal load demand in classical morning peak periods. In this case, a larger storage would be necessary and plant capital costs would increase. For this main reason, the optimal size of wind turbines is lower than the size of photovoltaic panels.
In Fig. 7 the mGT and the ICE thermal productions are shown; storage tank supply is also shown; thermal demand is represented by black line. During the winter season thermal power production (cogeneration effect) exceeds the demand at midday, when a peak of electrical demand is present, and in the late afternoon. Since the prime movers are co-generative, surplus heat produced is used to fill the storage tank. Likewise, during night hours, since ICE works to cover electrical demand, its thermal power production, which exceeds the demand, is sent to the storage. As Fig. 7 shows, the storage tank operates to cover thermal load in the early morning, when thermal demand presents a peak and the prime movers are not sufficient to satisfy the thermal demand, and in the afternoon, when electrical demand gets lower and co-generative prime movers are off (in the case of MCI) or work at strong part load (mGT).
Fig. 6: Renewable generators behaviour vs. time (note: since the wind model is fully random, electricity generated by wind turbines is not repetitive for different days or months of the year, while PV generation is mainly related to day and night solar insulation with possible cloud effects considered in the model).

In the other seasons, in particular in summer, thermal demand gets low: as consequence, a higher surplus thermal power amount is present, a fact that could bring strong economical penalties due to storage tank overfilling. Since hot and cooling thermal demands are complementary (see Fig. 3), a significant part of the thermal power surplus generated by mGT is used by adsorption chiller (see Fig. 4) in the summer, supplying the cooling demand and avoiding thermal storage overfilling. Thermal power used by chiller is represented in dashed lined yellow columns in Fig. 7.
Cooling thermal demand is supplied by the adsorption chiller which is powered directly by mGT. In the case under investigation, cooling power demand is zero and the adsorption chiller does not work during the winter. It is worth noting (see Fig. 3) that cooling power load demand is practically complementary to thermal load demand; it is the highest in summer (when thermal load demand is the lowest) and zeros out in winter (when thermal load is the highest). Thus, the surplus thermal power produced by mGT in summer time, that would not be necessary in the storage tank, finds an application in the adsorption chiller, allowing cooling power production without any additional fuel cost and obtaining revenue from the cooling water.

Storage tank behaviour during the year is shown in Fig. 8 where storage tank level vs. time for each season is reported. The poly-generation plant with renewable generators (continuous line) results are compared to the only fossil configuration (dashed line), investigated in a previous work [20], considering the same load curves shown in Fig. 8.
Without renewable generators, electrical energy is produced by traditional prime movers only (mGT and MCI). Due to their co-generative nature, all the heat produced is sent to storage tank, without any dissipation. Thus, as Fig. 8 shows, tank level gets higher and the minimum volume to avoid overfilling penalties is 45 m$^3$. Since renewable generators produce only electrical energy, their integration in the poly-generation system implies lower co-generative device utilization, thus a lower thermal storage maximum filling. As Fig. 8 shows, storage tank optimal dimension is reduced to 30 m$^3$, allowing a significant decrease in terms of volume and capital costs.

Analyzing the whole grid operating period, it is worth noting that starting level and final level are the same. This result proves how prime movers operation, renewable generators size and energy load demands are perfectly balanced. Storage tank levels get higher in summer, when the demand for heat is low, while it gets empty in winter, when the demand for heat is higher.
Fig. 9 compares economic results for the poly-generative system plant lay-out, considering two different configurations: (a) without any renewable production (configuration A); (b) including 17 kW photovoltaic panels and 3 kW wind turbine installation (configuration B). For both configurations, annual cash flows are reported, including revenues and costs: revenues (on the left) are represented by electrical, hot and cooling thermal energy selling, costs are composed of the depreciation rate, gas consumption and electricity purchased from external network.

It is worth noting that installation of renewable generators (configuration B) implies increasing revenues; moreover, it has primary effects on the whole thermo-economic annual results:

a) **Electrical energy increase**, since new configuration includes 17kWᵣ solar panels and 3kWₑ, installed power increase from 120 kW to 140 kW, thus during peak hours electricity purchased from the external grid is reduced;

b) **Fuel consumption decrease**, since prime movers average utilization is reduced

c) **Depreciation rate increase**, since capital costs are higher

To sum up, total annual costs are nearly constant, since effects (b) and (c) balance each other; on the other hand, revenues get higher since electrical energy increases, thus introducing the renewable generators is significant from a thermo-economic point of view. Moreover, annual fuel consumption is reduced by 13.5 tons: since specific methane emissions are 2.75 kg CO₂ / kg CH₄, about 37 tons/year of CO₂ can be avoided thanks to renewable generators production.

5. Conclusions

In this paper, the influence of renewable non-controllable sources (wind turbines or solar photovoltaic panels) on the thermo-economic performance of a poly-generation smart grid, similar to the one installed in the laboratory of the University of Genoa [5], has been investigated.

The analysis was carried out using a hierarchical thermo-economic approach for poly-generative system optimization, developed by the authors [18][19]. The code, named EPoMP, found the optimal size for both renewable generators and hot storage system.
The results allow the following main conclusions to be carried out:

▪ Two different renewable non-controllable generators were considered, specifically photovoltaic panels and wind turbines. Although both of them are random generators, photovoltaic panels follow at least the alternation night/day, a feature which allows a better matching with prime movers compared to wind turbine (i.e. the random grade is higher for wind energy). For this reason, the optimum result is a size of 17 kW for PV panels and only 3 kW for wind turbine.

▪ Since renewable generators considered here produce only electrical energy, optimal size has been optimized by also taking into account that the poly-generative system must satisfy thermal and cooling demands throughout the year too. Since they are non controllable, higher renewable sizes (i.e. 20 kW solar panels, 5 kW wind turbine) would not verify the problems constraints and costs (capital and variable) increase largely.

▪ Renewable generator utilization influences the storage system optimal size too; since co-generative prime movers working time and load (they operate largely at part load) is reduced due to RES generators, thermal energy production is lower and storage optimal volume decreases (in this case from 45 to 30 m$^3$) allowing a reduction in storage tank capital costs too.

▪ The optimized system integrated with non-controllable renewable generators was compared from a thermo-economic point of view with the same poly-generative plant without RES installed; the main result is that optimizing solar panels, wind turbine sizes and system management allows annual revenues increasing to be obtained, while annual costs are nearly constant (capital costs increase, but variable costs reduce).

It is worth noting that the model developed here can be applied to any poly-generative plant including any kind of renewable generator. Moreover, the presented results have a generalized value, since the method takes into account the type and size of the prime movers, the energy load profiles, the plant location, and the economic scenario.

Using the laboratory poly-generation grid at University of Genoa with a very flexible approach not only for generation aspects, but also for load profile generation, the EPOMP model will be validated and different options will be investigated such as low geothermal heat, solar panel, hydrogen generation and utilization with the intention to define case by case the best option for multi product distributed generation grids.

6. Nomenclature

Abbreviations:
ICE Internal Combustion Engine  
PV Photo Voltaic  
mGT Micro Gas Turbine  
RES Renewable Energy Sources

Symbols:
C Cost [€]  
E Electricity flow [kWh]  
P Power [kW]  
$\alpha$ RES availability coefficient  
c Specific cost [€/kWh]  
F Fuel consumption [kg]  
Q Heat flow [kWh]

Subscripts:
acq acquired
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References


Local stability analysis of a thermo-economic model of an irreversible heat engine working at different criteria of performance

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Abstract:
In recent works [1, 2], we reported a local stability analysis of a thermo-economic model of an irreversible heat engine working at maximum power conditions. In those works, we calculated the relaxation times in terms of $\tau$, $f$ and $R$ (a parameter which comes from the Clausius inequality and measures the degree of the internal irreversibilities). Besides $\tau = T_2/T_1$, with $T_1 > T_2$, being the temperatures of the external heat reservoirs and the parameter $f$ is the fractional fuel cost, which is associated to several energy resources considering energy sources where de investment is the preponderant cost ($f = 0$), until energy sources where the fuel is the predominant cost ($f = 1$). In those works, we showed that, after a small perturbation the system decays exponentially to the steady state determined by two different relaxation times. In this work, we extend the local stability analysis considering other regimes of performance: The Maximum Efficient Power and the Ecological Function regime. We show that the relaxation time under maximum ecological function conditions is lesser than the relaxation times under both maximum power and maximum efficient power, that is, under maximum ecological conditions we have better stability conditions than for the other two regimes. Besides, we observe that the stability of the system improves as $\tau$ increases whereas the steady-state energetic properties of the engine declines for all cases of energy sources.

Keywords:
Thermo-economics, Local stability, Irreversible heat engine.

1. Introduction
Since the work of Curzon–Ahlborn(CA) [3], most of the studies on Finite-Time Thermodynamics (FTT) have focused on steady-state energetic properties [3-8], which are important from the point of view of design. However, all thermal engine works in many cycles per unit time and they are never identical, that is, there exists intrinsic cyclic variability (CV) in any sequence of cycles. For example, in internal combustion engines, the CV is produced for incomplete combustion of fuel and other causes [9]. It is crucial to know how much each cycle allows external perturbations, while still preserving the steady-state regime that lets it carry out its function well. In order to have a well-designed system, it is important to analyse the effect of noisy perturbations on the stability of the system’s steady state. This study may allow us to guarantee proper dynamical behaviour of a system like stability and small relaxation times, or to warn about possible failure in the performance of a thermal engine. In 2001, Santillán et al. [10] studied the local stability analysis of an endo-reversible Curzon-Ahlborn heat engine operating under maximum power conditions. Later, Guzmán-Vargas et al. [11] investigated the effect of the heat transfer laws and the thermal conductances on the local stability of an endo-reversible heat engine. Recently, Páez-Hernández et al. [12], analyzed the local stability of a non-endo-reversible Curzon-Ahlborn cycle taking into
account the engine’s implicit time delays when operating at maximum power regime. However, the local stability analysis described in these works have not considered the effect of economical aspects. Within the context of Finite-Time Thermodynamics, the effect of the economical aspects was early introduced by De Vos [13] for the study of the thermo-economic performance of a model of power plant of the Novikov type [14,15]. Later, Sahin and Kodal [16] studied the thermo-economics of an endo-reversible heat engine in terms of the maximization of a profit function defined as the quotient of the power output and the annual investment and the full consumption costs. This thermo-economic performance analysis [17], consists in to maximize a benefit function in terms of the power output and the cost involved in the performance of the power plant. Recently, Barranco-Jiménez et al. [1,2] reported a local stability analysis of a thermo-economic model of an irreversible heat engine working under maximum power conditions. In those studies they used two different heat transfer laws, the Newton heat transfer law [1] and the Dulong-Petit heat transfer law [2]. In this work, we extend the local stability analysis considering other regimes of performance: The Maximum Efficient Power [18,19,20] and the Ecological Function regime [21-22]. We show that the relaxation time under maximum ecological function conditions is lesser than the relaxation times under both maximum power and maximum efficient power, that is, under maximum ecological conditions we have better stability conditions than for the other two regimes. The paper is organized as follows: In section 2, we present the thermo-economic analysis of the irreversible heat engine with different criteria of performance. In section 3, we describe the local stability analysis method applied to a two-dimensional system. In section 4, the local stability analysis of the irreversible heat engine is presented. Finally, en section 5, we present our conclusions.

2. Thermo-economic analysis of the steady-state irreversible heat engine under different criteria

In Fig. 1 a schematic diagram of the irreversible heat engine (Curzon-Ahlborn model) is shown. This engine consists in a Carnot-like thermal engine that works in irreversible cycles and exchanges heat with external thermal reservoirs at temperatures $T_1$ and $T_2$ ($T_1 > T_2$). In the steady state, the temperatures of the Carnot-like cycle isothermal branches are $\bar{x}$ and $\bar{y}$, here overbars are used to indicate the corresponding steady-state value. The steady-state heat flows from the hot to the cold thermal reservoirs are denoted as $\bar{J}_1$ and $\bar{J}_2$, respectively (Fig.1).

Fig. 1. Schematic representation of an irreversible heat engine. $R$ is a measure of the departure of the endo-reversible regime.
Applying the Clausius theorem and using the fact that the inner Carnot-like engine works in irreversible cycles, we get the following inequality,

\[
\frac{\mathcal{J}_1}{\bar{x}} - \frac{\mathcal{J}_2}{\bar{y}} < 0,
\]

this expression can be transformed into an equality by introducing a parameter \( R \) leading to,

\[
\frac{\mathcal{J}_1}{\bar{x}} = R \frac{\mathcal{J}_2}{\bar{y}},
\]

The parameter \( R \), which in principle is within the interval \( 0 \leq R \leq 1 \) \((R = 1\) for the endo-reversible limit), can be seen as a measure of the departure from the endo-reversible regime \([23-25]\). If we assume that the heat flows from \( \bar{x} \) to \( \bar{T}_1 \) and from \( \bar{y} \) to \( \bar{T}_2 \) are of the Newton type, then

\[
\begin{align*}
\mathcal{J}_1 &= \alpha (\bar{T}_1 - \bar{x}), \\
\mathcal{J}_2 &= \alpha (\bar{y} - \bar{T}_2),
\end{align*}
\]

where \( \alpha \) is the thermal conductance. For simplicity of the calculations, we have assumed that the heat exchanges take place in conductors with the same thermal conductance \( \alpha \); that is, the materials of both conductors are the same. The effect of the thermal conductances on the local stability of an endo-reversible heat engine was investigated by Guzmán-Vargas et al \([11]\), in fact the power is a strong function of the conductances ratio \([26]\). The system’s steady-state power output and the efficiency can be defined as,

\[
\bar{P} = \mathcal{J}_1 - \mathcal{J}_2,
\]

and

\[
\bar{\eta} = \frac{\bar{P}}{\mathcal{J}_1} = 1 - \frac{\mathcal{J}_2}{\mathcal{J}_1} = 1 - \frac{\bar{y}}{R \bar{x}}.
\]

By combining Eqs. (2), (3), (4) and (6), we can write \( \bar{x} \) and \( \bar{y} \) in terms of \( \bar{T}_1, \bar{T}_2, R \) and \( \bar{\eta} \) as,

\[
\begin{align*}
\bar{x} &= \frac{T_1}{1 + R} \left( 1 + \frac{\tau}{1 - \bar{\eta}} \right) \\
\bar{y} &= \frac{R}{1 + R} \bar{T}_1 \left( 1 + \frac{\tau}{1 - \bar{\eta}} \right) \left( 1 - \bar{\eta} \right)
\end{align*}
\]

where \( \tau = \bar{T}_2 / \bar{T}_1 \). The De Vos thermo-economical analysis considers a profit function \( \bar{F} \), which is maximized \([13]\). This profit function is given by the quotient of the power output \((\bar{P})\) and the total cost involved in the performance of the power plant \((\bar{C}_{tot})\), that is,

\[
\bar{F} = \frac{\bar{P}}{\bar{C}_{tot}}.
\]
In his early study, De Vos assumed that the running cost of the plant consists in two parts: a capital cost which is proportional to the investment and, therefore, to the size of the plant and, a fuel cost that is proportional to the fuel consumption and, therefore, to the heat input rate $\bar{J}_1$. Assuming that $\bar{J}_{\text{max}}$ is an appropriate measure for the size of the plant, the running cost of the plant exploitation is defined as [13],

$$\bar{C}_{\text{tot}} = a\bar{J}_{\text{max}} + b\bar{J}_1 = a\alpha T_i \left[ (1-\tau) + \beta \left( 1 - \frac{x}{T_i} \right) \right],$$

(10)

where the proportionality constants $a$ and $b$ have units of $$/Joule, $\beta = b/a$ and $\bar{J}_{\text{max}} = \alpha (T_1 - T_2)$ is the maximum heat that can be extracted from the heat reservoir without supplying work (see Fig. 1). By using Eqs. (3), (6), (7), (9) and (10), the profit function can be written as,

$$a\bar{F}_{\text{MP}} = \frac{\bar{\eta}}{1 + R} \left[ \frac{1}{1 + \frac{\beta}{1 - \bar{\eta}}} \left( R - \frac{\tau}{1 - \bar{\eta}} \right) \right].$$

(11)

If we calculate the derivative of $a\bar{F}$ with respect to $\bar{\eta}$, and we solving for the efficiency $\left[d(a\bar{F}_{\text{MP}})/d\bar{\eta}_{\bar{\eta}=\bar{\eta}^*} = 0\right]$ we get [27],

$$\bar{\eta}^*(\beta, \tau, R) = 1 - \sqrt{\frac{\tau}{R}} \frac{R(1-\tau) - \tau \beta}{(1-\tau) R \left[ 1 + \frac{\beta(R-\tau)}{(1-\tau) R} - \sqrt{R} \tau \beta \right]}.$$

(12)

We can observe in Eq. (12) that for $R=1$ the result obtained previously by De Vos [13] is recovered. Besides, when $\beta = 0$, we obtain $\bar{\eta}^*_{\text{opt}} = 1 - \sqrt{\tau / R}$, which was previously obtained by Wu and Kiang [28], and by Arias-Hernández et al. [25]. Instead of expressing the result in terms of the parameter $\beta$, a number that is difficult to obtain in the literature [13], we can also express the efficiency in terms of the fractional fuel cost which is defined as [13],

$$f = \frac{b\bar{J}_1}{a\bar{J}_{\text{max}} + b\bar{J}_1} = \frac{T_i \beta}{\bar{J}_{\text{max}} + \beta\bar{J}_1}. $$

(13)

The fractional fuel costs ($f$) for various technologies were reported by De Vos for different energy sources; that is, for example; renewable energy $f = 0$, for Uranium $f = 0.25$, for Coal $f = 0.35$ and for natural gas $f = 0.5$ [13]. By using Eqs. (3), (7) and (13), we can write the parameter $\beta$ in terms of the fractional fuel cost as follows [13],

$$\beta = \frac{f}{1-f} \frac{(1 + R)(1-\tau)}{R - \frac{\tau}{1-\bar{\eta}}}. $$

(14)

Therefore, the efficiency that maximizes the profit function is given by [27],
\[ \eta_{mp}(f, \tau, R) = 1 - \frac{f}{2R} \tau - \frac{\sqrt{4(1-f)\tau R + f^2\tau^2}}{2R}, \]  
(15)

and the power output is given by

\[ P = \alpha T_i \left[ (f - 2)\tau - \sqrt{4(1-f)\tau R + f^2\tau^2} \right] \left[ f\tau - 2R + \sqrt{4(1-f)\tau R + f^2\tau^2} \right] \]
\[ \frac{2(1+R)}{-f\tau - \sqrt{4(1-f)\tau R + f^2\tau^2}}. \]  
(16)

Eqs. (15) and (16), represent the steady-state efficiency (\( \eta \)) and power output (\( P \)) respectively. They are function of \( \tau \), \( f \) and \( R \) for a non-endoreversible Novikov-Curzon-Ahlborn heat engine working at the maximum-power regime. It is straightforward to show that both \( \tilde{\eta} \) and \( \tilde{P}/\alpha T_i \) are decreasing functions of \( \tau \) for every fixed value of \( R \) [1]. Analogously to Eq. (9), for our thermo-economic optimization approach, we define two objective functions in terms of the so-called Efficient Power [18, 19, 20] and the so-called Ecological Function [21, 22], both divided by the fractional fuel cost. The Maximum Efficient Power performance [18, 19] for heat engines was studied for Yilmaz [18] and previously defined for Stucki [20] as the product of power output \( (P) \) by the efficiency \( (\eta) \) in the context of the first order irreversible thermodynamics (FOIT) in 1980. The ecological optimization criterion for the FTT-thermal cycles was proposed by Angulo-Brown [21]. This criterion considers the maximization of a function \( E \) which represents a compromise between high power output \( (P) \) and low entropy production \( \Sigma \). The \( E \) function is given by \( E = P - T_2 \Sigma \), where \( P \) is the power output of the cycle, \( \Sigma \) the total entropy production per cycle and \( T_2 \) is the temperature of the cold reservoir. One of the most important characteristics of a \( CA \)-engine operating under maximum \( E \) conditions is that it produces around 80% of the maximum power and only 30% of the entropy produced in the maximum power regime [21, 22]. Another interesting property of the maximum- \( E \) regime is that the \( CA \)-engine’s efficiency in this regime, is given by

\[ \eta_E \approx (\eta_C + \eta_{CA})/2, \]

where \( \eta_C \) is the Carnot efficiency and \( \eta_{CA} \) the Curzon-Ahlborn efficiency [9].

For our thermo-economical approach, these objective functions are given by \( F_{EP} = \frac{\eta P}{C_{tot}} \) and

\[ F_E = \frac{P - T_2 \Sigma}{C_{tot}} \]

respectively. In the same way that Eq. (11) the profit function can be written as,

\[ aF_{EP} = \left( \eta \right) \left[ \frac{1}{1+R} \left( \frac{R - \tau}{1-\eta} \right) \right] \]
\[ \left( 1-\tau \right) + \frac{\beta}{1+R} \left( \frac{R - \tau}{1-\eta} \right), \]  
(17)

and
In Eq. (18) we have applied the second law of thermodynamics to calculate the total entropy production given by $\Sigma = \frac{\mathcal{T}_2 - \mathcal{T}_1}{T_2 - T_1}$ (see Fig. 1). Analogously to Eq. (11), if we calculate the derivatives of $a\mathcal{F}_E$ and $a\mathcal{F}_E$ with respect to $\tilde{\eta}$, and we solve for the efficiency
\[
\left. \frac{d(a\mathcal{F}_E)}{d\tilde{\eta}} \right|_{\tilde{\eta}=\tilde{\eta}^0} = 0 \quad \text{and} \quad \left. \frac{d(a\mathcal{F}_E)}{d\tilde{\eta}} \right|_{\tilde{\eta}=\tilde{\eta}^0} = 0
\]
and using Eq. (14) we get,
\[
\tilde{\eta}_{EP}(f, \tau, R) = 1 - \frac{(1+f)\tau}{4R} - \sqrt{\frac{(1-f)\tau}{2R} + \frac{(1+f)^2\tau^2}{16R^2}}.
\]
\[
\tilde{\eta}_E(f, \tau, R) = 1 - \frac{f\tau}{2R} - \sqrt{\frac{(1-f)(1+f)\tau}{2R} + \frac{f^2\tau^2}{4R^2}}.
\]

Eqs. (19) and (20), represent the steady-state efficiencies working both under maximum-efficient power ($\tilde{\eta}_{EP}$) and under maximum ecological function conditions ($\tilde{\eta}_E$), respectively. In analogous way to Eq. (12) or Eq. (15), for the endo-reversible case ($R = 1$), from Eqs. (19) and (20), when $f = 0$, ($\beta = 0$), we obtain $\tilde{\eta} = 1 - \tau/4 - \sqrt{\tau(8 + \tau)/4}$ and $\tilde{\eta} = 1 - \sqrt{\tau(\tau + 1)/2}$, respectively, which were previously obtained by Guzmán-Vargas et al. and Yılmaz [11, 18], and by Angulo-Brown [21], for the case of a Curzon-Ahlborn heat engine, working at maximum efficient power and maximum ecological function conditions respectively.

![Figure 2. The steady-state efficiencies working under maximum power output ($\tilde{\eta}_P$), maximum-efficient power ($\tilde{\eta}_{EP}$) and maximum ecological function conditions ($\tilde{\eta}_E$).](image)

We can see in Fig. 2, how the optimal efficiencies smoothly increase from the maximum efficiency point, $f = 0$, corresponding to energy sources where the investment is the preponderant cost up to the Carnot value for $f = 1$, that is, for energy sources where the fuel is the predominant cost [13].

3. Linearization and stability analysis
In this section, we present a brief description of the linear stability analysis of a two-dimensional system [29]. Consider the dynamical system,

$$\frac{dx}{dt} = h(x, y),$$  \hspace{1cm} (17) \\
$$\frac{dy}{dt} = g(x, y),$$  \hspace{1cm} (18) \\

where \( h \) and \( g \) are functions of \( x \) and \( y \). Let \((\bar{x}, \bar{y})\) be a fixed point such that \( h(\bar{x}, \bar{y}) = 0 \) and \( g(\bar{x}, \bar{y}) = 0 \). Consider a small perturbation around this fixed point and write \( x = \bar{x} + \delta x \) and \( y = \bar{y} + \delta y \), where \( \delta x \) and \( \delta y \) are small disturbances from the corresponding fixed point values. By substituting into equations (17) and (18), expanding \( h(\bar{x} + \delta x, \bar{y} + \delta y) \) and \( g(\bar{x} + \delta x, \bar{y} + \delta y) \) in a Taylor series, and using the fact that \( \delta x \) and \( \delta y \) are small to neglect quadratic terms, the following equations are obtained for the perturbations:

$$\begin{pmatrix} d\delta x \\ dt \\ d\delta y \\ dt \end{pmatrix} = \begin{pmatrix} h_x & h_y \\ g_x & g_y \end{pmatrix} \begin{pmatrix} \delta x \\ \delta y \end{pmatrix},$$  \hspace{1cm} (19) \\

where \( h_x = \frac{\partial h}{\partial x} \bigg|_{\bar{x}, \bar{y}}, \quad h_y = \frac{\partial h}{\partial y} \bigg|_{\bar{x}, \bar{y}}, \quad g_x = \frac{\partial g}{\partial x} \bigg|_{\bar{x}, \bar{y}}, \quad g_y = \frac{\partial g}{\partial y} \bigg|_{\bar{x}, \bar{y}} \). Equation (19) is a linear system of differential equations. Thus, we assume that the general solution of the system is of the form,

$$\delta \vec{r} = e^{\lambda t} \vec{u},$$  \hspace{1cm} (20) \\

with \( \delta \vec{r} = (\delta x, \delta y) \) and \( \vec{u} = (u_x, u_y) \). Substitution of the solution \( \delta \vec{r} \) into equation (19) yields the following eigenvalue equation:

$$A \delta \vec{r} = \lambda \delta \vec{u},$$  \hspace{1cm} (21) \\

where \( A \) is the matrix given by the first term on the right-hand-side of equation (19). The eigenvalues of this equation are the roots of the characteristic equation,

$$|A - \lambda I| = (h_x - \lambda)(g_y - \lambda) - g_x h_y = 0.$$  \hspace{1cm} (22) \\

If \( \lambda_1 \) and \( \lambda_2 \) are solutions of equation (22), the general solution of the system is

$$\delta \vec{r} = c_1 e^{\lambda_1 t} \vec{u}_1 + c_2 e^{\lambda_2 t} \vec{u}_2,$$  \hspace{1cm} (23) \\

where \( c_1 \) and \( c_2 \) are arbitrary constants and \( \vec{u}_1 \) and \( \vec{u}_2 \) are the eigenvectors corresponding to \( \lambda_1 \) and \( \lambda_2 \), respectively. To determine \( \vec{u}_1 \) and \( \vec{u}_2 \) we use equation (21) again for each eigenvalue.
Information about the stability of the system can be obtained from the eigenvalues $\lambda_1$ and $\lambda_2$. In general, $\lambda_1$ and $\lambda_2$ are complex numbers. If both $\lambda_1$ and $\lambda_2$ have negative real parts, the fixed point is stable. Moreover, if both eigenvalues are real and negative, the perturbations decay exponentially. In this last case, it is possible to identify relaxation times for each eigendirection as

$$t_1 = \frac{1}{|\lambda_1|},$$

$$t_2 = \frac{1}{|\lambda_2|},$$

### 4. Local stability analysis

Following Santillán et al [10], due to the reservoirs $x$ and $y$ are not real heat reservoirs but macroscopic objects (the working substance at the isothermal branches of the cycle) with heat capacity $C$. Their temperatures change according to the following differential equations:

$$\frac{dx}{dt} = \frac{1}{C}[\alpha(T_1 - x) - J_1],$$

$$\frac{dy}{dt} = \frac{1}{C}[J_2 - \alpha(y - T_2)],$$

where $J_1$ and $J_2$ are the heat flows from $x$ to the working substance and from the Carnot engine to $y$, respectively. According to the non-endoreversibility hypothesis [23,24], $J_1$ and $J_2$ are given by

$$J_1 = \frac{Rx}{Rx - y} P$$

and,

$$J_2 = \frac{y}{Rx - y} P,$$

On the other hand, we can use Eqs. (6) and (15) to construct the expression which relates the internal variables $x$ and $y$, to the external temperatures $T_1$ and $T_2$, in this case under maximum power conditions we get,

$$\tau = \frac{y^2}{x^2(1 - f)R + xyf}.$$

In a similar way to Eq. (30), by using Eqs. (7) and (15), we can obtain an expression for $T_1$, given by,
and the corresponding steady-state values of $\bar{x}$ and $\bar{y}$ as functions of $T_1$ and $T_2$ at maximum-profit function (in this case defined by Eq. (11)) are obtained by substituting Eq. (15) into Eqs. (7) and (8), respectively

$$\bar{x} = \frac{T_1}{1 + R} \left( 1 + \frac{2R \tau}{f \tau + \sqrt{4(1 - f) \pi R + f^2 \tau^2}} \right),$$

(32)

$$\bar{y} = \frac{RT_1}{2(1 + R)} \left[ (f + 2R) \tau + \sqrt{4(1 - f) \pi R + f^2 \tau^2} \right].$$

(33)

We can observe from Eqs. (30) - (33) that for the case $f = 0$ and $R = 1$, the results previously obtained by Santillán et al. [10] and Guzmán-Vargas et al. [11] are recovered. Finally, by substituting Eqs. (30) and (31) into Eq. (16), the steady-state power output can be expressed by,

$$\mathcal{P}(\bar{x}, \bar{y}, f, R) = \alpha \left[ \frac{x \Lambda + (f - 2) \sqrt{f y^2}}{(1 - f) R \bar{x} + f \bar{y}} \right] - \frac{2 \sqrt{y^2 + (f - 2) \sqrt{f y^2}}}{(1 - f) R \bar{x} + f \bar{y}},$$

(34)

where $\Lambda = \sqrt{x^2 \left[ f \right] (1 - f) R \bar{x} + f \bar{y}}^2$. Using the assumption [10] that out of the steady state but not too far away, the power output of a Curzon-Ahlborn heat engine depends on $x$ and $y$ in the same way that it depends on $\bar{x}$ and $\bar{y}$ at the steady-state $\mathcal{P}(\bar{x}, \bar{y}, f, R) \rightarrow \mathcal{P}(x, y, f, R)$, that is, this assumption is applicable only in the vicinity of the steady state, we can write the dynamical equations for $x$ and $y$ as follows:

$$\frac{dx}{dt} = \frac{1}{C} \left[ \alpha (T_1 - x) - \frac{Rx}{Rx - y} P(x, y, f, R) \right],$$

(36)

$$\frac{dy}{dt} = \frac{1}{C} \left[ \frac{y}{Rx - y} P(x, y, f, R) - \alpha (y - T_2) \right].$$

(37)

To analyze the system stability near the steady state, we proceed following the steps described in the previous section, where

$$h(x, y, f, R) = \frac{1}{C} \left[ \alpha (T_1 - x) - \frac{Rx}{Rx - y} P(x, y, f, R) \right],$$

(38)
After solving the corresponding eigenvalue equation, we find that both eigenvalues (\( \lambda_1 \) and \( \lambda_2 \)) are function of \( \alpha \), \( C \), \( \tau \), \( f \) and \( R \). The final expression and the algebraic details are not shown because they are quite lengthy and can be easily reproduced with the help of a symbolic algebra package. Moreover, our calculations show that both eigenvalues are real and negative. Thus, the steady state is stable because any perturbation would be decay exponentially. For the case \( f = 0 \), expressions for the eigenvalues previously obtained by Santillán et al. [10] and Guzmán-Vargas et al. [11] are recovered. In Figs. 3, 4 and 5, the relaxation times are plotted against \( \tau \) for different values of fractional fuel cost \( f \), for a fixed value of \( R \) (\( R = 1 \)), that is, the endo-reversible case. We observe that \( t_1 \) (Eq. 24) is a decreasing function of \( W \). This relaxation time decreases as the fuel cost increases, indicating a faster decay as \( f \to 1 \).

**Figure 3.** Plot of relaxation times under maximum power conditions (\( t_1 \) and \( t_2 \)) versus \( \tau \) for a) several values of the endorreversibility parameter and a value of the fractional fuel cost and b) for several values of the fractional fuel cost \( f \) in the endoreversible case (\( R = 1 \)).

**Figure 4.** Plot of relaxation times under maximum efficient power (\( t_1 \) and \( t_2 \)) versus \( \tau \) for a) several values of the endo-reversibility parameter and a value of the fractional fuel cost and b) for several values of the fractional fuel cost \( f \) in the endo-reversible case (\( R = 1 \)).
Figure 5. Plot of relaxation times under maximum ecological function conditions \((t_1 \text{ and } t_2)\) versus \(\tau\) for a) several values of the endo-reversibility parameter and a value of the fractional fuel cost and b) for several values of the fractional fuel cost \(f\) in the endo-reversible case \((R = 1)\).

For \(t_2\) (see Eq. 25), we observe that this relaxation time remains almost constant for \(f = 0\). As the fractional fuel cost \(f\) increases, \(t_2\) slowly increases too. We notice that in the limit \(f \to 1\), both relaxation times tend to be closer each other, but there is a stronger inequality \(t_2 < t_1\) in the interval \(0 < \tau < 1\). In Fig. 4 we also show the relaxation times as a function of \(\tau\), for several values of the parameter \(R\), and for a fixed value of the fractional fuel cost \(f\). We can see in Figs. 3, 4 and 5, that \(t_1\) is a decreasing function of \(\tau\) and decreases as the parameter \(R\) decreases. We also observe that \(t_2\) remains almost constant when the irreversibility parameter changes. From the findings of Figs. 3, 4 and 5, we can conclude that the system is stable for \(\tau > 0\). We notice that as the fractional fuel cost \(f\) increases, \(t_1\) decreases whereas \(t_2\) increases, for a given value of \(R\). In contrast, for a given value of \(f\), as the irreversibility parameter \(R\) decreases, \(t_1\) decreases whereas \(t_2\) increases. We also remark that the power output and the efficiency depend on \(\tau\) for the cases analyzed here, and both energetic quantities are decreasing functions of this parameter, that is, the system’s stability moves in the opposite direction to that of the steady – state as \(P\), \(\eta\) and \(\tau\) varies.

Additionally, in Fig. 6, for the cases of maximum power output, maximum efficient power and maximum ecological function conditions, we show the relaxation times versus fuel fractional cost for several values of \(\tau\). We can see, in this case, how the fast (slow) relaxation time slightly increases (decreases) as \(f\) changes from 0 to 1.

Conclusions

In this work, we present a local stability analysis of a thermo-economic model of an irreversible heat engine working at different regimes of performance: The Maximum Efficient Power and the Ecological Function regime and by considering a linear heat transfer law (the Newton law case). We show that the relaxation times are function of \(\alpha\), \(C\), \(\tau\), \(f\) and \(R\); that is, they depend on the materials that separate the working fluid form the reservoirs (through \(\alpha\)); on the working fluid (through \(C\)); on the reservoirs temperatures (through \(\tau\)); on fractional fuel costs (through \(\mathcal{F}\), which is associate to various energy sources from renewable energy until natural gas, see Table 1. reported by De Vos [13]) and on the internal irreversibilities (through \(R\)).
Figure 6. Relaxation times \( t_1 \) and \( t_2 \) in the endo-reversible case \( (R = 1) \) versus \( f \) for several values of \( \tau \) for a) Maximum Power output, b) Maximum Efficient Power conditions and c) Maximum Ecological Function.

After a small perturbation the system decays exponentially to the steady state determined by two different relaxation times. We show that the relaxation time under maximum ecological function conditions is lesser than the relaxation times under both maximum power and maximum efficient power, that is, under maximum ecological conditions we have better stability conditions than for the other two regimes. Besides, we observe that the stability of the system improve as \( \tau \) increases whereas the steady-state energetic properties of the engine declines for all cases of energy sources. Our cycle’s model is an FTT version of a Carnot-type engine, but considering in addition the fractional fuel costs. In this sense, it has the same general idealized characteristics of any other model stemming from FTT. However such as it has been showed by Fisher and Hoffmann [6] and Curto-Riso et al. [30], this kind of models are useful to describe yet very elaborated dynamical models of heat engines of the Otto-type for example.

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Nomenclature

\[ F \]
Profit function \((W/$$)

\[ P \]
Steady-state power output \((O$$)

\[ J_i \]
Steady-state heat flows \( i = 1,2 \)

\[ C_{\text{tot}} \]
Total costs \((O$$)
\[ f \quad \text{Fractional fuel cost} \]
\[ R \quad \text{Parameter of the internal irreversibilities} \]
\[ C \quad \text{Heat capacity} \]

**Greek symbols**

\[ \bar{\eta} \quad \text{Steady-state efficiency} \]
\[ \beta \quad \text{Economical parameter} \]
\[ \tau \quad \text{Temperature ratio} \]
\[ \alpha \quad \text{Thermal conductance} \]
\[ \lambda_i \quad \text{Relaxation times (} i = 1, 2 \text{)} \]

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MULTICRITERIA OPTIMIZATION OF A DISTRIBUTED ENERGY SUPPLY SYSTEM FOR AN INDUSTRIAL AREA

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Abstract:
In the paper a multi-objective optimization model for distributed energy supply systems optimization is presented. The superstructure of the system comprehends a district heating network that connects the users to each other, small scale CHP systems (e.g. micro gas turbines or internal combustion engine), conventional integration boilers, large centralized solar plant and a seasonal thermal storage. The optimization has to determine the optimal structure of the system, the size and the load of each component inside the optimal solution, as well as the optimal operation strategy. The multi-objective optimization is based on a Mixed Integer Linear Programming model (MILP) and it takes into account as objective function a linear combination of the total annual cost (for owning, maintaining and operating the whole system) and the CO\textsubscript{2} emissions amount, associated to the system operation. The model allows to obtain different optimal solutions by varying the relative weight of the economic and the environmental objectives. In this way the Pareto Front is identified and the possible improvements in both economic and environmental terms can be highlighted. The model has been applied as an example to a specific case study made of nine industrial facilities and it has been optimized for different superstructure configurations and for two different values of the electricity greenhouse emissions factor. The obtained results shows that the solar plant, coupled with the seasonal thermal storage, allows reaching both environmental and economic goals. If the centralized solar plant is not considered in the superstructure, CO\textsubscript{2} emissions related to electricity affect the optimal structure of the energy supply system.

Keywords:
Multicriteria optimization, District heating, Mixed Integer Linear Programming, Solar thermal plant, CHP

1 Introduction
Distributed energy systems have already been recognized as an efficient and reliable alternative to the traditional energy supply [1]. It must be said also that nowadays a purely economic analysis is not anymore sufficient due to growing environmental concerns like global warming and depletion of fossil fuel reserves. Therefore, the operation problems become more challenging when the environmental burdens should be minimized at the same time when costs, too, are to be minimized [2]. The reason is because the minimization of costs and pollutant emissions are normally contradictory objectives, as it is often expensive to utilize environmentally friendly technologies.

The multi-objective optimization can help to solve this problem combining the supply energy cost together with the environmental impact. These goals have to be both minimized. Multi-objective optimization tackles the issue of conflicting objective functions, finding a set of solutions by varying the impact of the single objective functions in the global optimum. For such solutions, called Pareto optimal solutions or non-dominated solutions, no improvement is possible without sacrificing the other objective functions [3]. Reviews on multi-criteria analysis and examples of multi-objective optimizations of distributed energy supply systems can be found in [4-9].
The paper proposes an optimization model that helps to determine the optimal configuration and operation of a distributed energy supply system. The developed model has been applied as an example to the energy supply system serving nine industrial facilities. They may be connected to each other through a district heating network of fixed size and layout. Heating and electric demands are known in advance and they can be satisfied by small-scale CHP systems (ICE or µTG), properly located at or near the end-users. Conventional integration boilers can also be installed inside the factories or in the centralized plant and each user is free to purchase and sell electricity from/to the national grid.

The paper also introduces the integration between conventional power sources and renewable energies, designing a solar district heating plant coupled with a long term thermal energy storage. This alternative is increasing in importance, as it is a valid solution to overcome the mismatch problem between the availability of the solar source and the energy user demands, especially for domestic applications[10-12]. As all users can be connected together through the DHN, the heat produced by the solar thermal plant and by the production units may be exchanged to one another or sent to the thermal storage.

The model used to solve the optimization problem is based on a Mixed Integer Linear Programming (MILP). In previous works of the authors, MILP models were developed to optimize the design and operation of distributed CHP systems in a tertiary sector scenario, considering different technologies and taking into account the effects of various economic support policies [13-16]. A similar model was applied to an industrial area considering also the thermal inertia of the network in [17].

The two objective functions of the model to be minimized consider the total operating $CO_2$ emissions and the total annual cost for owning, maintaining and operating the whole distributed system. The optimization is subject to the constraints that express component operation characteristics, energy balances of nodes and district heating network behavior. The optimization specifies the size, the kind and the location of cogeneration equipment and integration boilers, the size of the solar thermal plant and of the heat storage, as well as the optimal operation of each component included in the optimal solution.

In the specific application, the Pareto frontiers associated to four different system configurations and for two different values of the electricity carbon intensity, were evaluated. The results of the optimizations were used to identify the best trade-off solutions.

2 Optimization model

The proposed model aims at providing decision support to planners for selecting the configuration system and the operating levels of various generation units throughout the planning period.

Recently a lot of research has been carried out to optimize the design and operation of distributed energy supply systems [3, 18-20] integrated also with the district heating network [21-23]. The mathematical problem of optimizing the operation of an energy system composed of CHP units, solar thermal modules and DHN has to be generally regarded as a variational calculus problem, because the optimization variables expressing partial load operation of each CHP engine are time dependent. However, a realistic description of the system may be represented by an MILP formulation by properly discretizing the load curves (in each time interval the thermal and the electrical demands are assumed to be constant) and approximating performance maps with a set of linear functions [24, 25]. The thermal losses along pipelines have been approximated as a fixed fraction of the thermal energy transferred in each time interval. All other relations that describe the system (energy balances, load limits, cost of energy vectors) are intrinsically linear and they do not need to be approximated. A detailed description of the model and the approximation introduced with the linearization of the performance curves can be seen in [26].

The optimization of an energy supply system begins with the definition of the superstructure. The superstructure must include all components that can be potentially part of the optimal solution, so
that generally it depends on the specific application case. After the optimization process, the superstructure will be reduced to the optimal configuration.

Figure 1 shows the system superstructure. The distributed energy supply system has to supply the thermal and electrical energy required by a set of users. The electric energy can be produced by the CHP systems installed in each production unit or purchased from the electric grid. The thermal energy can be produced by the CHP systems, by conventional boilers or by a large centralized solar plant. All users can be connected to each other and to the solar field by a district heating network. Additionally, the superstructure includes also a hot water storage. Many large scale solar district heating systems have been built already in central and northern Europe, mainly in Sweden, Denmark, The Netherlands, Germany and Austria [27]. They consist of ground mounted collector fields integrated into a DHN for supplying heat to residential and industrial areas. The sizes of those plants allow lower specific investments compared to small applications. When the system is coupled with a heat water storage it is possible to reach solar fractions of approximately 50% [28].

In the superstructure, a typical user $k$ can be equipped with a cogeneration unit and a boiler. The central production unit includes a boiler and a solar field. The district heating network connects the users to one another and to the heat storage. The model is completely general and it can be applied to different applications with a similar superstructure, by changing the values that describe the components only. An MILP model has been used for properly describing the choice of centralized/decentralized components inside the system superstructure by means of binary variables, as well as the on/off operation of chosen components in the optimal operation strategy.

![Figure 1: Superstructure of the energy supply system described by the optimization model.](image)

### 2.1 Decision variables

The degrees of freedom that characterize the model are the decision variables, both binary and continuous. The optimization procedure finds the set of decision variables that allows the minimization of the objective function. The identified decision variables are:

- Existence and size of each component;
- Operation status and load level of each component in each time interval;
- Electricity to be exchanged with the electricity network;
- Thermal flows inside the district heating network;
- Size of the thermal energy storage;
- Level of the thermal energy stored in the storage.

Binary variables represent existence and operation status of components, while all other variables are continuous. The decision variables can be set in advance to describe cases when only a subset of the components included in the superstructure can be adopted. For example in the conventional
the district heating network has to be excluded by setting to zero the decision variable related to its existence.

2.2 Objective functions

The MILP model considers two different objective functions, which quantify the total annual cost and the environmental impact, both to be minimized. The environmental impact objective function attempts to capture the increasing awareness of environmental pollution resulting from energy generation.

The objectives have been kept separated inside the model. It allows to find and to rank the best integrated solutions from the superstructure, which are both economical convenient and less polluting. The solutions returned by the optimization are used to identify the Pareto Front. The Pareto Front solutions cannot be made less polluting without being more costly, or cheaper without emitting more.

2.2.1 Economic objective function

The economic objective function $C_{tot}$ considers the energy supply system in terms of the total annual cost for owning, operating and maintaining the whole system. The objective function is linear with respect to all decision variables and consists of:

$$C_{tot} = C_{inv} + C_{ope}$$

The sum of the investment costs of all components ($Inv$) multiplied by the related capital recovery factor $rf$ gives the annual investment cost $C_{inv}$. Capital recovery factor takes into account the interest rate $i$ and the life span $n$ of each components $j$ contained in the superstructure.

$$C_{inv} = \sum_j rf(j) \cdot Inv(j)$$

(2)

$$rf(j) = \frac{i \cdot (1 + i)^n(j)}{(1 + i)^n(j) - 1}$$

(3)

The whole year has been subdivided in a set of discrete time intervals $t$. The annual operating and maintaining cost $C_{ope}$ associated to the energy supply system is expressed by:

$$C_{ope} = \sum_t (c_{gas} \cdot F_{g}(t) + c_{sp} \cdot E_{p}(t) - c_{sz} \cdot E_{s}(t)) + \sum_{j} C_{man}(j,t)$$

(4)

The maintenance cost $C_{man}(j,t)$ of the $j^{th}$ component in the $i^{th}$ time interval is assumed to be proportional to the energy produced.

2.2.2 Environmental objective function

The environmental impact objective function $CE$ totalizes the CO$_2$ emissions of the electricity taken from the grid and of the consumed fuel, subtracted by the avoided CO$_2$ emissions related to the sold electricity:

$$CE = CE_{G} + CE_{Ep} - CE_{Es}$$

(5)

The CO$_2$ emissions from consumed fuels are calculated by multiplying the total amount of fuel consumption with the carbon intensity of the fuel:

$$CE_{G} = \sum_{t} F_{g}(t) \cdot CIE$$

(6)

The CO$_2$ emissions due to the usage of electricity from the grid are calculated by multiplying the total amount of purchased power by the carbon intensity of grid electricity $CIE$:
While the avoided CO₂ emissions related to the sold electricity are obtained by multiplying the total amount of the sold electricity by the carbon intensity of grid electricity:

\[ C_{\text{Ep}} = \sum_t E_p(t) \cdot CI_E \]  

(7)

The carbon intensity \( CI_E \) is the amount of CO₂ emissions per unit of electricity generated within the specific utility grid. It can be measured in kg/kWh and depends on the electricity mix of the electricity network.

### 2.3 Model Constraints

In the MILP optimization model, three main different categories of constraints can be identified:

- **Components constraints**: relate output and input energy of each component;
- **Energy balances**: ensure that the amount of input energy is equal to the output, for each time interval and for each node;
- **Network constraints**: describe thermal losses and the maximum thermal energy transfer from units to users.

#### 2.3.1 Component constraints

This kind of constraints have been introduced for each component. Equality constraints represent the relation between fuels, products and sub-products, while inequality constraints describe the load and size ranges. The thermal production of the solar thermal plant is related to the size of the plant. The energy production per surface unit has been supposed to be known in advance, for each time interval. This means that position and tilt angle are fixed in advance. A long term thermal storage is considered a component too, and the equality constraints relate the thermal level of the storage to the input/output flow, taking into account the thermal losses. The stored energy depends on the temperature of the medium multiplied by the volume contained in the storage, so that volume and temperature cannot be both decision variables because each relation inside the model has to be linear. The volume has been chosen as a decision variable, while the temperature is considered constant. This choice corresponds to the hypothesis of a perfect stratification of the fluid inside the heat storage, so that if the storage is not completely empty, the residual energy is stored at the same temperature required by the DHN.

#### 2.3.2 Energy balances

These constraints are equality constraints and represent the thermal and electric energy balances, in each time interval. Taking the thermal balance as an example, for each production unit the heat produced by the cogeneration unit and by the boiler has to be equal to the heat consumed by the local user and sent to other users through the DHN.

#### 2.3.3 Network constraints

These constraints describe the DHN and limit the thermal flows in each pipeline based on its size. Moreover they represent the thermal energy balance of the network taking into account the thermal losses along each pipeline.

### 2.4 Multicriteria optimization method

Steps towards the design of sustainable energy systems must include tools for simultaneously considering the broad range of criteria linked to the thermodynamic, economic and environmental performance assessment of a system. The increasing needs for more efficient systems, that are attractive both from the economical and environmental point of view, request the development of
new criteria and determine new design rules. In fact, up until recently, the main criterion to choose the best plant was the economic one. Introducing a new decision criterion that considers the impact of the system in the environment, it is obvious that the design of such a system is associated with conflicting objectives [4,5], as it is often expensive to utilize environmentally friendly technologies. Cogeneration systems are usually studied from an economic, energetic or environmental point of view, but not optimized. In the case of multiple objectives, there does not necessarily exist a solution that is the best with respect to all targets because of differentiation between objectives. A solution may be the best in one objective but worse in another. For this reason the same problem has a lot of optimal solutions that can be intended as a compromise between the objectives. For such solutions, forming the Pareto frontier, there is not any other solution that reduces one objective without increasing another. The Pareto frontier represents a set of alternatives among them the decision maker can choose the best solution suited to its needs.

There are a lot of methods for solving multi-objective optimization problems, such as compromise programming, global criterion method, and goal programming [29]. In this study, the compromise programming method has been employed to solve this problem through the implementation in the commercial optimization software X-press® [30].

In order to apply compromise programming, the optimization model has been modified including only one objective function. In this method the aim is to minimize the distance of the criterion values from their utopia point. Considering this, the objective function of the problem is formulated as follows:

\[ F_{obj} = \alpha \cdot C_{tot} + \beta \cdot CE \]  

(9)

Where \( C_{tot} \) and \( CE \) are the economic and environmental objective function, respectively.

The optimization of this problem for different values of \( \alpha \) and \( \beta \) gives the Pareto optimal solutions and the Pareto frontier can be obtained. For each combination of \( \alpha \) and \( \beta \) only one optimal solution exists and it is a weighted combination of economic and environmental benefits.

In order to reduce the number of iterations (infinite combinations of \( \alpha \) and \( \beta \)), a different formulation of the problem can be expressed as:

\[ \text{Min} F_{obj} = \varepsilon \cdot C_{tot} + (1 - \varepsilon) \cdot CE \]  

(10)

For \( \varepsilon \in \{1, 2, \ldots \} \) \( 0 \leq \varepsilon \leq 1 \)

\( \varepsilon \) represents the slope of the Pareto front tangent. For \( \varepsilon = 1 \) the solution that minimizes \( C_{tot} \) can be obtained, while the solution that minimizes \( CE \) can be obtained for \( \varepsilon = 0 \). Intermediate values of \( \varepsilon \) give other intermediate solutions appertaining to the Pareto front.

Usually, the single objective functions that constitute the overall objective function of the multicriteria optimization problem assume values that can have different orders of magnitude. In this situation the resulting optimization would be led by the objective function with the greatest order of magnitude, because the objective function with the lowest one would not affect the value of the objective function \( F_{obj} \). In order to avoid this a scale coefficient has to be multiplied by the objective functions \( C_{tot} \) and \( CE \) so that they have the same order of magnitude.

3 The case study

The nine users considered in the study belong to different economic sectors, like plastic, food, furniture, engineering and tertiary. Despite the heterogeneity of the goods produced, their energy consumptions show quite regular trends along the year. The electrical and heating demands have been evaluated by means of energy audits. Figure 2a represents a plan view of the whole industrial area. The blue line represents the layout of the main DHN that is 5 km long and provides the heat required by the users. The locations of the nine users are marked by red spots, while the yellow spot indicates the space available for positioning the central unit, the solar field and the heat storage.
Figure 2b shows the annual electric and heating load duration curves of the nine users. Electric load is higher than zero all year round. This is because a certain amount of electricity is always required, even when factories are closed. Heating load is higher than zero for about 7,000 hours, higher than 2 MW for almost 6,000 hours and higher than 4 MW for almost 3,000 hours.

Figure 3a shows the aggregated electric and heating demand of the nine users in a typical winter and summer week. The profile is quite predictable, with peaks during intensive working hours, low consumption during nights and a very low demand in the weekend, when the most of the factories are closed. The two trends are very similar; the difference is a higher consumption in summer because of the electricity required to power the air conditioning systems of the factories.

Figure 3b shows the aggregated heating demand of the nine users in a typical winter and summer week. It can be noted that heating load is slightly higher during coldest months, when space heating is operating. The Saturday heat consumption is very small, while in Sunday neither process heat nor space heating is required.

Figure 2a: Plan view of the industrial district. Figure 2b: Total annual energy load duration curves.

Figure 3: Total electric and thermal demand profile in a typical winter and summer week.
Table 1 shows the peak power and the yearly energy consumption of each user for both electric and heating demands. Total peak power is the actual maximum hourly energy demand of the all users and it is clearly lower than the sum of the single user peaks because they do not appear simultaneously.

In order to reduce the variables number and the model complexity, the whole year is represented by twelve typical weeks (1 week per month), each composed of seven days of 24 hours, for a total of 2,016 time intervals. This kind of discretization allows keeping a realistic picture of the actual annual behaviour of the whole system and also allows the solar district heating plant and the thermal storage producing their effects on the optimal operation.

Table 2 was produced considering the energy requirements and the energy load profiles of each user. For each component it shows: the life span, the possible location and the minimum and maximum sizes and costs [31]. Machine prices are considered linear with the size. It can be observed that units 3 through 9 have the same structure: a CHP system (ICE or µGT) and a boiler, both of variable sizes. Moreover, in unit 1 only a 500 kW GT can be installed, while unit 2 includes only a ICE of variable size.

Table 3 shows the energy prices and the solar plant costs used in the application, with reference to a current Italian market scenario. A life span of 20 years is considered for the solar thermal panel and 40 years for both the DHN and the seasonal storage. The unitary heat storage cost generally depends on its kind and size and can vary between 120 and 180 €/m$^3$[11, 32, 33]. An intermediate cost of 150 €/m$^3$ has been assumed in this case study in order to maintain the problem linear. An interest rate equal to 6% has been adopted for the calculation of all capital recovery factors.
Table 3. Energy and solar thermal plant prices

<table>
<thead>
<tr>
<th></th>
<th>PRICE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Electricity purchased</td>
<td>0.15 €/kWh</td>
</tr>
<tr>
<td>Electricity sold</td>
<td>0.12 €/kWh</td>
</tr>
<tr>
<td>Natural Gas</td>
<td>0.05 €/kWh</td>
</tr>
<tr>
<td>District Heating Network</td>
<td>4,500 k€</td>
</tr>
<tr>
<td>Heat storage</td>
<td>150 €/m³</td>
</tr>
<tr>
<td>Solar thermal panel</td>
<td>200 €/m³</td>
</tr>
</tbody>
</table>

In this application case the CO\textsubscript{2} emissions are associated with the carbon intensity of both natural gas and electricity because they are the only input fuels at the system boundary. While the natural gas has similar carbon intensity all around the world, the electricity carbon intensity strictly depends on the electricity mix of the network at which the system is connected. The optimizations has been conducted for a value 0.202 kg/kWh related to the emissions of burning natural gas [34, 35]. The greenhouse emissions of the electricity have been assumed equal to 0.504 and 0.356 kg/kWh. The first value is representative of the average electricity greenhouse emissions of the world in the years 2007-2009 and it is similar to the greenhouse emissions of the USA in 2009. The second value refers to the average CO\textsubscript{2} emissions in Europe in the years 2007-2009 and it is similar to the emissions of Italy in 2009 [36].

4 Results and discussion

The model defined for the specific case study has been optimized considering different superstructures and different values of the electricity carbon intensity. The optimization MILP model includes 160,000 decision variables and 240,000 constraints and can be solved in about two hours with a processor Intel Core Due T9400 @2.52 GHz, 4GB RAM, with the stopping criteria based on the GAP, that is the percentage difference between the current value of the objective function and the current value of the objective function of the relaxed problem. The GAP assumed in the calculation is equal to 0.05%.

In the specific, the Pareto frontiers have been obtained for the following four superstructures and for two different values of the electricity carbon intensity:

- Isolated Solution (IS): the users are not connected with the DHN, therefore heat has to be produced locally while the electricity can also be bought from the grid;
- Distributed Cogeneration Solution (DCS): the users can be connected to each other through the district heating network;
- Distributed Cogeneration Solution integrated with the Thermal Storage (DCS + TS): a thermal storage is added to the superstructure of the Distributed Cogeneration Solution;
- Distributed Renewable Solution (DRS): the superstructure includes also a large solar thermal plant.

Figure 4a and 4b report the Pareto frontiers obtained by varying the weight of the two objective functions and for two different values of the electricity carbon intensity. The purely environmental and purely economic optima can be obtained by setting to 0 and 1 respectively the value of $\omega$ in eq.10. Therefore, these optima correspond to the two extremities of each points series.

Comparing Figure 4a with Figure 4b a sensible reduction of the annual CO\textsubscript{2} emissions can be noted when the electricity carbon intensity is lowered. As the variation of the carbon intensity operate only in the environmental objective function, the purely economic optima in the two figures are the same, for each superstructure.
If the solar plant (DRS) is not taken into account, the best economic performance can be achieved with the Isolated Solution (IS), while including the DHN in the system (DCS), total annual costs increase allowing an improvement of the environmental benefits. The Pareto frontiers obtained optimizing the solution including also the thermal storage (DCS + TS) dominate the DCS Pareto frontiers, therefore the adoption of a thermal storage together with the District Heating Network allows achieving better results in terms of both economic and environmental benefits.

Figure 5 shows the values of the main energy parameters for each optimization performed. Each diagram refers to a specific Pareto frontiers, obtained by varying the coefficient $\epsilon$ from 0 to 1.

The IS optimal results show that the thermal energy produced by the CHP increases moving from the economic optimum ($\epsilon=1$) towards the environmental optimum ($\epsilon=0$), while the thermal energy produced by boilers decreases. When the district heating network is included in the superstructure (DCS), the thermal energy produced by CHP is greater, while the one produced by boilers is lower. This effect is more evident approaching the purely environmental optima, up to reach a complete absence of boilers. This trend is still more pronounced if the thermal storage is included in the superstructure (DCS + TS). In fact, the thermal energy produced by CHP is greater than in the other cases and quite constant, while the contributions of the boilers are negligible for a wide range of $\epsilon$ values.

Generally, by comparing the optimal solutions obtained for the electricity carbon intensity equal to 0.356 (figures on the left) with the ones obtained for a value of 0.504 (figures on the right), it can be pointed out that the former are characterized by a higher usage of boilers and a lower usage of cogeneration units in the environmental optima. This trend is due to the fact that the electric production by means of cogeneration units is more environmentally convenient when $CI_E$ is higher. An additional set of optimizations performed for a lower value of electricity greenhouse emissions ($CI_E=0.25$) showed that such a low value of emissions brings to the adoption of boilers instead of cogeneration units in the environmental optima, according to the results presented by Carvalho et al. [3]. This can be easily understood thinking at an extreme situation where the electric energy available from the grid were produced without CO$_2$ emissions, for example from nuclear or solar sources. In this case the usage of natural gas cogeneration units would certainly imply an increase of global CO$_2$ emissions.
Figure 5: Results of electricity, thermal energies, plant and thermal storage size varying the weight of the objective functions
Table 4: Optimal solutions resulting from the analysis of the Pareto Front

<table>
<thead>
<tr>
<th>Points in Figure 5:</th>
<th>IS</th>
<th>DCS</th>
<th>DCS + TS</th>
<th>DRS</th>
</tr>
</thead>
<tbody>
<tr>
<td>A1</td>
<td>1,703</td>
<td>2,118</td>
<td>3,139</td>
<td>4,090</td>
</tr>
<tr>
<td>A2</td>
<td>3,083</td>
<td>3,160</td>
<td></td>
<td></td>
</tr>
<tr>
<td>A3</td>
<td>4,032</td>
<td>4,643</td>
<td></td>
<td></td>
</tr>
<tr>
<td>A4</td>
<td>4,707</td>
<td>4,697</td>
<td></td>
<td></td>
</tr>
<tr>
<td>B1</td>
<td>2,318</td>
<td>2,973</td>
<td>3,241</td>
<td>2,065</td>
</tr>
<tr>
<td>B2</td>
<td>4,090</td>
<td>2,065</td>
<td></td>
<td></td>
</tr>
<tr>
<td>B3</td>
<td>677</td>
<td>0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>B4</td>
<td>2,322</td>
<td>1,462</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Cogeneration units power [kWel]</th>
<th>IS</th>
<th>DCS</th>
<th>DCS + TS</th>
<th>DRS</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cogeneration units number</td>
<td>2</td>
<td>3</td>
<td>4</td>
<td>7</td>
</tr>
<tr>
<td>Boilers power [kWth]</td>
<td>4,032</td>
<td>3,241</td>
<td>2,065</td>
<td></td>
</tr>
<tr>
<td>Boilers number</td>
<td>1</td>
<td>2</td>
<td>0</td>
<td>1</td>
</tr>
</tbody>
</table>

| Solar field [m2]               | 28,153 |
| Thermal Storage [m3]           | 28,147 |
| Total investment cost [k€/y]   | 11,528 |
| Total annual cost Ctot [k€/y]  | 4,777  |
| CS annual cost [k€/y]          | 4,864  |
| PBP                             | 5     |
| CO2 emission of the CS [ton/y] | 15,757 |
| Saved CO2 em. wrt CS [ton/y]   | 6,30  |

Better results can be achieved in terms of both economic and environmental benefits if the solar plant is included in the superstructure (DRS, Figure 4) and the total convenience of these integrated systems is more pronounced for low values of $CI_E$. This is because the recourse to renewable energy sources is more effective with respect to the usage of natural gas cogeneration, if the cogenerated electric energy implies a lower CO$_2$ emission reduction. Focusing on the main energy parameters obtained varying the $\epsilon$ coefficient (Figure 5) the DRS trends are significantly different from the others. In particular, approaching the environmental optima the usage of CHP decreases, thanks to the important contribution of the solar plant. For $\epsilon=0$ boilers and cogeneration units are not included in the optimal configurations and all thermal energy required by the users is produced by the solar plant. These results can be attained also thanks to the adoption of a large seasonal thermal storage. In the economic optima, the thermal energy produced by the solar plant covers about 60% of the thermal demand.

An analysis of Pareto front (Figure 4) and the related energy parameters (Figure 5) can lead to the identification of the most attractive economic/environmental compromise solutions for each plant configuration, that are indicated in Figure 4 with A1÷A4, B1÷B4. The relating main data concerning the configurations, the economic and the environmental performance are reported in Table 4.

Figure 6 summarizes the optimal operation results of the identified trade-off points, in terms of electric energy and thermal energy amounts (Figure 6a and b, respectively). Generally, a little amount of electricity is sold to the grid if the solar plant is not considered. When it is included in the configuration the CHP are less convenient and consequently no electricity is sold to the grid.

Focusing on Figure 6b it can be noted that for solutions 3 and 4, the thermal energy produced is greater than the thermal energy required by the users because of the thermal losses through the storage wall.
5 Conclusions

In this paper, a MILP multi-objective model has been developed for identifying the operational synthesis and operation of a distributed energy system, in order to meet the energy demands of a set of users while considering both economic and environmental objectives. The two objective functions of the model to be minimized consider the total annual cost for owning, maintaining and operating the whole distributed system and the total operating CO$_2$ emissions. The solution allows identifying the location, size and optimal operation of boilers and CHP, as well as the optimal size of the solar plant and of the seasonal thermal storage.

The model has been applied as an example to a specific case study made of nine industrial facilities. It has been optimized for four different configurations, beginning with a distributed cogeneration systems made up of CHP and boilers only, up to the more integrated configuration which includes the district heating network, the thermal storage and the solar field.

Without considering the solar field, the best economic performances can be achieved with the optimal mix of boilers and cogeneration units, without adopting neither the district heating network nor the thermal storage. The adoption of these components made the total annual cost increase, but allows a reduction of the operating CO$_2$ emissions. It is worth noting that the adoption of a thermal storage together with the District Heating Network allows achieving better results in term of both economic and environmental benefits, with respect to the district heating network alone. In the considered hypotheses, the optimal integrated solution which includes the solar field, turns out to be the most convenient and the most environmentally friendly, at the same time.

A comparison between optimal solutions obtained for two different values of the electricity greenhouse emission factor shows that lower value of this coefficient brings to a lower usage of cogeneration units and a higher usage of boilers. In addition, the lower the electricity carbon intensity is, the lower the environmental benefit of natural gas cogeneration is, so that the recourse to renewable energy sources (e.g. solar fields) becomes the most effective strategy for the emission reduction. The cogeneration could be still attractive if renewable sources had used as primary energy.

Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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</thead>
<tbody>
<tr>
<td>BOI</td>
<td>Boiler</td>
</tr>
<tr>
<td>CE</td>
<td>Total Annual CO2 Emission, ton/y</td>
</tr>
<tr>
<td>CE$_G$</td>
<td>Annual Natural Gas CO2 Emissions, ton/y</td>
</tr>
<tr>
<td>CE$_{Ep}$</td>
<td>Purchased Electricity CO2 Emissions, ton/y</td>
</tr>
<tr>
<td>c$_{ep}$</td>
<td>Purchase Price of Electricity, €/kWh</td>
</tr>
<tr>
<td>CEs</td>
<td>Sale Electricity CO2 Emissions, ton/y</td>
</tr>
<tr>
<td>c$_{es}$</td>
<td>Sale Price of electricity, €/kWh</td>
</tr>
<tr>
<td>c$_{gas}$</td>
<td>Purchase Price of Natural Gas, €/kWh</td>
</tr>
<tr>
<td>CHP</td>
<td>Combined Heat and Power</td>
</tr>
<tr>
<td>CI$_E$</td>
<td>Electricity Carbon Intensity, t/kWh</td>
</tr>
</tbody>
</table>
CI
Natural Gas Carbon Intensity, t/kWh

C_{inv}
Annual Investment Cost, €/y

C_{man}
Maintenance Cost, €

C_{ope}
Annual Operating Cost, €/y

CS
Conventional Solution

C_{tot}
Total Annual Cost, €/y

DCS
Distributed Cogeneration Solution

DHN
District Heating Network

DRS
Distributed Renewable Energy Solution

E_p
Purchased Electricity, kWh

E_s
Sold electricity, kWh

F_g
Natural Gas Consumption, kWh

F_{obj}
Objective Function

GT
Gas Turbine

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On the Effect of Eco-indicator Selection on the Conclusions Obtained from an Exergoenvironmental Analysis

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Abstract:
An exergoenvironmental analysis is conducted at the component level of a system and identifies (a) the relative importance of each component with respect to environmental impact, and (b) options for reducing the environmental impact associated with the overall system. In an exergoenvironmental analysis, a one-dimensional characterization indicator is obtained using a Life Cycle Assessment (LCA). An index (a single number) describes the overall environmental impact associated with system components and exergy carriers. It should be emphasized that the evaluation of environmental impacts will always be subjective to some degree and associated with some uncertainties. Exergoenvironmental analysis identifies the magnitude, location and causes of thermodynamic inefficiencies and environmental impacts. The information supplied by such exergy-based methods is very useful in understanding the operation of energy conversion systems and in developing strategies for improving them.

The paper presents the exergoenvironmental analysis applied to a gas-turbine cogeneration system (the CGAM problem that is used here as an example), and discusses the effect of the Eco-indicator used in an exergoenvironmental analysis on the conclusions obtained from the analysis. Five indicators are used here to conduct the LCA: Eco-indicator 95, Eco-indicator 99, Cumulative Exergy Consumption, a method developed at the Center Environmental Studies of the University of Leiden (The Netherlands), and ECO-factor 2006.

Keywords:
Exergy analysis, LCA, Eco-indicator, Exergoenvironmental Analysis.

1. Introduction

The detailed evaluation of an energy conversion system from the ecological point of view is a relative new area in engineering. The thermodynamic evaluation is part of the ecological evaluation: By increasing the thermodynamic efficiency, the fuel consumption is decreased and this affects positively the improvement of the energy system from the ecological and thermodynamic points of view. In general, the relationship between thermodynamic variables and ecological variables is complex and, and if we want to successfully reduce thermodynamic inefficiencies and environmental impacts, we must understand their formation process, i.e. we need a deep understanding of

- the real thermodynamic inefficiencies and the processes that cause them,
- the environmental impact associated with equipment and thermodynamic inefficiencies as well as the connection between these two sources of environmental impact,
- the interconnections among efficiency and component-related environmental impact associated with the selection of specific system components, and
- possible measures that would improve the efficiency and would reduce the environmental impact of the system being studied.
A thermodynamic analysis should be conducted in terms of exergy. The exergy destruction caused by the irreversibilities within the system being considered is the most important thermodynamic inefficiency and is identifiable with the aid of an exergetic analysis.

There are many publications where the exergetic analysis and the ecological (environmental) analysis are applied to energy-conversion and chemical systems. The most interesting conclusions can be obtained if the exergetic analysis and the environmental assessment are discussed simultaneously. Different approaches have been developed to combine these analyses, for example, cumulative exergy consumption [1], exergoeconomic analysis [2], extended exergy accounting [3], environomic analysis [4]. A recently developed exergoenvironmental analysis has been introduced in [5] and already applied to different energy-conversion systems [5-10].

An exergoenvironmental analysis consists of an exergetic analysis, a Life Cycle Assessment (LCA) of the environmental impact and an exergoenvironmental evaluation [5].

Life cycle assessment is a method used to evaluate the inputs and outputs of systems and to organize and convert those inputs and outputs into environmental impacts relative to resource use, human health and ecological areas [12]. The quantification of inputs and outputs of a system is called Life Cycle Inventory (LCI). At this stage, all emissions are reported on a volume or mass basis. Life Cycle Impact Assessment (LCIA) converts these flows into simpler indicators. The Eco-indicator of a material or process is a number that indicates the environmental impact of a material or process, based on data from a life cycle assessment. The higher the indicator, the greater the environmental impact.

During the last 15 years many LCIA methods have developed, for example, Eco-indicator 95 [13], Eco-indicator 99 [14], EDIP 97 [15], EDIP 2003 [16], EPS 2000d [17], Impact (2002)+ [18], JEPIX [19], LIME [20], CML [21], ECO-Factors 2006 [22], etc.

However, only few of them are applicable to an energy conversion system (because of the availability of data that can be used for this purpose), but almost all these methods have already been applied to estimate environmental impacts of land, processes in agriculture, transport, buildings, etc. In the previous publications related to an exergoenvironmental analysis only the Eco-indicator 99 has been used, for example in [5-8]. In recently published papers also the Eco-indicator 95 and the Cumulative Exergy Consumption method are used for the exergoenvironmental evaluation of a simple refrigeration machine [9] and an open-cycle gas-turbine system [11].

Since different approaches (from environmental impact point of view) are used to develop LCIA methods, it is interesting to analyze the effect of selection of the indicator to the results and conclusions obtained from the exergoenvironmental analysis.

2. Exergoenvironmental analysis

The exergoenvironmental costing principle which is used in an exergoenvironmental analysis, states that exergy is the only rational basis for assigning environmental impact to energy streams and to the thermodynamic inefficiencies within a system [5].

An exergoenvironmental analysis is conducted at the component level of a system and identifies (a) the relative importance of each component with respect to environmental impact, and (b) options for reducing the environmental impact associated with the overall system. In an exergoenvironmental analysis, a one-dimensional characterization indicator is obtained using an LCA. An index (a single number) describes the overall environmental impact associated with system components and exergy carriers.

It should be emphasized that the evaluation of environmental impacts will always be subjective and associated with uncertainties. However, the information extracted from an exergoenvironmental analysis is very useful.
2.1. Exergetic analysis

Using the exergy rates associated with fuel and product, \( \dot{E}_F \) and \( \dot{E}_P \) [24], respectively, the exergetic balances for the \( k \)th component and for the overall system are, respectively

\[
\dot{E}_{F,k} = \dot{E}_{P,k} + \dot{E}_{D,k} \quad \text{and} \quad \dot{E}_{F,\text{tot}} = \dot{E}_{P,\text{tot}} + \sum_n \dot{E}_{D,k} + \dot{E}_{L,\text{tot}}
\]

(1)

The exergetic efficiencies for the \( k \)th component and for the overall system are, respectively

\[
\varepsilon_k = \frac{\dot{E}_{P,k}}{\dot{E}_{F,k}} \quad \text{and} \quad \varepsilon_{\text{tot}} = \frac{\dot{E}_{P,\text{tot}}}{\dot{E}_{F,\text{tot}}}
\]

(2)

The exergy destruction ratios for the \( k \)th component and for the overall system are, respectively

\[
y_k = \frac{\dot{E}_{D,k}}{\dot{E}_{F,\text{tot}}} \quad \text{and} \quad y_{\text{tot}} = \frac{\dot{E}_{D,\text{tot}}}{\dot{E}_{F,\text{tot}}}
\]

(3)

2.2. Life Cycle Assessment (LCA)

In this paper the following five indicators (given with the corresponding units) are used for the analysis:

- Eco-indicator 95 (ECO-95) [13], with mPts (milipoints of the ECO-indicator 95);
- Eco-indicator 99 (ECO-99) [14], with mPts (milipoints of the ECO-indicator 99 that is different than milipoints of the ECO-indicator 95);
- Cumulative Exergy Consumption (CExC) [1], with kJ of cumulative exergy;
- Method of the Centre for Environmental Studies (CML) of the University of Leiden, The Netherlands, also known as “The Dutch Guide” [21], with ELU (environmental load units), and

2.3. Exergoenvironmental evaluation

The exergoenvironmental model of an energy conversion system consists of balances and auxiliary equations associated with environmental impact [5].

The \textit{environmental impact balances} are written for the \( k \)th system component in the following form

\[
\dot{B}_{P,k} = \dot{B}_{F,k} + \left( \dot{Y}_k + \dot{B}_{h}^{PF} \right) \quad \text{or} \quad b_{P,k} \dot{E}_{P,k} = b_{F,k} \dot{E}_{F,k} + \left( \dot{Y}_k + \dot{B}_{h}^{PF} \right)
\]

(4)

Here \( \dot{B}_{P,k} \) and \( \dot{B}_{F,k} \) are the environmental impact rates associated with the product and fuel respectively, and \( b_{P,k} \) and \( b_{F,k} \) are the corresponding environmental impacts per unit of exergy for product and fuel.

The component-related environmental impact \( \dot{Y}_k \), which considers the entire life cycle of the \( k \)th component, consists of the following contributions:

\[
\dot{Y}_k = \dot{Y}_k^{CO} + \dot{Y}_k^{OM} + \dot{Y}_k^{DI}
\]

(5)

Here \( \dot{Y}_k^{CO} \) is the environmental impact that is associated with construction, including manufacturing, transport and installation, \( \dot{Y}_k^{OM} \) is associated with operation and maintenance, including production of pollutants during operation, and \( \dot{Y}_k^{DI} \) refers to the environmental impact associated with disposal.
To account for pollutant formation within the $k$th component, a new variable was recently introduced $\dot{B}_{k}^{PF}$ [9]. This term $\dot{B}_{k}^{PF}$ is zero if no pollutants are formed within a process, i.e. for processes without a chemical reaction (compression, expansion, heat transfer, etc.). For components, where chemical reactions occur (combustion, for example), the value of $\dot{B}_{k}^{PF}$ is calculated from

$$\dot{B}_{k}^{PF} = \sum_{i} b_{i}^{PF} (m_{i, out} - m_{i, in}) \tag{6}$$

where only pollutant streams which finally will be emitted to the environment are taken into account: CO, CO$_2$, CH$_4$, N$_2$O, NO$_x$ and SO$_x$ [5].

The following variables may be used for evaluating and improving the $k$th component and the overall system [5]:

- Environmental impact rate associated with the exergy destruction within the $k$th component

$$\dot{B}_{D,k} = b_{F,k} \dot{E}_{D,k} \tag{7}$$

- Total environmental impact associated with a component $(\dot{Y}_{k} + \dot{B}_{k}^{PF} + \dot{B}_{D,k})$.

- Relative environmental impact difference

$$r_{b,k} = \frac{b_{p,k} - b_{F,k}}{b_{F,k}} = \frac{1 - \varepsilon_{k}}{\varepsilon_{k}} \frac{\dot{Y}_{k} + \dot{B}_{k}^{PF}}{\dot{B}_{D,k}} \tag{8}$$

- Exergoenvironmental factor

$$f_{b,k} = \frac{\dot{Y}_{k}^{CO} + \dot{B}_{k}^{PF}}{\dot{Y}_{k}^{CO} + \dot{B}_{k}^{PF} + \dot{B}_{D,k}} = \frac{\dot{Y}_{k}^{CO} + \dot{B}_{k}^{PF}}{\dot{Y}_{k}^{CO} + \dot{B}_{k}^{PF} + \dot{B}_{F,k} E_{D,k}} \tag{9}$$

3. Example

Figure 1 shows a cogeneration system based on an open-cycle gas-turbine power system. This cogeneration system is known as CGAM problem – an academic example used to demonstrate the different exergy-based methods for optimization [23] as well as the application of exergetic and exergoeconomic analyses [24]. An exergoenvironmental analysis for this system was already presented in [5,6]. Here only main data from References [6,23,24] are repeated, in order to understand the new results. The cogeneration system generates net power $\dot{W}_{net} = 30$ MW and saturated steam (stream 9) with $m_{water} = 14$ kg/s at a pressure of 20 bar.

Table 1 contains the main results of the simulation. Table 2 (first column) and Table 3 show the definition and the values of the exergy of fuel and exergy of product for the $k$th component of the cogeneration system and results obtained from the exergetic analysis.

The equations used for the $k$th component in the exergoenvironmental analysis are given in the second column of Table 2. The value of the environmental impact associated with the fuel (methane) $b_{i0}$ depends on the approach used for LCA. There are two approaches to estimate the value of $b_{i0}$.
• Approach 1: The value of $b_{10}$ associated only with the fuel itself without pollutant formation. The values of $b_{10}$ for the different eco-indicators are given in Table 4. The value of $\dot{B}_k^{PF}$ should be estimated separately using Eq. (6). For the system being analyzed, only CO$_2$ and NO$_x$ are taken into account, i.e. $\dot{B}_k^{PF} = b_{CO_2}^{PF} (\dot{m}_{CO_2,\text{out}} - \dot{m}_{CO_2,\text{in}}) + b_{NO_x}^{PF} (\dot{m}_{NO_x,\text{out}} - \dot{m}_{NO_x,\text{in}})$, with $\dot{m}_{CO_2,\text{in}} = 0.019$ kg/s, $\dot{m}_{CO_2,\text{out}} = 0.322$ kg/s, and $\dot{m}_{NO_x,\text{out}} = 0.001$ kg/s. The values of $b_{CO_2}^{PF}$ and $b_{NO_x}^{PF}$ for the different eco-indicators are also given in Table 4 as well as the values of $B_{CC}^{PF}$.

• Approach 2: The value of $b_{10}$ associated with the fuel includes the pollutant formation, for example, according to the average data for a combustion process. This data is available for few eco-indicators (Table 4). For this approach the value of $\dot{B}_k^{PF}$ is equal to 0. Note that the variable $\dot{B}_k^{PF}$ cannot be estimated and used in conjunction with the method of cumulative exergy consumption, because in CExC the effect of pollutants cannot be appropriately considered since exergy is the only measure of environmental impact.

![Diagram of a gas-turbine-based cogeneration system (CGAM problem)]

**Fig. 1. A gas-turbine-based cogeneration system (CGAM problem)**

**Table 1. Thermodynamic data for the cogeneration system shown in Figure 1**

<table>
<thead>
<tr>
<th>Stream</th>
<th>Material of stream</th>
<th>$\dot{m}$ [kg/s]</th>
<th>$T$ [K]</th>
<th>$p$ [bar]</th>
<th>$e$ [MJ/kg]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Air</td>
<td>91.28</td>
<td>298.1</td>
<td>1.01</td>
<td>0</td>
</tr>
<tr>
<td>2</td>
<td>Air</td>
<td>91.28</td>
<td>603.7</td>
<td>10.13</td>
<td>0.302</td>
</tr>
<tr>
<td>3</td>
<td>Air</td>
<td>91.28</td>
<td>850.0</td>
<td>9.62</td>
<td>0.459</td>
</tr>
<tr>
<td>4</td>
<td>CG</td>
<td>92.92</td>
<td>1520.0</td>
<td>9.14</td>
<td>1.092</td>
</tr>
<tr>
<td>5</td>
<td>CG</td>
<td>92.92</td>
<td>1006.2</td>
<td>1.10</td>
<td>0.417</td>
</tr>
<tr>
<td>6</td>
<td>CG</td>
<td>92.92</td>
<td>779.8</td>
<td>1.07</td>
<td>0.234</td>
</tr>
<tr>
<td>7</td>
<td>CG</td>
<td>92.92</td>
<td>426.9</td>
<td>1.01</td>
<td>0.030</td>
</tr>
<tr>
<td>8</td>
<td>Water</td>
<td>14.00</td>
<td>298.1</td>
<td>20</td>
<td>0.04</td>
</tr>
<tr>
<td>9</td>
<td>Water</td>
<td>14.00</td>
<td>485.6</td>
<td>20</td>
<td>0.915</td>
</tr>
<tr>
<td>10</td>
<td>CH4</td>
<td>1.64</td>
<td>298.1</td>
<td>12</td>
<td>51.825</td>
</tr>
<tr>
<td>11</td>
<td>Power to AC*)</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>12</td>
<td>Net power**)</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

*) $W_{AC} = 29.662$ MW, **) $W_{net} = 30$ MW

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<table>
<thead>
<tr>
<th>Component</th>
<th>Exergetic analysis</th>
<th>Exergoenvironmental analysis</th>
</tr>
</thead>
<tbody>
<tr>
<td>AC</td>
<td>$\dot{E}<em>{F,AC} = \bar{W}</em>{AC}$</td>
<td>$b_{\text{w}} \dot{W}<em>{AC} + \dot{Y}</em>{AC} = b_2 \dot{E}_2 - b_4 \dot{E}_1$</td>
</tr>
<tr>
<td></td>
<td>$\dot{E}_{P,AC} = \dot{E}_2 - \dot{E}_1$</td>
<td>with $b_1 = 0$</td>
</tr>
<tr>
<td>APH</td>
<td>$\dot{E}_{F,APH} = \dot{E}_5 - \dot{E}_6$</td>
<td>$(b_5 \dot{E}_5 - b_6 \dot{E}<em>6) + \dot{Y}</em>{APH} = b_9 \dot{E}_9 - b_2 \dot{E}_3$</td>
</tr>
<tr>
<td></td>
<td>$\dot{E}_{P,APH} = \dot{E}_3 - \dot{E}_2$</td>
<td>with $b_k = b_6$ (F rule [5])</td>
</tr>
<tr>
<td>CC</td>
<td>$\dot{E}<em>{F,CC} = \dot{E}</em>{10}$</td>
<td>$b_{10} \dot{E}<em>{10} + (\dot{Y}</em>{CC} + \dot{B}_{CC}^{\text{FP}}) = b_2 \dot{E}_4 - b_3 \dot{E}_3$</td>
</tr>
<tr>
<td></td>
<td>$\dot{E}_{P,CC} = \dot{E}_4 - \dot{E}_3$</td>
<td>$b_{10} =$ known (Table 4)</td>
</tr>
<tr>
<td>GT</td>
<td>$\dot{E}_{F,GT} = \dot{E}_4 - \dot{E}_5$</td>
<td>$(b_2 \dot{E}<em>4 - b_3 \dot{E}<em>3) + \dot{Y}</em>{GT} = b_9 \dot{W}</em>{GT}$</td>
</tr>
<tr>
<td></td>
<td>$\dot{E}<em>{P,GT} = \dot{W}</em>{GT}$</td>
<td>with $b_4 = b_3$ (F rule)</td>
</tr>
<tr>
<td>HRSG</td>
<td>$\dot{E}_{F,HRSG} = \dot{E}_6 - \dot{E}_7$</td>
<td>$(b_6 \dot{E}_6 - b_7 \dot{E}<em>7) + \dot{Y}</em>{HRSG} = b_9 \dot{E}_9 - b_8 \dot{E}_8$</td>
</tr>
<tr>
<td></td>
<td>$\dot{E}_{P,HRSG} = \dot{E}_9 - \dot{E}_8$</td>
<td>with $b_6 = b_7$ (F rule) and $b_8 = 0$ (arbitrary assumption)</td>
</tr>
</tbody>
</table>

Table 3. Exergetic analysis the cogeneration system shown in Figure 1

<table>
<thead>
<tr>
<th>Component</th>
<th>$\dot{E}_{F,k}^{\text{real}}$ [MW]</th>
<th>$\dot{E}_{P,k}^{\text{real}}$ [MW]</th>
<th>$\dot{E}_{D,k}^{\text{real}}$ [MW]</th>
<th>$\varepsilon_k$ [%]</th>
<th>$\eta_k$ [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>AC</td>
<td>29.66</td>
<td>27.54</td>
<td>2.12</td>
<td>92.84</td>
<td>2.50</td>
</tr>
<tr>
<td>APH</td>
<td>16.93</td>
<td>14.40</td>
<td>2.53</td>
<td>84.55</td>
<td>3.09</td>
</tr>
<tr>
<td>CC</td>
<td>85.00</td>
<td>59.52</td>
<td>25.48</td>
<td>80.37</td>
<td>29.98</td>
</tr>
<tr>
<td>GT</td>
<td>62.67</td>
<td>59.66</td>
<td>3.01</td>
<td>95.20</td>
<td>3.54</td>
</tr>
<tr>
<td>HRSG</td>
<td>18.98</td>
<td>12.75</td>
<td>6.23</td>
<td>67.17</td>
<td>7.33</td>
</tr>
<tr>
<td><strong>Overall system</strong></td>
<td><strong>85.000</strong></td>
<td><strong>42.750</strong></td>
<td><strong>39.370</strong></td>
<td><strong>50.3</strong></td>
<td><strong>46.3</strong></td>
</tr>
</tbody>
</table>

Table 4. Environmental impact of the fuel and pollutants

<table>
<thead>
<tr>
<th>Substance</th>
<th>ECO-95</th>
<th>ECO-99</th>
<th>ECO-99</th>
<th>CExC</th>
<th>CML</th>
<th>ECO-F2006</th>
<th>ECO-F2006</th>
</tr>
</thead>
<tbody>
<tr>
<td>Methane as fuel, $b_{10}$</td>
<td>0.063</td>
<td>3.5</td>
<td>5.38</td>
<td>0.00104</td>
<td>0.02671</td>
<td>0.97</td>
<td>3.3</td>
</tr>
<tr>
<td>$\text{CO}<em>2$ as pollutant, $b^{\text{FP}}</em>{\text{CO}_2}$</td>
<td>-</td>
<td>5.454</td>
<td>-</td>
<td>-</td>
<td>0.0636</td>
<td>310</td>
<td>-</td>
</tr>
<tr>
<td>NO$\text{x}$ as pollutant, $b^{\text{FP}}_{\text{NO}_x}$</td>
<td>-</td>
<td>2749.36</td>
<td>-</td>
<td>-</td>
<td>0.395</td>
<td>92000</td>
<td>-</td>
</tr>
<tr>
<td>$B_{\text{FF}}^{\text{CC}}$ based on Eq. (6)</td>
<td>-</td>
<td>4.402</td>
<td>-</td>
<td>-</td>
<td>0.01967</td>
<td>185.93</td>
<td>-</td>
</tr>
</tbody>
</table>

* average statistical data for a low NO$\text{x}$ combustion process
Approach 1 is more accurate, whereas Approach 2 can be used if the $b_i^{PF}$ data necessary for Approach 1 are not available.

$\dot{Y}_k$ is a term in the exergoenvironmental balance (Eqs. (4) and (5)). In the previous applications of the exergoenvironmental analysis [5-11], the value of $\dot{Y}_k$ was set equal to $\dot{Y}_k^{CO}$ that is the biggest contributor to $\dot{Y}_k$. In Ref. [6] the values of $\dot{Y}_k$ for the system being analyzed have already been discussed. In all publications related to an exergoenvironmental analysis [5-11] the authors conclude that the value of $\dot{Y}_k$ is very small compared with the value of the environmental impact associated with the exergy destruction ( $\dot{B}_{D,k}$) and can be neglected in the analysis without effect to the results. In this paper we considered the value of $\dot{Y}_k$ only in conjunction with ECO-99.

Table 5 presents the results obtained from the exergoenvironmental analysis using five eco-indicators and three approaches for considering the values of $\dot{B}_k^{PF}$ and $\dot{Y}_k$.

The environmental impact associated with the total product of the cogeneration system (electricity and heat) can be calculated from the environmental impact balance applied to the overall system as

$$b_{P, tot} \dot{E}_{P, tot} = b_{F, tot} \dot{E}_{F, tot} + \left( \dot{Y}_{tot} + \dot{B}_k^{PF} \right) - \dot{B}_{L, tot}$$  \hspace{1cm} (10)

When the environmental impact associated with the exergy losses of the overall system ($\dot{B}_{L, tot} = \dot{B}_g$) is charged to the product, we obtain

$$b^*_{P, tot} = \frac{b_{P, tot} \dot{E}_{P, tot} + \dot{B}_{L, tot}}{\dot{E}_{P, tot}}$$  \hspace{1cm} (11)

In order to estimate the environmental impacts associated with electricity and steam, the value of $\dot{B}_{L, tot}$ should be split using the terms $W_{net}$ and $(\dot{E}_g - \dot{E}_k)$ as weighting factors:

$$b_w^* = \frac{b_w W_{net} + \dot{B}_{L, tot} (\dot{E}_g - \dot{E}_k) + W_{net}}{W_{net}} = b_w + \frac{\dot{B}_{L, tot}}{(\dot{E}_g - \dot{E}_k) + W_{net}}$$  \hspace{1cm} (12)

The following data are obtained:

- Using ECO-95, we obtained $b^*_{w} = 0.39$ mPts/kWh while the average value for Europe measured in ECO-95 points is 0.75 mPts/kWh [13].
- Using ECO-99, we obtained three values $b^*_{w} = 2.54$ mPts/kWh if $\dot{B}_k^{PF} = 0$ (for the case with the pollutant formation included in the value of $b_{10}$), $b^*_{w} = 1.702$ mPts/kWh (with separate consideration of the formation of pollutants) and $b^*_{w} = 1.703$ mPts/kWh (with separate consideration of the formation of pollutants and consideration of the values $\dot{Y}_k$). The average value for electricity in Europe according to ECO-99 is 27 mPts/kWh [14].
- For CExC we obtain $b^*_{w} = 1.795$ kJ/kJ while the average environmental impact associated with electricity generation according to Ref. [25] is equal to 12.857 kJ/kJ. Note that in Ref. [25] a coal power plant is discussed. Apparently, coal power plants have a much higher relative environmental impact than gas-turbine power systems.
Table 5. Exergoenvironmental analysis of the cogeneration system shown in Figure 1

<table>
<thead>
<tr>
<th>Component</th>
<th>$\dot{Y}_k$</th>
<th>$\dot{B}_{D,k}$</th>
<th>$\dot{B}_{k,P}^{PF}$</th>
<th>$\dot{Y}<em>k + \dot{B}</em>{D,k}^{PF}$</th>
<th>$b_{F,k}$</th>
<th>$b_{P,k}$</th>
<th>$r_{h,k}$</th>
<th>$f_{h,k}$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>(mPts/h)</td>
<td>(mPts/h)</td>
<td>(mPts/h)</td>
<td>(mPts/h)</td>
<td>(mPts/MJ)</td>
<td>(mPts/MJ)</td>
<td>(%)</td>
<td>(%)</td>
</tr>
<tr>
<td>ECO-95</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>AC</td>
<td>–</td>
<td>780</td>
<td>–</td>
<td>780</td>
<td>0.1022</td>
<td>0.1132</td>
<td>10.73</td>
<td>0</td>
</tr>
<tr>
<td>APH</td>
<td>–</td>
<td>950</td>
<td>–</td>
<td>950</td>
<td>0.1003</td>
<td>0.1187</td>
<td>18.26</td>
<td>0</td>
</tr>
<tr>
<td>CC</td>
<td>–</td>
<td>5779</td>
<td>–</td>
<td>5779</td>
<td>0.0630</td>
<td>0.0900</td>
<td>42.81</td>
<td>0</td>
</tr>
<tr>
<td>GT</td>
<td>–</td>
<td>1087</td>
<td>–</td>
<td>1087</td>
<td>0.1003</td>
<td>0.1022</td>
<td>1.86</td>
<td>0</td>
</tr>
<tr>
<td>HRSG</td>
<td>–</td>
<td>2250</td>
<td>–</td>
<td>2250</td>
<td>0.1003</td>
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Table 5 (continuation)

| ECO – F2006 ($\hat{B}_k^{HF} \neq 0$, $\hat{Y}_k \neq 0$) |
|---------------------------------|-----------------|-----------------|-----------------|-----------------|-----------------|-----------------|-----------------|
| (mPts/h)                        | (mPts/h)        | (mPts/h)        | (mPts/MJ)       | (mPts/MJ)       | (%)             | (%)             |
| AC                              | 126             | 43989           | –               | 44115           | 5.764           | 6.382           | 10.73           | 0.03            |
| APH                             | 145             | 53571           | –               | 53716           | 5.658           | 6.694           | 18.31           | 0.27            |
| CC                              | 20              | 321048          | 15847           | 336915          | 3.500           | 5.072           | 44.92           | 4.71            |
| GT                              | 58              | 61311           | –               | 61369           | 5.658           | 5.764           | 1.87            | 0.09            |
| HRSG                            | 119             | 126900          | –               | 127019          | 5.658           | 8.426           | 48.92           | 0.09            |
| Overall system                  | 468             | 497322          | 15847           | 513637          | 3.500           | 6.925           | 97.86           | 3.18            |

Cumulative exergy consumption (CExC)

| Cumulative exergy consumption (CExC) |
|-------------------------------------|-----------------|-----------------|-----------------|-----------------|-----------------|-----------------|
| (kJ/h)                              | (kJ/h)          | (kJ/h)          | (kJ/MJ)         | (kJ/MJ)         | (%)             | (%)             |
| AC                                  | –               | 12.88           | –               | 12.88           | 0.00169         | 0.00187         | 10.73           | 0               |
| APH                                 | –               | 15.68           | –               | 15.68           | 0.00166         | 0.00196         | 18.26           | 0               |
| CC                                  | –               | 95.40           | –               | 95.40           | 0.00104         | 0.00148         | 42.81           | 0               |
| GT                                  | –               | 17.95           | –               | 17.95           | 0.00166         | 0.00169         | 1.86            | 0               |
| HRSG                                | –               | 37.15           | –               | 37.15           | 0.00166         | 0.00247         | 48.88           | 0               |
| Overall system                      | –               | 147.8           | –               | 147.8           | 0.00104         | 0.00203         | 95.19           | 0               |

CM L

| CM L                               |
|------------------------------------|-----------------|-----------------|-----------------|-----------------|-----------------|-----------------|
| (ELU/h)                            | (ELU/h)         | (ELU/h)         | (ELU/MJ)        | (ELU/MJ)        | (%)             | (%)             |
| AC                                 | –               | 334             | –               | 334             | 0.0437          | 0.0484          | 10.73           | 0               |
| APH                                | –               | 406             | –               | 406             | 0.0429          | 0.0507          | 18.26           | 0               |
| CC                                 | –               | 2450            | 71              | 2521            | 0.0267          | 0.0385          | 44.05           | 2.81            |
| GT                                 | –               | 465             | –               | 465             | 0.0429          | 0.0437          | 1.86            | 0               |
| HRSG                               | –               | 962             | –               | 962             | 0.0429          | 0.0639          | 48.88           | 0               |
| Overall system                      | –               | 3795            | 71              | 3866            | 0.0267          | 0.0525          | 96.63           | 1.84            |

ECO – F2006

| ECO – F2006                        |
|------------------------------------|-----------------|-----------------|-----------------|-----------------|-----------------|-----------------|
| AC                                 | –               | 40861           | –               | 40861           | 5.354           | 5.928           | 10.73           | 0               |
| APH                                | –               | 49764           | –               | 49764           | 5.256           | 6.216           | 18.26           | 0               |
| CC                                 | –               | 302702          | –               | 302702          | 3.300           | 4.713           | 42.81           | 0               |
| GT                                 | –               | 56954           | –               | 56954           | 5.256           | 5.354           | 1.86            | 0               |
| HRSG                               | –               | 117881          | –               | 117881          | 5.256           | 7.825           | 48.88           | 0               |
| Overall system                      | –               | 468904          | 468904          | 3.300           | 6.432           | 94.91           | 0               |
Table 5 (continuation)

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</table>

- Using the CML method, the environmental impact associated with the electricity generation is $b^*_w = 0.0129$ ELU/kWh, and
- Using the ECO-factor 2006 $b^*_w = 1.582$ EP/kWh (for the case with the pollutant formation included in the value of $b_{10}$) and $b^*_w = 1.514$ EP/kWh (with separate consideration of the formation of pollutants).

Note that for the CML and the ECO-factor 2006 average data for the environmental impact associated with electricity generation are not reported; therefore, we cannot compare the obtained results with the data mentioned in References [21] and Ref.[22].

If $\dot{Y}_k$ and $\dot{B}_k^{PF}$ are not considered (ECO-95, ECO-99, CE\text{x}C and ECO-F2006), then the following conclusions can be obtained: Independently of the used method for the LCA, the absolute results of the exergoenvironmental analysis are different, but the relative values remain the same (for example, values $r_{b,k}$ and $r_{b,\text{tot}}$). Therefore, the conclusions (recommendations for improving the system) are independent of the method used for the LCA. The variable $f_{b,k}$ cannot be meaningfully used in this approach.

If $\dot{Y}_k$ is considered (ECO-99), this value does not significantly affect the results, and the conclusions from the analysis are the same without considering the value $\dot{Y}_k$.

If the value of $\dot{B}_k^{PF}$ is considered as a separate term in Eq. (4), then different methods used for the LCA lead to different results: For example, (a) using ECO-99, $\dot{B}_{CC}^{PF} < \dot{B}_{D,CC}$ and $r_{b,\text{tot}} = 98\%$, (b) using CML, $\dot{B}_{CC}^{PF} << \dot{B}_{D,CC}$ and $r_{b,\text{tot}} = 97\%$, whereas (c) using ECO-F2006 $\dot{B}_{CC}^{PF} >> \dot{B}_{D,CC}$ and $r_{b,\text{tot}} = 534\%$.

**Conclusions**

An exergoenvironmental analysis demonstrates the formation of environmental impacts associated with energy conversion systems at the component level.

This paper deals with the effect of the selected eco-indicator on the results and the conclusions obtained from the exergoenvironmental analysis. Five eco-indicators (ECO-95, ECO-99, CE\text{x}C, CML and ECO-F2006) with two approaches for estimating the pollutants formation process were used for the environmental evaluation of a gas-turbine system (the so-called CGAM problem).
The total environmental impact associated with a component of the overall system consists in three terms: component-related environmental impact, environmental impact associated with the pollutant formation (only for the combustion chamber) and environmental impact associated with exergy destructions (e.g., Ref.[5]). The value of the environmental impact of pollutant formation can be calculated only if the used eco-indicator provides the necessary data. Calculating the value of the component-related environmental impact is a difficult task because many data related to the material consumption for manufacturing the equipment must be estimated. For the exergo-environmental analysis, however, this value can in the majority of the energy conversion systems be neglected (independently of the selected eco-indicator). The value of environmental impact associated with exergy destruction and, where appropriate, the value of pollutant formation should be considered in the analysis. Finally we can conclude that in many cases an exergy conversion system can be improved from the environmental point of view simply by improving its thermodynamic efficiency because the lower the exergy destruction within a component, the lower the environmental impact associated with it. Note that the environmental impact associated with pollutant formation is calculated only for the “pollutant producing” components.

Nomenclature

\( b \) \hspace{1cm} \text{environmental impact per unit of exergy, Pts/J, kJ/J, ELU/J, EP/J or per unit of mass, Pts/kg, kJ/kg, ELU/kg, EP/kg}

\( \dot{B} \) \hspace{1cm} \text{environmental impact rate associated with exergy, Pts/s, kJ/s, ELU/s, EP/s}

\( e \) \hspace{1cm} \text{specific exergy, J/kg}

\( \dot{E} \) \hspace{1cm} \text{exergy rate, W}

\( j \) \hspace{1cm} \text{\( j \) th stream}

\( f_{b} \) \hspace{1cm} \text{exergoenvironmental factor, %}

\( k \) \hspace{1cm} \text{\( k \) th component}

\( m \) \hspace{1cm} \text{mass flow rate, kg/s}

\( p \) \hspace{1cm} \text{pressure, bar}

\( r_{b} \) \hspace{1cm} \text{relative environmental impact difference, %}

\( T \) \hspace{1cm} \text{temperature, °C}

\( W \) \hspace{1cm} \text{power, W}

\( y \) \hspace{1cm} \text{exergy destruction ratio, %}

\( \dot{Y} \) \hspace{1cm} \text{environmental impact, Pts/s, kJ/s, ELU/s, EP/s}

Greek symbols

\( \Delta \) \hspace{1cm} \text{difference}

\( \varepsilon \) \hspace{1cm} \text{exergetic efficiency, %}

Superscripts

\( \dot{} \) \hspace{1cm} \text{time rate}

\( BF \) \hspace{1cm} \text{pollutants formation}

Subscripts

\( D \) \hspace{1cm} \text{exergy destruction}

\( F \) \hspace{1cm} \text{exergy of fuel}

\( j \) \hspace{1cm} \text{\( j \) th stream}
$k$  $k$th component

$P$  exergy of product

**Abbreviations**

AC  air compressor
APH  air preheater
CC  combustion chamber
HRSG  heat-recovery steam generator
GT  gas turbine

**References**


Optimisation of supply temperature and mass flow rate for a district heating network

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Abstract:
Heat losses and pump electrical energy consumption are inevitable during the operation of a district heating (DH) network. Therefore, the appropriate design and operating strategy of a DH network is required in order to reduce the heat losses and pump energy consumption. This study is aimed at optimising the supply water temperature and mass flow rate in a DH network, according to the heating load variations. For this purpose, two different design cases of a DH pipe network were considered. An optimisation model was developed using FICO\textsuperscript{TM} Xpress optimisation tool. Hydraulic and thermal calculations of the model were validated with the commercial software PSS SINCAL. The DH networks were optimised with respect to the annual energy (heat losses and pump energy consumption), and associated costs. The optimum supply temperature and mass (volume) flow rate were obtained for each case. Consequently, optimum heat losses and pump power were calculated.

Keywords:
District heating network, Modelling, Design, Optimisation, Supply temperature

1. Introduction
District heating (DH) systems provide multiple buildings or dwellings with heat and domestic hot water (DHW) from a central energy centre. Heat is transferred from the central energy centre through a network of insulated pipes, carrying the hot water to each building. DH systems, particularly for densely populated urban environments, have proved to be sustainable and efficient systems compared with individual boilers [1]. However operational costs, related to the pump’s power and heat losses in DH pipe networks, present the major drawback of DH compared to the individual heating system. The operational costs can be reduced by energy efficient design as well as a better energy management during the operation. Therefore the reduction of heat energy losses and pump energy consumption is one of the most important tasks, which reduces costs and improves the efficiency of a DH system.

Modelling and operation of a DH network is addressed in a number of studies. Short-term optimal operation of a DH system to find the most economic way of fulfilling consumer’s heat requirement, is presented in ref [2-6]. An optimisation model, considering the dynamic of the network, which incorporates the consumers, the district heating network and the production plant was developed to minimise operational costs [2-3]. An equivalent model of DH network with regards to on-line optimisation of the operational costs of the complete system was developed [4]. An optimisation model, taking into account the dynamic character of the DH network, was formulated to minimise operating costs and maximise the profit of the system [5]. Reference [6] specifically addresses the modelling and optimal operation of Micro-grid system.

Determination of pipe sizes is a first task in a DH pipe network. Pipe diameters are usually chosen based on maximum flow and pressure loss. Target pressure loss (TPL) is a common design parameter of DH pipe networks. DH networks are designed using pressure losses of 50-200 Pa/m [7]. Many DH networks in Denmark and other EU countries have been designed based on pressure loss of 100 Pa/m [8]. However, in newer studies much higher pressure loss is used. In a study of Li
et al. [9] for the determination of pipe size, the pressure loss of 500 Pa/m was used for main pipes. In the meantime, pressure loss of 1500 Pa/m was used as the dimension criteria for the street pipes [9]. A much higher pressure loss of 2000 Pa/m was used for low energy DH systems in energy efficient building area [10-11].

In this study two different design cases of a DH pipe network were prepared using SINCAL. Optimal annual operation of each DH design case was carried out, using FICO™ Xpress. The objective of the optimisation was to determine the optimum annual supply temperature and flow rate. Consequently, using obtained temperature and flow rate, optimum heat losses and pump energy consumption were calculated. The obtained results were compared and analysed.

2. District heating model

District energy model based upon a real project redevelopment in South Wales, UK, was applied [12]. A simplified diagram of the DH pipe connection is shown in Fig.1. Consumers were geographically split into a set of clusters. Consumers have different occupancy type and building size within a cluster.

![Simplified diagram of DH case study](image)

Each cluster was connected to the DH network using heating substation. Maximum energy demand for space heating (SH) and domestic hot water (DHW) was calculated based on estimated area for each building [13-14].

Mean daily DHW requirement calculated for each building was assumed to be constant over the whole year. Energy demand for SH varied proportionally to the outdoor temperature. Daily SH demand was calculated using the concept of heating degree days (HDD) [15-17]. The base temperature of 15.5 °C was assumed and the minimum outdoor temperature was considered to be -3 °C in this study. The total heating demand of SH was calculated for each day over the year. The heating load was divided into two main seasons. The winter season lasts for 182 days and includes energy demand for SH and DHW. For the rest of the year (summer season) only energy demand for DHW was taken into account. The total heat demand (right) and annual load duration curve (left) are shown in Fig.2.
2.1. Design cases
In this study, two approaches were used. First, system was designed based on low pressure loss (Case 1: TPL of 100 Pa/m). Then, the design was reiterated for much higher pressure loss (Case 2: TPL of 1200 Pa/m). The pipe diameters and heating system design parameters for both design cases are given in Table 1, and Table 2.
The difference between TPL and actual pressure loss is due to the selection of standard pipe sizes available in the market [18]. Standard sizes of pre-insulated steel pipes were selected. Pipes diameters, calculated in PSS SINCAL, were different compared with the standard size. The pipe with the diameter which was closest to the calculated pipe diameter was selected. By reiterating the calculation with actual pipes diameter the actual pressure loss in the system was obtained.

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<th>Table 1. DH design</th>
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<tr>
<td>11</td>
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<tr>
<td>12</td>
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</tbody>
</table>
2.2. Optimisation model
The optimisation model determines the optimum system flow rate and system supply temperature at each time step over a year, at a steady state condition.

The suitable method to describe a DH pipe network is to use the concept of graph theory [19]. A graph is commonly defined as a combination of

- A set of nodes
- A set of branches, and
- An incident relation

Each branch within the graph is connected to a pair of nodes, the node where the branch starts and the node where it ends. A DH network can be treated as a network graph. The branches represent pipes and nodes symbolise the points where pipes are connected. In the study the DH network graph consists of 13 nodes and 12 pipes (Fig.1.)

The connectivity matrix was created to explain the DH network with \( n \) nodes and \( m \) flow stream (branches). The program stores the network graph in a matrix data structure such as \( (A) \) which is a \( kn \times mf \) matrix, containing the incidence elements \( (a_{kj}) \).

\[
\begin{align*}
a_{kj} &= -1 & \text{If pipe } j \text{ starts at node } k, \\
a_{kj} &= 1 & \text{If pipe } j \text{ ends at node } k, \\
a_{kj} &= 1 & \text{If source } j \text{ ends at node } k, \\
a_{kj} &= -1 & \text{If load } j \text{ starts at node } k, \\
a_{kj} &= 0 & \text{Otherwise}
\end{align*}
\]

The connectivity matrix has one column for each flow stream, and one row for each node. The connectivity matrix \( (A) \) was used to calculate flow in supply pipes. The flow rates in the return pipes are calculated in the same manner as in supply pipes. For the calculation of flow in the return pipes, the connectivity matrix was simply multiplied by -1. In this study, since supply and return pipes are assumed to be identical, it was found that flow and pressure loss in the supply and return pipes were nearly the same.

The energy flow and flow rate in each branch, temperature (supply and return), and pressure at each node were calculated using equations found in the thermal engineering textbooks [20-21].

The total heat supplied (SH and DHW) to the network was calculated using the following equation.

\[
Q_{SH,DHW} = \eta \left( T_s - T_r \right) \tag{1}
\]

Heat supply at each branch was obtained using Kirchhoff rule at nodes.

<table>
<thead>
<tr>
<th>Case</th>
<th>( T_{s,\text{max}} / T_{r,\text{max}} ) (°C)</th>
<th>( V_{\text{max}} ) (m³/s)</th>
<th>( P_{\text{i loss,\max}} ) (Pa/m)</th>
<th>( \Delta P_{p,\text{max}} ) (kPa)</th>
<th>( P_{p,\text{max}} ) (kW)</th>
<th>( v_{\text{max}} ) (m/s)</th>
</tr>
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<tbody>
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<td>99</td>
<td>158</td>
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<td>1.7</td>
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<td>1276</td>
<td>1454</td>
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</table>

Table 2. Heating system design parameters
A linear approximation of the exponential decay of temperature was used to calculate temperature at the outlet of the pipes [6].

\[ T_{out,j} = \begin{cases} 
(T_{in,j} - T_g)(1 - \frac{U_j}{cm_j}) + T_g & \frac{U_j}{cm_j} \leq 1 \\
T_g & \frac{U_j}{cm_j} \geq 1 
\end{cases} \]  

At nodes with more than one branches or supply sources, the nodes temperature was calculated as the mixed of incoming streams temperatures to the node [19-20]. For example, if pipe 1 receives hot water from pipe 2 and 3, then temperature at node 1 is:

\[ T_k = \frac{\sum_{j=1}^{m} \dot{m}_j T_{out,j}}{\sum_{j=1}^{m} \dot{m}_j} = \frac{\dot{m}_2 T_{out,2} + \dot{m}_3 T_{out,3}}{\dot{m}_2 + \dot{m}_3} = T_i \]  

Heat loss in each pipe section was obtained by the equation:

\[ Q_{loss,j} = cm_j (T_{in,j} - T_{out,j}) \quad \text{for} \quad j = 1, 2, 3, \ldots, m \]  

The total heat loss in the DH pipe network is the sum of the heat loss in each pipe section of supply and return pipes. According to the law of conservation of mass, flow in each pipe was calculated using the Kirchhoff law.

\[ \sum_{j=1}^{m} \dot{m}_{jk} = 0 \]  

The pump head of variable speed pump was obtained using the equation [22]:

\[ \Delta p_p = \left( \frac{\dot{m}}{\dot{m}_{max}} \right)^2 \cdot \Delta p_{p, max} \]  

Pump power (in kW) was calculated by the following equation:

\[ P_p = \frac{\Delta p_p \dot{m}}{1000 \rho \eta_p} \]  

### 2.3. Validation

Model validation, guarantees the credibility of the results. Hence, the heating calculations of the model were validated with the commercial software PSS SINCAL. For this purpose, for both cases the heat losses and pump power over the whole year calculated in Xpress model were validated with the commercial software PSS SINCAL. Model validation is shown in Fig.3. In terms of credibility of the result, it is seen that Xpress model provides
approximately the same result as PSS SINCAL model. Therefore Xpress model was used for further analysis of the DH network.

Fig. 3. Model validation

2.4. Objective function

The optimisation model determines the optimum system flow rate and system supply temperature when total annual energy consumption and losses of the main pipe network is minimal. Detailed modelling of the secondary network was not considered.

In a DH network, total energy consumption and losses include pump electrical energy requirement and heat energy losses to the ground. Therefore, total operational costs consist of pumping cost and heat losses cost. The objective function was:

\[
\begin{align*}
\text{Min} & \sum_{t=1}^{n} \left[24 \times \left( P_{p,t} + Q_{loss, t} \right) \right] \\
\text{Min} & \sum_{t=1}^{n} \left[24 \times \left( P_{p,t} \times CE_{t} + Q_{loss, t} \times CH_{t} \right) \right]
\end{align*}
\]

Two objective functions were considered. The equation (9) considers the minimisation of energy losses and pump energy consumption, regardless of the energy price. Exergy losses were not calculated in this study. In equation (10) minimisation of operational costs is addressed. The only difference between equation (9) and equation (10) is the inclusion of energy price.

Supply and return temperature of the pipes was within the permissible range:

\[
\begin{align*}
T_{s, \text{min}} & \leq T_s \leq T_{s, \text{max}} \\
T_{r, \text{min}} & \leq T_r \leq T_{r, \text{max}}
\end{align*}
\]

Maximum supply temperature of 120 °C during winter season was considered. For the entire summer regime, supply temperature was fixed at 70 °C in both cases, as there was only demand for DHW. For both cases, the return temperature was assumed to be constant in the consumer’s substations. For the entire winter season, return temperature was assumed to be 40 °C in all substations. In the summer season, the return temperature of 30 °C was taken into account.
The system flow rate (total flow in the source side) was required to be within minimum and maximum limits.

\[ \dot{m}_{\text{min}} \leq \dot{m} \leq \dot{m}_{\text{max}} \]  

(13)

The maximum differential pressure of pump was equal or higher than the maximum pressure drop in the network.

\[ \Delta p_p \leq \Delta p_{p,\text{max}} \]  

(14)

The optimisation model was formulated as a nonlinear objective function with nonlinear constraints, using FICO™ Xpress optimisation. Xpress optimisation allows modelling and solving of large and complex optimisation problems. For the study, successive (sequential) linear programming (Xpress-SLP) was used. Xpress-SLP is a solver for nonlinear optimisation problems. It uses successive linear approximation which has been developed from techniques used in the process industries and it can solve large problems with many thousands of variables [23, 24].

3. Results

3.1. Minimisation of energy consumption

The results of the optimisation based on minimisation of energy losses and consumption are shown in Fig.4. The energy demand for heating decreases over the year (according to the outdoor temperature). Therefore, the temperature and the flow rate decrease. As a result, heat losses and pump power reduce in both cases. However, the temperature and flow rate profiles of different cases are very different.

![Fig.4. Optimal DH network parameters, based on minimising energy](image)
The difference is rooted in the design of the DH pipe network. For the case 1, since pipes with larger diameters were selected, pressure loss and pump energy consumption is less compared with the case 2. Heat losses are higher for larger diameter pipes. Therefore, in case 1, larger flow rate and lower supply temperature were obtained compared with the case 2. Due to lower supply temperature, heat losses decrease in case 1 while due to lower flow rate in case 2, the pump energy consumption is reduced.

In case 1, higher heat losses and lower pump power compared with the case 2 is observed, due to the different design conditions. It is clear that in terms of energy losses and consumption, heat losses in both cases are more important than pump energy requirement. In both cases during summer season, as a result of reduction in flow rate pump energy consumption is very low (Case 1: 8 W, Case 2: 65 W), since demand for DHW is less than demand for SH (about 7 % of the total peak demand). However, because supply temperature is still relatively high, heat losses are substantial in both cases.

3.2. Minimisation of costs

For the optimisation based on minimisation of costs, electricity and heat price were assumed to be constant over the year. An average price per kWh was considered for this purpose. The price of 0.095 (£/kWh) was taken into account for electricity [25, 26], while heat price was assumed to be 0.07 (£/kWh) over the whole year [27]. The results of the optimisation are shown in Fig.5.

![Fig.5. Optimal DH network parameters, based on minimising costs](image)

It is seen that as the energy demand changes over the year and supply temperature and flow rate change accordingly. As energy demand decreases heat losses and pump power requirement decrease, due to the reduction in supply temperature and flow rate. However, for different DH pipe networks, different results were obtained. As it was explained earlier, this is the consequence of the selection of pipe with different diameters and different size of pumps.
3.3. Comparison
Optimisation results for both approaches, based on minimisation of energy and costs, are shown in Fig.6.
A slight difference is seen when system is optimised based on minimisation of energy and costs. For the optimisation based on minimising costs a slightly higher temperature and smaller flow rate are observed in both cases. This is due to the price of heat energy losses and pump electricity consumption. Since the price of electricity which is consumed by pump is higher than the heat price, the optimiser increases supply temperature and reduces flow rate to avoid excessive pumping costs. Hence, the amount of optimal supply temperature, flow rate, heat losses and pump power can vary with regards to the energy price. When electricity price is higher than the heat price, it is better to reduce flow rate and increase supply temperature to avoid higher pumping cost. In the mean time, when heat price is more than the electricity price, it may be beneficial to reduce supply temperature and increase flow rate to avoid higher heat losses cost. It is worth mentioning, that results of the both optimisation approaches (minimisation of energy and costs) show similar trends of the change in supply temperature and flow rate. As the heat demand changes according to outdoor temperature, system supply temperature and flow rate change to balance the production and demand.

Fig.6. Optimal supply temperate and flow based on minimisation of energy and costs

4. Conclusion
In this study, an optimisation model was developed to optimise supply temperature and flow rate for a DH network. Optimal heating system parameters, temperature and flow rate, were found and validated. Due to the uncertainty in the price of heat and electricity, two objective functions were taken into account, one based on minimisation of energy losses and energy consumption, and another based on minimisation of operational costs. In addition, two different DH networks were designed and then optimised.
It was shown that for the DH pipe networks, optimisation results were different, due to the selection of pipes and pumps with different sizes. For a DH pipe network designed based on low pressure loss (100 Pa/m), pump energy consumption was not substantial and high heat losses were due to the selection of pipes with larger diameters. Therefore, in order to reduce heat losses, supply temperature had to be reduced and flow rate had to be increased. When the DH pipe network was designed using high pressure loss (1276 Pa/m), pipes with smaller diameters were selected. Hence, reduction of pump energy consumption was achieved by increasing supply temperature and reducing flow rate.
Heat losses were higher than pump energy consumption in both cases as well as for both optimisation approaches. It was observed, that the design case based on high pressure drop performed better since lower heat losses were obtained. Finally, it was found that price of energy (electricity and heat) have impact on optimum supply temperature and flow rate and consequently heat losses and pump energy consumption.

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Nomenclature

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
</tr>
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<tbody>
<tr>
<td>CE</td>
<td>electricity price, £/(kWh)</td>
</tr>
<tr>
<td>CH</td>
<td>heat price, £/(kWh)</td>
</tr>
<tr>
<td>c</td>
<td>specific heat capacity, kJ/(kg K)</td>
</tr>
<tr>
<td>l</td>
<td>pipe length, m</td>
</tr>
<tr>
<td>m</td>
<td>mass flow rate, kg/s</td>
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<tr>
<td>p</td>
<td>pressure, Pa</td>
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<tr>
<td>Δp</td>
<td>differential pressure, Pa</td>
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<tr>
<td>P</td>
<td>pump power, kW</td>
</tr>
<tr>
<td>Q</td>
<td>thermal power, kW</td>
</tr>
<tr>
<td>T</td>
<td>temperature, °C</td>
</tr>
<tr>
<td>(\dot{V})</td>
<td>volume flow rate, m³/s</td>
</tr>
<tr>
<td>v</td>
<td>velocity, m/s</td>
</tr>
<tr>
<td>U</td>
<td>heat transition coefficient, W/(m K)</td>
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</table>

Greek symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>(\eta)</td>
<td>efficiency</td>
</tr>
<tr>
<td>(\rho)</td>
<td>water density, kg/m³</td>
</tr>
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</table>

Subscripts and superscripts

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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</thead>
<tbody>
<tr>
<td>g</td>
<td>Ground</td>
</tr>
<tr>
<td>in</td>
<td>Inlet</td>
</tr>
<tr>
<td>j</td>
<td>Index for pipe</td>
</tr>
<tr>
<td>k</td>
<td>Index for node</td>
</tr>
<tr>
<td>max</td>
<td>Maximum</td>
</tr>
<tr>
<td>min</td>
<td>Minimum</td>
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<td>p</td>
<td>Pump</td>
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<td>Return</td>
</tr>
<tr>
<td>s</td>
<td>Supply</td>
</tr>
<tr>
<td>t</td>
<td>Index for time step</td>
</tr>
</tbody>
</table>
References


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Optimization of Energy Supply Systems in Consideration of Hierarchical Relationship Between Design and Operation

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Abstract:
To attain the highest performance of energy supply systems, it is necessary to rationally determine types, capacities, and numbers of equipment in consideration of their operational strategies corresponding to seasonal and hourly variations in energy demands. Combinatorial optimization methods based on the mixed-integer linear programming (MILP) have been proposed to solve this design problem, and integer variables are used to express the selection, number, and on/off status of operation of equipment. However, the number of these variables increases with the numbers of equipment and periods for variations in energy demands, and affects the computation efficiency significantly. In this paper, a MILP method utilizing the hierarchical relationship between design and operation variables is proposed to solve the optimal design problem of energy supply systems efficiently: At the upper level, the values of design variables are optimized by the conventional branch and bound method; At the lower level, the values of operation variables are optimized independently at each period by the conventional branch and bound method; Both the levels are connected with each other to exchange data for optimization calculations. The proposed method is applied to a numerical study on the optimal design of a gas engine cogeneration system for electric power and hot water supply, and its validity and effectiveness are clarified.

Keywords:
Energy supply systems, Design, Operation, Optimization, Mixed-integer linear programming, Branch and bound method, Hierarchical approach.

1. Introduction

In designing energy supply systems, it is important to rationally determine their structures by selecting energy producing and conversion equipment from many alternatives so that they match energy demand requirements. It is also important to rationally determine capacities and numbers of selected equipment in consideration of their operational strategies such as on/off status of operation and load allocation corresponding to seasonal and hourly variations in energy demands.

Recently, distributed energy supply systems have been widespread and diversified, and many types of equipment have been installed into them, which means that many alternatives for system design and operation have arisen. Thus, it has become more and more difficult for designers to design the systems properly in consideration of their operational strategies only with their experiences. In addition, not only reliability in energy supply but also economics, energy saving, and environmental impact have become more and more important criteria for system design and operation, with which designers have been burdened more heavily. For the purpose of assisting designers in system design and operation, therefore, it is necessary to develop a tool for providing rational design and operation solutions flexibly and automatically.

One of the ways to rationally determine the aforementioned design and operation items of energy supply systems is to use combinatorial optimization methods, which are based on the mathematical programming such as the mixed-integer linear programming (MILP) \cite{1-6} and the mixed-integer nonlinear programming \cite{7}, as well as the meta heuristics such as the simulated annealing \cite{8} and...
the genetic algorithm [9]. For example, in the method based on the MILP, the selection, numbers, and on/off status of operation of equipment are expressed by integer variables, and the capacities and load allocation of equipment by continuous ones. However, the optimal design problem has often been treated in consideration of single-period operation [1], or multi-period one for a small number of periods [2], to avoid excessive difficulty of the problem. This is because the number of integer variables increases with those of equipment and periods, and it becomes difficult to obtain the optimal solution in a practical computation time using the conventional solution algorithm for the MILP which combines the branch and bound method with the simplex one.

Some efforts have been made to treat the optimal design problem in consideration of multi-period operation for a larger number of periods [3–5]. Nevertheless, equipment capacities have still been treated as continuous variables, and correspondingly performance characteristics and capital costs of equipment have been assumed to be continuous functions with respect to their capacities. This is because if equipment capacities are treated discretely, the number of integer variables increases drastically, and the problem becomes too difficult to solve. As a result, the treatment of equipment capacities as continuous variables causes discrepancies between existing and optimized values of capacities, and expresses the dependence of performance characteristics and capital costs on capacities with worse approximations. On the other hand, an optimal design method has been proposed in consideration of discreteness of equipment capacities [6]. In this method, a formulation for keeping the number of integer variables as small as possible has been presented to solve the optimal design problem easily. However, the aforementioned difficulty in the MILP method still exists essentially. Even commercial MILP solvers which are recently available may not derive the optimal solutions in practical computation times.

In this paper, a MILP method utilizing the hierarchical relationship between design and operation variables is proposed to efficiently solve the optimal design problem of energy supply systems in consideration of discreteness of equipment capacities: At the upper level, the values of the design variables are optimized by the conventional branch and bound method; At the lower level, the values of the operation variables are optimized independently at each period by the conventional branch and bound method; Both the levels are connected with each other to exchange data for optimization calculations. Since this MILP method utilizing the hierarchical relationship cannot be incorporated into commercial MILP solvers currently, it is incorporated into an open solver at the initial stage. Finally, a numerical study on the optimal design of a gas engine cogeneration system for electric power and hot water supply is conducted using the proposed method, and its validity and effectiveness are investigated.

2. Formulation of optimal design problem

2.1. Summary of optimal design problem

To consider seasonal and hourly variations in energy demands, a typical year is divided into multiple periods, and energy demands are estimated for each period. As shown in Fig. 1, a super structure for an energy supply system is created to match energy demand requirements. The super structure is composed of all the units of equipment considered as candidates for selection, and a real structure is created by selecting some units of equipment from the candidates. Furthermore, some units of equipment are operated to satisfy energy demands for each period. The selection, capacities, and numbers of equipment are considered as design variables, and the on/off status of operation and load allocation of equipment as operation ones. The hierarchical relationship between the design and operation variables is shown in Fig. 2. The selection and capacities are expressed by binary variables, the numbers and on/off status of operation by integer ones, and the load allocation by continuous ones.

As fundamental constraints, performance characteristics of equipment and energy balance relationships are considered. If necessary, other constraints such as relationships between maximum
contract demands and consumptions of purchased energy, and operational restrictions are considered.

Fig. 1. Concept of super structure.

Fig. 2. Hierarchical relationship between design and operation variables

As the objective function to be minimized, the annual total cost is adopted typically, and is evaluated as the sum of annual capital cost of equipment and annual operational cost of purchased energy.

These constraints and objective function are expressed as functions with respect to the design and operation variables.

In the following, an optimal design problem is formulated for the energy supply system with a simple super structure shown in Fig. 3. The formulation can easily be extended to energy supply systems with complex super structures.

2.2. Selection, capacities, and numbers of equipment

The energy supply system is composed of $I$ blocks, each of which corresponds to a type of equipment. The capacity of the $i$th type of equipment is selected from its $J_i$ candidates. In addition, the number of the $i$th type and the $j$th capacity of equipment is determined within its maximum $N_{ij}$. 
The selection and number of the $i$th type and the $j$th capacity of equipment are designated by the binary variable $\gamma_{ij}$ and the integer variable $\eta_{ij}$, respectively. By these definitions, the following equations are obtained:

\[
\begin{align*}
\eta_{ij}/N_{ij} & \leq \gamma_{ij} \leq \eta_{ij} \quad (j = 1, 2, \ldots, J_i) \\
\sum_{j=1}^{J_i} \gamma_{ij} & \leq 1 \\
\gamma_{ij} & \in \{0, 1\} \quad (j = 1, 2, \ldots, J_i) \\
\eta_{ij} & \in \{0, 1, \ldots, N_{ij}\} \quad (j = 1, 2, \ldots, J_i)
\end{align*}
\]

Here, it is assumed that multiple units with the same capacity can be selected for a type of equipment. To select multiple units with different capacities for a type of equipment, multiple blocks for the type of equipment should be included in the system.

### 2.3. Performance characteristics of equipment

A relationship between the flow rates of input and output energy is shown in Fig. 4 as performance characteristics of a piece of the $i$th type of equipment. Here, the discontinuity of the relationship due
to the number of equipment at the on status of operation is expressed by an integer variable, and the relationship for all of equipment is approximated by a linear equation as follows:

\[
\begin{align*}
\delta_i(k) &= P_i x_i(k) + Q_i \delta_i(k) \\
X_i \delta_i(k) &\leq x_i(k) \leq X_i \delta_i(k) \\
\delta_i(k) &\in \{0, 1, \ldots, \max_{1 \leq j \leq J_i} N_{ij}\} \\
\end{align*}
\]

(2)

where \(\delta_i\) is the integer variable for the number of equipment at the on status of operation. Here, it is assumed that \(\delta_i\) units of equipment are operated at the same load level, and the sums of the flow rates of input and output energy are expressed by the continuous variables \(x_i\) and \(y_i\), respectively.

This assumption is validated if the simple performance characteristics expressed by Eq. (2) are used. \(P_i, Q_i, X_i, X_i\) and \(\bar{X}_i\) are the performance characteristic values, i.e., \(P_i\) and \(Q_i\) are the slope and intercept, respectively, of the linear relationship between the flow rates of input and output energy for a piece of equipment at the on status of operation, and \(X_i\) and \(\bar{X}_i\) are the lower and upper limits, respectively, for the flow rate of input energy for a piece of equipment at the on status of operation. The argument \(k\) is the index for periods, and \(K\) is the number of periods. The first equation in Eq. (2) expresses the flow rate of output energy as a function with respect to that of input energy when a part of equipment are at the on status of operation, and makes the flow rate of output energy zero when all of equipment are at the off status of operation. The second equation in Eq. (2) makes the flow rate of input energy within its lower and upper limits when a part of equipment are at the on status of operation, and zero when all of equipment are at the off status of operation. The third equation in Eq. (2) means that the number of equipment at the on status of operation may not be larger than that selected. Since \(\delta_i\) is common to all the capacities, \(\max_{1 \leq j \leq J_i} N_{ij}\) is used as its maximum in this equation.

The values of \(P_i, Q_i, X_i,\) and \(\bar{X}_i\) depend on the selected capacity, and are expressed as follows:

\[
\begin{align*}
P_i &= \sum_{j=1}^{J_i} p_{ij} \gamma_{ij} \\
Q_i &= \sum_{j=1}^{J_i} q_{ij} \gamma_{ij} \\
X_i &= \sum_{j=1}^{J_i} x_{ij} \gamma_{ij} \\
\bar{X}_i &= \sum_{j=1}^{J_i} \bar{x}_{ij} \gamma_{ij} \\
\end{align*}
\]

(3)

where \(p_{ij}, q_{ij}, x_{ij},\) and \(\bar{x}_{ij}\) are the performance characteristic values of the \(i\)th type and the \(j\)th capacity of equipment, i.e., \(p_{ij}\) and \(q_{ij}\) are the slope and intercept, respectively, of the linear relationship between the flow rates of input and output energy, and \(x_{ij}\) and \(\bar{x}_{ij}\) are the lower and upper limits, respectively, for the flow rate of input energy. In addition, the number of equipment at the on status of operation is smaller than or equal to that selected, and the following equation is obtained:

\[
\delta_i(k) \leq \sum_{j=1}^{J_i} \eta_{ij} \quad (i = 1, 2, \ldots, I; k = 1, 2, \ldots, K)
\]

(4)
Therefore, Eqs. (2) to (4) result in

\[
\begin{align*}
  y_i(k) &= \sum_{j=1}^{J_i} p_{ij}\gamma_i x_i(k) + \sum_{j=1}^{J_i} q_{ij}\gamma_i \delta_i(k) \\
  \sum_{j=1}^{J_i} x_{ij}\gamma_i \delta_i(k) &\leq x_i(k) \leq \sum_{j=1}^{J_i} x_{ij}\gamma_i \delta_i(k) \\
  \delta_i(k) &\leq \sum_{j=1}^{J_i} \eta_{ij} \\
  \delta_i(k) &\in \{0, 1, \ldots, \max_{1 \leq j \leq J_i} N_{ij}\}
\end{align*}
\]

Since \( \delta_i \) is common to all the capacities, this formulation keeps the number of integer variables as small as possible, which makes the computation time as short as possible.

### 2.4. Capital costs of equipment

The capital cost of each unit of equipment depends on its capacity and performance characteristics. The capital cost of the \( i \)-th type of equipment \( C_i \) is calculated as follows:

\[
C_i = \sum_{j} c_{ij} \eta_{ij} \quad (i = 1, 2, \ldots, I)
\]

where \( c_{ij} \) is the capital cost of the \( i \)-th type and the \( j \)-th capacity of equipment.

### 2.5. Objective function and energy balance relationship

As mentioned previously, the annual total cost is adopted as the objective function \( z \) to be minimized, and is expressed by

\[
z = \sum_{i=1}^{I} \left( R C_i + \varphi_i \sum_{k=1}^{K} T(k) x_i(k) \right)
\]

\[
= \sum_{i=1}^{I} \left( R \sum_{j} c_{ij} \eta_{ij} + \varphi_i \sum_{k=1}^{K} T(k) x_i(k) \right)
\]

where \( R \) is the capital recovery factor, \( \varphi_i \) is the unit cost for energy charge of the input energy consumed by the \( i \)-th type of equipment, and \( T \) is the duration per year of each period.

As the energy balance relationship, the following equation is considered:

\[
\sum_{i=1}^{I} y_i(k) = Y(k) \quad (k = 1, 2, \ldots, K)
\]

where \( Y \) is the energy demand for each period.

### 3. Solution of optimal design problem

#### 3.1. Linearization of nonlinear terms

The aforementioned formulation leads to the following optimal design problem:
Find \[
\gamma_{ij}, \eta_{ij}, \delta_i(k), \pi_i(k), \text{ and } y_i(k)
\]
which minimize \[z\] of Eq. (7)
subject to Eqs. (1), (5) and (8).

To reformulate this optimal design problem as a MILP one, the nonlinear terms due to the products of \(\gamma_{ij}\) and \(x_{ij}\), \(\eta_{ij}\) and and in Eq. (5) are replaced by the continuous variables \(\xi_{ij}\) and \(\zeta_{ij}\), respectively, as follows:

\[
\begin{align*}
\xi_{ij}(k) &= \gamma_{ij}x_{ij}(k) \quad (i = 1, 2, \ldots, I; j = 1, 2, \ldots, J; k = 1, 2, \ldots, K) \\
\zeta_{ij}(k) &= \eta_{ij}\delta_i(k) \quad (i = 1, 2, \ldots, I; j = 1, 2, \ldots, J; k = 1, 2, \ldots, K)
\end{align*}
\]

As a result, Eq. (5) is reduced to

\[
\begin{align*}
y_i(k) &= \sum_{j=1}^{J_i} p_{ij} \xi_{ij}(k) + \sum_{j=1}^{J_i} q_{ij} \zeta_{ij}(k) \\
\sum_{j=1}^{J_i} x_{ij}\xi_{ij}(k) &\leq x_i(k) \leq \sum_{j=1}^{J_i} x_{ij}\zeta_{ij}(k) \\
\delta_i(k) &\leq \sum_{j=1}^{J_i} \eta_{ij} \\
\delta_i(k) &\in \{0, 1, \ldots, \max_{1 \leq j \leq J_i} N_{ij}\}
\end{align*}
\]

In addition, the following constraints are employed:

\[
\begin{align*}
x_i(k)\gamma_{ij} &\leq \xi_{ij}(k) \leq \bar{x}_i(k)\gamma_{ij} \\
x_i(k) - \bar{x}_i(k)(1 - \gamma_{ij}) &\leq \xi_{ij}(k) \leq x_i(k) \\
\delta_i(k)\gamma_{ij} &\leq \zeta_{ij}(k) \leq \bar{\delta}_i(k)\gamma_{ij} \\
\delta_i(k) - \bar{\delta}_i(k)(1 - \gamma_{ij}) &\leq \zeta_{ij}(k) \leq \delta_i(k)
\end{align*}
\]

where \(\gamma_{ij}\) and \(\gamma_{ij}\) are lower and upper bounds, respectively, and these values can be set as follows:

\[
\begin{align*}
x_i(k) &= 0 \\
x_i(k) &= \max_{1 \leq j \leq J_i} N_{ij}x_{ij} \\
\delta_i(k) &= 0 \\
\delta_i(k) &= \max_{1 \leq j \leq J_i} N_{ij}
\end{align*}
\]
The validity of the constraints of Eqs. (12) and (13) are shown as follows: Eq. (12) means that if \( \gamma_{ij} = 0 \), then \( \xi_{ij}(k) = 0 \), and that else if \( \gamma_{ij} = 1 \), then \( \xi_{ij}(k) = x_i(k) \), which makes Eq. (9) valid indirectly; Similarly, Eq. (13) means that if \( \gamma_{ij} = 0 \), then \( \xi_{ij} = 0 \), and that else if \( \gamma_{ij} = 1 \), then \( \xi_{ij} = \delta_{ij}(k) \), which makes Eq. (10) valid indirectly.

This procedure can linearize the nonlinear terms without any approximations and transform the optimal design problem into a MILP one as follows:

Find: 
\[
\gamma_{ij}, \eta_{ij}, \delta_{ij}(k), x_{ij}(k), y_{ij}(k), \xi_{ij}(k), \text{and} \xi_{ij}(k)
\]
subject to: 
Equations (1) to (13).

### 3.2. Solution in consideration of hierarchical relationship

Some commercial MILP solvers which are recently available can solve large scale problems in practical computation times [10]. The reformulated MILP problem may be solved by such solvers. However, the MILP problem under consideration has the feature that it becomes extremely large scale with increases in the numbers of equipment and periods, \( I, J \), and \( K \). In such cases, even commercial MILP solvers may not derive the optimal solutions in practical computation times. In this paper, a special solution method is developed in consideration of the hierarchical relationship between the design and operation variables. However, this special solution method cannot be incorporated into commercial MILP solvers currently, because they are not open for revision. Therefore, the special solution method is incorporated into a MILP solver published in reference [11], because it is open for revision.

The original optimal design problem has the hierarchical relationship between the design and operation variables as shown in Fig. 2. The reformulated MILP problem also has a similar relationship.

Namely, the design variables at the upper level are the binary and integer variables, \( \gamma_{ij} \) and \( \eta_{ij} \), respectively, while the operation variables at the lower level are the integer variable \( \delta_{ij}(k) \) as well as the continuous variables \( x_{ij}(k), y_{ij}(k), \xi_{ij}(k), \) and \( \xi_{ij}(k) \) at each period \( k \). The values of these design and operation variables at all the periods should be optimized simultaneously. However, if the values of the design variables are assumed tentatively at the upper level, the values of the operation variables can be optimized independently at each period at the lower level. This feature leads to the following hierarchical solution process as shown in Fig. 5.

In place of the reformulated MILP problem, the design and operation problems at the upper and lower levels, respectively, are defined as follows:

![Fig. 5. Solution process in consideration of hierarchical relationship between design and operation variables.](image-url)
The design problem at the upper level is defined by relaxing $\delta_i(k)$ as a continuous variable in the reformulated MILP problem, while the operation problem at the lower level is defined at each period by adopting $\sum_{i=1}^{I} \varphi_i T(k) x_i(k)$ as the objective function and giving the values of $\gamma_{ij}$ and $\eta_{ij}$.

The optimal values of the design variables, $\gamma_{ij}$ and $\eta_{ij}$, are searched by the branching and bounding operations used in the conventional branch and bound method. In the proposed method, however, when the branching operation is conducted for all the design variables and their values are assumed tentatively, the operation problem at the lower level is solved independently at each period by the conventional branch and bound method, and its result is returned to the design problem at the upper level as follows: If an operation problem at a period is infeasible, or the deficit in energy supply arises, the tentative values of the design variables cannot become the optimal solution, and therefore the bounding operation is conducted; If the operation problems at all the periods are feasible, the values of the operation variables, $\delta_i(k), x_i(k), y_i(k), \xi_{ij}(k),$ and $\zeta_{ij}(k)$, and are determined, and a part of the objective function $\sum_{i=1}^{I} \sum_{k=1}^{K} \varphi_i T(k) x_i(k)$ is evaluated correspondingly; Then, the value of the objective function $z$ is evaluated using the tentative value of $\eta_{ij}$; If $z$ is larger than that for the tentative optimal solution obtained previously, the bounding operation is conducted; Else if $z$ is smaller than that for the tentative optimal solution, this solution is a new candidate for the optimal solution, and the tentative optimal solution is replaced with it.

The number of all the variables in the design problem is the same as that in the reformulated MILP problem. However, the number of the binary and integer variables in the design problem is much smaller than that in the reformulated MILP problem. Therefore, the design problem needs a smaller memory size as well as a shorter computation time for conducting the branching and bounding operations. In addition, the number of the variables of the operation problem at each period is quite small, and the operation problem can be solved easily. As a result, the proposed method has better features in memory size and computation time as compared with the direct solution of the reformulated MILP problem.

![Super structure for gas engine cogeneration system](image)
The aforementioned proposed method is incorporated into the open solver in the following way: Two sets of general programs by the conventional branch and bound method are prepared for both the design and operation problems; Specific programs for creating matrices for simplex tableaus are also prepared for both the design and operation problems. These specific programs are generated automatically from the equations for the objective function and constraints in consideration of the structure of an energy supply system and the number of periods. All the programs are connected with one another to exchange data for conducting the branching and bounding operations in the design and operation problems.

Table 1. Capacities of equipment for selection

<table>
<thead>
<tr>
<th>Equipment</th>
<th>Capacity</th>
<th>Item</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gas engine generator</td>
<td>1</td>
<td>Maximum power output</td>
<td>25.0 kW</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Maximum hot water output</td>
<td>138.2 MJ/h</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>Maximum power output</td>
<td>35.0 kW</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Maximum hot water output</td>
<td>189.6 MJ/h</td>
</tr>
<tr>
<td>Gas-fired boiler</td>
<td>1</td>
<td>Maximum hot water output</td>
<td>356.4 MJ/h</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>Maximum hot water output</td>
<td>712.9 MJ/h</td>
</tr>
</tbody>
</table>

4. Numerical study on optimal design of cogeneration system

4.1. Conditions

The proposed method is applied to a numerical study on the optimal design of a gas engine cogeneration system for electric power and hot water supply. Figure 6 shows the super structure for the system, which has two gas engine generators with a same capacity and two gas-fired boilers with a same capacity. Table 1 shows the capacities of the gas engine generators and gas-fired boilers to be selected. In addition to the equipment, the maximum contract demands of electric power and city gas purchased from outside utility companies are also determined. However, the proposed method can treat only binary and integer variables in the design problem at the upper level. Thus, the maximum contract demands of electric power and city gas are treated using integer variables, and are selected among discrete values by 10.0 kW and 1.0 Nm³/h, respectively. Table 2 shows the capital costs of equipment as well as the unit costs for demand and energy charges of utilities. The capital recovery factor is set at 0.7782 by assuming the interest rate and life of equipment as 0.02 and 15 y respectively.

Table 2. Capital costs of equipment and unit costs for demand and energy charges of utilities

<table>
<thead>
<tr>
<th>Equipment/Utility</th>
<th>Item</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gas engine generator</td>
<td>Capital cost</td>
<td>225.0 x 10³ yen/kW</td>
</tr>
<tr>
<td>Gas-fired boiler</td>
<td>Capital cost</td>
<td>2.5 x 10³ yen/kW</td>
</tr>
<tr>
<td>Electric power</td>
<td>Unit cost for demand charge</td>
<td>1685 yen/(kW·month)</td>
</tr>
<tr>
<td></td>
<td>Unit cost for energy charge</td>
<td>12.08 yen/kWh</td>
</tr>
<tr>
<td>City gas</td>
<td>Unit cost for demand charge</td>
<td>630 yen/(Nm³/h·month)</td>
</tr>
<tr>
<td></td>
<td>Unit cost for energy charge</td>
<td>60 yen/Nm³</td>
</tr>
</tbody>
</table>
A hotel with the total floor area of 3000 $\text{m}^2$ is selected as the building which is supplied with electric power and hot water by the cogeneration system. To take account of seasonal and hourly variations in energy demands, a typical year is divided into three representative days in summer, midseason, and winter whose numbers of days per year are set as 122, 122, and 121 d/y, respectively, and each day is further divided into 3, 6, and 12 sampling time intervals with 8, 4, and 2 h, respectively. Thus, the year is divided into 9, 18, and 36 periods correspondingly. Figure 7 shows the hourly changes in energy demands in each season for the case with 12 sampling time intervals per day.

![Fig. 7. Energy demands in three seasons at hotel: a) electric power, b) hot water.](image)

Table 3. Results by optimization calculations

<table>
<thead>
<tr>
<th>Number of periods</th>
<th>Solution and Objective $\times 10^6$ yen/y</th>
<th>Computation time</th>
<th>Solution and Objective $\times 10^6$ yen/y</th>
<th>Computation time</th>
<th>Capacity and number of gas engine generator kW</th>
<th>Capacity and number of gas-fired boiler MJ/h</th>
<th>Electric Power contract demand kW</th>
<th>City gas contract demand Nm$^3$/h</th>
</tr>
</thead>
<tbody>
<tr>
<td>9 (3 x 3)</td>
<td>Optimal 10.45</td>
<td>3072 s</td>
<td>Optimal 10.45</td>
<td>2 s</td>
<td>25.0 x 2</td>
<td>—</td>
<td>50</td>
<td>13</td>
</tr>
<tr>
<td>18 (6 x 3)</td>
<td>Feasible 10.54</td>
<td>3600 s*</td>
<td>Optimal 10.44</td>
<td>6 s</td>
<td>25.0 x 1</td>
<td>356.4 x 1</td>
<td>80</td>
<td>10</td>
</tr>
<tr>
<td>36 (12 x 3)</td>
<td>Feasible 10.90</td>
<td>3600 s*</td>
<td>Optimal 10.49</td>
<td>192 s</td>
<td>25.0 x 1</td>
<td>356.4 x 1</td>
<td>80</td>
<td>13</td>
</tr>
</tbody>
</table>

*Attains limit for computation time.

4.2. Results and discussion

Table 3 shows the results obtained by the optimization calculations on a MacBook Air with Mac OS X 10.6.7. The optimal design problem is solved by both the conventional and proposed methods. Both the design and operation variables are simultaneously optimized by the conventional method. This calculation is conducted using the same program as that used for the design problem by the proposed method. The results obtained by both the methods are compared with each other in terms of solutions and computation times. In addition, the values of the design variables obtained by the proposed method are also shown.

In the case with the number of periods of 9, both the methods derive the optimal solution. However, the conventional method needs a much longer computation time. In the cases with the number of periods of 18 and 36, the proposed method derives the optimal solutions in practical computation times. However, the conventional method does not derive the optimal solutions but only feasible ones within the limit for the computation time. In the case with the number of periods of 9, the
optimal solution adopts two smaller gas engine generators. In the cases with the number of periods of 18 and 36, the optimal solutions adopt one smaller gas engine generator and one smaller gas-fired boiler. These are because in the former case the electric power and hot water demands are averaged and resultantly balanced, which is advantageous to cogeneration, and in the latter cases the electric power and hot water demands are not balanced in some periods.

As an example, Fig. 8 shows the allocation in electric power and hot water supply in summer in the case with the number of periods of 36. The gas engine generator is stopped, is operated at a part load status, or is operated at the rated load status, depending on the electric power and hot water demands. The heat flow rate of hot water generated by the gas engine generator is larger than the hot water demand during 10:00 to 18:00, and its excess is disposed of. This is because the purchased electric power attains its contract demand. In addition, the heat flow rate of hot water generated by the gas-fired boiler is larger than the hot water demand during 2:00 to 6:00, and its excess is disposed of. This is because the lower limit for the hot water output of the gas-fired boiler is larger than the hot water demand.

![Fig. 8. Allocation in energy supply in summer: a) electric power, b) hot water](image)

5. Conclusions

A MILP method utilizing the hierarchical relationship between design and operation variables has been proposed to efficiently solve the optimal design problem of energy supply systems in consideration of discreteness of equipment capacities: At the upper level, the values of the design variables are optimized by the conventional branch and bound method; At the lower level, the values of the operation variables are optimized independently at each period by the conventional branch and bound method; Both the levels are connected with each other to exchange data for optimization calculations. This MILP method utilizing the hierarchical relationship has been incorporated into an open solver at the initial stage. Finally, the proposed method has been applied to a numerical study on the optimal design of a gas engine cogeneration system for electric power and hot water supply. The optimal design problem has been solved with a change in the number of periods by both the conventional and proposed methods. The results obtained by both the methods have been compared with each other in terms of solutions and computation times. Through the study, it has turned out that the proposed method is much superior to the conventional one in terms of computation time. In addition, some features concerning the optimal design and operation of the cogeneration system have been clarified. As a subsequent subject, the proposed method should be incorporated into commercial solvers so that much larger scale optimal design problems can be solved in practical computation times.
Nomenclature

C, c : capital cost of equipment
I : number of types of equipment
J : number of capacities of equipment
K : number of periods
k : index for periods
N : maximum number of selected equipment
P, p : slope of linear relationship between flow rates of input and output energy of equipment
Q, q : intercept of linear relationship between flow rates of input and output energy of equipment
R : capital recovery factor
T : duration per year of period
X, x : lower limit for flow rate of input energy of equipment
X, x : upper limit for flow rate of input energy of equipment
x : flow rate of input energy
Y : energy demand
y : flow rate of output energy
z : annual total cost
γ : selection of equipment
δ : number of equipment at on status of operation
ζ : product of γ and δ
η : number of selected equipment
ξ : product of γ and x
φ : unit cost for energy charge of input energy
( ) : lower bound
( ^ ) : upper bound

Subscripts
i : index for types of equipment
j : index for capacities of equipment

References


The Fuel Impact Formula Revisited

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Abstract:

Energy systems comprise complex networks structures, where the irreversibility of a process depends on those of other processes. Thermoeconomics provides tools to identify and to quantify the additional consumption of resources caused by an irreversibility increase of some component affecting the other components. The fuel impact formula is the mathematical expression of the non-equivalence of energy losses in energy systems, i.e. the same irreversibility change on several system components has a different effect on the corresponding change on the total irreversibility of the system. This formula is used for diagnosis of malfunctions in energy plants and also for evaluating the potential energy saving that could be obtained by the efficiency improving in one or more processes. For this last purpose simulations or approximation formulae are used, but error estimation has not yet been studied.

The aim of this paper is to review the fundamentals of the principle of non-equivalence of the exergy losses and its connection with the thermoeconomic isolation principle, which allows to analyse the potential savings of each individual process instead on the overall system. A revised exact expression of the global exergy savings due to the improvement of the efficiency of a component and a new indicator of the thermoeconomic isolation of process within the global system are obtained. These results provide a better understanding of the effects of local savings and the productive structure in the total energy saving of any energy system.

Keywords:

Irreversibility, Fuel Impact Formula, Thermoeconomic isolation, Energy Savings

1. Introduction

According to the exergy cost theory \cite{1}, the fuel impact is the effect on total resources consumption, which can be achieved if the efficiency of a process, or equivalently its unit consumption, is improved. It will be assessed by multiplying its local exergy saving $\Delta I_i = P_i \Delta k_i$, by the unit exergy cost of the fuel consumed by this process, and defined as:

$$A_{T,i} = k^*_{F,i} P_i \Delta k_i$$  \hspace{1cm} (1)

The fuel impact formula (1) states that the location of an irreversibility do not coincide with the location of the causes that provoke these losses. When a system’s process degrades, we will need more local resources to obtain the same required production. These resources are product of some upstream process that will readapt its production, and subsequently increasing its irreversibility. Moreover, the increase of the unit exergy consumption of a process causes the increase of its unit production cost and consequently the increase of the unit production costs of all process downstream. The unit exergy cost of fuel, say $k^*_{F,i}$, is the measure of this propagation. The fuel impact formula expresses the non-equivalence of irreversibilities in different components of a system.

Consider the case of a sequential system, like the one shown in the figure 1. In this example, which corresponds to a simplified steam cycle, the variation of irreversibility of 1 MW on each process
has different fuel impacts, 2.96 MW in case the generator, 2.70 MW in case of the turbine and only 1MW in case of de boiler. However, a relative variation of its unit exergy consumption, for example $\% \Delta k = 1\%$, has the same fuel impact 0.92 MW for all components.

It is because the local irreversibility change on a process does not have the same equivalence in relative efficiency change, 1 MW in the turbine is a relative change of 3%, meanwhile in the boiler is only 1%.

![Exergy cost diagram of a simplified steam cycle](image)

**Figure 1. Exergy cost diagram of a simplified steam cycle**

The fuel impact of an irreversibility variation does not depend only for the unit exergy cost of fuel, but also the amount of resources consumed by the process. In fact equation (1) could be rearranged as:

$$A_{T,i} = F_i^* % \Delta k_i$$  \hspace{1cm} \text{(2)}

in which $F_i^*$ is the cost of the resources consumed, and $% \Delta k_i = \Delta k_i / k_i$.

We could see, in case of a sequential system, eq. (2) measures the real fuel impact $\Delta F_{T,i}$, due to a variation in the $i$-th process efficiency, if the efficiency of the remaining components doesn’t change.

For more complex systems, this expression has been proved [2] that it is a reasonable approximation for small variations. But, several theoretical questions arise here: could we find an estimation of the relative error between $A_{T,i}$ and $\Delta F_{T,i}$, which explain the discrepancies? Is there an exact expression for fuel impact formula?

The fuel impact formula is the basis of Thermoeconomic Diagnosis. It was proposed by Valero et al. [3], and developed by several authors [4-8]. The formula permits to compute the fuel impact generated by each process, as a function of its unit exergy consumption changes.

### 2. The fuel impact formula revisited

In this section we will obtain an exact expression for external resources variation due to the efficiency variation of an individual process, with the condition that the unit exergy consumption ratios of the rest of processes do not change.

Using Thermoeconomic Input-Output (see Appendix A), Torres et al. [9] introduce an exact formula, for the fuel impact generated by process, at constant production, when the unit exergy consumption of the processes $\Delta k_{ij}$ change:

$$\Delta F_T = \sum_{i=1}^{n} \left( \Delta k_{0i} + \sum_{j=1}^{n} k_{ij}^* (\bar{x}) \Delta k_j \right) P_i(x_0)$$ \hspace{1cm} \text{(3)}

To compute the fuel impact it is required to know two states of the system, a reference state, say $x_0$, and a current state $\bar{x}$, and compute the unit production cost at the current state. Equation (3) will be our starting point to answer the question posed above.
Let’s consider now a relative change \( \% \Delta k_i \) on the unit consumption of the \( i \)-th process, under the condition the unit exergy consumption of the rest of components do not change, i.e. \( \Delta k_j = 0 \) for \( l \neq i \). Hence, only the \( i \)-th column, say \( k_i \), of the matrix \( \langle KP \rangle \) is change and \( \Delta k_i = k_i' - k_i \).

Under these conditions (3) is rewritten as:

\[
\Delta F_{T,i} = \left( k_{i0} + \sum_{j=1}^{n} k_{p,j} k_i' \right) P_i \% \Delta k_i
\]

(4)

and:

\[
\Delta F_{T,j} - A_{T,j} = \left( \sum_{j=1}^{n} \Delta k_{p,j} k_i' \right) P_i \% \Delta k_i
\]

(5)

Therefore, to solve the problem we should compute \( \Delta k_{p,j}' \), for a relative change \( \% \Delta k_i \).

According to equations of Thermoeconomic Input-Output (see appendix A), the unit production cost satisfy:

\[
\top^* k_p = \top^* k_p \left| \mathbf{P} \right|
\]

(6)

Let’s \( \mathbf{K}_p \) to be the unit production cost in the new state \( \mathbf{x} \), when the efficiency or technical coefficients of one or several processes change, therefore:

\[
\top^* \mathbf{K}_p = \left( \top^* k_p + \Delta \top^* k_p \right) \left| \mathbf{P} \right|
\]

(7)

in which, the production matrix \( \left| \mathbf{P} \right| \) in the new state is:

\[
\left| \mathbf{P} \right| = \left| \mathbf{P} \right| \left( \mathbf{U} - \Delta \langle \mathbf{KP} \rangle \left| \mathbf{P} \right| \right)^{-1}
\]

(8)

Under our problem conditions:

\[
\Delta \top^* k_p = \% \Delta k_i \mathbf{k}_0 \mathbf{p}_i
\]

\[
\Delta \langle \mathbf{KP} \rangle \left| \mathbf{P} \right| = \% \Delta k_i \mathbf{k}_i \otimes \mathbf{p}_i
\]

where \( \mathbf{p}_i \) denote the \( i \)-th row of matrix \( \left| \mathbf{P} \right| \). Therefore, applying the Sherman-Morrison formula (see Appendix C), we get:

\[
\left( \mathbf{U} - \Delta \langle \mathbf{KP} \rangle \left| \mathbf{P} \right| \right)^{-1} = \mathbf{U} + \frac{\% \Delta k_i}{1 - \tau_i \% \Delta k_i} \mathbf{k}_i \otimes \mathbf{p}_i
\]

(9)

Note that the quotient of the right side equation is a scalar, and \( \tau_i \) is also a scalar defined as:

\[
\tau_i = \mathbf{p}_i \cdot \mathbf{k}_i = \sum_{j} p_{ij} k_{ij} = p_u - 1
\]

(10)

Substitution of (9) into (7) yields:

\[
\top^* \mathbf{K}_p - \top^* k_p' = \Delta \top^* k_p' = \frac{\% \Delta k_i}{1 - \tau_i \% \Delta k_i} k_{p,j} k_j \mathbf{p}_i
\]

(11)

and, substitution of (11) into (5), follows:

\[
\Delta F_{T,j} - A_{T,j} = F_j \frac{\tau_i \% \Delta k_i}{1 - \tau_i \% \Delta k_i} \% \Delta k_i
\]

(12)

Therefore, we get an exact expression of the fuel impact formula, as a function of the relative variation of the efficiency:
\[ \Delta F_{r,i} = F_i \left( \frac{\%\Delta k_i}{1 - \tau_i \%\Delta k_i} \right) \]  

where the relative error is given by:

\[ \delta = \frac{\Delta F_{r,i} - A_{r,i}}{A_{r,i}} = \frac{\tau_i \%\Delta k_i}{1 - \tau_i \%\Delta k_i} \approx \tau_i \%\Delta k_i \]  

(14)

It means that the relative error between the original fuel impact formula and the new expression is linear, and it depends on the value of \( \tau_i \), and as consequence also depends on \( p_{ni} \).

If the graph which represents the productive structure of the system is acyclic (see Appendix B), it means that without recirculation, the diagonal elements of the Leontief inverse matrix are \( p_{ni} = 1 \), and consequently \( \tau_i = 0 \). In these conditions, as we stated in the introduction section, the original fuel impact formula gives the correct value. Hence, the parameter \( \tau_i \) measures the recirculation degree of a process and we will call it, recirculation factor. If a process has no feedback flows then its recirculation factor is zero. The elements of matrix \( |P| \) represents the total requirements of product of process \( i \) to obtain a unit of the product of the process \( j \). In particular, if \( p_{ni} > 1 \), it means that there is a feedback in process \( i \). The larger the feedback of a process, the larger the recirculation factor is.

To simplify previous equations, we will denote \( \%\Delta \vec{k} = \%\Delta k_i / (1 - \tau_i \%\Delta k_i) \), which measure the feedback effect on the relative change of the unit exergy consumption of a process. Observe that, if \( \tau_i = 0 \) then \( \%\Delta \vec{k} = \%\Delta k_i \).

Therefore the fuel impact formula (2) could be expressed as:

\[ \vec{A}_{r,i} = \Delta F_{r,i} = F_i \cdot \%\Delta \vec{k} \]  

(15)

As a conclusion, if a process has no feedback the fuel impact formula coincides with the real external resources change due to a relative change of the efficiency of such process. If the system has cycles the formula maintains the same structure and only is modified by a correction factor \( 1/(1 - \tau_i \%\Delta k_i) \), which depends on the recirculation factor and the amplitude of the change.

<table>
<thead>
<tr>
<th>Nr</th>
<th>Process</th>
<th>( A_r )</th>
<th>( \Delta F_r )</th>
<th>( \delta )</th>
<th>( \tau )</th>
<th>%( \Delta \vec{k} )</th>
<th>%( \Delta F_r )</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Combustor</td>
<td>2198.8</td>
<td>2198.8</td>
<td>0,000%</td>
<td>0,000</td>
<td>1,000%</td>
<td>1,000%</td>
</tr>
<tr>
<td>2</td>
<td>Compressor</td>
<td>1960.886</td>
<td>1978.531</td>
<td>0,892%</td>
<td>0,892</td>
<td>1,009%</td>
<td>0,900%</td>
</tr>
<tr>
<td>3</td>
<td>Gas Turbine</td>
<td>3219.867</td>
<td>3248.840</td>
<td>0,892%</td>
<td>0,892</td>
<td>1,009%</td>
<td>1,478%</td>
</tr>
<tr>
<td>4</td>
<td>HRSG</td>
<td>939,819</td>
<td>939,819</td>
<td>0,000%</td>
<td>0,000</td>
<td>1,000%</td>
<td>0,427%</td>
</tr>
<tr>
<td>5</td>
<td>Steam Turbine</td>
<td>943,692</td>
<td>943,709</td>
<td>0,002%</td>
<td>0,002</td>
<td>1,000%</td>
<td>0,429%</td>
</tr>
<tr>
<td>6</td>
<td>Pump</td>
<td>3,87306</td>
<td>3,87313</td>
<td>0,002%</td>
<td>0,002</td>
<td>1,000%</td>
<td>0,002%</td>
</tr>
<tr>
<td>7</td>
<td>Generator</td>
<td>2202,673</td>
<td>2202,712</td>
<td>0,002%</td>
<td>0,002</td>
<td>1,000%</td>
<td>1,002%</td>
</tr>
</tbody>
</table>

Table 1, shows the fuel impact and associated parameters for a deviation \( \%\Delta k_i = 1% \) in each process of the combined cycle power plant case study. Appendix D.

Note that, the relative error “\( \delta \)” is directly related with the recirculation factor \( \tau \). If we observe the corresponding fuel-product diagram of the plant, see figure 2, the combustor and the HRSG has no feedback, thus, their recirculation factors are zero. The steam cycle has a small recirculation for the feedwater pump, whose fuel is provided by the generator, and its recirculation factor is very small.
and then $A_{T5}$ provides a good value for the fuel impact of steam turbine. Compressor and gas turbine have a strong feedback loop, therefore its recirculation factor is high, and then the relative error is near to 1%. In general, for each process of this plant, as it has been put forward in the introduction, the fuel impact $A_T$ is a reasonable approximation to the real value. But now we know how good the approximation is and why. It depends of the productive structure of the system, and in particular of their feedbacks.

![Figure 3. Fuel-Product Diagram of the Combined Cycle Power Plant](image)

Obviously, we can use the revised fuel impact formula (13), we do not need additional calculation to compute the corrected value, the recirculation factor is obtained from the production matrix, required to calculate the unit production cost.

In last column it is shown the relative change of the total fuel due to a process efficiency change. This parameter, called total fuel elasticity factor, shows the importance of a process on the overall system, and the sensitivity of the system efficiency besides an efficiency change in a local process.

### 3. Production cost stability matrix

As application of the fuel impact formula, we developed in the previous section, we will determine the effect of process efficiency on the variation of the production cost, that could be expressed by means of its elasticity matrix of the type: $\% \Delta k^*_P = [\epsilon^*_A] \% \Delta k$.

From (11) the unit production cost variation of process $j$ is related with the efficiency variation of process $i$, as:

$$\Delta k^*_P, j = k^*_P, j \% \Delta \bar{k}_j$$

(16)

Therefore, if a process has no recycling its unit production cost variation is $\% \Delta k^*_P, i = \% \Delta k_i$, it means that a variation of 1% in its unit consumption represents 1% in the unit production cost variation. But if the process has feedbacks the effect of a positive variation (malfunction) on the unit production cost variation is bigger: $\% \Delta k^*_P, i = p_0 \% \Delta \bar{k}_i > \% \Delta k_i$, because $p_0 > 1$. 211
Let denote \( \mathbf{P}^* = \left[ p_{ij}^* \right] \) the matrix of cost normalized total requirement values defined as 
\[ p_{ij}^* = k_{p,j}^* p_0 / k_{p,j}^* \], therefore the elasticity matrix is 
\( \mathbf{\varepsilon}_A^* = \mathbf{P}^* \) and:
\[
\% \Delta k_{p,j}^* = p_{ij}^* \% \Delta \bar{k}_j
\] (17)

In systems without flow recirculation, the elasticity matrix is lower triangular, therefore an efficiency change only affects to downstream process, and every element is lower or equal to one.

In real plants, the measures used to calculate the efficiency of a process have a certain measurement error, and usually average values with error deviation are used. This elasticity matrix could be used to analyse the sensitivity of cost against efficiency measurement uncertainty.

In our example, the elasticity matrix of the combined cycle is:

\[
\begin{bmatrix}
1 & 1.909 & 1.909 & 0 & 0 & 0 & 0 \\
1 & 0.9 & 1.909 & 0 & 0 & 0 & 0 \\
1 & 0.9 & 0.9 & 1 & 0 & 0 & 0 \\
1 & 0.9 & 0.902 & 1 & 1.002 & 0.004 & 0.004 \\
1 & 0.9 & 1.478 & 0.427 & 0.429 & 1.002 & 1.002 \\
1 & 0.9 & 1.478 & 0.427 & 0.429 & 0.002 & 1.002 \\
\end{bmatrix}
\]

In bold characters, the values corresponding to processes with recirculating flows are shown. In the case of the gas turbine cycle (rows 2 and 3), the values are high and the process is very sensitive to changes on its efficiency, meanwhile in the Rankine cycle these values are not significant. For example an increase of 1% in the unit consumption of the gas turbine, increases 1.9% the unit cost of the compressor, meanwhile in the rest of processes an increase of 1% in the unit consumption increase the unit cost of the affected process less or equal than 1%, and recirculation in the Rankine cycle affects less than 0.005%. The last row, which corresponds to the electric generator, it shows the effect of the variation in the efficiency of each process to the production cost of the electricity, and indicates the weight of each process on the system. A malfunction in the gas turbine, column #3, has more effect that the rest of processes, meanwhile the feeder pump, column #6, has no effect.

4. Thermoeconomic Isolation

According to Evans and Frangopoulos [10, 11] a process of an energy system is thermoeconomically isolated (TI), from the rest of the system if the product of the process and the unit cost of the products used as fuel in the process are constant and have known quantities. As a consequence, if a process is thermoeconomically isolated from the rest of system, then may optimized “by itself”, i.e. without considering the variation made in the rest of the system processes, and the optimum solution thus obtained for the process coincides with the optimum solution for the system as a whole.

In this section we will show that the recycling factor could be used as an indicator of the thermoeconomic isolation degree of a process, and give some criteria to improve the local optimization methodology.

The first premise for thermoeconomic isolation is equivalent to say that the fuel unit cost of a process remains constant when its efficiency is modified.

The unit cost of fuel variation could be written as:
\[
\Delta k_{f,j}^* = \frac{1}{k_j} \sum_{j=1}^{n} \Delta k_{p,j}^* \kappa_j
\] (18)
and applying (16) we obtain:
\[ \Delta k^*_j = \tau_j k^*_{p,j} \% \Delta k_i \] (19)

Therefore, if the recirculation factor of a process is zero, its unit cost of fuel is constant and the first condition for TI is satisfied. If a process has no feedback, the production matrix \(|P|\) is upper triangular, therefore, from (16) only the processes downstream modify their unit production cost.

The second premise for thermoeconomic isolation says that if the efficiency of the process varies the production remains constant, or an efficiency change in a process only modified the production of the processes upstream it.

The production variation is \( \Delta P = \Delta |P| \omega_j \), and applying (8), we obtain:
\[ \Delta |P| = \frac{\% \Delta k_i}{1 - \tau_i \% \Delta k} |P| k_i \otimes p_i \] (20)

therefore the production change is given as:
\[ \Delta P = P_i \% \Delta k_i \tau_i \] (21)

where \( \tau_i \) is the i-th column of matrix of matrix \(|P| (KP) = |P| - U \). Hence the production change in a process due to a change in its unit consumption is:
\[ \Delta P_i = \tau_i P \% \Delta k_i \] (22)

Equation (22) shows that the production of a process remains constant when its efficiency changes, if and only if the recirculation ratio of the process is zero. This means that the process has no recirculations or feedbacks. For the rest of processes their production changes deal \( \Delta P_i = p_{ij} P \% \Delta k_i \), thus if the recirculation factor of the i-th process is zero, then the production of every process downstream does not change.

Hence, from both results we can state that a process is thermoeconomically isolated if and only if the recirculation ratio of the process is zero. Furthermore, to fulfil this condition in a real case, a completely disjoint set of mutually independent free variables affect unit consumption for each process, is required. It means that all interdependences between processes must be banished from the thermodynamic model.

Finally as a corollary of this result, we will prove that if a process is TI, then the local fuel impact is equal to the total fuel impact \( \Delta F_{r,i} = \Delta F_i \). In fact, the local fuel impact of a process is given by:
\[ \Delta F_i^* = \Delta P_i^* = k^*_{p,i} P_i + k^*_{i} \Delta P_i \] (23)

If the process is TI then \( \Delta P_i = 0 \), applying (16) we have: \( \Delta F_i^* = p_{ij} F_i^* \% \Delta k_i \), and applying that \( \tau_i = 0 \), then \( p_{ij} = 1 \) and \( \% \Delta k_i = \% \Delta k_j \), so this results in:
\[ \Delta F_{r,i} = \Delta F_i^* = F_i^* \% \Delta k_i \] (24)

Thermoeconomic Isolation is an ideal condition which cannot be achieved in most of real systems, but the closer to zero of the recirculation factor is the closer to TI conditions will be, and the less iteration loops will be needed in the local optimization procedure [12].

In our study case combustor and HRSG satisfy the TI conditions. The processes of the steam cycles have recirculation factors close to zero and could be consider thermoeconomically isolated. Here, the gas turbine and compressor do not satisfy TI conditions. Systems as Rankine cycles are near the TI conditions and this method works well. However, other systems based on gas turbine cycles have strong feedbacks and then high recirculation factors, in these cases TI is not satisfied and the convergence of local optimization procedures will not be good.
By means of graph theory algorithms [13], we can determine which processes are strongly connected and determine its recirculation factor. To apply local optimization procedures these processes should be aggregated. In case of the example of the paper, the gas turbine system should be considered as a component in order to apply local optimization.

5. Conclusions

The fuel impact formula is the mathematical representation of the principle of non-equivalence of the irreversibilities [14], therefore it needs a rigorous formalization. In this paper it has been shown that the fuel impact formula is a linear approximation to the fuel impact due to the increase of the local irreversibility of an individual process of a system. Moreover, a revised formula for the real fuel impact has been obtained, which coincides with the original exergy cost theory formula (1), when the system is sequential. For a non-sequential branched system including feedbacks, this expression becomes more complex, but essentially maintains the same idea. There exists a relationship between local irreversibility and the additional consumption of resources. Feedbacks or recyclings amplify the effect of malfunctions or local savings, thus we must pay more attention on processes where recycling plays an important role.

The importance of the fuel impact formula is paramount, because it justifies the practical reason for internal cost accounting. Or in other words, it answers the question of how many additional expenses we must pay because of presence of malfunctions in a process [15]. If the local investment $\Delta Z_i$ is related with the irreversibility that could be achieved, by means of the saving-investment formula, then a potential local investment is feasible if $\Delta Z_i < A_{x_i}$.

The paper also introduces a new parameter, the recirculation factor, which measures the degree of thermoeconomic isolation of a process and could be used to improve the Local-Global Optimization methodologies.

The main drawback to use the Fuel Impact Formula to analyse the feasibility of local improvements, lies in the interdependence between the processes efficiencies, which are not taken into consideration in the formula, which is based on the hypothesis that local changes do not affect the local behaviour of the rest of processes.

From the thermodynamic model of a system, the unit consumption of a process depends of a set of free variables. But, in general, several processes could have common free variables. Insofar as the number of parameters of the thermodynamic model is greater than the number of parameters of the thermoeconomic model, i.e. unit consumptions, the former will be more sensitive to the calculation of marginal costs of the type $(\partial \text{Resources}/\partial \text{Parameter})$, than the information of the fuel impact formula.

However, as it is shown in reference [16], when the system structure is complex, the change of characteristic parameters of a process have mainly local not global effects on the rest of processes, and structural effects predominate over local effects. As far as the processes are independent, i.e. $\partial x_{ji}/\partial x_i \approx 0$, the accuracy of the fuel input formula is improved, and gives a good picture of the plant behaviour.

Appendix

This appendix shows a brief review of some mathematical questions required to understand the paper development, and a description of the plant used as study case.

A. Thermoeconomic Input-Output Analysis
Thermoeconomic input-output [17] is an extension of the Input-Output analysis [18], based on the second law of thermodynamics and the concept of productive purpose [19]. An energy system is formed by, say, \( n \) processes. Let \( P_i \) denote the exergy of the \( i \)-th process production, which is used in part to meet the intermediate requirement as input resources of other processes and in part to meet the final demand of the system. If \( E_{ij} \) denotes the exergy of process \( i \) uses as resource for process \( j \), and \( E_{io} \) denotes the final demand produced in process \( i \), we have the following condition:

\[
P_i = E_{io} + E_{ii} + \cdots + E_{in} \quad i = 1, \ldots, n
\]

(25)

On the other hand, the input resources of each process \( i \), say \( F_i \), are in part coming from external resources and in part from the production of other processes. If \( E_{oi} \) the exergy of the external resources used in process \( i \), we have the condition:

\[
F_i = E_{oi} + E_{ui} + \cdots + E_{ni} \quad i = 1, \ldots, n
\]

(26)

Furthermore, Second Law states \( F_i - P_i = I_i \geq 0 \), where \( I_i \) denotes the irreversibility of the process \( i \) and \( k_i = F_i / P_i \geq 1 \) represents the unit exergy consumption of process \( i \). Now let \( \kappa_{ij} \) denote the quantity (exergy) of the process \( i \) production, which is used to obtain one unit of product in process \( j \), then we have \( E_{ij} = \kappa_{ij} P_j \), these ratios are calling technical coefficients, and satisfy:

\[
k_i = \kappa_{ii} + \kappa_{ui} + \cdots + \kappa_{ni}
\]

(27)

Therefore eq.(25) could be written in terms of technical coefficients as follows:

\[
P_i = E_{io} + \kappa_{i1} P_1 + \cdots + \kappa_{in} P_n \quad i = 1, \ldots, n
\]

(28)

Let \( F_i^* \) and \( P_i^* \) denote respectively the exergy cost of resources and product of a process, i.e. the amount of external resources, measured in terms of exergy, required to produce a unit of exergy. Then, the cost balance established:

\[
P_i^* = F_i^* = E_{io} + k_{p,i} E_{ui} + \cdots + k_{p,n} E_{ni} \quad i = 1, \ldots, n
\]

(29)

where \( k_{p,i} \) denotes the cost per production unit of process \( i \), applying the definition of technical coefficient to eq. (29), we obtain a system of \( n \) simultaneous equations, which let to determine the unit production cost of each process given the values of technical coefficients:

\[
k_{p,i} = \kappa_{ii} + k_{p,i} \kappa_{ui} + \cdots + k_{p,n} \kappa_{ni} \quad i = 1, \ldots, n
\]

(30)

In matrix notation, eq. (28) may be compactly expressed as:

\[
P = \omega_s + \langle KP \rangle P
\]

(31)

in which \( \langle KP \rangle \equiv [\kappa_{ij}] \) is the \((n \times n)\) matrix of technical coefficients, \( P \equiv [P_i] \) is the \((n \times 1)\) column vector of process production, and \( \omega_s \equiv [E_{io}] \) is the \((n \times 1)\) column vector of final demand. The matrix \( \langle KP \rangle \) is, in the input-output methodology, the matrix of direct requirement because it shows the quantity of product \( i \) required directly in the production of one unit of process \( j \).

To solve this system for the production vector \( P \), known the final demand vector \( \omega_s \), we have:

\[
P - \langle KP \rangle P = (U - \langle KP \rangle) P = \omega_s
\]

(32)

If the demand vector is positive, the matrix \( U - \langle KP \rangle \) is non-singular, then the desired production vector is evaluated as a function of the final demand and the technical coefficients:

\[
P = (U - \langle KP \rangle)^{-1} \omega_s
\]

(33)

In a similar way, (30) may be compactly expressed as:
where \( \mathbf{k}_p^+ = \left[ k^+_{p,j} \right] \) is the \((n \times 1)\) column vector of unit production costs and \( \mathbf{k}_0 = \left[ k_0 \right] \) is the \((n \times 1)\) column vector of external resources direct requirements. Therefore the unit production exergy cost vector could be obtained as a function of the technical coefficients:

\[
\mathbf{k}_p^+ = \mathbf{T} \left( \mathbf{U} - \mathbf{KP} \right)^{-1} \mathbf{k}_0
\]  

(35)

The total production requirement of process \( i \) in the production of a unit of process \( j \), both direct plus indirect, is revealed by, say \( p_{ij} \), the element \( i, j \)-th of the Leontief inverse matrix \( \mathbf{P} \equiv \left( \mathbf{U} - \mathbf{KP} \right)^{-1} \), which verifies \( \langle \mathbf{KP} \rangle \mathbf{P} = \mathbf{P} - \mathbf{U} \).

The total exergy of the external resources could be written as:

\[
F_t = \mathbf{T} \mathbf{k}_e \mathbf{P} = \mathbf{T} \mathbf{k}_p^+ \mathbf{e}_t
\]  

(36)

B. Directed Graphs

The productive structure of an energy system could be represented by a weight directed graph, where the \((n \times n)\) matrix \( \mathbf{E} \equiv \mathbf{E}_{ij} \) is its adjacency matrix. Here we review some interesting results related with directed graphs [20], applied on this paper.

A directed graph or digraph is a pair \( \mathbf{G}=\langle \mathbf{V}, \mathbf{E} \rangle \) of a set \( \mathbf{V} \), whose elements are called nodes, and a set \( \mathbf{E} \) of ordered pairs of nodes call edges. (loops are not allowed in simply digraph). A weighted digraph or network is a digraph with weights assigned for its edges. The adjacency matrix of a digraph is a matrix with rows and columns corresponding to the digraph nodes, where a non-diagonal entry \( a_{ij} \) is the weight of arcs from node \( i \) to node \( j \), and the diagonal entry \( a_{ii} = 0 \) if loops are not allowed.

A path in a graph is a sequence of nodes such that from each of its nodes there is an edge to the next vertex in the sequence. A cycle is a path such that the start vertex and end vertex are the same. The vertices of a directed cycle are said to be strongly connected. A digraph with no cycles is called directed acyclic graph (DAG). The strongly connected components of a directed graph are their maximal strongly connected subgraphs. If each strongly connected component is contracted to a single vertex, the resulting graph is a DAG.

A topological ordering of a directed graph is a linear ordering of its nodes such that, for every edge \( (u,v) \), \( u \) comes before \( v \) in the ordering. A topological ordering is possible if and only if the graph has is a DAG, and any DAG has at least one topological ordering. For this topological ordering the adjacency matrix of the DAG is strictly upper triangular, it means all the entries below the main diagonal are zero: \( a_{ij} = 0 \) \( \forall i \geq j \).

If the graph which represents an energy system is acyclic then its adjacency matrix \( \mathbf{E} \), is strictly upper triangular, and consequently \( \langle \mathbf{KP} \rangle \) is also strictly upper triangular. Therefore, the Leontief inverse matrix \( \mathbf{P} \) is upper triangular and its main diagonal is the unity vector.

C. The Sherman-Morrison Formula

Given a non-singular matrix \( \mathbf{A} \) and its inverse \( \mathbf{L} \equiv \mathbf{A}^{-1} \), assume that several elements of \( \mathbf{A} \) are changed, i.e. \( \bar{a}_{ij} = a_{ij} + \Delta a_{ij} \), producing \( \bar{\mathbf{A}} = \mathbf{A} + \Delta \mathbf{A} \), therefore the new inverse matrix \( \bar{\mathbf{A}}^{-1} \) can be found by adjusting the known matrix \( \mathbf{A}^{-1} \), by means of:

\[
\bar{\mathbf{L}} \equiv \bar{\mathbf{A}}^{-1} = \mathbf{L} - \frac{\mathbf{LA}\Delta \mathbf{L}}{\mathbf{U} + \Delta \mathbf{L}} = \frac{\mathbf{L}}{\mathbf{U} + \Delta \mathbf{L}}
\]  

(37)
If the change is on the form $\Delta A = v \otimes w$, for some vectors $v$ and $w$, where the outer product $v \otimes w$ is defined as $\Delta a_{ij} = v_i w_j$, then previous equation is simplified as:

$$
\mathbf{L} = \mathbf{L} - \frac{(Lv) \otimes (TwL)}{1 + \lambda}
$$

(38)

where $\lambda = ^T wL v$ is a scalar. This equation is called [21] Sherman-Morrison formula.

If $v$ is a unit vector $u_i = (0, \ldots, 1, \ldots 0)$ then $v \otimes w$ adds the elements of $w$ to the $i$-th row of $A$, meanwhile if $w$ is the unit vector $u_i$ then $v \otimes w$ adds the elements of $v$ to the $i$-th column of $A$. This formula let to compute easily the inverse of a matrix when only the elements of a row or column are modified.

**D. Thermoeconomic model of a combined cycle**

This appendix describes the model of a simple combined cycle power plant used a case study though this paper. Figure 3, shows its physical structure and the exergy of their flows.

![Figure 3. Physical diagram of study case, a combined cycle power plant.](image)

<table>
<thead>
<tr>
<th>Flow</th>
<th>Exergy (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.00</td>
</tr>
<tr>
<td>2</td>
<td>88091.00</td>
</tr>
<tr>
<td>3</td>
<td>224086.00</td>
</tr>
<tr>
<td>4</td>
<td>60266.00</td>
</tr>
<tr>
<td>5</td>
<td>12450.00</td>
</tr>
<tr>
<td>6</td>
<td>95289.00</td>
</tr>
<tr>
<td>7</td>
<td>61180.00</td>
</tr>
<tr>
<td>8</td>
<td>219880.00</td>
</tr>
<tr>
<td>9</td>
<td>3723.00</td>
</tr>
<tr>
<td>10</td>
<td>65.00</td>
</tr>
<tr>
<td>11</td>
<td>193.00</td>
</tr>
<tr>
<td>12</td>
<td>38817.00</td>
</tr>
<tr>
<td>13</td>
<td>31767.00</td>
</tr>
<tr>
<td>14</td>
<td>158.53</td>
</tr>
<tr>
<td>15</td>
<td>90000.00</td>
</tr>
<tr>
<td>16</td>
<td>3633.00</td>
</tr>
</tbody>
</table>

Its basic operational parameters are:

- The system is fueled by natural gas, with a LHV=45000 kJ/kg
- Net electric power of 90 MW, of which about 60 MW come from gas turbine and 30 MW come from steam turbine.
- Life steam conditions: 40 bar and 420°C
- Gas turbine inlet temperature: 870°C
- Gas turbine outlet temperature: 450°C
- Compression rate: 9

The Fuel-Product definition and the thermoeconomic properties of the plant are shown in Table 2, and have been obtained using TAESS [22]. The gases leaving the HRSG and the waste heat of condenser are considered as external exergy losses.
Table 2. Thermoeconomic properties of the combined cycle power plant

<table>
<thead>
<tr>
<th>Nr</th>
<th>Process</th>
<th>Fuel</th>
<th>Product</th>
<th>F (kW)</th>
<th>P (kW)</th>
<th>k</th>
<th>$k_p^*$</th>
<th>$k_p^*$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Combustor</td>
<td>$E_8$</td>
<td>$E_3-E_2$</td>
<td>219880</td>
<td>128439.3</td>
<td>1.7119</td>
<td>1.0000</td>
<td>1.7119</td>
</tr>
<tr>
<td>2</td>
<td>Compressor</td>
<td>$E_6$</td>
<td>$E_2-E_1$</td>
<td>95289</td>
<td>83196.7</td>
<td>1.1453</td>
<td>2.0578</td>
<td>2.3569</td>
</tr>
<tr>
<td>3</td>
<td>Gas Turbine</td>
<td>$E_3-E_4$</td>
<td>$E_7+E_6$</td>
<td>163820</td>
<td>156469.0</td>
<td>1.0470</td>
<td>1.9655</td>
<td>2.0578</td>
</tr>
<tr>
<td>4</td>
<td>HRSG</td>
<td>$E_4-E_5$</td>
<td>$E_12-E_{11}$</td>
<td>47816</td>
<td>34978.1</td>
<td>1.3670</td>
<td>1.9655</td>
<td>2.6869</td>
</tr>
<tr>
<td>5</td>
<td>Steam Turbine</td>
<td>$E_{12}-E_9$</td>
<td>$E_{13}$</td>
<td>35094</td>
<td>31767.0</td>
<td>1.1047</td>
<td>2.6890</td>
<td>2.9707</td>
</tr>
<tr>
<td>6</td>
<td>Pump</td>
<td>$E_{14}$</td>
<td>$E_{11}-E_{10}$</td>
<td>158.53</td>
<td>115.9</td>
<td>1.3676</td>
<td>2.4431</td>
<td>3.3412</td>
</tr>
<tr>
<td>7</td>
<td>Generator</td>
<td>$E_{7}+E_{13}$</td>
<td>$E_{14}+E_{15}$</td>
<td>92947</td>
<td>90158.5</td>
<td>1.0309</td>
<td>2.3698</td>
<td>2.4431</td>
</tr>
</tbody>
</table>

Nomenclature

$n$  Number of processes
$E$  Exergy of a flow (kW)
$F$  Fuel exergy of a process (kW)
$P$  Product exergy of a process (kW)
$I$  Irreversibility of a process
$Z$  Equipment investment (€/year)
$k$  Unit exergy consumption (kW/kW)
$k_p^*$  Unit cost of product (kW/kW)
$k_p^*$  Unit cost of fuel (kW/kW)
$F_T$  Total Fuel of the system (kW)
$A_x$  Fuel Impact (kW)

Greek symbols

$\Delta$  increment
$\kappa$  technical coefficients
$\tau$  recirculation factor
$\omega$  System output

Subscripts and superscripts

*  Exergy Cost
T  Transpose Matrix

Matrix and vectors

$\langle KP \rangle$  Matrix of technical or direct requirement coefficients ($n \times n$)
$|P|$  Production or total requirement coefficients matrix ($n \times n$)
$|P^*|$  Cost normalized production matrix ($n \times n$)

References


The introduction of exergy analysis to the thermo-economic modelling and optimisation of a marine combined cycle system

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Abstract:

Combined cycle systems are promising solutions for the efficient, cost-effective and environmentally friendly power generation and propulsion onboard ocean-going ships. However, their complexity is significantly increased by machinery space and weight limitations, multiple safety and operational constraints and inherently higher capital costs. To address concurrently these issues, a thermo-economic approach is required that is able to take into account the design, operation and control of the marine energy system throughout its mission profile in an integrated manner. In addition to that, the use of exergy analysis could further strengthen such a thermo-economic approach by associating the exergetic losses of each component to the overall system irreversibility and, thus, revealing and quantifying the internal sources of thermodynamic losses.

In this paper, we present an exergy analysis approach in thermo-economic modelling and optimisation of an advanced marine combined cycle system for an ocean-going container vessel. A thermo-economic model based on energy (first-law) analysis, that is used for model-based design and operation optimisation, is coupled with exergy analysis with the objective to: a) gain better insight of the complex energy conversion process through the identification and ranking of the components which contribute most to the overall system exergy destruction at each operating mode, and, b) to enhance the optimisation of the system by retrofitting/re-designing selected high irreversibility components.

The introduction of exergy analysis to the thermo-economic modelling of the combined cycle system resulted in better insight and understanding of its energy conversion processes, and of the sources of exergy losses. The contribution of components to the total system irreversibility was found to vary with the mode of operation of the vessel. The components that were identified to contribute most to the total irreversibility and that were able to be further re-designed and/or retrofitted, were allowed for further model-based design and operation optimisation. The optimisation results yielded an improved and more cost-effective combined cycle design. The presented approach of integrating exergy analysis in thermo-economic modelling and optimisation is generic and can be applied to various marine energy systems aiding significantly system designers and operators.

Keywords:
Marine energy systems, Thermo-economics, Exergy analysis, Optimisation.

1. Introduction

Combined cycle systems are promising integrated solutions for increasing the overall efficiency, reducing fuel costs and emissions of marine powerplants. However, there are certain challenges to handle including higher capital costs, space and weight requirements, operational and safety constraints and complexity. Advanced thermo-economic modelling, simulation and optimisation approaches have the ability to shed light on such issues.

Exergy analysis is a well-established method for the assessment, improvement and optimization of energy systems [1-4]. Exergy-based approaches have been widely applied on a range of energy systems applications spanning from optimisation [5-11], system assessment and improvement [12-20], monitoring and diagnostics [21-24] to multidisciplinary optimisation of complex aero-
thermodynamic processes [25, 26]. The level of complexity of these exergy-based approaches ranges from simple exergy analysis to more advanced approaches that incorporate categorisation of exergy losses and monetary costs.

This work builds upon the thermo-economic modelling and optimisation of an advanced marine combined cycle system for an ocean-going containership that was presented in [27]. Herein we extend the study with exergy analysis so as to gain better insight of the energy conversion process by identifying and ranking the components of the system with respect to their contribution to the overall exergy destruction. Subsequently, the optimisation problem has been solved again, this time retrofitting/re-designing selected high irreversibility components.

The following sections present the marine combined system description, the exergy analysis and its results and the second round of the thermo-economic design and operation optimisation of the system. Finally, the results and the benefits of this approach are discussed with respect to the potential added value to marine energy systems designers and operators.

2. Marine combined cycle system modelling and optimisation

2.1. System description

We consider a marine combined cycle system for a containership. The generic system model flowsheet and its mission profile are shown in Fig. 1, while the main application characteristics are given in Table 1. The detailed description of this system and its modelling can be found in [27]. The system consists of a marine Diesel engine for propulsion with its exhaust gases driving a power turbine and a dual pressure heat recovery module. The steam produced is expanded through a steam turbine. The steam and power turbines are in a single shaft arrangement with an electricity generator. The produced electric power is supplied to the ship’s grid to cover the demand. In addition, an electric motor is coupled to the main engine shaft-line, able to supply the propulsion shaft with excess power from the combined cycle. In cases where the ship’s electricity demand is not covered by the combined cycle system, the Diesel gen-sets supply the rest of the electricity.

There are two key features of the combined cycle system of Fig. 1: the turbocharger exhaust gas by-pass and the integration of water pre-heating with charge air cooling. A part of the cylinders exhaust by-passes the turbocharger to feed the high pressure superheater and the power turbine. The by-pass stream is then mixed with the bulk flow after the turbocharger to feed the rest of the heat recovery
module. This concept offers advanced heat recovery potential at the expense of reduced charge air pressure which has a negative impact on the engine’s efficiency. This trade-off is subject to the overall optimisation of the system. The water condensate is preheated in three-stages: a) in the engine’s lubricating oil cooler, b) in a stage of one of the charge air-coolers of the main engine, and c) the high pressure condensate is further preheated in a similar charge air-cooler stages. This integration of pre-heating with the engine cooling system can offer significant efficiency improvements in the steam cycle. 

A generic model of the marine combined cycle system has been computer implemented to our in house developed modelling framework DNV COSSMOS. COSSMOS is an acronym for Complex Ship Systems MOdelling and Simulation. We have developed a modular library of reconfigurable generic component models suitable for design, performance and transient operation analyses, and optimisation of ship machinery systems in a hierarchical process modelling environment. Our methodology is based on the mathematical modelling of the steady-state and dynamic thermofluid behaviour of marine machinery components resulting in systems of non-linear Partial Differential and Algebraic Equations (PDAE), subject to initial and boundary conditions. The DNV COSSMOS modelling framework is described in detail in [27-29].

Table 1. Principal application characteristics.

<table>
<thead>
<tr>
<th>Ship</th>
<th>Main engine</th>
<th>Diesel generators</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ship Type</td>
<td>Engine Type</td>
<td>Generator-sets type</td>
</tr>
<tr>
<td>Containership</td>
<td>2S-Diesel</td>
<td>4S-Diesel</td>
</tr>
<tr>
<td>Capacity</td>
<td>Number Type</td>
<td>Number</td>
</tr>
<tr>
<td>4500 TEU</td>
<td>of cylinders</td>
<td></td>
</tr>
<tr>
<td>Length overall</td>
<td>Number of TCs</td>
<td>Nominal power</td>
</tr>
<tr>
<td>260.05 m</td>
<td>3</td>
<td>1700 kW</td>
</tr>
<tr>
<td>Breadth</td>
<td>Bore / Stroke</td>
<td>Nominal speed</td>
</tr>
<tr>
<td>32.25 m</td>
<td>0.9m / 2.3m</td>
<td>720 rpm</td>
</tr>
<tr>
<td>Depth</td>
<td>Nominal power</td>
<td>Fuel consumption</td>
</tr>
<tr>
<td>19.30 m</td>
<td>36540 kW</td>
<td>191 gr/kWh</td>
</tr>
<tr>
<td>Draft</td>
<td>Nominal speed</td>
<td></td>
</tr>
<tr>
<td>12.60 m</td>
<td>104 rpm</td>
<td></td>
</tr>
<tr>
<td>Service speed</td>
<td>Exhaust gas flow</td>
<td></td>
</tr>
<tr>
<td>24.5 kn</td>
<td>92.5 kg/s</td>
<td></td>
</tr>
<tr>
<td>Year of built / Class</td>
<td>Exh. gas temperature</td>
<td></td>
</tr>
<tr>
<td>2004 / DNV</td>
<td>275°C</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Fuel consumption</td>
<td></td>
</tr>
<tr>
<td></td>
<td>186 gr/kWh</td>
<td></td>
</tr>
</tbody>
</table>

2.2. Optimisation problem formulation

In [27] a thermo-economic design and time-varying operation optimisation problem was formulated based on an energy (first-law) analysis model. The time-varying operational profile and power demand for a typical trading route of the vessel (Fig. 1) were also incorporated in the thermo-economic model.

The optimisation objective was the maximisation of the Net Present Value (NPV) of the investment on the combined cycle system compared to the traditional / baseline system configuration of the propulsion and electric power production, consisting of a propulsion marine Diesel engine and independent auxiliary Diesel generator sets:

$$\max_x(NPV) = -\phi \cdot \sum C_Y + \sum_{t=1}^{N_T} \left( \sum_{i=1}^{N_Y} \left( C_{op,\text{base},t} - C_{op,i} \right) \cdot \Delta \tau_t \cdot c_{TEU} \cdot \Delta(TEU) \right) \left(1 + ir\right)$$

where $Y\{\text{SUP}_{HP}, \text{SUP}_{LP}, \text{EVAP}_{HP}, \text{EVAP}_{LP}, \text{ST}, \text{PT}, \text{CND}, \text{HP pump}, \text{LP pump}, \text{HP piping}, \text{LP piping}, \text{Exhaust path piping}, \text{GEN}, \text{SM}\}$. The cost of components, appearing both in the heat
recovery and in the baseline system (main engine, gen-sets, etc.), is not included in the capital cost formulation since the NPV was calculated on a comparative basis.

The set $X$ of independent decision variables was divided in three subsets: $X_1$ with heat exchanger geometry variables, $X_2$ nominal performance variables and $X_3$ time-varying operation variables:

$$X = [X_1, X_2, X_3]^T$$

$$X_1 = \left[ D_i, L_i, l_p, l_b, h_f, N_{tubes, row}, N_{tubes, col} \right]_{i=\{HP, HP_{evap}, LP, LP_{evap}\}}$$

$$X_2 = \left[ P_{HP}, P_{LP}, \pi_{PT,n}, \eta_{is, PT,n}, P_{CND} \right]$$

$$X_3 = \left[ \dot{m}_{f, ME}, x_{HP} \right]_{t=1,\ldots, N_T}$$

This optimisation problem formulation captured successfully the main techno-economic drivers and technical challenges relevant to the introduction of the marine combined cycle when compared to a baseline/ not integrated system; namely:

- The additional capital of combined cycle components (by the use of component cost functions).
- The potential gain in operational expenses in fuel and lubricating oil consumptions by covering electric needs with the combined cycle additional output and utilising less the Diesel gen-sets.
- The potential loss of profit from the reduction of containers onboard due to the larger engine room size to host the combined cycle system.
- The influence of the time-varying operational strategy.

Finally, the optimisation problem was subject to numerous design, space, demand, operability and safety equality and inequality constraints, the most important of which are:

- Coverage of all of the onboard propulsion and electric energy needs;
- Operation of the turbocharger within safe surge margin limits (exhaust gas by-pass results to operation closer to the surge limit);
- Overall exhaust gas pressure drop (heat recovery sections) under an allowable limit to ensure safe operation of the main engine;
- Exhaust gas velocity within allowable limits in all operating modes;
- Outlet exhaust gas temperature higher than a minimum to avoid sulphuric acid condensation and subsequent corrosion;
- Dimensions of heat exchange components bounded by engine room space constraints.

This formulation resulted in a mixed integer non-linear problem (MINLP) consisting of 44 independent optimisation variables, 8 equality constraints and 35 inequality constraints. It is also noted that the combined cycle process model flowsheet, depicted in Fig. 1, consists of 9,674 equations.

2.3. Thermo-economic optimal solution

The results of the first-law thermo-economic optimisation of [27] are presented in Tables 2 and 3. The combined cycle system has an NPV of about 3.6 million USD, over 25 years lifetime, with a market interest rate of 10% and base fuel price 600 USD/ton. This corresponds to approximately 8 years discounted pay-back period. The overall system energy efficiency is 51.31% which represents an 11% increase compared to the baseline system efficiency of 46.19% [27]. The system delivers 5.9 MW additional electric power at nominal load. In all operating modes the use of the Diesel gen-sets is minimised, resulting in additional fuel and maintenance cost savings. In high loads (>85%) the combined cycle system supports the propulsion power plant, through the shaft motor, delivering
approximately 3% of the demand. Finally, for main engine loads below 50%, the exhaust by-pass is closed and the power turbine as well as the HP superheater is shut-down.

3. Introduction of exergy analysis

In this paper the exergy analysis is introduced to the thermo-economic model of the marine combined cycle system of [27]. For every node of the system process streams the specific physical, chemical exergy and exergy rate are evaluated from the relevant physical properties and process variables. Based on these figures the exergetic efficiency and irreversibility of the major system components are derived. For clarity, the numbering of stream nodes and short names of system components are depicted in Fig. 2 and the listing of system components is given in Table 4.

In the system flowsheet of Fig. 2 three types of process streams appear: air flowing into the Diesel engine, exhaust gas supplied to the heat recovery sections and water both in vapour and liquid phases. In addition, there are also mechanical connections in all rotating components for the torque (i.e. mechanical power) production. It is noted that the exergy rate of mechanical and electrical connection nodes is equal to the respective mechanical or electric power. The specific physical and chemical exergy at any node of a process stream are:

\[
\varepsilon_{ph} = h - h_0 - T_0 \cdot (s - s_0) \tag{4}
\]

\[
\varepsilon_{ch} = \left[ \sum_{i=1}^{N_{sp}} x_i \cdot \varepsilon_{ch,0} + RT_0 \cdot \sum_{i=1}^{N_{sp}} x_i \cdot \ln(x_i) \right] / MW \quad \forall i = 1, ..., N_{sp} \tag{5}
\]

The specific chemical exergy is calculated under the assumption of perfect gas mixtures [2]. The reference state is set at a pressure of \( p_0 = 1.013 \text{ bar} \) and temperature \( T_0 = 298.15 \text{ K} \).

For liquid hydrocarbons of the form \( C_xH_yO_zN_wS_v \) the following empirical relation is valid [3]:

\[
\varepsilon_{ch,f} = \left[ 1.0401 + 0.1728 \frac{y}{x} + 0.0432 \frac{z}{x} + 0.2169 \frac{v}{x} \left( 1 - 2.0628 \frac{y}{x} \right) \right] \cdot H_u \tag{6}
\]

The total exergy rate at each node of the process flowsheet is calculated:

\[
\dot{E} = \dot{m} \cdot (\varepsilon_{ph} + \varepsilon_{ch}) \tag{7}
\]

Table 2. Thermo-economic optimal combined cycle system design [27].

<table>
<thead>
<tr>
<th>Component Geometry</th>
<th>HP sup. (m)</th>
<th>HP evap. (m)</th>
<th>LP sup. (m)</th>
<th>LP evap. (m)</th>
<th>CND</th>
<th>Nominal Characteristics</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>( D_i )</td>
<td>0.049</td>
<td>0.0168</td>
<td>0.0173</td>
<td>0.018</td>
<td>0.030</td>
<td>( p_{HP} ) (bar)</td>
<td>10.210</td>
</tr>
<tr>
<td>( L )</td>
<td>6.000</td>
<td>3.876</td>
<td>3.000</td>
<td>4.014</td>
<td>5.000</td>
<td>( p_{LP} ) (bar)</td>
<td>2.388</td>
</tr>
<tr>
<td>( l_p )</td>
<td>0.098</td>
<td>0.200</td>
<td>0.200</td>
<td>0.200</td>
<td>-</td>
<td>( W_{ST,n} ) (kW)</td>
<td>2670</td>
</tr>
<tr>
<td>( l_b )</td>
<td>6.000</td>
<td>1.229</td>
<td>3.000</td>
<td>1.108</td>
<td>-</td>
<td>( p_{CND} ) (bar)</td>
<td>0.050</td>
</tr>
<tr>
<td>( h_f )</td>
<td>0.003</td>
<td>0.000</td>
<td>0.000</td>
<td>0.00003</td>
<td>-</td>
<td>( \pi_{PT,n} )</td>
<td>3.533</td>
</tr>
<tr>
<td>( N_{tubex,row} )</td>
<td>12</td>
<td>28</td>
<td>25</td>
<td>28</td>
<td>1000</td>
<td>( \dot{m}_{PT,n} )</td>
<td>18.590</td>
</tr>
<tr>
<td>( N_{tubex,col} )</td>
<td>2</td>
<td>28</td>
<td>13</td>
<td>27</td>
<td>-</td>
<td>( \eta_{PT,n} )</td>
<td>0.8596</td>
</tr>
<tr>
<td>( C_{\text{capital}} )</td>
<td>( = 2.631 \cdot 10^6 \text{USD} )</td>
<td>( A_{HRB} = 36.533 \text{m}^2 )</td>
<td>( H_{HRB} = 14.826 \text{m} )</td>
<td>( W_{PT,n} ) (kW)</td>
<td>3318</td>
<td></td>
<td></td>
</tr>
<tr>
<td>( \text{Profit} )</td>
<td>( = 682 \cdot 10^3 \text{USD/ year} )</td>
<td>( m_{\text{HP},n} ) (kg/s)</td>
<td>( = 2.631 )</td>
<td>( W_{HRB} ) (kW)</td>
<td>3318</td>
<td></td>
<td></td>
</tr>
<tr>
<td>( \text{Payback period} )</td>
<td>( = 8.16 \text{ years} )</td>
<td>( \eta_{\text{tot}} = 0.5131 )</td>
<td>( m_{\text{LP},n} ) (kg/s)</td>
<td>( = 2.338 )</td>
<td>( \text{USDNPV} )</td>
<td>( = 3.561 \cdot 10^6 \text{USD} )</td>
<td></td>
</tr>
</tbody>
</table>
Table 3. Thermo-economic optimal combined cycle operation [27].

<table>
<thead>
<tr>
<th>Operating Mode</th>
<th>Operating characteristics</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$m_{f, ME}$</td>
</tr>
<tr>
<td>High speed transit</td>
<td>1.5795</td>
</tr>
<tr>
<td>Normal speed</td>
<td>1.2521</td>
</tr>
<tr>
<td>Ballast transit</td>
<td>0.9032</td>
</tr>
<tr>
<td>Manoeuvring</td>
<td>0.7926</td>
</tr>
</tbody>
</table>

Fig. 2. Combined cycle system flowsheet with listing of components and numbering of stream nodes used in the exergy analysis.

Table 4. Combined cycle system components listing.

<table>
<thead>
<tr>
<th>Component</th>
<th>Short-name</th>
<th>Component</th>
<th>Short-name</th>
<th>Component</th>
<th>Short-name</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diesel engine</td>
<td>DE</td>
<td>Power turbine</td>
<td>PT</td>
<td>HP steam turbine</td>
<td>ST&lt;sub&gt;HP&lt;/sub&gt;</td>
</tr>
<tr>
<td>TC compressor</td>
<td>C</td>
<td>Exhaust gas mixer</td>
<td>MIX&lt;sub&gt;EXH&lt;/sub&gt;</td>
<td>LP steam turbine</td>
<td>ST&lt;sub&gt;LP&lt;/sub&gt;</td>
</tr>
<tr>
<td>TC turbine</td>
<td>T</td>
<td>HP superheater</td>
<td>SUP&lt;sub&gt;HP&lt;/sub&gt;</td>
<td>HP/LP steam mixer</td>
<td>MIX&lt;sub&gt;ST&lt;/sub&gt;</td>
</tr>
<tr>
<td>Diesel generators</td>
<td>DG</td>
<td>LP superheater</td>
<td>SUP&lt;sub&gt;LP&lt;/sub&gt;</td>
<td>Vacuum condenser</td>
<td>CND</td>
</tr>
<tr>
<td>Charge air cooler</td>
<td>HP</td>
<td>HP evaporator</td>
<td>EVAP&lt;sub&gt;HP&lt;/sub&gt;</td>
<td>Exhaust gas outlet</td>
<td>OUT&lt;sub&gt;EXH&lt;/sub&gt;</td>
</tr>
<tr>
<td>Charge air cooler</td>
<td>LP</td>
<td>LP evaporator</td>
<td>EVAP&lt;sub&gt;LP&lt;/sub&gt;</td>
<td>Generator/ shaft motor</td>
<td>GEN</td>
</tr>
<tr>
<td>Water pre-heater</td>
<td>HP</td>
<td>HP steam valve</td>
<td>VLV&lt;sub&gt;HP&lt;/sub&gt;</td>
<td>Shaft lines</td>
<td>SFT</td>
</tr>
<tr>
<td>Water pre-heater</td>
<td>LP</td>
<td>LP steam valve</td>
<td>VLV&lt;sub&gt;LP&lt;/sub&gt;</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
The total exergy rate of the fuel input to the system is:

$$\dot{E}_{\text{f,net}} = \dot{E}_{\text{f,DE}} + \dot{E}_{\text{f,DG}} = \left( m_{\text{f,DE}} + m_{\text{f,DG}} \right) e_{\text{ch,f}}$$  \hspace{1cm} (8)

For the heat exchange components the definitions for the exergetic efficiency and irreversibility rate depend on their purpose, i.e. heating or cooling [1]. For heating components, i.e CPH\textsubscript{HP}, CPH\textsubscript{LP}, SUP\textsubscript{HP}, SUP\textsubscript{LP}, EVAP\textsubscript{HP} and EVAP\textsubscript{LP}, the exergetic efficiency and the irreversibility rate are:

$$\zeta = \frac{\dot{E}_{\text{Cold, out}} - \dot{E}_{\text{Cold, in}}}{\dot{E}_{\text{Hot, in}} - \dot{E}_{\text{Hot, out}}}$$  \hspace{1cm} (9)

$$i = \left( \dot{E}_{\text{Hot, in}} - \dot{E}_{\text{Hot, out}} \right) - \left( \dot{E}_{\text{Cold, out}} - \dot{E}_{\text{Cold, in}} \right)$$  \hspace{1cm} (10)

with the indexes “Hot” and “Cold” denoting the heating and heated streams, respectively, while indexes “in” and “out” denoting the inlet and outlet from the component.

In a similar fashion, for the cooling components, i.e. CAC\textsubscript{HP}, CAC\textsubscript{LP} and CND:

$$\zeta = \frac{\dot{E}_{\text{Cold, in}} - \dot{E}_{\text{Cold, out}}}{\dot{E}_{\text{Hot, out}} - \dot{E}_{\text{Hot, in}}}$$  \hspace{1cm} (11)

$$i = \left( \dot{E}_{\text{Hot, out}} - \dot{E}_{\text{Hot, in}} \right) - \left( \dot{E}_{\text{Cold, in}} - \dot{E}_{\text{Cold, out}} \right)$$  \hspace{1cm} (12)

with the indexes “Hot” and “Cold” denoting the cooling and cooled streams, respectively.

For the expansion components in the system, i.e. T, PT, ST\textsubscript{HP} and ST\textsubscript{LP} the exergetic efficiency and the irreversibility rate are [1]:

$$\zeta = \frac{\dot{W}}{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}$$  \hspace{1cm} (13)

$$i = \left( \dot{E}_{\text{in}} - \dot{E}_{\text{out}} \right) - \dot{W}$$  \hspace{1cm} (14)

with $\dot{W}$ the expansion work produced per unit of time.

For the turbocharger compressor:

$$\zeta = \frac{\dot{E}_{\text{out}} - \dot{E}_{\text{in}}}{\dot{W}}$$  \hspace{1cm} (15)

$$i = \dot{W} - \left( \dot{E}_{\text{in}} - \dot{E}_{\text{out}} \right)$$  \hspace{1cm} (16)

The two junctions MIX\textsubscript{EXH} and MIX\textsubscript{ST} that serve the mixing of exhaust gas prior to the SUP\textsubscript{LP} and of HP with LP steam prior to ST\textsubscript{LP}, respectively, have:

$$\zeta = \frac{\dot{E}_{\text{out}}}{\dot{E}_{\text{in},1} + \dot{E}_{\text{in},2} + ...}$$  \hspace{1cm} (17)

$$i = \left( \dot{E}_{\text{in},1} + \dot{E}_{\text{in},2} + ... \right) - \dot{E}_{\text{out}}$$  \hspace{1cm} (18)

The HP/LP steam valves (VLV\textsubscript{HP} and VLV\textsubscript{LP}) have:

$$\zeta = \frac{\dot{E}_{\text{out}}}{\dot{E}_{\text{in}}}$$  \hspace{1cm} (19)
\[ i = \dot{E}_{\text{in}} - \dot{E}_{\text{out}} \tag{20} \]

The exergetic efficiency and the irreversibility rate of mechanical and electrical components, i.e. SFT and GEN, are equal to the mechanical efficiency and losses, respectively. The form of the respective calculation equations is that of Eqs. (19) and (20).

In the exhaust gas outlet, OUT\text{EXH}, the stream is dumped to the environment. Therefore, the exergetic efficiency is considered to be equal to zero. Similarly, the irreversibility rate is equal to the exergy of the outlet stream (Fig. 2):

\[ \dot{i}_{\text{EXH,out}} = \dot{E}_{13} \tag{21} \]

For the combustion block of the main engine (DE) an exergy balance is formulated (Fig. 2) and the irreversibility rate and exergetic efficiency are derived:

\[ \dot{E}_{\text{f,DE}} + \dot{E}_{\text{e}} = W_{\text{DE}} + \dot{E}_{7a} + \dot{E}_{7b} + \dot{i}_{\text{DE}} \tag{22} \]

\[ \zeta_{\text{DE}} = \frac{W_{\text{DE}}}{\dot{E}_{\text{f,DE}} + \dot{E}_{\text{e}}} \tag{23} \]

Finally, for the Diesel generators a simpler (“black-box”) exergy balance is employed yielding:

\[ \dot{i}_{\text{DG}} = \dot{E}_{\text{f,DG}} - \dot{W}_{\text{DG}} \tag{24} \]

\[ \zeta_{\text{DG}} = \frac{\dot{W}_{\text{DG}}}{\dot{E}_{\text{f,DG}}} \tag{25} \]

The irreversibility ratio (\( \delta \)) of each component is defined:

\[ \delta = \frac{\dot{i}_i}{\dot{E}_{\text{f,\text{tot}}}}, \text{ for each component } i \tag{26} \]

4. Exergy analysis results

We performed exergy analysis to the optimal marine combined cycle design of [27], which is briefly presented in Tables 2 and 3. The analysis covered the entire mission profile of the system. In Table 5 the component exergetic efficiencies, irreversibility rates, irreversibility ratios and component contributions to total system irreversibility are given. The results give a clear insight of the energy conversion processes within the system. The exergy losses are obtained on a component basis for each operating mode of the system. Therefore, the components that contribute the most to the overall exergy losses of the system are readily identified for the various operating modes. The analysis of the results on a component level reveals:

- The combustion block of the main propulsion engine (DE) is the biggest contributor to the overall exergy losses with an irreversibility of 49 to 50% of the chemical exergy of the fuel input for all modes of operation.
- The Diesel gen-sets (DG) have low exergetic efficiency causing high exergy losses. Their contribution to the overall irreversibility depends on their utilisation which is high at low loads.
- Excluding the combustion engines (i.e. DE and DG), the rest of the system components have a total irreversibility of 8.8 to 10.5% of the chemical exergy of the fuel input for all modes of operation. This represents a contribution to the overall exergy losses of 13 to 18% increasing with load demand.
The main engine charge air cooler stages CAC\textsubscript{HP} and CAC\textsubscript{LP} have very low exergetic efficiency in all operating modes. This is due to the heat rejection, i.e. exergy destruction, to the sea water cooling circuit. This allows for further low temperature heat recovery in these units as a subsequent improvement to the system. A first step towards that direction is the integrated condensate pre-heaters CPH\textsubscript{HP} and CPH\textsubscript{LP}, already installed. The exergy losses in CAC\textsubscript{HP} and CAC\textsubscript{LP} correspond to 0.7 to 3.7\% of the total system irreversibility, having a greater impact on high loads (i.e High speed transit and Normal speed modes). Thus, exergy analysis offers a formal justification for the introduction of the integrated condensate pre-heaters CPH\textsubscript{HP} and CPH\textsubscript{LP} as a means to minimise the exergy destruction in the charge air cooling module.

The condenser has also low exergetic efficiency since heat is again rejected to sea cooling water. However the condenser’s contribution to the overall system irreversibility is relatively low.

The steam valves VLV\textsubscript{HP} and VLV\textsubscript{LP} have a minimal contribution to the overall system irreversibility, since the amount of throttling is kept at a minimum, as their purpose is the HP/LP pressure regulation prior to admission to the steam turbine.

The mixing process at the two junctions MIX\textsubscript{EXH} and MIX\textsubscript{ST} has a very low (even negligible) contribution to the overall exergy losses, despite the fact that mixing is a usual source of high irreversibility in most cases. This is mainly due that in both junctions the inlet streams have very similar pressure and temperature and the same composition.

The exhaust gas outlet to the environment causes the loss of roughly 3 to 4\% of the chemical exergy of the fuel input. The contribution of OUT\textsubscript{EXH} to the total system irreversibility varies from 4\% to 6\%. This source of losses cannot easily be alleviated due to minimum exhaust temperature constraints. However, the use of low sulphur fuels or SO\textsubscript{X} scrubbers will result into lower minimum permissible exhaust temperatures and, therefore, the exergy losses will be lower.

The LP section of the steam turbine contributes 1.2 to 1.6\% to the total system irreversibility, with the higher rates at low loads. On the other hand, the HP section has a relatively low contribution to the total exergy losses. The ST\textsubscript{LP} high contribution is primarily due to its lower isentropic efficiency (compared to the HP section). In addition, at low loads the exhaust gas by-pass is closed and the PT and SUP\textsubscript{HP} are switched-off. Therefore, the LP steam is then becoming more crucial to the combined cycle power production.

At low loads (i.e. Ballast transit and Manoeuvring modes), when the exhaust gas by-pass is closed and the PT and SUP\textsubscript{HP} are switched-off, the contribution of the HP evaporator in the overall system irreversibility is increased up to 1.6\%. Therefore, its importance for efficient low load operation is high.

The main engine turbocharger (C and T) is responsible for the loss of roughly 2\% of the exergy of the fuel input. This amounts to a contribution to the overall irreversibility of 2.7 to 3.9\%, depending on the operating mode. Therefore, the turbocharger is an important sub-system with a high potential for improvement through re-designing and matching with the main engine.

On a system level, the exergetic efficiency at each operating mode ranges from 34 to 40\%, being minimum at low loads. It is noted that the system energy efficiencies (Table 3) range from 45 to 54\%. Hence, there is a significant difference between the energetic and exergetic approach in the system analysis. As shown in Table 5, the combined cycle system efficiencies are significantly higher to those of the baseline system (i.e. independent power production), yielding an improvement from 18 to 25\%, increasing at low loads. This strongly supports the marine combined cycle system concept as a means to save primary energy onboard. Further to the results of Table 5, the system overall exergetic efficiency, considering the entire mission profile is: $\dot{\xi}_{\text{tot}} = 37.83\%$, whereas the overall exergetic efficiency of the baseline system is 32.39\%. It is noted that the comparison between combined cycle and baseline system using first-law energy efficiencies (paragraph 2.3), yielded an improvement of 11\%. 

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<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>DE</td>
<td>32.53 52338</td>
<td>48.998 81.63</td>
<td>49.168 82.38</td>
<td>32.70 6010</td>
<td>52.227 82.42</td>
</tr>
<tr>
<td>C</td>
<td>88.61 1794</td>
<td>1.680 2.80</td>
<td>1.578 2.64</td>
<td>85.47 1205</td>
<td>82.38 1.90</td>
</tr>
<tr>
<td>T</td>
<td>95.67 713</td>
<td>0.668 1.11</td>
<td>0.686 1.15</td>
<td>86.68 1273</td>
<td>2.01 0.47</td>
</tr>
<tr>
<td>CAC_{HP}</td>
<td>11.65 1874</td>
<td>1.754 2.92</td>
<td>1.542 2.58</td>
<td>8.72 773</td>
<td>1.128 1.05</td>
</tr>
<tr>
<td>CAC_{LP}</td>
<td>11.30 814</td>
<td>0.762 1.27</td>
<td>0.672 1.13</td>
<td>6.99 150</td>
<td>0.301 0.47</td>
</tr>
<tr>
<td>CPH_{HP}</td>
<td>78.60 61</td>
<td>0.057 0.10</td>
<td>0.036 0.06</td>
<td>76.88 7</td>
<td>0.015 0.02</td>
</tr>
<tr>
<td>CPH_{LP}</td>
<td>53.86 132</td>
<td>0.124 0.21</td>
<td>0.086 0.14</td>
<td>70.82 10</td>
<td>0.019 0.03</td>
</tr>
<tr>
<td>PT</td>
<td>87.57 473</td>
<td>0.442 0.74</td>
<td>0.367 0.62</td>
<td>85.94 206</td>
<td>0.300 0.50</td>
</tr>
<tr>
<td>MIX_{EXH}</td>
<td>99.99</td>
<td>0.000 0.00</td>
<td>0.000 0.00</td>
<td>100.00 0</td>
<td>0.000 0.00</td>
</tr>
<tr>
<td>SUP_{HP}</td>
<td>64.03 91</td>
<td>0.085 0.14</td>
<td>0.059 0.10</td>
<td>69.18 34</td>
<td>0.050 0.08</td>
</tr>
<tr>
<td>SUP_{LP}</td>
<td>55.84 33</td>
<td>0.031 0.05</td>
<td>0.027 0.05</td>
<td>62.04 20</td>
<td>0.029 0.05</td>
</tr>
<tr>
<td>EVAP_{HP}</td>
<td>84.07 351</td>
<td>0.329 0.55</td>
<td>0.253 0.42</td>
<td>85.04 215</td>
<td>0.314 0.52</td>
</tr>
<tr>
<td>EVAP_{LP}</td>
<td>76.11 407</td>
<td>0.381 0.64</td>
<td>0.381 0.64</td>
<td>77.11 276</td>
<td>0.402 0.66</td>
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<tr>
<td>VLV_{HP}</td>
<td>99.14 20</td>
<td>0.019 0.03</td>
<td>0.016 0.03</td>
<td>99.13 12</td>
<td>0.018 0.03</td>
</tr>
<tr>
<td>VLV_{LP}</td>
<td>97.05 43</td>
<td>0.040 0.07</td>
<td>0.042 0.07</td>
<td>97.09 30</td>
<td>0.043 0.07</td>
</tr>
<tr>
<td>ST_{HP}</td>
<td>85.53 110</td>
<td>0.103 0.17</td>
<td>0.110 0.18</td>
<td>79.75 91</td>
<td>0.132 0.22</td>
</tr>
<tr>
<td>ST_{LP}</td>
<td>73.31 723</td>
<td>0.677 1.13</td>
<td>0.703 1.18</td>
<td>68.98 549</td>
<td>0.800 1.32</td>
</tr>
<tr>
<td>MIX_{ST}</td>
<td>99.93 2</td>
<td>0.002 0.00</td>
<td>0.001 0.00</td>
<td>99.91 2</td>
<td>0.003 0.00</td>
</tr>
<tr>
<td>CND</td>
<td>23.38 215</td>
<td>0.201 0.34</td>
<td>0.182 0.31</td>
<td>29.70 130</td>
<td>0.190 0.31</td>
</tr>
<tr>
<td>GEN</td>
<td>90.00 285</td>
<td>0.267 0.44</td>
<td>0.113 0.19</td>
<td>0.00 0.00</td>
<td>0.000 0.00</td>
</tr>
<tr>
<td>SFT</td>
<td>99.00 60</td>
<td>0.056 0.09</td>
<td>0.047 0.08</td>
<td>99.00 28</td>
<td>0.041 0.07</td>
</tr>
<tr>
<td>OUT_{EXH}</td>
<td>0.00 3573</td>
<td>3.345 5.57</td>
<td>3.617 6.06</td>
<td>0.00 3259</td>
<td>3.734 6.17</td>
</tr>
<tr>
<td>DG</td>
<td>0.00 0</td>
<td>0.000 0.00</td>
<td>0.000 0.00</td>
<td>32.96 497</td>
<td>0.725 1.20</td>
</tr>
<tr>
<td>System</td>
<td>39.98 64114</td>
<td>60.023 100</td>
<td>51455 100</td>
<td>39.53 41450</td>
<td>60.472 100</td>
</tr>
<tr>
<td>Baseline</td>
<td>34.11</td>
<td>34.19</td>
<td>33.18</td>
<td>31.48</td>
<td>27.15</td>
</tr>
</tbody>
</table>
The exergetic efficiency of the combined cycle is improved by 17%, compared to the baseline system. Therefore, exergy analysis provides an improved quantitative argument in favour of the combined cycle system.

A final step to utilise and exploit further the exergy analysis results of Table 5, is the ranking of the system components based on their contribution to the overall system irreversibility \( \left( \delta_i/\delta_{\text{tot}} \right) \) weighted over the operating hours of each mode of the system’s mission.

\[
\left( \frac{\delta_i}{\delta_{\text{tot}}} \right)_w = \frac{\sum_{t=1}^{N_t} \left( \frac{\delta_i}{\delta_{\text{tot}}} \right)_t \cdot \Delta \tau_t}{\sum_{t=1}^{N_t} \Delta \tau_t}, \quad \forall \text{ component } i
\]  

The weighted component contributions to the overall system irreversibility are given in Table 6. Via this ranking the individual components that should be carefully considered when designing and operating marine combined cycle systems are identified. The combustion engines, the exhaust losses, the turbocharging system, the charge air coolers (HP and LP) and the steam evaporators (HP and LP) are among the largest 10 contributors to the exergy losses of the system.

5. System re-design and optimisation

Based on the ranking results of Table 6, further system re-design and optimisation is considered, focusing on the components with the higher contribution to the overall system irreversibility. The improvement of the combustion engines (ranking positions 1 and 3) requires detailed modelling of engine design and combustion, which is a very broad field, yet outside the scope of this system-oriented study. The exhaust losses (rank 2) are already incorporated in the thermo-economic modelling and optimisation through appropriate minimum exhaust gas temperature constraints [27]. In addition, the HP and LP evaporators are also incorporated in the optimisation problem formulation, with their design characteristics being independent decision variables. Similarly the heat integration of the charge air coolers (ranking positions 6 and 10) with the condensate water pre-heaters is already implemented in the marine combine cycle system. Finally, only the turbocharger compressor and turbine (ranking positions 4 and 5) was selected for re-design and optimisation in order to improve further the combined cycle system.

<table>
<thead>
<tr>
<th>Component</th>
<th>Rank</th>
<th>( \left( \frac{\delta_i}{\delta_{\text{tot}}} \right)_w ) [%]</th>
<th>Component</th>
<th>Rank</th>
<th>( \left( \frac{\delta_i}{\delta_{\text{tot}}} \right)_w ) [%]</th>
<th>Component</th>
<th>Rank</th>
<th>( \left( \frac{\delta_i}{\delta_{\text{tot}}} \right)_w ) [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>DE</td>
<td>1</td>
<td>81.776</td>
<td>EVAP_{LP}</td>
<td>9</td>
<td>0.695</td>
<td>SUP_{LP}</td>
<td>17</td>
<td>0.055</td>
</tr>
<tr>
<td>OUT_{EXH}</td>
<td>2</td>
<td>6.110</td>
<td>CAC_{LP}</td>
<td>10</td>
<td>0.675</td>
<td>SUP_{HP}</td>
<td>18</td>
<td>0.045</td>
</tr>
<tr>
<td>DG</td>
<td>3</td>
<td>2.150</td>
<td>CND</td>
<td>11</td>
<td>0.330</td>
<td>CPH_{HP}</td>
<td>19</td>
<td>0.035</td>
</tr>
<tr>
<td>C</td>
<td>4</td>
<td>2.098</td>
<td>PT</td>
<td>12</td>
<td>0.276</td>
<td>VLV_{HP}</td>
<td>20</td>
<td>0.034</td>
</tr>
<tr>
<td>T</td>
<td>5</td>
<td>1.531</td>
<td>ST_{HP}</td>
<td>13</td>
<td>0.253</td>
<td>GEN</td>
<td>21</td>
<td>0.028</td>
</tr>
<tr>
<td>CAC_{HP}</td>
<td>6</td>
<td>1.523</td>
<td>VLV_{LP}</td>
<td>14</td>
<td>0.072</td>
<td>MIX_{ST}</td>
<td>22</td>
<td>0.021</td>
</tr>
<tr>
<td>ST_{LP}</td>
<td>7</td>
<td>1.435</td>
<td>CPH_{LP}</td>
<td>15</td>
<td>0.066</td>
<td>MIX_{EXH}</td>
<td>23</td>
<td>0.000</td>
</tr>
<tr>
<td>EVAP_{HP}</td>
<td>8</td>
<td>0.733</td>
<td>SFT</td>
<td>16</td>
<td>0.057</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

In the initial study [27], the turbocharger design and matching with the main engine in the combined cycle system are the same as in the baseline system. However, for marine combined cycle systems with exhaust gas by-pass it is strongly recommended in the technical literature [30-32] that the turbocharger selection and matching processes are revised. The exergy analysis results (Section
4) highlight and justify in a formal way this technical guideline to re-match the turbocharger for exhaust-gas by-pass operation.

In order to introduce the turbocharger re-design to a new thermo-economic optimisation formulation, the compressor and turbine maps [27] are expressed in non-dimensional form as functions of their nominal point. Then, a set of scaling factors are introduced on the nominal compressor and turbine flow rate and pressure ratio. By changing these scaling factors new compressor and turbine nominal points are created and, in turn, new component performance maps are generated. These scaling factors are introduced as new independent decision variables to the thermo-economic optimisation problem formulation. It is noted that the isentropic efficiency maps have not been scaled, under the assumption that the current and the re-designed (by scaling) turbocharger have the same efficiency. In addition, it is assumed that no additional capital cost is introduced by the turbocharger re-design.

Let \( x_{C,m}, x_{C,p}, x_{T,m} \) and \( x_{T,p} \) be the aforementioned scaling factors. In the optimisation problem formulation of Eqs. (1) – (3), the nominal performance independent variables \( \mathbf{X}_2 \) of Eq. (3) are now given by:

\[
\mathbf{X}_{2,\text{new}} = \left[ p_{HP}, p_{LP}, p_{\alpha,PT,n}, p_{CND}, x_{C,m}, x_{C,p}, x_{T,m}, x_{T,p} \right]
\] (28)

The optimisation problem is solved again subject to the same constraints as in Section 2.2. The new optimisation results are briefly presented in Table 7. It is noted that the geometry of the heat exchange components is not given in Table 7, since there were only minor differences to the previous results of Table 2. The new optimal solution yields a significant improvement of the cost-effectiveness of the system. The new \( \text{NPV}^\ast = 4.8 \text{ million USD} \) represents more than 30% improvement than the optimal results without turbocharger re-design. This halves the payback period of the investment, to 4 years. The overall energy efficiency is now 51.73%, improved by 0.8% from the previous configuration, while the overall exergy efficiency is 38.34%, improved by 1.3%. Once again the exergetic efficiency is able to capture effectively the improvement to the performance of the system. The results indicate that the fuel consumption (hence the efficiency) is the main and very strong driver to the economic performance of such systems. An exergy efficiency relative improvement of 1.3% leads to cost savings of 20%.

<table>
<thead>
<tr>
<th>Nominal Characteristics</th>
<th>Value</th>
<th>Nominal Characteristics</th>
<th>Value</th>
<th>Operating characteristics</th>
</tr>
</thead>
<tbody>
<tr>
<td>( p_{HP} ) (bar)</td>
<td>10.050</td>
<td>( W_{PT,n} ) (kW)</td>
<td>3325</td>
<td>( \dot{m}_{f,ME} ) (kg/s)</td>
</tr>
<tr>
<td>( p_{LP} ) (bar)</td>
<td>2.410</td>
<td>( \dot{m}_{rt,HP,n} ) (kg/s)</td>
<td>2.645</td>
<td>1.5624 0.1776</td>
</tr>
<tr>
<td>( W_{ST,n} ) (kW)</td>
<td>2701</td>
<td>( \dot{m}_{rt,LP,n} ) (kg/s)</td>
<td>2.334</td>
<td>1.2387 0.1617</td>
</tr>
<tr>
<td>( p_{CND} ) (bar)</td>
<td>0.050</td>
<td>( x_{C,m} ) (-)</td>
<td>0.9900</td>
<td>0.8931 0</td>
</tr>
<tr>
<td>( \pi_{PT,n} )</td>
<td>3.560</td>
<td>( x_{C,p} ) (-)</td>
<td>0.9968</td>
<td>0.7826 0</td>
</tr>
<tr>
<td>( \dot{m}_{PT,n} ) (kg/s)</td>
<td>18.526</td>
<td>( x_{T,m} ) (-)</td>
<td>0.9900</td>
<td></td>
</tr>
<tr>
<td>( \eta_{\alpha,PT,n} )</td>
<td>0.8586</td>
<td>( x_{T,p} ) (-)</td>
<td>1.0000</td>
<td></td>
</tr>
</tbody>
</table>

System characteristics

\[
\text{Cost}_{\text{capital}}^\ast = 2.638 \times 10^6 \text{USD} \quad \text{NPV}^\ast = 4.787 \times 10^6 \text{USD}
\]

\[
\text{Profit}^\ast = 818 \times 10^3 \text{USD/ year} \quad \eta_{\text{tot}}^\ast = 0.5173 \quad \zeta_{\text{tot}}^\ast = 0.3834
\]

Payback period = 4.09 years

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The results also indicate that the system efficiency is very sensitive to the turbocharger performance. The new maps, scaled at a maximum of 1% relative to the original ones, are able to yield the aforementioned improvements in both efficiency and cost-effectiveness of the system.

6. Conclusions

In this work, exergy analysis was introduced to the thermo-economic modelling and optimisation of a marine combined cycle system. With the aid of exergy analysis, a better insight of the system was gained and the internal sources of exergy losses were identified. The analysis provided a formal and quantifiable basis of ranking the importance of the system components with respect to their contributions to the overall irreversibility of the system. The most important components proved to be the combustion engine(s), turbocharging modules, charge air coolers, steam evaporators and steam turbines, with their contribution varying with the mode of operation of the vessel.

As a result of the exergy analysis, the re-design of the main engine turbochargers was formally justified. Therefore, a new thermo-economic design and operation optimisation problem incorporating the turbocharger re-design was solved. This resulted into a significantly improved system design and operation. By re-design of the turbocharger, a 30% improvement on the optimum NPV of the investment is attained with a payback period of 4 years. The overall exergetic efficiency of the system is also improved by 1.3%, reaching 38.3%. These results confirm the key importance of the turbocharging module as it was assessed and quantified through the exergy analysis.

The insights from the exergy analysis provide the system designers and operators with a clear picture of the internal energy conversion process and the potential focus areas for overall system improvements. In addition, the importance of the individual components is quantified in terms of their irreversibility. The results of the exergy analysis provide an improved quantitative argument in favour of the marine combined cycle system as a viable and cost-effective concept to save primary energy onboard vessels.

Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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</thead>
<tbody>
<tr>
<td>A</td>
<td>Area, m²</td>
</tr>
<tr>
<td>C</td>
<td>Cost, USD</td>
</tr>
<tr>
<td>c</td>
<td>Specific cost, USD / (kW or kWh or kg)</td>
</tr>
<tr>
<td>D_i</td>
<td>Internal diameter, m</td>
</tr>
<tr>
<td>E</td>
<td>Exergy rate, kW</td>
</tr>
<tr>
<td>f_L</td>
<td>Load factor</td>
</tr>
<tr>
<td>h</td>
<td>Specific enthalpy, kJ/kg</td>
</tr>
<tr>
<td>h_f</td>
<td>Tube fin height, m</td>
</tr>
<tr>
<td>H</td>
<td>Height, m</td>
</tr>
<tr>
<td>H_u</td>
<td>Lower heating value, kJ/kg</td>
</tr>
<tr>
<td>İ</td>
<td>Irreversibility rate, kW</td>
</tr>
<tr>
<td>ir</td>
<td>Market interest rate</td>
</tr>
<tr>
<td>L</td>
<td>Length, m</td>
</tr>
<tr>
<td>l_p</td>
<td>Tube pitch, m</td>
</tr>
<tr>
<td>l_b</td>
<td>Buffle spacing, m</td>
</tr>
<tr>
<td>m</td>
<td>Mass flow rate, kg/s</td>
</tr>
<tr>
<td>MW</td>
<td>Molecular weight, kg/kmol</td>
</tr>
<tr>
<td>N_sp</td>
<td>Number of species in a mixture</td>
</tr>
<tr>
<td>N_T</td>
<td>Number of time intervals</td>
</tr>
<tr>
<td>N_years</td>
<td>Economic life of the investment, years</td>
</tr>
<tr>
<td>p</td>
<td>Pressure, bar</td>
</tr>
<tr>
<td>R_g</td>
<td>Gas constant, kJ/(kmolK)</td>
</tr>
<tr>
<td>s</td>
<td>Specific entropy, kJ/(kgK)</td>
</tr>
<tr>
<td>SM</td>
<td>Surge margin</td>
</tr>
<tr>
<td>T</td>
<td>Temperature, K</td>
</tr>
<tr>
<td>W</td>
<td>Power, kW</td>
</tr>
<tr>
<td>x</td>
<td>Molar composition</td>
</tr>
<tr>
<td>X</td>
<td>Set of independent optimisation variables</td>
</tr>
<tr>
<td>x_bp</td>
<td>Exhaust gas by-pass ratio</td>
</tr>
</tbody>
</table>

Greek Symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>𝞀</td>
<td>Irreversibility to chemical exergy of the fuel ratio</td>
</tr>
<tr>
<td>Δt</td>
<td>Time interval duration, hours</td>
</tr>
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<td>ε</td>
<td>Specific exergy, (kJ/kg)</td>
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<tr>
<td>η</td>
<td>Energy efficiency</td>
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<tr>
<td>ζ</td>
<td>Exergy efficiency</td>
</tr>
<tr>
<td>π</td>
<td>Pressure ratio</td>
</tr>
<tr>
<td>φ</td>
<td>Capital cost scaling factor</td>
</tr>
</tbody>
</table>
Subscripts
0 reference
a Air
AUX Auxiliaries
base Baseline system
C Compressor
CAC Charge air cooler
ch Chemical
CND Condenser
del Diesel engine
dg Diesel generator
electric electric
exh Exhaust (gas)
f Fuel
GEN Generator
HP High pressure

Superscripts
* Optimum

References

The Relationship Between Costs and Environmental Impacts in Power Plants: An Exergy-Based Study

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Abstract:
Exergy-based methods (exergetic, exergoeconomic and exergoenvironmental analyses) are powerful tools for developing, evaluating and improving energy-conversion systems. In an exergoeconomic analysis, thermodynamic inefficiencies – represented by exergy destruction – are used together with investment cost to calculate the “cost-optimal” alternative for the overall plant. Analogously, in an exergoenvironmental analysis, the aim is to reduce the total environmental impact. Normally, exergoeconomic and exergoenvironmental analyses are conducted independently of each other. Thus, until now, the improvement of a plant has been considered in terms of reduction of either costs or environmental impact. To simultaneously decrease the investment costs and the component-related (manufacturing or construction-related) environmental impacts, their relationship with exergy destruction must be studied in parallel. This paper examines the relationship between exergoeconomic and exergoenvironmental data under various plant operating conditions. A combined-cycle power plant is analyzed and options for a simultaneous improvement from the economic and environmental viewpoints are discussed.

Keywords:
Exergy analysis, Exergoeconomic Analysis, Exergoenvironmental Analysis.

1. Introduction

Thermodynamic, economic, and environmental-impact analyses are three tools used for the evaluation and improvement (optimization) of an energy conversion system. These analyses reveal
(a) The real thermodynamic inefficiencies and the processes that cause them,
(b) The costs associated with equipment and thermodynamic inefficiencies as well as the connection between these two important factors,
(c) The environmental impact associated with equipment and thermodynamic inefficiencies as well as the connection between these two sources of environmental impact, and
(d) Possible measures that would improve the efficiency and the cost effectiveness and would reduce the environmental impact of the system being studied.

An exergoeconomic analysis [1-4] consists of an exergetic analysis, an economic analysis, and an exergoeconomic evaluation. An exergoenvironmental analysis [4,5] consists of an exergetic analysis, a life cycle assessment (LCA) of the environmental impact and an exergoenvironmental evaluation conducted in analogy with the exergoeconomic one.

In the exergoeconomic and exergoenvironmental analyses (which are already known as powerful tools for analyzing, evaluating and improving energy-conversion systems) the economic analysis and the LCA (therefore the exergoeconomic and the exergoenvironmental analysis) are conducted independently of each other. Obviously, then the conclusions from these analyses are also obtained independently.
In this paper we try to obtain consistent conclusions on how to improve an energy conversion system by reducing simultaneously cost and environmental impact. Note that we do not want to assign cost values to environmental impacts (or vice versa) because of the arbitrariness associated with this procedure. As before, the main assumption is that data obtained from an LCA and from a cost analysis are independent from each other.

2. Exergy-based Analyses

2.1 Exergetic analysis

Using the exergy rates associated with fuel and product [1-3], \( \dot{E}_{F,k} \) and \( \dot{E}_{P,k} \), respectively, the exergetic balance for the \( k \)-th component is

\[
\dot{E}_{F,k} = \dot{E}_{P,k} + \dot{E}_{D,k}
\]  

(1)

The total exergy destruction within the \( k \)-th component (\( \dot{E}_{D,k} \)) can be determined through an exergy balance.

The exergetic efficiency for the \( k \)-th component is

\[
\eta_k = \frac{\dot{E}_{P,k}}{\dot{E}_{F,k}} = 1 - \frac{\dot{E}_{D,k}}{\dot{E}_{F,k}}
\]

(2)

Additional variables used in the exergetic analysis can be found in many publications, for example, [2-4].

2.2 Exergoeconomic analysis

The exergoeconomic analysis is an exergy-based method that identifies and calculates the location, magnitude, causes and costs of thermodynamic inefficiencies in an energy conversion system. An exergoeconomic analysis is conducted at the component level of a system and reveals (a) the relative cost importance of each component, and (b) options for improving the overall cost effectiveness.

The exergoeconomic model of an energy conversion system [2-4] consists of:

(a) Cost balances written for each system component

\[
\hat{C}_{P,k} = \hat{C}_{F,k} + \hat{Z}_k
\]

(3a)

or

\[
c_{P,k} \dot{E}_{P,k} = c_{F,k} \dot{E}_{F,k} + \dot{Z}_k
\]

(3b)

Here \( \hat{C}_{P,k} \) and \( \hat{C}_{F,k} \) are the cost rates associated with fuel and product, whereas \( c_{P,k} \) and \( c_{F,k} \) are the corresponding costs per unit of exergy. Finally \( \dot{Z}_k \) is the sum of cost rates associated with capital investment (\( CI \)) and operating & maintenance (\( O&M \)) expenditures for the \( k \)-th component

\[
\dot{Z}_k = \dot{Z}_{k}^{CI} + \dot{Z}_{k}^{OM}
\]

(4)
To simplify the discussion, we assumed in the present paper, that the contribution of $\dot{Z}_k^{OM}$ remains constant when the design is changed, and, therefore, the changes in the value of $\dot{Z}_k$ are associated only with changes in the capital investment cost $\dot{Z}_k^{CI}$.

(b) **Auxiliary costing equations** based on the P-rule and the F-rule as, they have been finalized in [3].

The following exergoeconomic variables may be used for improving the overall effectiveness of the $k$-th component in an iterative optimization:

- Cost rate associated with the exergy destruction within the $k$-th component
  $$\dot{C}_{D,k} = c_{P,k} \dot{E}_{D,k}$$  (5)

- Total costs associated with the component, which are the sum ($\dot{Z}_k^{CI} + \dot{C}_{D,k}$)

- Relative cost difference
  $$r_k = \frac{c_{P,k} - c_{F,k}}{c_{F,k}} = \frac{1 - \dot{e}_k}{\dot{e}_k} \frac{\dot{Z}_k}{\dot{C}_{D,k}}$$  (6)

- Exergoeconomic factor
  $$f_k = \frac{\dot{Z}_k^{CI}}{\dot{Z}_k^{CI} + \dot{C}_{D,k}} = \frac{\dot{Z}_k^{CI}}{\dot{Z}_k^{CI} + c_{F,k} \dot{E}_{D,k}}$$  (7)

### 2.3 Exergoenvironmental analysis

An exergoenvironmental analysis is an exergy-based method that identifies and calculates the location, magnitude, causes and environmental impact of thermodynamic inefficiencies in an energy conversion system [4,5]. An exergoenvironmental analysis is also conducted at the component level of a system and identifies (a) the relative importance of each component with respect to environmental impact, and (b) options for reducing the environmental impact associated with the overall system. In an exergoenvironmental analysis a one-dimensional characterization indicator is obtained using a Life Cycle Assessment (LCA). This indicator is used in a similar way as the cost is used in exergoeconomics. An index (a single number) describes the overall environmental impact associated with system components and exergy carriers. The Eco-indicator 99 [6] is an example of such an index and is used here. It should be emphasized that the evaluation of environmental impacts will always be subjective and associated with uncertainties. However, the information extracted from the exergoenvironmental analysis is very useful, and future work should also focus on reducing the arbitrariness associated with the LCA and the index used in the analysis.

The exergoenvironmental model of an energy conversion system consists of:

(a) **The environmental impact balances** written for each system component
  $$\dot{B}_{P,k} = \dot{B}_{F,k} + (\dot{Y}_k + \dot{B}_k^{PR})$$  (8a)
or

\[ b_{P,k} \dot{E}_{P,k} = b_{F,k} \dot{E}_{F,k} + (\dot{Y}_k + \dot{B}_{k}^{PF}). \]  

(8b)

Here \( \dot{Y}_k \) is associated with the component-related (manufacturing, operation and retirement) environmental impact [5]; \( \dot{B}_{P,k} \) and \( \dot{B}_{F,k} \) are the environmental impact rates associated with product and fuel respectively, and \( b_{P,k} \) and \( b_{F,k} \) are the corresponding environmental impacts per unit of exergy for product and fuel [5]. To separately account for pollutant formation within the \( k \)th component during system operation, a new variable was introduced \( \dot{B}_{k}^{PF} \) [7]. This term \( \dot{B}_{k}^{PF} \) is zero if no pollutants are formed within a process, i.e. for processes without a chemical reaction (compression, expansion, heat transfer, etc.). For components, where chemical reactions occur (for example, combustion), the rule on how to calculate the value of \( \dot{B}_{k}^{PF} \) is described in detail in [7].

(b) Auxiliary environmental impact equations based on the P-rule and the F-rule, which are applied in analogy to exergoeconomics [4,5].

The following exergoenvironmental variables may be used for reducing the environmental impact associated with the \( k \)-th component:

- Environmental impact rate associated with the exergy destruction within the \( k \)-th component

\[ \dot{B}_{D,k} = b_{P,k} \dot{E}_{D,k} \]  

(11)

- Relative environmental impact difference

\[ r_{b,k} = \frac{b_{P,k} - b_{F,k}}{b_{F,k}} = \frac{1 - \varepsilon_k}{\varepsilon_k} \frac{\dot{Y}_k}{\dot{B}_{D,k}} \]  

(12)

- Exergoenvironmental factor

\[ f_{b,k} = \frac{\dot{Y}_k}{Y_k^{CO} + \dot{B}_{D,k}} = \frac{\dot{Y}_k^{CO}}{Y_k^{CO} + b_{F,k} \dot{E}_{D,k}} \]  

(13)

2.4 3D Analysis

Figure 1 shows some possible relationships among exergy destruction, capital investment cost and construction-of-component-related environmental impact [8]. The effect of component size is taken into consideration in this figure by relating \( \dot{E}_{D,k} \), \( Y_k^{CI} \) and \( Y_k^{CO} \) to the product exergy rate associated with the same component at the given operation conditions (\( \dot{E}_{P,k} \)).

In Fig.1 single curves are shown for simplicity. In reality each curve should be replaced by a rather wide area representing the fact that for each value of relative exergy destruction (\( \dot{E}_{D,k} / \dot{E}_{P,k} \)), both the \( Y_k^{CI} / \dot{E}_{P,k} \), and the \( Y_k^{CO} / \dot{E}_{P,k} \) values can vary within a rather wide range.

The values of \( \dot{Y}_k^{CO} / \dot{E}_{P,k} \) shown in the lower left part of each plot (quarter II) in Fig.1 could have different shapes since some design changes might correspond to entirely different materials and/or
manufacturing methods being used for the construction of component $k$, and, thus, to various curves for the environmental impact. Until now the character of this curve has not been studied, therefore the four curves (a-d) shown here in quarters II are just some examples of possible options. The resulting functions given in the upper right part of each plot (quarter III) are of particular importance for the simultaneous reduction of investment cost and environmental impact.

In this paper, and for the first time, we study the relationship among three variables: $\dot{E}_{D_k}/\dot{E}_{P_k}$, $\dot{Z}^{Cl}_{k}/\dot{E}_{P_k}$, and $\dot{Y}^{CO}_{k}/\dot{E}_{P_k}$ using a particular example (a combined-cycle power plant).

![Figure 1: Expected relationships among capital investment, construction-of-component-related environmental impact and exergy destruction for the $k$th component of an energy conversion system](image)

3. Study Case

The power plant studied in this paper is a three-pressure-level combined-cycle plant with one reheat stage. The plant has one product – electricity – and works with natural gas that was assumed here to be pure methane. The configuration of the process is shown in Figure 2. The thermodynamic variables for selected streams of the plant are shown in Table 2. The total exergy, $\dot{E}_{tot,j}$, includes both the chemical and physical exergy of each material stream $j$.

High-temperature flue gas with a mass flow rate of 628 kg/s exits the plant’s gas turbine (GT) and is led to the heat recovery steam generator (HRSG), where it provides thermal energy to produce steam at three different pressure levels, 124, 22, and 4.1 bar. The combustion products enter the HRSG with a pressure of 1.058 bar at 580°C and are exhausted to the atmosphere at 95°C. The
high-pressure steam at 560°C is expanded to 23 bar in the high-pressure steam turbine (HPST) and returns to the HRSG, where it is reheated to 560°C. The reheated steam is sent to the intermediate-pressure steam turbine (IPST), where it is expanded to 4.1 bar. This low-pressure steam is mixed with low-pressure superheated steam and it is then led to the low-pressure steam turbine (LPST), where it is expanded to 0.05 bar. The steam is condensed in the condenser, preheated, led to the deaerator of the plant and further conveyed to the feedwater pumps to continue the cycle.

Results from the exergetic, exergoeconomic and exergoenvironmental analyses of this combined-cycle power plant have been reported in several publications (for example, [9]), where the performance of the complex combined cycle power plant including different systems for CO₂ capture has been discussed. Some data obtained from the exergetic, exergoeconomic and exergoenvironmental analyses for selected components of the power plant (Figure 2) are presented in Table 2. For the base case, the cost of electricity is equal to 7.39 €/kWh, whereas the environmental impact associated with electricity production is equal to 14.69 mPts/kWh.

From the economic point of view, we need to find a way to reduce the total cost associated with the component, i.e. the sum \(Z_k + \ell_k\); from the environmental viewpoint we want to reduce the total environmental impact associated with the component, i.e. the sum \(Y_k + B_k\). The main final goal for the improvements is to decrease the cost of the final product, electricity. Afterwards, the effect of the economics-based modifications on the environmental impact of the final product should also be investigated. Taking into account these modifications, an exergoenvironmental analysis has been carried out to obtain the total environmental impact of the produced electricity.

**Table 2: Calculated thermodynamic variables for selected material streams [9].**

<table>
<thead>
<tr>
<th>Stream, (j)</th>
<th>(\dot{m}_j) [kg/s]</th>
<th>(T_j) [°C]</th>
<th>(p_j) [bar]</th>
<th>(\dot{E}_{\text{tot},j}) [MW]</th>
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<tr>
<td>1</td>
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<th>Stream, (j)</th>
<th>(\dot{m}_j) [kg/s]</th>
<th>(T_j) [°C]</th>
<th>(p_j) [bar]</th>
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Table 3: Some data obtained from the exergetic, exergoeconomic and exergoenvironmental analyses for the selected components of the power plant shown in Figure 2.

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<th>Component</th>
<th>Exergetic analysis</th>
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<th>Exergoenvironmental analysis</th>
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<td>$\dot{C}_{D,k}$ [€/h]</td>
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4. Sensitivity Analysis

The improvement suggestions can be divided into two groups:

- Design changes that lead to an increase in the exergetic efficiency of the components (to a decrease in the exergy destruction), i.e., to a decrease in the values of $\dot{C}_{D,k}$ and $B_{D,k}$, or

- Design changes that lead to a decrease in the values of $\dot{Z}_k$ and $\dot{Y}_k$ by decreasing the exergetic efficiency and increasing the exergy destruction.

The following assumptions for the sensitivity analysis were used.
4.1 Combustion Chamber
From the viewpoint of all three analyses, the combustion chamber can be improved by increasing its exergetic efficiency. In order to achieve this goal, the following effects have been studied:

- Effect of excess air (Figure 3a). For the sensitivity analysis, we assumed $\lambda=2.1; 2.05$ (Base Case), 1.95 and 1.9.
- Effect of the fuel inlet temperature (Figure 3b). For the sensitivity analysis, we assumed for $T_4$ the values 15°C (Base Case), 50°C, 100°C and 150°C.

4.2 Gas Turbine
For the sensitivity analysis, we assumed that the isentropic efficiency of the gas turbine remains constant and an improvement of this component is possible only through modification of the operation conditions, for example, through a change in the temperature of the combustion gases entering the gas turbine. For the sensitivity analysis (Figure 4) we assumed for $T_5$ the values 1264°C (Base Case), 1300°C, 1324°C and 1350°C.

4.3 Compressor
The two effects were considered for the improvement of the compressor:

- Effect of the isentropic efficiency (Figure 5a). For the sensitivity analysis, we assumed $\eta_{CM}=0.91, 0.915$ (Base Case), 0.92 and 0.925.
- Effect of the pressure ratio (Figure 5b). For the sensitivity analysis we assumed $p_2/p_1=16/1.013, 17/1.013$ (Base Case), 18/1.013 and 19/1.013.

4.4 High-pressure superheater
The modifications can be conducted by changing the temperature difference between hot inlet and cold outlet streams (Figure 6). For the sensitivity analysis we assumed $(T_9-T_43)=15K, 20K$ (Base Case), 25K and 30K.

4.5 High-pressure evaporator
The modifications can be done by changing the pinch temperature. For the sensitivity analysis (Figure 7) we assumed $\Delta T_{pinch}=9K, 10K$ (Base Case), 15K and 20K.

4.6 Low-pressure steam turbine
The following two options were considered for the improvement of the low-pressure steam turbine:

- Effect of the isentropic efficiency (Figure 8a). For the sensitivity analysis we assumed $\eta_{CM}=0.85, 0.88$ (Base Case), 0.90, 0.91 and 0.915.
- Effect of the pressure at the outlet from the turbine (Figure 8b). For the sensitivity analysis we assumed $p_2/p_1=125$ bar, 130.5 bar (Base Case), 135 bar and 140 bar.

4.7 Overall system
The effects of the improvement of the selected components to the overall cost and the environmental impact of the electricity are shown in Figures 9 and 10. For this analysis we selected only variables that have a positive effect to the reduction of the cost and the environmental impact of electricity.
Figure 3: Combustion Chamber: (a) Effect of the excess air, and (b) effect of the inlet temperature of the fuel.

Figure 4: Gas turbine (Effect of the temperature at the inlet).

Figure 5: Compressor: (a) Effect of the isentropic efficiency, and (b) effect of the pressure ratio.

Figure 6: High-pressure superheater (Effect of the minimum temperature difference between hot stream inlet and cold stream outlet).

Figure 7: High-pressure evaporator (effect of the pinch temperature difference).
Figure 8: Low-pressure steam turbine: (a) Effect of the isentropic efficiency, and (b) effect of the pressure at the outlet from the turbine.

5. Results and Discussion

The results of the sensitivity analysis (Figures 3 through 8) for the selected components of the analyzed combined-cycle power plant show, that for the assumptions made, the results obtained from the exergoeconomic and the exergoenvironmental analyses are qualitatively the same. We have two types of curves:

- Rising curves - for the combustion chamber (Figures 3a and 3b), compressor (Figure 5b), and low-pressure steam turbine (Figure 8a), and
- Falling curves – for the gas turbine (Figure 4), compressor (Figure 5a), High-pressure superheater (Figure 6), high-pressure evaporator (Figure 7), and low-pressure steam turbine (Figure 8b).

With respect to the character of the resulting curves among the three variables: $\frac{\dot{E}_{D,k}}{\dot{E}_{P,k}}$, $\frac{\dot{Z}_{k}^{CI}}{\dot{E}_{P,k}}$, and $\frac{\dot{Y}_{k}^{CO}}{\dot{E}_{P,k}}$ (Figure 1), the case of a rising curve corresponds to Type I (Figure 1a) and the case of a falling curve corresponds to Type II (Figure 1b).

If during the variation of a given process variable we obtain a Type I curve, then optimization is not required, because the lower the value of $\frac{\dot{E}_{D,k}}{\dot{E}_{P,k}}$ (i.e., of the thermodynamic inefficiencies), the lower the values of $\frac{\dot{Z}_{k}^{CI}}{\dot{E}_{P,k}}$ and/or $\frac{\dot{Y}_{k}^{CO}}{\dot{E}_{P,k}}$ (i.e., of the investment costs and/or of the component-related environmental impact. It is apparent that in this case we would select the most efficient option that happens to exhibit also the lowest investment cost and/or the lowest environmental impact. Thus, for example, for the combustion chamber and according to the assumptions made in this paper, we would select the lowest possible amount of excess air and the highest possible preheating temperature of the fuel (see Figures 3a and 3b). Along the same line for the compressor we would select the highest possible pressure ratio, and for the low-pressure steam turbine we would select the highest possible isentropic efficiency.

If during the variation of a given process variable we obtain a Type II curve, then optimization is necessary, because the higher the value of $\frac{\dot{E}_{D,k}}{\dot{E}_{P,k}}$ the lower the values of $\frac{\dot{Z}_{k}^{CI}}{\dot{E}_{P,k}}$ and/or $\frac{\dot{Y}_{k}^{CO}}{\dot{E}_{P,k}}$. Thus, an optimal value needs to be determined, for example, for the temperature of the
Figure 9: Specific cost of the generated electricity, $c_{P,\text{tot}}$ [€/kWh]. The effects of improving selected components on the overall cost (Base Case 7.39 €/kWh).

Figure 10: Specific environmental impact of the generated electricity, $b_{P,\text{tot}}$ [mPts/kWh]. The effects of improving selected components on the overall environmental impact (Base Case 14.69 mPts/kWh).
combustion gases at the inlet of the gas turbine, for the isentropic efficiency of the compressor, for
the minimum temperature difference between hot inlet and cold outlet streams in the high-pressure
superheater, the pinch temperature difference for the high-pressure evaporator, and for the inlet
pressure in the low-pressure steam turbine.

Figures 9 and 10 show how the considered options for improving the components of the combined-
cycle power plant affect the cost and the environmental impact of the generated electricity.
Significant improvements are achieved when improving the performance of the combustion
chamber and the turbomachinery.

6. Conclusions

A complex energy conversion system (a three-pressure-level combined cycle power plant) was used
in this paper to study the interdependencies among costs, environmental impacts and
thermodynamic inefficiencies. Exergoeconomic and exergoenvironmental analyses and
considerations have been applied. The data obtained from these analyses under various plant
operating conditions have been considered simultaneously.

The results demonstrate that, for the three-pressure-level combined cycle power plant,
improvements in efficiency result - in most cases - in the decreases of both costs and
environmental impact. However, the trends of the functions $\dot{Z}_k/\dot{E}_{P,k}$ and $\dot{B}_{D,k}/\dot{E}_{P,k}$ are not always
similar. The analysis presented here suggests ways for improving a three-pressure-level combined
cycle power plant simultaneously from the thermodynamic, economic and ecological viewpoints.
Future work should focus on more detailed analysis such as the examination of the relationship
between the three functions: $\dot{E}_{D,k}/\dot{E}_{P,k}$, $(\dot{Z}_k + \dot{C}_{D,k})/\dot{E}_{P,k}$ (i.e., consideration of the total cost
associated with the component) and $(\dot{Y}_k + \dot{B}_{D,k})/\dot{E}_{P,k}$ (i.e. consideration of the total environmental
impact associated with the component).

Nomenclature

$B$ environmental impact associated with an exergy stream, Points

$b$ environmental impact per unit of exergy, Points/J

$C$ cost associated with an exergy stream, €

$c$ cost per unit of exergy, €/J

$E$ exergy, J

$e$ specific exergy, J/kg

$f$ exergoeconomic factor

$f_e$ exergoenvironmental factor

$k$ k th component

$m$ mass, kg

$p$ pressure, bar

$r$ relative cost difference

$r_e$ relative environmental impact difference

$T$ temperature [K]

$Y$ construction-of-component-related environmental impact, Points
Z  cost associated with investment expenditures, €

**Greek symbols**

ε  exergetic efficiency

η  isentropic efficiency

λ  stoichiometric amount of air

**Subscripts**

D  refers to exergy destruction

F  fuel

P  product

tot  refers to the total system

**References**


Thermo-Ecological Evaluation of Biomass Integrated Gasification Gas Turbine Based Cogeneration Technology

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Abstract:
Biomass integrated gasification combined cycle cogeneration is nowadays considered as one of the most attractive technology for CO\(_2\) emission reduction and non-renewable fuel savings. Different criterions such as primary energy savings or emission reduction are usually taken into account in order to evaluate potential energy and environmental benefits resulting from the technology. On the other hand investment decision is in most cases based on financial profitability of a project. Nowadays the biomass energy conversion plants, especially the integrated gasification ones, cannot compete effectively with fossil fuel fired technologies without an effective financial support. Therefore in many countries, in order to satisfy political priorities, there have been established supportive mechanisms that are based on different forms of financial subsidies. The subsidies are usually allocated between projects according to the amount of electricity generated, therefore promoting rather power generation than its efficiency or the influence on the depletion of non-renewable resources. In the face of growing scarcity of non-renewable natural resources, it seems to be reasonable that additional criterion is applied to evaluate biomass conversion technologies. The Thermo-Ecological Cost (TEC), which expresses the cumulative consumption of non-renewable exergy, seems to be an appropriate indicator in this matter. Moreover, to express the total effect of considered energy conversion systems the TEC should be supplemented with the data resulting from Life Cycle Analysis (LCA). In this paper the proposed methodology has been applied to the analysis of a gas turbine based cogeneration plant integrated with gasification of biomass. There are investigated different gasification technologies and configurations of CHP plant. There are taken into account atmospheric fluidized bed gasification (AFB), pressurized fluidized bed gasification (PFB) and allothermal gasification using pure steam as gasification agent (FICFB) as well as simple and combined power cycles based on a recuperated gas turbine. The performance of the plant has been investigated using the combined model developed using Engineering Equation Solver and GateCycle software. The results reveal that simple cycle with gas turbine and waste heat recovery water boiler offers better effects than combined cycle configuration. The best performance has been reported for pressurized gasification technology.

Keywords:

1. Introduction

Depletion of natural non-renewable resources is accelerated by continuously increasing energy consumption. From the economic point of view the increase of consumption is a base for further development [1] of societies. On the other hand, it provides an ecological threat to the existence of future generations. Environmental risks associated with the growth of consumption can be divided into two groups: depletion of non-renewable natural resources and discharge of harmful waste products into the environment. Damages caused by the waste products can also be expressed through the impact on the depletion of non-renewable natural resources as the losses should be compensated or prevented. In order to reduce the consumption of non-renewable natural resources, the use of renewable ones should increase. The replacement of non-renewable fuels by biomass is an option. However, the effects of biomass utilisation in the energy sector should be evaluated using system analysis, not pure direct effects appearing at the boundary of a conversion plant. Moreover,
it is highly recommended to take into account beside the classical economic criterion the one based on physical laws in non-renewable management. The global effects of natural resources management can be investigated applying the methods of cumulative calculus and Life Cycle Assessment (LCA) [2,3]. The calculation of the cumulative coefficients was initiated by Chapman, who introduced the concept of energy cost [4-6]. The theory of the energy cost of useful products was developed by Boustead and Hancock [7]. Szargut introduced the important concept of cumulative exergy consumption (CExC) [8,9], and then based on CExC the Thermo-Ecological Cost (TEC), which extends the applications of exergy analysis onto the field of environmental considerations. The TEC expresses the cumulative consumption of non-renewable exergy of natural resources [10,11]. The Szargut's method in comparison with other methods of ecological assessment can bring all environmental impacts to one measure which is the exergy. The minimization of the TEC [13,14] ensures a mitigation of the depletion of non-renewable resources. Such optimisation can be a base of ecological economy that is in line with the concept of sustainable development.

The original Szargut's TEC analysis concerns only the operational phase of a production plant. In order to evaluate a global (or total) impact of a considered production process the method should be supplemented with tools offered by the LCA. Nowadays the problem of emissions becomes a very important one. Therefore Szargut proposed adaptation of the TEC algorithm for the calculation of the cumulative emissions. In the case of green house gasses (GHG) he proposed a new term that is the Thermo-Climatic Cost (TCC) [14,15]. The combination of the TEC, LCA and TCC methods can be a comprehensive tool for sustainable management of non-renewable resources. In this paper there is proposed the algorithm of thermo-ecological evaluation based on TEC, TCC and LCA principles for environmental evaluation of biomass to energy conversion technology.

2. Exergetic cost of natural resources

Szargut [15] defined the TEC as the cumulative consumption of non-renewable exergy connected with the fabrication of a particular product, including also an additional consumption, that results from the necessity of compensation of environmental losses caused by rejection of harmful substances into the environment. The index of operational TEC can be determined by solving a set of balance equations. The equations are formulated using the scheme presented in Fig. 1.

The physical cost of any product expressed by the TEC is mainly the consumption of chemical exergy \( b_s \) extracted directly from the nature as fuels, mineral ores, nuclear ores or fresh water. However this consumption appears only in the production processes directly connected with an extraction of a substance from the natal deposits, e.g. in coal mine. Not all branches of economy are directly connected to the nature. However, due to interconnections in the production systems, the TEC is generated also by consumption of domestic semi-finished products \( a_{ij} \) exchanged between branches of the system. Additionally, if within the national system there are some imported and exported goods a part of TEC can result from an interregional exchange. For example, in Polish conditions 70% of natural gas is imported from outside the balance boundary of domestic economy. In this case the total TEC of a \( j \)-th product results from consumption of a \( r \)-th imported good in a \( j \)-th branch \( a_{rj} \). In some branches a by-production can appear. By-products replace main products in other branches and for this reason the value of TEC of a considered main product is reduced. In the balance presented in Fig. 1 by-products are taken into account by the coefficient of by-production \( f_{ij} \). TEC of useful by-product should be determined using the principles of avoided exergetic cost method. The balance of TEC of a \( j \)-th production branch includes also an additional consumption of resources \( \psi \) connected with rejection of wastes into the environment. This additional consumption is connected with the consumption at an abatement installation \( \psi' \), as well as it results from the necessity of compensation of losses in the environment. By the definition the exergetic TEC takes into account only the consumption of non-renewable resources.
Fig. 1. Idea of TEC balance equation

Therefore if a process driven by a renewable energy is being examined the direct consumption of chemical exergy should not be taken into account in the balances of TEC. This is because the consumption of renewable exergy, e.g. biomass, doesn’t make any deleterious impact on depletion of deposits of limited natural resources. However, it does not mean the TEC of biomass should be equal to zero. The TEC of biomass is positive and it results from cumulation of some external expenses of non-renewable natural resources appearing in the total life cycle of a conversion technology. For this reason, the original TEC algorithm should be supplemented with LCA tools and new comparative indices of technology performance should be derived. According to the balance scheme given in Fig. 1 the equations for calculation of the operational TEC take the form [10,11,15]:

\[ \rho_j + \sum_i (f_{ij} - a_{ij}) \rho_i - \sum_r a_{rj} \rho_r = \sum_s b_{sj} + \psi_{j0} \]  

(1)

The last part of the Eq.(1) includes the additional expenses of cumulative exergy of non-renewable resources arising from the formation of waste products within the \( j \)-th production process. These additional expenses \( \psi_{j0} = \psi_{j0}' + \psi_{j0}'' \) are divided between abatement of wastes and losses in environment:

\[ \psi_{j0}' = \sum_k p'_{kj} \cdot \sigma_k \]  

(2)

An abatement of a negative impact of \( k \)-th waste substance requires the consumption of additional quantities of cumulative exergy of non-renewable resources and can be expressed as follows:

\[ \psi_{j0}'' = \sum_k (p''_{kj} - p'_{kj}) \cdot \zeta_k \]  

(3)
The concept of evaluation of exergetic abatement cost has been exhaustively described by Valero and Botero in [29] and this method has been adopted within the TEC algorithms. A residual amount of harmful \( k \)-th substance \((p''_k - p'_k)\) that is not removed in an abatement installation is transferred into the environment and causes damages. The additional thermo-ecological cost arising from the environmental losses results from necessity of either compensation or repair. Determination of the exergetic cost of compensation \( \zeta_k \) is one of the most difficult task within the analysis of TEC [30]. Szargut [20] proposed the simplified method based on the knowledge of external impacts of the process using so called monetary indices of harmfulness:

\[
\zeta_k = \frac{Bw_k}{PKB + \sum P_k w_k}
\]

Practical application of this method is shown by Stanek and Czarnowska in [16,17]. The results of \( \zeta_k \) obtained using external costs \( w_k \) for main harmful gaseous substances are presented in Table 1.

### Table 1. External cost and TEC of major air pollution

<table>
<thead>
<tr>
<th>No.</th>
<th>Indicator</th>
<th>Units</th>
<th>Harmful substance</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.</td>
<td>Monetary external cost, ( w_k )</td>
<td>€2008/kg</td>
<td>SO(_2) 12.81 NO(_X) 9.41 PM 7.00</td>
</tr>
<tr>
<td>2.</td>
<td>Thermo-Ecological Cost, ( \zeta_k )</td>
<td>MJ(_{ex})/kg</td>
<td>SO(_2) 97.82 NO(_X) 71.88 PM 53.42</td>
</tr>
</tbody>
</table>

The exergetic abatement cost \( \sigma_k \) can be determined on the basis of mass and energy balances for abatement technology. In the case of CO\(_2\) removal by MEA the obtained value of abatement cost [19] is \( \sigma_{CO2} = 4.4 \) MJ/kg.

### 3. Thermo-Climatic Cost

The methodology for determination of depletion of non-renewable resources (Eq. 1) can be also used for investigation of cumulative emissions. It seems to be an appropriate approach in the case of GHG emissions, because these gases are supposed to be responsible for global warming effect. Therefore the TCC has been defined as the cumulative emission of the GHG. In order to modify the TEC balance for the analysis of GHG cumulative emissions, first of all the specific consumption of exergy \( b_s \) should be replaced by the total direct emission \( e_{GHG,j} \) of GHG from a considered process. In the energy conversion processes the direct emission consists mainly of:

- anthropogenic CO\(_2\) emission from combustion processes \( e_{CO2} \),
- methane emission in the processes of fossil fuels delivery and processing \( e_{CH4} \).

The direct anthropogenic CO\(_2\) emission is closely dependent on the energy efficiency of the transformation of chemical energy of fuels. Direct emission of carbon dioxide can be evaluated assuming complete combustion of fuels. Basing on the law of conservation the specific emission of CO\(_2\) related to the chemical energy of a fuel can be calculated using simple stoichiometric formula:

\[
e_{CO2} = \frac{M_{CO2}}{M_c} \cdot \frac{c}{LHV} = 3.667 \cdot \frac{c}{LHV} \cdot \frac{kg \ CO_2}{MJ}
\]

The calculated direct emission of CO\(_2\) associated with combustion of primary fuels is shown in Table 2. It is important to emphasis the fact that wrong conclusions can be drawn if the impact of different fuels on global warming is analysed using the direct emission indicators. Such approach shows that combustion of coal gives almost two times higher emission than combustion of natural gas. This results from the fact that the additional external emissions are not taken into account. To
have a complete picture of the influence of a given technology on the GHG emission, the emission in all links of the process of energy conversion should be considered. A solution to this problem is to apply cumulative emission calculus [14,18] and LCA [13,19,20]. The approach allows for the inclusion of greenhouse gases in the entire cycle of investment and operation of a considered production system e.g. power plant.

The external emissions result from consumption $a_{ij}$ of $i$-th semi-finished product in considered $j$-th branch or from by-production $f_{ij}$ of $i$-th product per unit of main $j$-th product. The direct emission is a sum of specific emission of $k$-th GHG in $j$-th branch $e_{jk}$ multiplied by its global warming potential (GWP)$_k$. Moreover, if the balance is formulated in global scale the part concerning imported goods $a_{ij}$ is not necessary. The formula of the TCC takes the form:

$$e_j^* = \sum_i (a_{ij} - f_{ij}) e_i^* + \sum_k (\text{GWP})_k e_{jk}$$

(6)

Coefficients $a_{ij}$ and $f_{ij}$ in Eq. 6 have the same meanings and take the same values like in the case of Eq. 1.

The GHG cumulative emission $e_j^*$ connected with coal and natural gas has been investigated by Stanek and Białecki [21]. Selected results concerning fuels consumed in Poland are given in Table 2 (rows 2,3 for coal and rows 5-10 for imported natural gas).

<table>
<thead>
<tr>
<th>No.</th>
<th>Variant</th>
<th>Emission, t CO$_2$/TJ</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Coal – direct emission</td>
<td>92</td>
</tr>
<tr>
<td>2</td>
<td>Coal – cumulative emission</td>
<td>96</td>
</tr>
<tr>
<td>3</td>
<td>Coal – cumulative emission with leakage of methane from coal mine</td>
<td>108</td>
</tr>
<tr>
<td>4</td>
<td>Natural gas – direct emission</td>
<td>56</td>
</tr>
<tr>
<td>5</td>
<td>Gas variant 1: 21 GWP + 0,11 % natural gas leakage form mine to consumer</td>
<td>72</td>
</tr>
<tr>
<td>6</td>
<td>Gas variant 2: 21 GWP + 3,25 % natural gas leakage form mine to consumer</td>
<td>85</td>
</tr>
<tr>
<td>7</td>
<td>Gas variant 3: 30 GWP + 0,11 % natural gas leakage form mine to consumer</td>
<td>72</td>
</tr>
<tr>
<td>8</td>
<td>Gas variant 4: 30 GWP = 1,5 % natural gas leakage form mine to consumer</td>
<td>80</td>
</tr>
<tr>
<td>9</td>
<td>Gas variant 5: 30 GWP + 3,5 % natural gas leakage form mine to consumer</td>
<td>91</td>
</tr>
<tr>
<td>10</td>
<td>Gas variant 6: 30 GWP + 4,2 % natural gas leakage form mine to consumer</td>
<td>97</td>
</tr>
</tbody>
</table>

If the utilisation of biomass is considered it is important to take into account the savings in the TCC. First of all the direct emission of CO$_2$ from biomass combustion is not taken into account because it is closed in the biomass cultivation life cycle. The only emission burdening the biomass utilisation is the external emission resulting from delivery and processing of biomass and production of fertilizers for cultivation purposes. The savings caused by replacement of fossil fuels by biomass result from the avoided cumulative emissions related to the chemical energy of replaced fuels, according to the results data given in Table 2. Two possibilities can be taken into account – replacement of coal or replacement of natural gas.

4. Life cycle exergetic cost

The Thermo Ecological Cost Life Cycle Assessment (LC-TEC) should cover the following phases:
1. **Construction Phase**, that encompasses project, extraction of raw materials, semi-finished products fabrications, transport expenditures in the construction phase. All these expenses influence the final thermo-ecological cost burdening the final useful product. This phase has a significant contribution in case the processes is based on renewable sources of energy. For instance, in case of wind power plant the thermo-ecological cost results mainly from expenses in construction phase.

2. **Operational phase**, that is defined as a period between the end of construction phase and the beginning of decommissioning phase. In processes utilising non-renewable resources, this phase is predominant in the cumulative consumption of natural resources, mainly chemical exergy of fossil fuels.

3. **Decommissioning phase** of plant concerns the period at the end of plant's life. In this phase, the thermo-ecological cost results from expenditures connected with retirement as for instance for recultivation of terrain and disposal of remains.

General form of the equation for determination of LC-TEC has been formulated by Szargut and [15]. The methodology has been used so far for Thermo-Ecological optimisation of solar collector. Results of the optimisation have been presented by Szargut and Stanek in [12,13]. The LC-TEC expresses an annual thermo-ecological cost of investigated useful product with inclusion of the whole life of this product. The formula takes the following form:

\[
\Theta_{LCA} = r_n \left( \sum_j \tilde{G}_j \rho_j + \sum_k \tilde{P}_k \tilde{\xi}_k - \sum_u \tilde{G}_u \rho_u \tilde{\eta}_u \right) + \frac{1}{\tau} \left( \sum_m G_m \rho_m (1 - u_m) + \sum_r G_r \rho_r \right)
\]

(7)

5. **Thermo-ecological indicators of environmental benefits**

The indices of operational TEC: \( \rho_i, \rho_j, \rho_m, \rho_r \), that appear in Eq. 7, are determined by means of TEC balance (Eq. 1). Equation (7) can be applied to optimize construction and operational parameters of different resources intensive systems and can be a supplementary criterion in ecological economy. After the specific emission \( e_k \) in Eq. 7 had been introduced instead of the TEC the formula for calculation of cumulative emission was obtained. The methodology of TEC, TCC and LC-TEC, briefly explained in previous sections, has been applied for the evaluation of the influence of utilisation of biomass in CHP plant on the depletion of non-renewable resources. Also the emissions on the entire life cycle have been considered.

In order to evaluate the ecological advantages of biomass utilisation for energy production three additional indicators are being proposed. The indicators are based on the results of analysis of TEC, TCC or LC-TEC. These are as follows:

- **Index of natural resources savings (NRS):**

\[
NRS = \frac{\Delta m_F \cdot (\rho_F + \sum_k p_{k,F} \cdot \zeta_k + p_{CO2} \cdot \sigma_{CO2})}{m_{bio} \cdot (\rho_{bio,LCA} + \sum_k p_{k,bio} \cdot \zeta_k)}
\]

(8)

This NRS index expresses the ratio of decrease of consumption of exergy of non-renewable resources, that results from the replacement of \( \Delta m_F \) of fossil fuel by biomass in the amount of \( m_{bio} \). This index can be interpreted as the cumulative efficiency of non-renewable exergy savings thanks to application of biomass.

- **Index of greenhouse gasses cumulative emission reduction \( \Delta(\text{GHG}) \):**
\[ \Delta(GHG) = \frac{\Delta m_F \cdot e_{GHG}^*}{m_{bio} \cdot (e_{GHG}^* - e_{CO2}^*)} \] (9)

The \( \Delta(GHG) \) is expressed as the ratio of cumulative reduction of GHG emissions resulting from decrease of consumption of fossil fuel \( \Delta m_F \) to increase of external emission of GHG \( (e_{GHG}^* - e_{CO2}^*) \) burdening the unit of biomass \( m_{bio} \). The external emissions of GHG results e.g. from transport or cultivation of biomass whereas it does not take into account the direct emissions of CO2.

- Index of natural resources savings resulting from decrease of GHG emissions \( \Delta(NRS)_{GHG} \):

\[ NRS_{GHG} = \frac{\Delta m_F \cdot e_{GHG}^* \cdot \sigma_{CO2}}{m_{bio} \cdot \rho_{bio,LCA}} \] (10)

The reduction of GHG emissions expressed by the \( \Delta(GHG) \) leads not only to the decrease of GHG emission expressed in mass units but also as a final result to the savings of natural resources. The avoided \( e_{GHG}^* \) emissions of CO2 lead to the savings of exergy of non-renewable natural resources for CO2 removal. For this reason it leads finally to the reduction of consumption of non-renewable resources in the amount \( e_{GHG}^* \cdot \sigma_{CO2} \) per unit of a fossil fuel saved. The natural resources that we have to pay for this savings is equal to the cumulative exergy consumption that has to be spend to produce unit of biomass for energetic purposes.

6. LC-TEC analysis of gasification and gas turbine based cogeneration technology for biomass utilisation

The methodology given in the previous sections has been applied to the analysis of biomass powered cogeneration plant. All stages of constructing, producing and dismantling of the plant have been considered. Additionally, the strongly connected branches have been introduced within LC-TEC procedure. The boundary of the investigated system includes biomass farming and harvesting, fertilizers production and all connections by transport. These connected processes with indication of supply of energy carriers, non-energetic materials and unavoidable emissions of harmful substances are presented in Fig. 2.

![Fig. 2. LC-TEC system boundary of CHP fed with biomass](image)
Different configurations of a cogeneration plant fed with biomass are taken into account. There are also taken into account different biomass gasification technologies. The total number of seven alternative designs of the system, that are numbered A.1 to A.3.1 (Table 3), are examined.

The example system presented in Fig. 3 is made up of fluidized bed gasifier, Mercury 50 Recuperated Gas Turbine and either a bottoming steam cycle or heat recovery water boiler. Detailed energy analysis of different configurations of the plant has been presented in [22, 23].

Within the first alternative design (named A.1) an autothermal Atmospheric Fluidized Bed (AFB) gasification technology and combined gas and steam turbine cycle are taken into account. The disadvantages of this technology are considerable gas cleaning requirements and high consumption of power for compression of large amount of the low calorific value GT fuel gas. The parameters of steam within the bottoming steam turbine cycle are relatively low due to the low temperature of GT exhaust gas. The scheme of the plant within the configuration A.1 is presented in Fig. 3.

**Fig. 3. Scheme of IBGCHP plant with atmospheric fluidised bed gasifier, gas turbine and bottoming steam cycle (design no. A.1)[22,23]**

Within the alternative configuration A.1.1 there is a supplementary firing applied. It leads to an increased electric and heating power of the plant. On the other hand however there is an increased demand for wet biomass and considerable heat consumption for drying purposes.

In another alternative design A.1.2 a simple GT cycle is coupled to a heat recovery water boiler. Within this configuration there is no bottoming steam cycle, what results in a lower investment cost.

Configurations number A.2 and A.2.1 are designed basing on pressurized fluidized bed gasification technology (PFB). There is no demand for producer gas compression as well as gas cleaning requirements are significantly reduced. In the configuration A.2 a combined cycle is taken into account while in A.2.1 a simple cycle with boiler is assumed.

In the alternative designs No. A.3 and A.3.1 there is applied the allothermal gasification technology using pure steam as gasification agent (FICF). This technology leads to much higher calorific value of the GT fuel gas. Therefore the flow of fuel gas is smaller and the power consumption for compression is reduced. Using the integrated gasification and combustion double circulating fluidized bed reactor technology helps to solve gasification waste substances related problems.

The highest level of net efficiency of electricity generation (in full cogeneration mode) is reached in alternative design A.2 (combined cycle integrated PFB gasification technology). On the other hand GT simple cycle in the alternative design A.2.1 offers the highest value of biomass energy utilization factor. These two design alternatives also lead to high values of non-renewable energy
replacement index (ERI) and specific global CO₂ emission reduction within the regional energy system (calculated per GJ of biomass energy consumed). The simple cycle configuration with waste heat recovery water boiler offers slightly better energy and emission savings.

The weakest performance of the plant is represented by the alternative design A.1.1 (AFB gasification technology integrated with combined cycle plant with supplementary firing). All indices that are taken into account have the lowest value in this case. Although the configuration A.1.1 leads to the highest electric power of the system the performance is poor due to high demand for wet biomass, significant drying requirements and relatively high fraction of electricity generated at the low efficient steam cycle.

An interesting conclusion is that the FICFB gasification technology offers the energy and environmental effects better than the AFB technology but slightly worse than PFB technology. The FICFB technology offers relatively high caloric value of the product gas. On the other hand the high demand for gas recirculation into the combustion zone of the reactor results in the net value of cold gas efficiency at the level of other gasification technologies.

Table 3. Main results of annual mass and energy balance [22]

<table>
<thead>
<tr>
<th>Gasification configuration</th>
<th>Alternative configuration</th>
</tr>
</thead>
<tbody>
<tr>
<td>Plant own electricity consumption, MWh</td>
<td>A.1</td>
</tr>
<tr>
<td>AFB</td>
<td>AFB</td>
</tr>
<tr>
<td>Nett electricity to grid, MWh</td>
<td>38 783</td>
</tr>
<tr>
<td>Network heat, GJ</td>
<td>127 352</td>
</tr>
<tr>
<td>Network heat from cogeneration, GJ</td>
<td>110 436</td>
</tr>
<tr>
<td>Electricity from cogeneration, MWh</td>
<td>23 380</td>
</tr>
<tr>
<td>Wet biomass consumed m_{bio}, tons</td>
<td>44 278</td>
</tr>
<tr>
<td>Biomass energy consumed, GJ</td>
<td>498 635</td>
</tr>
<tr>
<td>Saved coal Δm_{c}, tons</td>
<td>6 896</td>
</tr>
<tr>
<td>CO₂ emission reduced Δm_{F eCO₂}, tons</td>
<td>52 092</td>
</tr>
<tr>
<td>Non-renewable energy saved within the regional energy system (loco generation plants), GJ</td>
<td>543 134</td>
</tr>
</tbody>
</table>

Investigation of LC-TEC and evaluation of indices defined by the authors (Eqs. 8 – 10) requires the initial calculations of TEC of fertilizers, TEC of biomass cultivation and TEC of biomass transport. The TEC of nitrogen fertilizer is calculated using data from Polish big fertilizers factory [24]. Ammonia takes part of the reactions for forming the N-fertilizer. The ammonia production process of producing very energy consuming and for this reason the TEC of N-fertilizer is high. According to [25,26] the TEC of phosphorus fertilizer, potassium fertilizer and lime is established for further calculations. Table 4 presents the LC-TEC of fertilizers production using the balance of TEC expressed by Eq. 1.

Table 4. Unit TEC of fertilizers production and CO₂ direct emissions

<table>
<thead>
<tr>
<th>Useful product</th>
<th>TEC MJ/kg fertilizer</th>
<th>CO₂ direct emissions kg emission/kg fertilizer</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nitrogen fertilizer</td>
<td>49.15</td>
<td>2.6410</td>
</tr>
<tr>
<td>Phosphorus fertilizer</td>
<td>15.11</td>
<td>0.6735</td>
</tr>
<tr>
<td>Potassium fertilizer</td>
<td>10.39</td>
<td>0.4656</td>
</tr>
<tr>
<td>Lime production</td>
<td>1.32</td>
<td>0.0079</td>
</tr>
</tbody>
</table>
According to [27] the demand of fertilizers for biomass production is presented in Table 5.

**Table 5. Fertilizer requirements for production of different biomass**

<table>
<thead>
<tr>
<th>Fertilizer</th>
<th>Units</th>
<th>Bundles, short-rotation wood</th>
<th>Miscanthus bales</th>
<th>Wheat straw, bales</th>
</tr>
</thead>
<tbody>
<tr>
<td>N-fertilizer</td>
<td>kg$<em>{fertilizer}$/kg$</em>{DB}$</td>
<td>0.0052</td>
<td>0.0040</td>
<td>0.0022</td>
</tr>
<tr>
<td>P2O5-fertilizer</td>
<td>kg$<em>{fertilizer}$/kg$</em>{DB}$</td>
<td>0.0040</td>
<td>0.0031</td>
<td>0.0011</td>
</tr>
<tr>
<td>K2O-fertilizer</td>
<td>kg$<em>{fertilizer}$/kg$</em>{DB}$</td>
<td>0.0064</td>
<td>0.0051</td>
<td>0.0009</td>
</tr>
<tr>
<td>Lime</td>
<td>kg$<em>{fertilizer}$/kg$</em>{DB}$</td>
<td>0.0065</td>
<td>0.0036</td>
<td>0.0044</td>
</tr>
</tbody>
</table>

The indices of fertilizes specific consumption included in Table 5 and the TEC of fertilizers (Table 4) are the basic information required for LC-TEC of biomass calculation. Additionally according to Fig. 3 the TEC resulting from transport has to be taken into account. The authors assumed that biomass is transported by Lorries. The TEC of lorry production is presented in Table 6. The values are calculated using Life Cycle Inventory [28] and TEC of useful products [19].

**Table 6. Unit TEC and CO$_2$ direct emission burdening car transport**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Units</th>
<th>Lorry</th>
<th>Lorry</th>
<th>Lorry</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Load capacity Mg</td>
<td>16</td>
<td>28</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Useful truck life years</td>
<td>15</td>
<td>15</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Annual mileage km</td>
<td>50 000</td>
<td>50 000</td>
</tr>
<tr>
<td></td>
<td></td>
<td>CO$_2$ direct emissions kg</td>
<td>633</td>
<td>633</td>
</tr>
<tr>
<td>Results of TEC calculations</td>
<td></td>
<td>TEC MJ$_{ex}$/km</td>
<td>0.603</td>
<td>0.863</td>
</tr>
<tr>
<td></td>
<td></td>
<td>TEC MJ$_{ex}$/(km&amp;Mg)</td>
<td>0.038</td>
<td>0.054</td>
</tr>
</tbody>
</table>

The TEC of biomass cultivation is presented in Table 7. It was calculated taking into account fertilizer production (Table 4), fertilizer transportation (Table 6) and usage of this fertilizer for growing three different crops (Table 5).

**Table 7. TEC and CO$_2$ emission of biomass cultivation**

<table>
<thead>
<tr>
<th>Units</th>
<th>Bundles, short rotation wood</th>
<th>Miscanthus bales</th>
<th>Wheat straw, bales</th>
</tr>
</thead>
<tbody>
<tr>
<td>MJ$<em>{ex}$/kg$</em>{DB}$</td>
<td>0.3965</td>
<td>0.3949</td>
<td>0.3932</td>
</tr>
<tr>
<td>kgCO$<em>2$/Mg$</em>{DB}$</td>
<td>20.04</td>
<td>15.52</td>
<td>6.99</td>
</tr>
</tbody>
</table>

For the process of biomass cultivation the TEC is on very similar level for different types of biomass (Table 7). The TEC index for example non-renewable fuel as coal can be on the level 27.0 MJ/kg and results mainly from chemical exergy of coal (about 90%). One can easily see that the (TEC) burdening the cultivation of bio-fuels is significantly lower than TEC resulting from chemical exergy of non-renewable fuels. TEC of biomass besides cultivation is influenced by the needs for transport. First of all the transportation needs can be higher than in the case of fossil fuels because of relatively low energy density of biomass (esp. wet biomass). Moreover, TEC of biomass can be dependent on the distance of transportation. Table 8 presents the results of biomass resulting from transport taking into account the distances between 25 and 100 km.
Table 8 TEC of biomass transportation as a function of distance

<table>
<thead>
<tr>
<th>Distance (km)</th>
<th>TEC (MJ/kg&lt;sub&gt;DB&lt;/sub&gt;)</th>
<th>CO&lt;sub&gt;2&lt;/sub&gt; emissions (kg&lt;sub&gt;CO2&lt;/sub&gt;/Mg&lt;sub&gt;DB&lt;/sub&gt;)</th>
</tr>
</thead>
<tbody>
<tr>
<td>25</td>
<td>0.031</td>
<td>1.838</td>
</tr>
<tr>
<td>50</td>
<td>0.063</td>
<td>3.676</td>
</tr>
<tr>
<td>75</td>
<td>0.094</td>
<td>5.514</td>
</tr>
<tr>
<td>100</td>
<td>0.125</td>
<td>7.353</td>
</tr>
</tbody>
</table>

Similarly as in the case of the TEC resulting from cultivation also the part burdening the transport of biomass (Table 8) is significantly lower than TEC index burdening non-renewable fuels. Even if we consider the transportation distance of about 100 km the TEC on the level of 0.125 MJ/kg<sub>DB</sub> is significantly lower than for instance chemical exergy of coal which can be about 24 MJ/kg.

TEC burdening the construction of the system has to be also introduced into the modified LC-TEC algorithm proposed in this work. Table 9 includes the investment part of LC-TEC of different power plants. This investment part is divided between construction and dismantling. The second one can decrease the total LC-TEC if the factor of recovery of m-th material appearing in Eq. 8 \( u_m > 0 \).

Table 9. TEC of power plants

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Units</th>
<th>Hard Coal</th>
<th>Hard Coal</th>
<th>Natural Gas/Sng</th>
<th>Lignite</th>
<th>Hard Coal/ Wood Co-Firing</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power</td>
<td>MW</td>
<td>400</td>
<td>800</td>
<td>950</td>
<td>950</td>
<td>400</td>
</tr>
<tr>
<td>TEC construction</td>
<td>MJ&lt;sub&gt;ex&lt;/sub&gt;/MJ&lt;sub&gt;el&lt;/sub&gt;</td>
<td>0.65</td>
<td>0.58</td>
<td>0.29</td>
<td>0.88</td>
<td>0.87</td>
</tr>
<tr>
<td>TEC dismantling</td>
<td>MJ&lt;sub&gt;ex&lt;/sub&gt;/MJ&lt;sub&gt;el&lt;/sub&gt;</td>
<td>0.0020</td>
<td>0.0018</td>
<td>0.0015</td>
<td>0.0075</td>
<td>0.0043</td>
</tr>
</tbody>
</table>

Basing on the results enclosed in table 9 it can be concluded that for two systems taken into account as a power technologies prepared for energetic conversion of biomass, the net TEC index resulting from construction decreased by TEC of recycled materials is as follows:
- synthetic natural gas power plant TEC = 0.288 MJ<sub>ex</sub>/MJ<sub>el</sub>,
- hard coal / wood cofiring power plant TEC = 0.866 MJ<sub>ex</sub>/MJ<sub>el</sub>.

Also in this case the TEC is relatively low, but not negligible. For example the TEC burdening generation of electricity in Polish conditions is on the level of about 3.6 MJ/MJ. The constructional net TEC mentioned above represents about 8% in the case of natural gas power, 14% in the case of hard coal power plant and 17% in the case of lignite power plant. Analysing the obtained results (Table 7 – 9) one can also conclude that the investment part can be the most important in (LC-TEC) of biomass. Nevertheless, the total (LC-TEC) is significantly lower in the case of biomass than that of non-renewable fed energy system. Such presumptions are confirmed by obtained results that are presented in Table 10.

Table 10. Greenhouse gas avoidance and natural resources avoidance in comparison to coal

<table>
<thead>
<tr>
<th>Gasification system</th>
<th>A.1</th>
<th>A.1.1</th>
<th>A.1.2</th>
<th>A.2</th>
<th>A.2.1</th>
<th>A.3</th>
<th>A.3.1</th>
</tr>
</thead>
<tbody>
<tr>
<td>NRS</td>
<td>6.0</td>
<td>8.7</td>
<td>10.1</td>
<td>7.5</td>
<td>10.7</td>
<td>7.8</td>
<td>10.5</td>
</tr>
<tr>
<td>( \Delta \text{GHG} )</td>
<td>22.39</td>
<td>27.04</td>
<td>31.43</td>
<td>23.30</td>
<td>33.26</td>
<td>24.18</td>
<td>32.60</td>
</tr>
<tr>
<td>( NRS_{\text{GHG}} )</td>
<td>4.93</td>
<td>5.96</td>
<td>6.92</td>
<td>5.13</td>
<td>7.32</td>
<td>5.33</td>
<td>7.18</td>
</tr>
</tbody>
</table>

It should indicate that the proposed indices of decreasing the GHG emissions (Eq. 9) and of natural resources savings (Eq. 8 and 10) should confirm high profitability of biomass utilisation from the
point of view of greenhouse effects as well as from the point of view of non-renewable resources savings.

7. Summary and conclusions

The methodology of thermo-ecological evaluation of biomass utilization in energy production sector has been presented in the paper. The performance indices of an energy conversion technology have been introduced taking into account reduction of the GHG emissions as well as the savings of non-renewable natural resources.

The comparison of the proposed cogeneration plant technologies in the aspect of LC-TEC revealed that the best environmental performance is obtained for the alternative design no. A.2.1. (Table 10). The previous study [22], where the economic effectiveness of an investment project was examined, showed that the best values of Net Present Value (NPV) and Internal Rate of Return (IRR) were obtained in the case of the variant A.1.2. However, the difference in profitability between A.1.2. and A.2.1 was not significant. The general conclusion from present and previous works is that in the case of the gas turbine based biomass-to-energy conversion technology in relatively small-scale of the plant the simple cycle is more effective than the combined one. On the other hand in Europe the combined cycle configurations have been tested so far in ARBRE and Värnamo projects [22].

The presented results confirm the high ecological profitability of biomass conversions. In the case of savings of natural non-renewable resources the TEC of biomass plant is 6.0 to 10.7 times lower than that in a plant fed with non-renewable chemical energy. It leads to the obvious conclusion that the share of biomass power plants and CHP plants should be maximised within the area of economic profitability. In other words the results of optimisation are only economically constrained. In the Polish conditions the economic profitability of renewable power plants strongly depends on subsidies (green certificates). But in the practice this system does not take into account that different renewable systems have different influence on depletion or savings in non-renewable resources, as has been widely demonstrated by the authors. The results presented in table 10 of this paper clearly show such influence. Moreover the overestimated prices of certificates can lead to the competition in agriculture between production of food and production of “food for fuels”. For this reason the proposed LC-TEC algorithm can be also used for division of green subsidies between renewable power plants basing on objective criterions as is the NRS – non-renewable resources savings. The same algorithm can be used for other renewable power systems e.g. wind turbines or photo-voltaic. One of the most important issues in the application of biomass energy is GHG emission. The index Α(GHG) is between 4.9 and 7.3. It is scientifically confirmed that the problem is global but the technology selection based on single economic criterion of profitability maximisation limits this problem often to local scale. Frankly speaking the biomass is transported for thousand kilometres and these external emission are not taken into account in the system of subsidies because the external expenditures that have been discussed in this paper are intentionally neglected.

Nomenclature

\begin{align*}
  a_{ij} & \text{ coefficient of the consumption of the } i\text{-th product per unit of the } j\text{-th major product, e.g in kg/kg or kg/MJ}, \\
  a_{rj} & \text{ coefficient of the consumption of the } r\text{-th imported product per unit of the } j\text{-th major product, e.g in kg/kg or kg/MJ}, \\
  B & \text{ exergy extracted per year from the domestic non-renewable natural resources}, \\
  b_{sj} & \text{ exergy of the } s\text{-th non-renewable natural resource immediately consumed in the process under consideration per unit of the } j\text{-th product, MJ/kg}, \\
  c & \text{ mass fraction of carbon element in the fuel in kg/kg or in kg/kmol};
\end{align*}
$e^{GHG}$ index of cumulative GHG emissions,  
$e^{i}$ coefficient of cumulative emission of greenhouse gases burdening $i$-th useful product,  
$e^{j}$ cumulative emission of greenhouse gases in $j$-th production branch,  
$e^{CO2}$ index of CO$_2$ direct emission.  
$e^{jk}$ coefficient of direct emission of $k$-th greenhouse gas in $j$-th production branch.  
$f_{ij}$ coefficient of the consumption and by production of the $i$-th product per unit of the $j$-th major product, e.g in kg/kg or kg/MJ,  
$GDP$ Gross Domestic Product,  
$G_m$ consumption of $m$th material or energy carrier used for construction of installation,  
$G_j$ nominal flow rate of the $j$th major product,  
$G_u$ nominal flow rate of the useful $u$th by-product,  
$GWP_k$ coefficient of global warming potential of $k$-th gas (1 for CO$_2$ and depending on source 21 – 30 for methane),  
$LHV$ lower heating value of the fuel in MJ/kg,  
$M_{CO2}, M_C$ molar mass of CO$_2$ and of element C, kg/kmol.  
$p^{'}_{kj}$ amount of $k$-th waste product abated from waste products.  
$P_k$ annual amount of $k$-th waste product,  
$P_k$ nominal flow rate of the $k$th deleterious waste product rejected to the environment,  
$p_{k,bio}, p_{k,F}$ amount of waste $k$-th substance rejected to the environment burdening the combustion of conventional (F) and renewable (bio) fuel,  
$P_{kj}$ total amount of $k$-th waste product generated in $j$-th production branch.  
$s_{iu}$ replacement ratio in units of the $i$th replaced product per unit of the $u$th by-product,  
$u_m$ expected recovery factor of the $m$th material,  
$w_k$ monetary factor of harmfulness of $k$-th substances.  
$\Delta m_F$ decrease of non-renewable fuel consumption,  
$\zeta_k$ the total expenditure of non-renewable resources exergy for compensation loses in environment caused by rejection of $k$-th contaminant,  
$\rho_{bio,LCA}$ LC-TEC of biomass,  
$\rho_f$ TEC of avoided consumption of non-renewable fuel,  
$\rho_i, \rho_j, \rho_r$ specific thermo-ecological cost of the $i$-th, $j$-th domestic product and the $r$-th imported good, e.g. in MJ/kg,  
$\rho_m$ thermoecological cost of $m$th material or energy carrier used for construction of installation,  
$\sigma_k$ cumulative exergy consumption of non-renewable resources due to the removing of $k$-th aggressive product from wastes in abatement installation (abatement TEC),  
$\tau$ nominal life time of installation,  
$\tau_n$ annual operation time with nominal capacity.  
$\psi_{j0}$ requirement for natural resources exergy to compensate or to avoid the environmental losses resulting from operation of $j$-th production process, MJ/kg of $k$-th waste product.

**Acknowledgments**

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Thermo-ecological optimization of a heat exchanger through empirical modelling

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Abstract:
The thermodynamic optimization of thermal devices requires information about the influence of operational and structural parameters on its behaviour. The interrelations among parameters can be estimated by different tools such as computational fluid dynamics (CFD) or experimental statistics of the device. Although precise and comprehensive results can be obtained by CFD, the time required for the computations is relatively long. This disadvantage often cannot be tolerated in the case of optimization or online control. In contrast to CFD, neural network and regression methods are characterized by short computational times, but these techniques do not take into account the physical phenomena underlying the investigated process. For this reason, the results of some CFD physical models can be transferred to empirical models. CFD models are used in such a case as a numerical experiment that generates a grid of operational states of the device under consideration. In this study, a CFD model of a heat exchanger (HEX) is built using the commercial package Fluent/Ansys. Based on a set of pseudo-measurements generated with an exact CFD model, an empirical HEX model is developed by applying regression and neural networks. In this case, the heat exchanger is treated as a black-box that connects parameters such as the tube diameter, number and length, the thermal power of the heat exchanger, the pressure drop and the Reynolds number. The results obtained by training and verifying the applied models are discussed. Furthermore, the authors demonstrate the usage of the developed empirical HEX model for the minimisation of thermo-ecological cost (TEC). The TEC expresses the cumulative exergy consumption of non-renewable resources. The example presented in this paper concerns the optimisation of the operational parameters of a heat exchanger. Calculations concerning the minimisation of the TEC are based on the objective function formulated by Szargut. This function adheres to the rules of life cycle analysis (LCA) because it includes the investment expenditures (measured by the cumulative exergy consumption of non-renewable natural resources), the device operation and the final effects of decommissioning the installation.

Keywords:
Heat exchanger, CFD, Optimization, Neural Networks, Thermo-Ecological Cost

1. Introduction

Each technological process requires different devices and activities. In many cases, the proper selection of devices requires the application of an optimization procedure, which determines what type of device will assure optimal operational results. After the device is chosen, its parameters can also be optimized. Of course, the result of optimization is strongly dependent on the optimization criterion. For the criterion formulation, various approaches can be used, such as those based on economic or thermodynamic considerations. As a thermodynamic criterion, the minimisation of entropy generation is often used [1,2]. This method is useful, but its application is very limited because it is a process, not a system, approach. One possible solution is to use the thermo-ecological cost (TEC) as the desired criterion [3]. Such approaches make use of exergy as a thermodynamic measure of the quality of resources [4-
Szargut [11] defined TEC as the cumulative consumption of non-renewable exergy associated with producing a particular useful product. In general, a TEC analysis can be applied for the solution of the following problems [7,11]:

- the examination of the influence of the operational parameters of energy and technological systems on the depletion of non-renewable natural resources,
- the selection of technology that ensures the minimal consumption of non-renewable natural resources,
- the optimisation of construction and operational parameters to ensure the minimum depletion of natural resources,
- the evaluation of harmful impacts from waste products, i.e., exergetic externalities,
- the investigation of the influence of interregional exchange on the depletion of domestic natural resources,
- the evaluation of the ecological harmfulness of particular useful goods over their entire lifetime (thermo-ecological life cycle analysis),
- the determination of pro-ecological taxes.

The results of calculations for thermo-ecological costs have been presented, for example, in [3,12,13,14]. In [7,13], Szargut proposed an extension of the analysis of operational TEC to other phases of production systems. For this purpose, an objective function was formulated to fulfil the rules of life cycle analysis (LCA) because such an approach includes both the investment expenditures (measured by the cumulative exergy consumption of non-renewable natural resources) and the final effects of decommissioning the installation. Moreover, the proposed TEC minimisation goes a step beyond the classic LCA by introducing a common measure for different ecological impacts. The application of the proposed LC-TEC objective function for the optimization of a solar collector is presented in [13]. However, this optimisation is based on a simplified mathematical model of the collector and it should be noted that the description of the optimized device plays an important role in the optimisation procedure. This description, which can be called the device characterisation, should simulate the operation of the device, namely, it should be able to generate the values of the output parameters (temperature, pressure of the working fluids, heat loads, etc.) based on the device input parameters.

Different types of characterisation can be provided. Currently, the most popular type employs experimental characteristics based on the measurements of real devices. Such an attempt has its drawbacks: it is expensive during the creation phase (a number of devices should be thoroughly investigated) and this method cannot be used for new constructions. Another method involves approximate characteristics based on measurements and approximate theoretical relationships. This solution is cheaper, but its accuracy is, in many cases, unacceptable. Currently, due to the rapid development of computer technology and advances in numerical modelling, another solution can be considered. In the place of device characteristics, a numerical model can be inserted. The numerical model simulates the characteristics of a device; it returns the output parameters based on the values of input parameters, supplemented with boundary and initial conditions. The most precise and sophisticated approach is computational fluid dynamics (CFD), in which thermal devices are efficiently modelled [15]. Although precise and comprehensive results can be obtained by CFD, the time required for these computations is relatively long; moreover, advanced computer equipment is required. These disadvantages are often unacceptable for optimization or online control. The alternative of using models directly based on statistical data does not take into account the physical phenomena occurring in the device under investigation. In contrast to CFD, neural networks are characterized by a short
computational time and high precision. For this reason, some CFD results can be transferred to empirical models. CFD models can be used in a numerical experiment, generating a grid of operational states of the considered devices. Based on this set of pseudo-measurements, an empirical model can be identified. In this paper, a heat exchanger is presented as an example of a thermal device. In this case, the heat exchanger (HEX) is treated as a black-box that connects parameters such as the tube diameter, number and length, the thermal power of the heat exchanger, the pressure drop and the Reynolds number. The neural network and regression methods are assessed for selected heat exchanger parameters. Additionally, the results obtained by training and verifying the applied models are discussed. Finally, the possibility of applying the developed models for optimization purposes is described.

2. Minimisation of thermo-ecological cost

The index of operational TEC can be determined by solving the TEC balance, as presented in Fig. 1. The equation for the balance of operational takes the following form [11,14]:

$$\rho_j + \sum_i (f_{ij} - a_{ij})\rho_i - \sum_r a_{ij} \rho_r = \sum_s b_{ij} + \psi_j$$

This set should comprise all branches of domestic economy. However, it would be difficult to solve such a problem. For this reason, practical calculations only consider strongly connected production [7,12]. To express the total natural resource expenditures, the TEC method should include the total life of the installation [4]. A thermo-ecological life cycle assessment (LC-TEC) consists of three main parts:

1. **The construction phase** encompasses the project, the extraction of raw materials, semi-finished product fabrication and transport expenditures in the construction phase. All of these expenses influence the TEC burdening the final useful consumptive product. This phase can have a significant contribution to the TEC in the case of processes based on renewable energy. For instance, the TEC of a wind power plant results mainly from expenses in the construction phase.

2. **The operation phase** is defined as the period of time between the end of the construction phase and the beginning of the decommissioning phase. In processes utilising non-renewable resources, this phase is the predominant consumer of natural resources, mainly energy carriers.

3. **The decommissioning phase** concerns the period at the end of installation. The TEC in this phase results from expenditures for developing the remnants of the system and for terrain reclamation.
The general form of the objective function, based on the LC-TEC concept of TEC minimisation, taking into account the lifetime of the product, was formulated by Szargut [7,8] and applied for a sample investigation in the work of Szargut and Stanek [13]. This function has the following form:

\[
\text{TEC} = \tau \left( \sum_j \dot{G}_j + \sum_k \dot{P}_k \zeta_k - \sum_a \dot{G}_a s_{ia} \right) + \frac{1}{\tau} \left( \sum_m G_m \rho_m (1 - u_m) + \sum_r G_r \right)
\]  \hspace{1cm} (2)

The presented formula expresses the yearly TEC of a given product, with a consideration of its complete lifetime. Equation (2) can also be used for the optimisation of construction and operational parameters of different resource-intensive systems. In this case, the function should be minimised: \( \text{TEC} \rightarrow \min \).

The objective function (2) can also be used for economic LCA optimization. In such cases, the indices \( \rho \) and \( \zeta \) should be expressed as monetary costs. TEC optimisation based on Equation (2) requires a mathematical model of the process or device.
3. Numerical simulation of a heat exchanger using CFD

A numerical model of a device can be developed from the physical laws appropriate for the considered device. In thermal technology, the numerical description of devices is quite often based on conservation equations. One can enumerate the mass balance, the energy conservation equation and the momentum equation. These equations cover the phenomena found in the solid parts of a device (e.g., heat transfer through the solid wall) as well as in the working media (fluid flow in engines, turbines, etc.). The branch of fluid mechanics that utilizes numerical methods to solve such problems is called computational fluid dynamics (CFD) and can be successfully employed to create numerical models of thermal devices. With CFD, in many cases, empirical equations for modelling thermal devices are being used [15].

The procedure of CFD model creation can be divided into several steps:

- geometry creation,
- meshing,
- computation,
- result processing.

The first three steps can be called pre-processing, while the fourth step can be called post-processing. The first step, geometry creation, is usually accomplished by using specialized commercial software, e.g., CATIA or Design Modeler. Depending on the complexity of the considered device, this step can be very time-consuming [16].

The second step divides the created geometry into finite volumes, for which partial equations are solved. Meshing (discretization) is usually performed automatically by sophisticated software, but the results should be carefully checked to avoid computational errors resulting from an improper mesh. The time required for meshing also depends on the complexity of the object and the computational power of the machine used. Within the computation step, the discretized governing equations are solved, returning discrete values of the desired parameters. The time required for this step is strongly dependent on the number of finite volumes under study (which is a function of the complexity of the geometry) and the power of the machine. The computational time can be significantly shortened by using parallel computations in computational clusters or multi-core processors.

For each device, each step of the procedure must be completed. If the aim is to optimize the device’s working parameters, step one need only is performed once at the beginning of the optimization process. However, the shape often needs to be changed during device optimisation. This requires a reconstruction of the geometry, which results in some difficulties in process automation. Fortunately, contemporary software makes it possible to automatically redefine the geometry. This feature allows CFD models to be used in optimization procedures, as will be described later in this paper.

As mentioned above, the computational time is strongly dependent on the complexity of the object and the computational power of the machine used. In more complex cases, this time can be counted in days or even in weeks. This significantly decreases the appeal of employing CDF in optimization loops. This disadvantage can be circumvented by applying another modelling method (a neural network). In the place of the CFD model, a neural algorithm can be used, which significantly decreases the computational time. Neural algorithms are quick and
straightforward. The cloud of data with which the neural algorithm operates can be generated by the CFD model for an entire series of types of the optimized device. Of course, the CFD generation of the input data is still time-consuming, but it can be performed outside the optimization loop and can be used in the optimization of different devices from the same series of types. During this generation, the need for geometry reconstruction may also arise.

The geometry reconstruction can be automated by utilizing the new features of commercial CFD software. In this paper, the commercial software Ansys was used. The new version of the Ansys CFD package is called Workbench [15]. This software allows one to design the CFD process, including loops and process flow control. Both the computational parameters and the geometry can be altered. This is performed through a so-called parameterization of the geometry. During the CFD process, the geometrical parameters can be modified, which, in fact, results in a redefinition of the entire geometry. Then, the model is remeshed and the computations can be repeated many times. The tool used by the authors is semi-automatic, which is one of the main advantages of using this application. At the beginning of the process, all of the parameters, such as those mentioned above, e.g., the geometrical and physical parameters, must be set by the user. After that, all of the computations, which may take up to one month or more, can be performed without the user. In this paper, as an example, the data generation for a tube and shell heat exchanger is presented. In this section, a sample algorithm of data generation, demonstrating the need for a neural algorithm, is presented. A heat exchanger is a device that is used, among other purposes, for exchanging heat between two media: one hot and one cold. Here, a tube and shell heat exchanger working with two streams of water was chosen as a model. Such devices consist of several pipes surrounded by a steel shell. One medium flows within the pipes, while the second flows within the shell outside the pipes. The pipes are fixed by one or two perforated bottoms, depending on the exchanger construction. The heat flows through the heat transfer area, which, in this case, is constituted by the pipes. Due to variations in material consumption, this area significantly influences the cost of the apparatus. A schematic of the analysed heat exchanger is shown in Fig. 2.

The amount of heat exchanged strongly depends on the size of this area and on the medium temperature difference, as governed by the Peclet equation [18]:

\[ \dot{Q} = kA \Delta t_m \]  

(3)

To evaluate the heat load, one should know the values of the heat transfer area \( A \) and the heat transfer coefficient \( k \). Both quantities are based, among others, on the geometry of the exchanger. The size of the inner tubes directly influences the heat transfer area and the flow inside and outside the tubes, which is strongly related to the overall heat transfer coefficient. Thus, for each geometrical configuration, the velocity, pressure and temperature fields should be computed. This step can be performed with any CFD solver; in the sample case, the Ansys Fluent solver was employed [19].
In the computations, the following parameters were assumed to be constant:

- number of tubes,
- outer diameter of tubes,
- hot fluid inlet temperature and mass flow rate,
- cold fluid inlet temperature and mass flow rate.

During data generation, the thickness and length of the tubes were changed. As a result, different heat loads, outlet temperatures and material consumption values were computed.

As the geometry was modified, the flow in the pipes was altered, which influenced the heat transfer coefficient of the pipes and, consequently, the overall heat transfer coefficient. This also changed the heat transfer area, which directly influenced the heat load of the exchanger.

A flow diagram of the data generation process is shown in Fig. 3. In each step, due to the variable geometrical parameters, the whole geometry was rebuilt and, as a consequence, was remeshed in each step. This was performed in a fully automatic way, without user interaction. An example of the geometry mesh is shown in Fig. 4.

In each step, after the geometry and mesh generation, the velocity, pressure and temperature fields of both fluids were computed using the commercial code Fluent. The resulting data were stored for later use with the neural algorithm. In Fig. 4, a sample mesh for one of the cases is shown.

The computational time depends on the power of the machine performing the computations. The analysed case (number of finite volumes ~1 million) required approximately 8 hours on an 8-processor machine and approximately 13 hours on a 4-processor machine. However, due to the full automation of the computations, no interventions by the user were required. This relatively long computational time is not acceptable for optimisation procedures. For this reason, the results of CFD modelling were used for empirical modelling of the HEX.

4. Empirical model of a HEX

In general, the empirical models, both the regression and neural models, belong to the group of black-box models. Such models are purely statistical models without any knowledge about the physical phenomenon proceeding in the investigated system [13, 20-22]. The concept of a black-box model is presented in Fig. 6.
Fig. 3 Flow diagram of the data generation process. (n – number of the tubes, d - external diameter of the tubes, δ - wall thickness of the tubes, l - length of the tubes, i, j, p, s, k, m, t, w - variables).
Figure 5 presents the heat load of the series of types of heat exchangers analysed as a function of the internal diameter of the pipes and the length of the exchanger.

In the case of a HEX, the thermal power of the HEX, the inner and outer diameters of the inner tubes and the inner temperature of both fluids are generally independent variables.

The identification of a black-box model comprises two main steps:

- calibration of the model,
- verification of the model.
It is important that the verification of the model be based on a statistical dataset other than that used for calibration. The verification step is the base for determining the possibility of usage of the model. Empirical models of the black-box type are characterised by the following features:

- short calculation time – for this reason, they can be used online and for optimisation purposes,
- lack of physics knowledge,
- strong influence of the quality of data used for calibration on the quality of the model,
- calibration is simple, with commonly accessible commercial computer codes.

Although the authors developed an exact physical model, its usage for TEC analysis was inconvenient due to a relatively long computational time. For this reason, the development of faster models is necessary. This aim can be obtained through empirical modelling. The authors compared two methods – regression and neural networks – to choose the more efficient model for describing the influence of operational and constructional parameters on the pressure drop of fluids in a heat exchanger. Better results were achieved in the case of neural network models. It should be noted that the models describing the pressure drop of agents in heat exchangers are very important for the objective function and the results of optimisation. The product that is most particularly burdened by cumulative exergy consumption is electricity. Pressure drops in a HEX directly influence the demand for electricity for pumping purposes. For this reason, an efficient model for the pressure drop is crucial to optimisation.

4.1 Regression
The choice of the model structure is the aim of structural identification [22]. Most often, a linear structure is applied in relation to the estimated parameters. The most general type of linear model, the multi-input single output (MISO) approach, can be described by the following equation:

$$y_i = m_0 + m_1 x_{i1} + \ldots + m_k x_{ik} + \epsilon_i$$

(3)

Estimators for the coefficients of the linear regression model can be determined by means of the following formula:

$$m = (X^T X)^{-1} X^T y$$

(4)
Each evaluation of the model’s parameters introduces errors. To determine the relation between \( y \) and \( \hat{y} \), the coefficient of multiple determinations \( R^2 \) is applied [22]:

\[
R^2 = \frac{R_R^2}{R_T^2} = \frac{\sum_{i=1}^{n} y_i}{n} - \frac{\left( \sum_{i=1}^{n} y_i \right)^2}{n} \quad (5)
\]

The coefficient of multiple determinations \( R^2 \) is applied as a measure of the quality of the model with respect to the data variability. The model’s accuracy is higher if the value of the coefficient of determination is closer to 1. However, as more independent variables are added, the \( R^2 \) coefficient increases, even though the dependence can be smaller.

4.2 Neural network

Several signals \( x_k \) of the model are supplied to the neural inputs. These signals come either from the inputs of the neural network or from the outputs of other neurons. Each signal reaching the neuron is multiplied by its weight \( w_k \) and then, all of the signals are summed up. The sum of the input signals \( \varphi \) multiplied by their weights represents the argument of the neuron activation function. The output (answer) of a neuron is the result of the neuron activation function \( y = f(\varphi) \). Figure 7 presents the scheme of a single artificial neuron.

The possibilities of applying a single neuron as a model are considerably limited due to its constrained computation capacity. To obtain advanced computation possibilities and a higher calculation accuracy, the single neurons are interconnected into a net [20,22]. The calculation of the neural network is called the learning process.

The Lavenberg Marquardt algorithm has been investigated as a training method [23].

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Fig. 7 Model of an artificial neuron.
Neural simulations were carried out ten times for \( j=5,10,\ldots,50 \) neurons for each layer with \( k=1,2,3 \) layers. The best \( R^2 \) coefficient for the hot stream was obtained for the case of one layer with 15 neurons; however, for the cool stream, the best value was obtained in the case of two layers with 15 neurons each. In the case of length, the best correlation was obtained for one layer with 15 neurons.

Regression was conducted twice for the hot and cool streams and the length. Firstly, regression was carried out for all CFD data, as shown in Fig. 9 – 11, designated as full regression; secondly, regression was conducted for the data used in training the neural network. In both cases, the correlation coefficient was high.

The correlation coefficients for the regressions and neural networks are presented in Table 1 and the adjustment is shown in Fig. 9 – 11.

In sum, the results of the pressure drops in the hot and cool streams for the neural network and regression methods are similar to those found by the CFD model.

<table>
<thead>
<tr>
<th>Table 1 Correlation coefficients for the neural network and regression methods</th>
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<tr>
<td>Layer</td>
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<tr>
<td>Neurons in layer</td>
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<td>Correlation coefficient</td>
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<td>Regression</td>
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<td>Full regression</td>
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Fig. 9 Adjustment of neural network and regression to CFD simulation results for the hot stream.

Fig. 10 Adjustment of neural network and regression to CFD simulation results for the cool stream.
5. TEC optimisation of heat exchangers

The yearly TEC of the heat exchanger operation can be expressed as follows [7,20,24]:

\[
(TEC) = \frac{\rho_\text{el}}{\eta_\rho} \int_0^\infty \left( V_1 \cdot \dot{\phi}_1 + V_2 \cdot \dot{\phi}_2 \right) d\tau + \frac{\rho_\text{el} \cdot (1-u_\text{el})}{\tau_\text{el}} \cdot (G_{p} + G_{s}^*) + E_{el} \cdot \rho_{el} \rightarrow \min
\]  

(6)

The amount of steel needed for the pipe in the heat exchanger is determined from the following equation [13]:

\[
G_p = \left( \frac{\pi}{4} (D + \kappa)^2 - \frac{\pi \sigma^2}{4} \right) L_{sg},
\]

(7)

It can be assumed [13] that the diameter of the heat exchanger jacket is proportional to the diameter of the tubes and the square root of the number of tubes:

\[
G_s^* = \sigma D \sqrt{n}(1 + \kappa L)
\]

(8)

The electricity consumption for the construction of the heat exchanger should be a function of influential variables, as follows [13]:

\[
E_{el} = \mu D n + \nu D \sqrt{n}
\]

(9)

The following data have been assumed:

- TEC index of electricity [13] \(\rho_{el}\), as shown in Fig. 12
- TEC index of steel products \(\rho_s = 35.7 \text{ MJ/MJ}\)
- Empirical coefficients in Eqs. (7) – (9):
  \[\sigma = 700 \text{ kg/m}; \kappa = 0.151 /\text{m}; \mu = 0.08 \text{ kJ/m}; \nu = 0.3 \text{ kJ/m}\]

The example of a tailed structure of the TEC of electricity calculated by the algorithm presented in Fig. 1 is illustrated in Fig. 12. The highest value arises for the exergy of the resources bar, corresponding to the exploitation cost of the selected system.
Equation (6) contains two main components:

1. the investment component, including the TEC of the materials used for the construction of the exchanger and
2. the operational components expressing the electricity consumption of the pump.

The investment component takes into account the lifetime $\tau_Z$ of the installation and the possibility of reusing the steel after the completion of the installation lifecycle. The operational components take into account the yearly operation time $\tau_P$ of the pump. The entire optimisation procedure is shown in Fig. 13. In the present work, to shorten the calculation time of the optimisation procedure, the CFD model is replaced by the empirical model.

The applied method of TEC analysis is directly connected with the assumption that the boundary is not local, but system-wide. However, the authors consider the heat exchanger boundary to be much broader. For example, there is a power requirement for pumping agents that exchange heat. It is assumed that the pumps are driven by an electric motor. For this reason, the boundary is extended for the process of electricity generation, which is included in the TEC of electricity. This index is introduced as $\rho_{el}$ in the objective function (6/10). Similarly, some materials are used for the construction of the exchanger. In this case, we use steel. The cost of steel is introduced in the index of $\rho_s$, the TEC of steel. The introduction of this cost is equivalent to the assumption of a balance boundary comprising the processes of steel production.
Sample results for the minimisation of the heat exchanger TEC are shown in Fig. 14.

![Diagram of the optimisation procedure](image_url)

*Fig. 13. Schematic of the optimisation procedure.*

*Fig. 14. Yearly TEC of heat exchanger exploitation as a function of the Reynolds number.*
Figures 14 and 15 illustrate the results of the optimisation function, Eq. 6, in conjunction with the TEC of electricity presented in Fig. 12. The calculation results plotted in red in Fig. 14 correspond to rather low levels of electricity TEC ($\rho_e$ in Eq. 6) in the range of 0.2 – 0.3 MJ/MJ. Such values would be obtained for the production of electricity from renewable sources and result only from the construction of a power plant (compare with Fig. 8). However, in the domestic energy market, non-renewable power plants fired with solid fuels have the dominant share. These plants are characterised by a TEC on the level of 3.4 – 3.8. The highest TEC (approximately 5 MJ/MJ) for electricity would appear in the case of CCS technologies. It can be observed that the results of minimisation are strongly dependent on the TEC index. In the case of a lower TEC, the minimum corresponds to higher $Re$ numbers and the curves are rather flat, while the obtained values of the optimal TEC result mainly from the TEC of materials used for heat exchanger construction. In the case of higher values of electricity TEC, the operation expenditures dominate and the minimum TEC values shift toward lower $Re$ numbers. For all cases of higher electricity TEC, the heat exchanger minimum corresponds to an $Re$ number of approximately 50000.

6. Summary and conclusion

The results of preliminary attempts to utilise CFD models in thermo-ecological optimization procedures are quite encouraging. The replacement of the experimental characteristics of devices with numerical models decreases the costs of determining such characteristics and extends the possible application of this method to new devices. The main problem lies in the computational time, which significantly increases the time of optimization. The rapid development in computer technology gives hope for machines that will be, in the not so distant future, able to solve such complicated problems in a reasonable amount of time. Currently, this problem can be overcome by the use of neural networks, which operate on a data cloud generated by a multi-variant numerical model.
Neural network modelling is rapid and, for trained, validated and tested networks with high correlation coefficients for the given range of parameters, is an easy and fast way to obtain the desired results. This article shows the adjustment of points with neural network models and multi-regression to the results of CFD modelling. In this case, for pressure drops and tube lengths, a high correlation coefficient was obtained between the input and output parameters. Both the regression and neural network methods are very useful in determining the correlation between variables and the results can be obtained very quickly.

In this paper, a CFD algorithm to generate data for a neural algorithm is presented. Due to the new features of the commercial package Ansys Workbench, geometry modifications can be provided automatically, which significantly shortens the time required for data generation.

Sample computations of a heat exchanger were carried out. The sample results and a discussion concerning the computational time are given. The presented algorithm can be used on any device for which one can construct a CFD model.

7. Nomenclature

\( A \) heat transfer area,
\( a_{ij} \) coefficient of the consumption of the \( i \)-th product per unit of the \( j \)-th major product
\( a_{ij} \) coefficient of the consumption of the \( r \)-th imported product per unit of the \( j \)-th major product
\( b_{sj} \) exergy of the \( s \)-th non-renewable natural resource immediately consumed in the process under consideration per unit of the \( j \)-th product
\( d \) inner dimension
\( D \) inner diameter of tubes
\( E_{el} \) electricity consumption during exchanger construction
\( f_{ij} \) coefficient of by-production of the \( i \)-th product per unit of the \( j \)-th major product
\( G_j \) nominal flow rate of the \( j \)-th major product
\( G_u \) nominal flow rate of the useful \( u \)-th by-product
\( G_m \) consumption of the \( m \)-th material or energy carrier used for the construction of installation
\( G_{ip}, G' \) mass of steel tubes and exchanger jacket
\( k \) overall heat transfer coefficient
\( L \) length of the tube
\( m_0, m_1, m_2, ..., m_k \) regression coefficients, determined during the calibration of the model
\( n \) number of tubes
\( n - p \) degrees of freedom
\( \delta p \) difference between inlet and outlet total pressure
\( p_{ki} \) amount of the \( k \)-th aggressive component of waste products emitted to the environment per unit of the \( j \)-th product
\( P_k \) nominal flow rate of the \( k \)-th deleterious waste product emitted to the environment
\( \dot{Q} \) heat load of the apparatus
\( \text{Re} \) Reynolds number
\( R \) regression sum of squares
\( R_f \) regression total sum of squares
\( s_{iu} \) replacement ratio in units of the \( i \)-th replaced product per unit of the \( u \)-th by-product
\( t \) thickness of pipe wall
\( \Delta t_m \) logarithmic temperature difference in the exchanger
$u_m$ expected recovery factor of the $m$-th material

$y_i$ actual observation from measurements

$V$ volumetric flow rate

$z_{lj}$ amount of the $l$-th aggressive component of waste products entering the cleaning installation

**Greek symbols**

$\gamma_s$ density of steel

$\varepsilon$ error, uncorrelated random variable

$\sigma, \kappa$ coefficients resulting from construction of the exchanger

$\sigma_i$ cumulative exergy consumption of non-renewable resources due to the removal of the $k$-th aggressive product from wastes

$\zeta_k$ cumulative exergy consumption of non-renewable resources due to the emission of one unit of the $k$-th waste product

$\eta_p$ electric efficiency of the pump and electric engine

$\psi_{j0}$ requirement for natural resource exergy to compensate or to avoid environmental losses resulting from operation of the $j$-th production process

$\mu, \nu$ proportionality coefficients (as determined by the producer of the exchanger, for example)

$\rho_{el}$ unit thermo-ecological cost of electricity

$\rho_s$ thermo-ecological cost of steel

$\rho_i, \rho_j$ thermo-ecological cost of the $i$-th product

$\rho_m$ thermo-ecological cost of the $m$-th material or energy carrier used for the construction of installation

$\rho_r$ specific thermo-ecological cost of the $r$-th imported good

$\tau$ lifetime of installation

$\tau_n$ annual operation time with nominal capacity

$\tau_z$ nominal lifetime of installation

$j_1$ hot medium

$j_2$ cold medium

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**References**


THERMOECONOMIC ANALYSIS AND OPTIMIZATION IN A COMBINED CYCLE POWER PLANT INCLUDING A HEAT TRANSFORMER FOR ENERGY SAVING

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Abstract:
Nowadays, the use of renewable energy is not capable of replacing the fossil energy, so, new technologies such as the combined cycle power, have been developed using the minimum fuel to obtain the maximum performance.
In this research, through the exergy analysis and thermoeconomics techniques a predictive analysis was developed according two possible scenarios: the sulfur content increase in the fuel composition and the implementation of an innovative heat recycling technology such as the heat transformers.
Exergy analysis was applied to each of equipment in an existing combined power cycle plant of 531 MW of power capacity, the individual and global efficiency were obtained, as well as, the effects of different concentrations of sulfur in natural gas composition before and during combustion were analyzed, and also the implementation of a heat transformer was proposed in order to reduce waste heat discharged to the atmosphere.
As a result of the study, the use of a heat transformer was suggested in the gas turbines, where the biggest exergy loss was located with the purpose to reduce the consumed fuel in an order of 4.5% with important economic savings. Also a predictive model of the corrosion in thermoeconomics terms was obtained, allowing determining the irreversibility cost due to corroded pipes and the presence of sulfur in the fuel composition, which will increase due to the location of new fossil fuel reserves in inaccessible sites will be expected, so that fossil fuels with higher impurities will be obtained.
Predictive models, related to exergy consumed and the implementation of new technologies are useful to estimate in a quick and easy way the main variables in a process.

Keywords:

1. Introduction

In general, electric power generation consists in the transformation of some kind of energy like chemical, mechanical or thermal, into electrical energy. Combined Cycle Power Plant (CCPP) stands out among the technologies developed for electric generation, because of the high energy efficiency obtained through the co-existence of two thermodynamic cycles in one system, using steam and combustion gases as working fluids and natural gas (NG) as fuel for power generation.
During the last decades, joined to the technology development, different strategies for the energetic optimization have emerged as response to the increase in the energy demand. Being the exergy analysis, one of the energy integration methods more useful because, the energy quality and the maximum work obtained have been utilized in the design and selection of the best equipments such as gas turbines [1], to establish the best operation parameters in a feasible way [2] considering the technical limits [3] and optimization of process and equipment with good results [4], moreover, the
combination of exergy analysis with economy has allowed the development of disciplines like thermoeconomic, which, is one of the most used at industrial level [5] due to the correlation exergy-cost, allows to realize a complete analysis according the available technology [6], to predict the operational performance [7] in thermal devices, and other applications more, with the purpose to suggest modifications in the current operational parameters and the implementation of new equipment existent [8] or in recent development [9].

In this research, a predictive analysis was developed, considering the operational performance range, construction material, fuel characteristics and energy use as main factors to be considered through the application of exergy analysis and thermoeconomic optimization in an existing CCPP, which produces 531 MW of electricity. Exergy efficiency was obtained and analyzed for each equipment and the global system; likewise the best operation conditions for a double heat absorption heat transformer implementation were determined, economic savings in order of 14,810,240.00 USD/year were obtained with the respective reductions in the NG and energy utilized. A thermoeconomic predictive model was developed to estimate the corrosion inside the Heat Recovery Steam Generators (HRSG), at different concentrations of sulfhydric acid (H₂S) in NG, because of the future expectatives about this pollutant, which estimate NG content will be increased because the lack of fossil resources and high costs of extraction and refinement. Therefore, it is important to analyze the effects of increasing H₂S concentration to suggest preventive actions against the acid corrosion since it constitutes a serious and expensive maintenance problem.

2. System description

2.1 A combined cycle power plant

The study system is a 2x1 CCPP, because two combustion turbines and one steam turbine conform the generation system of 531 MW. The main equipments considered in the combined cycle power flowsheet are shown in Fig.1. The two combustion turbines were designed to operate with NG as fuel and are capable to produce together a combined total of 336 MW. Steam is generated in two Heat Recovery Steam Generators (HRSGs) by heat exchange between the combustion gases (CG) and water. The HRSG is a heat recovery unit with natural circulation, which contains 6 superheaters, 3 evaporators, 3 drums, 4 economizers and 2 reheates. The components are included within an outer isolated cover, designed to receive hot exhaust gas from the combustion turbine at 550°C and are released to the environment from the chimneys at 90°C. The internal components of the HRSG are arranged to obtain a CG horizontal flow through vertical pipes. This equipment operates with three different pressure levels: low pressure (LP) 393 kPa, intermediate pressure (IP) 2000 kPa, and high pressure (HP) 9545 kPa. Each pressure level has the same sections: economizer or pre heater, evaporator and superheater. The economizer heats water at temperature near the saturation, which varies according to the vapor pressure in the corresponding evaporators. In evaporators, saturated steam is generated. The superheaters raise the temperature of vapor above the saturation temperature.

The feed water for the LP level is provided by the condenser at 40°C to the LP drum, passing or bypassing the LP economizer. The LP drum supplies water to IP and HP levels where fed water is evaporated and superheated at 294 °C and 526 °C respectively and receives the steam from the LP section. All the steam produced in both HRSGs is fed to the Steam Turbine (ST) with capacity of 195 MW. The steam leaves the turbine in a saturated state and then passes through the condenser at 10 kPa where all the steam is condensed and it is send to the HRSGs to restart the cycle. Note: The data above indicated are average measured values from an existing CCPP.
2.2 Corrosion

In general, fossil fuels cannot be considered as free of sulfur with the exception of the wood. Small amounts of sulfur have been found in commercial NG components, or in significant amounts in liquid fuels such as diesel or fuel oil [10].

The corrosion is a latent problem in HRSGs, it can be developed inside the pipes where, the protective magnetite layer could be dissolved by the ammonia concentration in water and the operation temperatures. Outside the pipes, corrosion could take place by acid depositions on the metal surfaces, caused by the impurites content and temperature of CG, specifically concentrations of NOx and SO2, due to the proximity of dewpoint of NOx, SO2 and water in shut down and star up. Corrosion by SO3 is possible in plants in operation, where more serious consequences occur when the gas temperature is below the dew point within the process. Corrosion is an inevitable consequence provided the hot sulfuric acid is deposited on a metal surface. SO3 formation occurs when the sulfur content in any fuel is burned and oxidized to SO2 (1). If oxygen is presented, a fraction of SO2 is oxidized to form SO3 (2). In most cases, SO3 forms a small but significant fraction of oxidized sulfur. If water vapor is present and the gas temperature is below the acid dew point this can react with SO3 to produce H2SO4 in spray form (3), and a film formed will also be deposited onto any exposed surface to a temperature below the dew point inside the HRSGs [11].

\[
\begin{align*}
H_2S + O_2 & \rightarrow SO_2 + H_2 \\
2SO_2 + O_2 & \rightarrow 2SO_3 \\
SO_3 + H_2O & \leftrightarrow H_2SO_4
\end{align*}
\]

Sulfuric acid is condensed at temperatures near 100 °C. In general, it is preferable to maintain the process temperature above the dew point, however, during the star up and shut down is inevitably a temperature decrease below the dew point of sulfuric acid or water and condensation will occur, so that, corrosion of sulfuric acid may not always be avoided [12], due to CCPP operators need to decide between low flue gas temperatures that may cause H2SO4 deposition, with consequent corrosion, and high flue gas temperatures, which may avoid corrosion but increase heat losses and reduce efficiency [13,14].

Corrosion by formation of SO2 and H2SO4 has been documented [15], but there has not considered the corrosion from the point of view of H2S content in the NG and the operational times, being
these, important factors due to the efficiency and replace equipment are affected by the exposition time, material composition and temperature [16,17]. In literature, there was not developed a thermoeconomic analysis like the developed in this research, considering corrosion rates and natural gas composition in a CCPP, like this analysis so important due to allows estimating and determining the energy losses and costs in a more realistic form.

2.3 Heat Transformers
There is a growing interest in using heat transformers systems in industrial and domestic sectors. A heat transformer uses low grade calorific energy such as waste heat from industries, solar, geothermal to obtain energy with a greater potential for industrial applications. Single-stage heat transformers can be used to retrieve industrial waste heat at intermediate temperatures when 50 °C gross lift temperature or less are required in an industrial process. However, when higher gross lift temperatures are required, advanced absorption heat transformers, such as two-stage heat transformers and double absorption heat transformer must be considered [18].

This equipment has been analyzed in changing operating and design parameters as measure to increase efficiency [19], in the industrial sector, at pilot scale has been used in the waste heat recovery [20], in the production of environmentally clean steam [21], also the simulation and analysis of its implementation in industrial processes such as pulp and paper effluent in the area [22] has been done, and its optimization in the process area, or integration with other devices distillation processes [23] has been considered.

2.3.1 Double absorption heat transformers
A double absorption heat transformer consists of a generator, a condenser, an evaporator, an absorber, an absorber/evaporator and an economizer Fig.2. A heat source is supplied to separate the working fluid in the generator at an intermediate temperature; the working fluid is evaporated and condensed at a low temperature. Then, the condensed working fluid is split into two streams, one is pumped to the evaporator where is vaporized at intermediate temperature and pressure. The other stream is pumped at a higher pressure to be evaporated in the absorber/evaporator. The vaporized working fluid is absorbed at an elevated temperature in the absorber, by a rich salt solution coming from the generator. The salt solution at intermediate concentration is split into two streams; one goes to the generator to preheat the rich solution in the heat exchanger, the other stream is fed to the absorber/evaporator, where the vaporized working fluid coming from the evaporator and delivers an amount of heat. Finally, the salt solution at a low concentration leaves the absorber/evaporator and is pumped to the generator to restart the cycle [24].

3. Exergy and thermoeconomic analysis

3.1 Methodology
A new methodology developed recently by the authors [9] allows to simulate and optimize the energy and economic sources considering the integration of several the methods above mentioned with pinch technology to obtain the best scenario in an industrial process. This was used in this predictive analysis as follows:
1. Establishing the dead state, temperature and pressure of reference, surroundings and volume control.
2. The balances of mass and energy in the system and the conceptual flowsheet was obtained, to establish the real energy and mass flow.
3. The global system was divided in subsystems, according to the exergy flow, complexity and the different involved processes.
The initial conditions of the global system and the selected subsystems were determined through the exergy analysis, such as exergy flow, efficiency, and the irreversibility. The sensibility analysis was developed to determine the possible causes of irreversibility in the main components, subsystems and the integrated system. Several mathematical models of the considered subsystems in efficiency and fuel terms and the relationship among the global efficiency were obtained, which allowed to establish the relationship between the main variable of the global system and each subsystem. The models obtained made possible to propose improvements such as the modification of the operating conditions and the implementation of new equipment. New exergy analyses were carried out again for each improvement choosing the ones with lower irreversibility to establish the operation conditions modifications. The Pinch technology was useful to determine the location of the new equipment. Once the previous steps were finished, and thermoeconomic analysis were carried out by means of optimization techniques developing and solving the objective function which was the optimum operation conditions of the existing equipment and the design of new equipment increasing the global efficiency at lowest cost.

4. Mathematical Models

According to the outlined methodology, atmospheric air with the conditions of temperature, pressure, and relative humidity of $T_0 = 303.15$ K, $P_0 = 101.3$ KPa and $H_r = 90\%$ respectively was chosen as dead state. The global system was made up for: two gas turbines, one steam turbine, two HRSGs and one condenser. It was a simple system, so it was divided in the four main subsystems, and the surroundings were chosen as from each subsystem to the part of the near atmosphere to the same ones. In a global form, the equations and concepts used in the exergy and exergoeconomic analysis are described as follows:

The exergy, also well-known as availability it is a measure of the useful work that can be obtained of the system in a state given in a specific atmosphere. The exergy balance for an open system that experiences physical and chemical processes can be written in the following way (4):
Where $I$ is the irreversibility, the first two terms in the right side to the equation are referred to the mass flow entering and leaving the system. The following terms are related to the input and output of work ($W_{\text{in}}, W_{\text{out}}$) and heat transfer ($Q_{\text{in}}, Q_{\text{out}}$) respectively, at a specified temperature ($T$). Therefore, energy balances were obtained applying the (4) to the main components considered in the studio system, the equations obtained are as follows (5-8).

**Gas Turbines:**

$$I = (E_{\text{NG}} + E_{\text{Air}}) - E_{CG\text{in}} - W_{\text{GTE}}$$

(5)

**Heat Recovery Steam Generators:**

$$I = (E_{CG\text{in}} + E_{\text{CW}} + E_{\text{CRH}}) - (E_{\text{LPS}} + E_{\text{IPS}} + E_{\text{HPS}} + E_{\text{HRH}}) - Q_{\text{loss}}$$

(6)

**Steam Turbine:**

$$I = (E_{\text{LPS}} + E_{\text{IPS}} + E_{\text{HPS}} + E_{\text{HRH}}) - (E_{\text{SST}} + E_{\text{Condensate}} + E_{\text{CRH}}) - W_{\text{STE}}$$

(7)

**Condenser:**

$$I = E_{\text{SST}} - Q_{\text{cond}}$$

(8)

Where: $E_{\text{NG}}$ and $E_{CG}$ is the Exergy content in NG and GC and $E_{\text{LPS}}, E_{\text{IPS}}, E_{\text{HPS}}$ are referred to the Exergy streams at Low, Intermediate, and High Pressure respectively; $E_{\text{CRH}}, E_{\text{HRH}}$ are the Exergy of Cold Reheat and Hot Reheat Steams Exergy respectively; $E_{\text{Condensate}}$ and $E_{\text{SST}}$ is the Condensate and Saturated Steam Exergy; $Q_{\text{Loss}}, Q_{\text{Cond}}$ is the exergy associated with the heat transfer from the HRSG and the Condenser; $W_{\text{GTE}}, W_{\text{STE}}$ are the Exergy associated with the work transfer produced from the Gas and Steam Turbines, all in [kW] units.

The exergetic efficiency ($\eta$) is obtained through the following equation (9):

$$\eta = \frac{-\left(1 - \frac{T_{0}}{T_{i}}\right) \times Q_{i}}{-\left(1 - \frac{T_{0}}{T_{i}}\right) \times Q_{i} - LW}$$

(9)

Where: $T_{0}$ and $T_{i}$ is the initial and final Temperature from the heat transfer [kW] and $Q_{i}$ and LW is the heat transfer and lost work respectively [kW].

Thermoeconomics provides to the system designer or operator, information crucial to the design and operation of a cost effective system. The cost balance for a system is as follows (10):

$$\sum_{e} \dot{C}_{e,k} + \dot{C}_{w,k} = \dot{C}_{q,k} + \sum_{e} \dot{C}_{i,k} + Z_{k}$$

(10)

where: $\dot{C}_{e,k}$ and $C_{i,k}$ are the average costs per unit of exergy in dollars per gigajoule ($$/GJ$$), $\dot{C}_{w,k}$ and $\dot{C}_{q,k}$ are the costs associated with a work and heat transfer respectively and $Z_{k}$ is the cost rate associated with capital investment or operating and maintenance.

From Eq. (10) and using the auxiliary equation cost for each one of the equipment, it was possible to determine the global system cost and the exergoeconomic optimization for the main components and the whole plant (11-14).
Gas Turbines:
\[ C_{CGin} + C_{GTE} = C_{NG} \]  

Heat Recovery Steam Generators:
\[ C_{LPS} + C_{IPS} + C_{HPS} + C_{CRH} + C_{Loss} + C_{CGin} + C_{CHR} + C_{CW} \]  

Steam Turbine:
\[ C_{SST} + C_{Condensate} + C_{STE} = C_{HPS} + C_{IPS} + C_{LPS} + C_{HRH} \]  

Condenser:
\[ C_{RW} + C_{CW} = C_{Condensate} + C_{SST} + C_{RW} \]  

Where: \( C_{NG}, C_{CG}, C_{LPS}, C_{IPS}, C_{HPS} \) are the NG, CG and Low, Intermediate and High Pressure Steams costs respectively; being \( C_{CRH}, C_{HRH}, C_{Condensate} \) and \( C_{SST} \) the costs referred to Cold Reheat and Hot Reheat Steams and Condensate and Saturated Steam costs; \( C_{Loss} \) are the costs associated with the heat transfer from the HRSG and \( C_{GTE}, C_{STE} \) are the costs associated with the work transfer produced from the Gas and Steam Turbines, all in \([$/kW]\) units. Being the water, the electric energy and the NG costs, known values.

5. Results and discussions

The current irreversibility distribution into the study system was determined by the Exergy analysis, with a 0 % H\(_2\)S composition in the NG (Fig.3), being the greatest irreversibility (65 %) located in the gas turbine due to the inherent irreversibility in the combustion chamber, however, despite the value obtained of irreversibility, none changes were suggested in the combustion process due to the efficiency is an acceptable value in these power systems. Therefore, the whole system was analyzed and as a result of a sensibility analysis, two situations were considered: the increase of sulfur content in NG and the implementation of a new thermal device according the developed model by Rivera [18].

Fig.3. Grassman diagram of the combined cycle power plant without a heat transformer.
The corrosion model development was initiated through the determination of the acid dewpoint temperatures according the H$_2$S content as well as the H$_2$SO$_4$ deposition percentage expected according the operation conditions into the equipment analyzed.

The H$_2$S content in NG was related with the H$_2$SO$_4$ dew point temperature, an exponential function was obtained (15) with a R$^2$ of 0.9996. Being 72.61°C the temperature initial value at 0% H$_2$S (This is the current situation in the existing CCPP analyzed) (Fig 4).

$$T_R = 72.61y_{H_2S}^{0.0637}$$  \hspace{1cm} (15)

![Fig. 4. Dewpoint temperature of H$_2$SO$_4$ from the content of H$_2$S in the NG.](image)

Through the temperature profile obtained, inside the HRSG was the corrosion prone zone, specifically the economizer located in LP zone (Fig. 5) and starting from a H$_2$S concentration of 25 ppm in NG, corrosion will occur in operational times with 65% in volume of H$_2$SO$_4$ in solution.

![Fig. 5. Temperature profile of CG through the HRSG.](image)

According to the equations proposed by Kiang, Okkes and Land [25,26,27] a model was obtained, considering the total number of tubes to be corroded (NT) being the two most important variables: water fed temperature in °C to the HRSG (T$_{CW}$) and the concentration of H$_2$S in percentage in GN ($y_{H_2S}$), the best fit (16) was obtained using linear regression with a 0.999968 R$^2$. The model obtained allows determining the amount of corroded pipes, which will increase while the concentration is increasing and the temperature is decreasing (Fig 6). It is worth to note that the pipes of the HRSG are circular finned pipes, with an area of 46.5 m$^2$ and height of 18 m, the nominal diameter and thickness are 0.044 m and 0.002 m respectively, the construction material is SA178A.
By Thermoeconomic analysis, was possible to quantify the economic losses associated to the energy required to form the corrosion in situ (deposition stage) and the corrosion attack (H$_2$SO$_4$ formation). The models above developed allowed to establish a model, which predicts the irreversibility cost of the system in MUSD/year ($C_i$) considering two of the most important variables: the number of pipes to be entirely corroded (NP) and the H$_2$S composition in percentage in the NG (y$_{H2S}$), the behavior obtained shows two stages, the first one is related to the formation and deposition of condensed sulfuric acid, it refers to incipient corrosion in which the irreversibility and number of affected pipes remains constant, occurring corrosion in situ, in the second stage exists an linear increasing behavior of the irreversibility and the number of affected pipes Fig. 7. The model (17) was obtained from a linear regression with an R-Squared of 0.999 in the software NCSS 2007 ©, it is worth to note that according the H$_2$S current concentration in NG, there is not corrosion inside the HRSGs and the model obtained will allow to estimate the costs related to the pipes replacement according the sulfur in NG in operational times.

$$C_i = 92.70 + 19.39(y_{H2S}) - 2.18\times10^{-2}(PN) + 5.40(y_{H2S})(PN)$$

Fig. 6. Response surface for corrosion in the HRSG.

Fig. 7. Thermoeconomic analysis of corrosion in the HRSG.
By using pinch technology, the chimney exit, was selected as the best site for the implementation of a double absorption Heat Transformer (HT), to improve the plant efficiency, where the HT will use the residual energy of the CG at the chimneys exit (90 °C), preheating 91.5% of the total water fed by the condenser from 40 °C to 120 °C, being the CG output at 40°C. The Exergy analysis was developed and the Grassman diagrams with the HT implementation (Fig 8), the electric production from the ST was increased in 0.25% while the CG exergy at the exit was decreased in 0.47 %, when the HT was implemented. The whole system efficiency was slightly increased in 0.5 % in the H2S scenarios considered with the HT. With the thermoeconomic analysis, it was possible to quantify the economic losses by CG thrown to the environment, which were reduced by the implementation of the HT being this the main advantage due to a 4.5% reduction in fossil fuel consumption, leading economic savings in order of 14,810,240.00 USD/year.

Fig.8. Grassman diagram of the combined cycle power plant with a heat transformer.

Fig.9. Exergetic efficiency and Cost of the CG with (w/HT) and without (w/o HT) the Heat Transformer.
6. Conclusions and recommendations

A predictive mathematical model was developed capable to predict the irreversibility costs in an existent CCPP, based on the quality of the NG and the corrosive effects inside the HRSGs, being the thermoeconomics a useful technique for real process with good results.

With the implementation of the HT savings about of $14,810,240.00 dollar/year savings will be expected, allowing suggest the use and implementation of this new thermal devices in industrial processes, since they improve the efficiency and decrease costs and irreversibility.

The study of complex systems demands a good methodology as the used in this research, is necessary to consider the use of different energy integration techniques to obtain the best operating conditions and to determine the major irreversibility within a system, considering several scenarios.

References


THERMOECONOMIC ANALYSIS AND OPTIMIZATION OF A HYBRID SOLAR-ELECTRIC HEATING IN A FLUIDIZED BED DRYER

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Abstract:
In the last decades, the technological development based in alternative energy sources has been increased. Hybrid systems are still an emerging technological option and it is expected this technology will continue to evolve in the future, so that it will have wider applicability and lower costs, therefore, the application of techniques such as exergy analysis and thermoeconomic are useful to determine the basic energy cost and the possible competitiveness of these systems.

In this research, a pilot prototype of a hybrid solar – electric system for air heating using a low temperature collector was developed as part of a redesign in the habanero chili drying, where a pre treatment by a solar dryer and a fluidized bed dryer were combined. Habanero chili is an important seasonal food product, so drying is so important for the Mexican producers. Exergy analysis was carried out to identify the operation conditions for the whole system and the potential savings, which can be made, such as, the air temperature and velocity (333 K and 6 m/s), while the product quality was preserved.

As a demo for the ECOS 2012 template, text has been uploaded from the Universal Declaration of Human Rights. Detailed info on paper preparation can be found from Section 2 on.

Whereas recognition of the inherent dignity and of the equal and inalienable rights of all members of the human family is the foundation of freedom, justice and peace in the world,

The results showed that the rates of heat and moisture transfer are related to the velocity and temperature of the drying air utilized. A reduction in the average drying time of 50% and in the energy usage of 77% in the fluidized bed dryer were obtained, with an exergy saving of 2.6 KW that represents 0.52 kg of natural gas per kg of fresh habanero chili with the final moisture required of 5%. A thermoeconomic model was developed for the whole system to determine the optimum operation conditions; also, it predicts the savings in energy cost terms at industrial level. An innovative sustainable and efficient process was developed in drying, one of the most demanding and wasteful of energy in the industrial sector.

Keywords:
Hybrid system, Solar Drying, Thermoeconomics, Optimization, Habanero chili

1. Introduction

1.1 Hybrid power systems
In the present times, a considerable interest has emerged in combined or ‘hybrid’ energy systems.

The term of hybrid energy systems refers to applications, where, different types of energy converters are involved for the same requirement.

These systems are usually isolated applications and at least one renewable energy source is included. The hybrid energy systems are used as an alternative operation form of conventional systems, which typically use energy from fossil fuels. The general definition refers to "Hybrid energy systems are a combination of two or more energy sources working in an integrated manner". Nowadays, there are several reasons to use hybrid systems. The main goal is the reduction in fossil fuel usage; also, there is an increase in the energy efficiency.
However, these systems are more expensive in installation and operation aspects, compared to common devices that use petroleum products like gasoline, coal or natural gas as fuels, when in the design stage, the cost of energy of a hybrid system is not considered, which, it is determined primarily by two factors: the cost of the system and the amount of useful energy that is produced. Other factors that are also important include the value of the energy, the cost of conventional energy, lifetime of the system, maintenance costs, and financial costs.

The cost of a hybrid energy system is first of all affected by the cost of the individual components that make up the system. Installation of the components and integrating them into a functioning unit will also contribute to the total cost. The cost of the useful energy will depend on the type of application and the nature of the resources available [1].

Therefore, when the equipment and energy available have a low or zero cost, an important economic advantage is been obtained by the hybrid systems over the conventional energy systems. Solar energy stands out from other renewable energy, being more available, accessible and free, and it is considered such as one of the greatest technological potential areas. Conversion technologies from solar energy are focused into two main branches: solar-thermal and solar electric. The solar-electric technology is performed by photovoltaic cells to obtain electric power by using materials such as silicon and germanium. The solar-thermal technology is performed with solar collectors, which are devices designed to capture solar radiation, transform it into thermal energy to raise the temperature of a fluid for later use. Solar collectors are divided into two groups: without and with energy concentration, the first ones allow obtaining temperatures in the order of 100 °C, while with the last ones, temperatures above of 100ºC are reached using methods of optical concentration. The materials used in the collectors have lower cost than solar cells and their applications are diverse, so, the solar-power tends to have more costly components, this in relation to copper used in solar-thermal technology also has solar thermal applications varied, so it has become a future field of study, within the most common applications, where solar collectors are used, are: the heating of water for domestic use, thermal conditioning of swimming pools, drying of leave tea and power generation.

In many practical applications, the drying process consumes a high amount of energy, due to high latent heat of water, and low energy efficiency of industrial dryers. Has been reported that industrial dryers consume on average 12% of the total energy used in manufacturing processes [2]. In the drying industry the main energy sources to operate the dryers are fossil fuels, industrial oil, natural gas, electric Power, waste materials (biomass) and solar energy [3]. So, drying process is one of the most demanding and wasteful of energy in the industrial sector being an opportunity area to redesign through a hybrid system, due to the low temperature requirements.

In this research, a low temperature solar collector (flat plate) and a open sky dryer were chosen between the technologies available due to its low cost and the maximum daily solar radiation of 560 W/m² existing in Yucatan, Mexico, the first one was coupled with an electric system for air heating in a pilot fluidized bed dryer and due to the habanero chili is a hygroscopic material there are practical restrictions to be fluidizable (up 90% humidity) was necessary to include a solar drying pretreatment (open sky dryer) to dry habanero chili into the fluidized bed dryer. Being the useful energy determine by the exergy analysis and the best conditions of temperature, air velocity and exergetic efficiency was obtained by the thermoeconomic optimization according the primary energy cost to heat air. Therefore, a reduction in fossil fuel consumption was obtained through the redesign of a drying system with important economic savings.

### 1.2 Solar drying

A true sustainable development requires the efficient usage of energy and resources, so that, solar drying is a sustainable process [4].
Solar drying is used in fruits and vegetables because good results are obtained, also it is efficient, providing a longer product life and preserving properties and vitamins. Solar dryers can be classified into three groups, which are defined based on the power source to operate, such as
1. Solar dryers using environmental natural energy only.
2. Semi-artificial solar dryers with a fan and a motor that maintains continuous airflow on drying area.
3. Artificial Solar-assisted drying, they use electricity as backup energy power.

1.3 Fluidized bed dryer
Fluidized bed dryer is an emerging technology also is one of the most used in the industry because of it allows drying a wide variety of products like granular materials, cereals, polymers, chemical compounds, pharmaceutical compounds, fertilizers, crystalline products and minerals. Fluidized bed dryer can operate in batch and continuous form compared with other drying techniques, also it has great advantages, such as: high transfer of energy and matter between gas and solid particles is obtained, good and rapid mixing of solids, a control in the temperature within the bed is possible, the dry sample is handled in an easy way to dry and the construction is simple. [5]

1.4 Habanero Chili
Mexico has the highest genetic diversity of Capsicum and ranks second after China in terms of world production. Among the biological properties of the chilies, highlight the antioxidant activity, anticancer properties, antimicrobial properties, anti-inflammatory and analgesic property of the hot varieties, because of the capsaicin presence. [6]

The habanero chili (Capsicum chinense Jacq) is a crop of great economic importance for Mexican producers in the state of Yucatán (Fig.1). The major crop is located in the north of the state and contributes over 90% of the volume of state production, which is for local marketing and some is used in industry as raw material for hot sauce preparation. Local consumers prefer the orange fresh chili [7]. The condiments production is the main application of dehydrated habanero chili, as well as the extraction of capsaicin, which is the substance that gives the itch to chili, and may also have other uses, including the making of tear gas, insects or rodents repellents. Traditionally dehydration of habanero chili is done in ovens wood, built by the same producers. These furnaces are characterized by being built in a rudimentary form and consequently have many energy losses. The drying process can take several hours, requiring a constant supply of wood, causing the unwanted emissions of pollutants to the environment. [8] Industrial drying of habanero chili takes place in trays, the disadvantage of this equipment is the high maintenance and operation costs and the average drying time is 12 hours, this process is wasteful of energy to get a moisture content of 5% wet basis. That is the main reason to use a new technology to energy saving like a hybrid energy system. Using the solar energy to pre-treatment the habanero chili and the fluidized bed drying can take place using a system solar electric for air heating, the optimum energy use in the global process was obtained, according the environmental and climatic conditions existing in Yucatan, Mexico, where a monthly average number of solar hours per day, is between 6 and 8.5 hours, an average ambient temperature per day is over 30 ° C from March to October, with a critical period in the months of April, May and June with average maximum temperature of 37 ° C and maximum 40 ° C. extreme and there is an availability of solar radiation per hour in the summer period in May and June, with a maximum daily solar radiation of 560 W/m2 integrated [9].
Fig 1. Fresh habanero chili (Capsicum chinense Jacq) native of the Yucatan Peninsula.

2. Exergy Analysis and Thermoeconomic

A sustainable energy system is more efficient, safer and friendlier to the environment compared to a system that operates with conventional energy for local use, being the exergy analysis widely used for design, simulation and performance of these energy systems. [10]

The exergy analysis is applied to several system for studying the efficiency of photovoltaic cells and heaters, to increase efficiency [11] of the drying of porous materials, different studies have been developed to analyzed the energy and exergy flow and it was found that many variables need to be measured such as the ambient humidity, moisture and texture of the product and air heating temperature [12], particularly in the food area, the energy applied to the drying process, is a greater amount than the needed to dry the product, so that, dryers use a great quantity of energy making the process inefficient.

The exergy flow in the drying process of food has been analyzed, concluding that the main variables are the total drying time, the amount of initial moisture of the product concerned as well as operation temperature of the drying air because of they are the most representative variables of the process [13-14].

In the solar area, several energy and exergy analysis have been conducted, [15] in an open solar dryer using mint leaves, exergy efficiencies were high, since the amount of useful energy was greater than in conventional drying, being the solar dryer temperatures used from 35°C to 60 ° C, the best results were obtained with temperature of 60 ° C, and the drying time was 8 hours to dehydrate mint leaves from 84.7% to 10% of wet basis moisture, [16] another research establishing that the characteristics of the solar collector, such as the length of collector, its surface as well as, the operating conditions, are the main variables in the heat transfer process and exergy efficiencies were found due to the low value of usefulness of energy sun, because only a little part of it is captured [17]. Also the coupling of a solar collector to drying system has been analyzed and it increased the exergetic efficiency values of the process [18].

In general, an efficient drying process has been obtained through the use of low-temperature solar collectors, being the temperatures close to 60 ° C, with a better yield in energy use and a good quality dry product. [19]

From the economic point of view a solar pretreatment in a drying process offers many economical savings, also reduces the residence time in the final dryer, obtaining a process more efficient and sustainable. Energy utilization in this treatment type, increase the energetic and exergetic efficiency of drying process, this potential decreases with increasing temperature and drying time.[15-18]

Several analyses have been realized to the habanero chili to determine its content features like pungency, capsaicin, water content, antioxidant and nourishing properties. [7-8], also 50°C-70°C has been established as the range of temperature operation for drying process, to avoid thermal degradation of product assuring final product quality. The drying kinetics of the habanero chili normally is realized in times greater than 20 hours in rotary dryers [20]. The process, reflects a high
operating cost because it is a continuous process where the amount to be treated is in function of the
capacity of the dryer.

3. Methodology

The methodology used for thermoeconomic optimization of the hybrid solar-electric heating system
in this research has been implemented previously. (Fig.2) [21]
1. Temperature, pressure and relative humidity of the surroundings were established as the
reference dead state, which are $T_o = 308.15 \, K$, $P = 1 \, \text{bar}$, $\text{RH} = 70\%$ respectively. The fluidized bed
dryer was selected as the control volume.
2. Mass and energy balances were developed for the control volume, to determine the exergy flow,
efficiency and irreversibility.
3. A sensibility analysis was carried out in order to determine the possible causes of the
irreversibility.
4. The global system was divided in two subsystems, mass and energy balances were obtained for
two subsystems, which are as follow:
Subsystem 1: Open Sky Solar Dryer.
Subsystem 2: Fluidized Bed Dryer.
5. New exergy analyses were carried out again for each subsystem, to establish the modifications in
the operating conditions. Once the previous steps were finished, a thermoeconomic analysis was
carried out by means of optimization techniques solving the objective function.
As a result of this, the optimum operation conditions of the old and new equipment were
determined, reducing the costs and increasing the global efficiency.

3.1. Mathematical Model

3.1.1 Initial system and hybrid system

For overall system, the exergy balance for an open system can be written as (1).

$$I = \sum (mE)_w + \sum (mE)_{out} + \sum W_w + \sum W_{out} + \sum \left[ Q\left(1 - \frac{T_o}{T}\right)\right]_w - \sum \left[ Q\left(1 - \frac{T_o}{T}\right)\right]_{out}$$

(1)

Where I is the irreversibility, the first two term in the right side to the equation are related with the
mass flow entering $(mE)_w$ and leaving $(mE)_{out}$ the system. The following two terms are related to
the work $(W_w, W_{out})$ and the two last ones with the heat transfer to the system and from the system
respectively $(Q)$. Therefore making exergy balances in the main components of the system the
following equations are obtained.

Applying the general equation to the drying column, considering that the main heat transfer is due
to heat of evaporation between the solid and the drying air, and also, the heat transfer to the
surroundings and the kinetic and potential energy are neglected, being the mass flow of dry air and
the mass of dry material within the control volume, constants with respect to time, the following
form (2) was obtained:
Or the above equation could be expressed as follows (3):

\[ \Delta E_{x_{1-2}} = \Delta E_{x_{air1-2}} + E_{x_{evap}} - E_{x_{loss}} - E_{x_D} \]  

(3)

Where is \( \Delta E_{x_{1-2}} \) the exergy loss in the column of fluidized bed dryer, \( \Delta E_{x_{air1-2}} \) is the difference in exergy of the air at the inlet and outlet of the column, \( E_{x_{evap}} \) the exergy of evaporation of water in the system, \( E_{x_{loss}} \) the exergy loss in the environment and the \( E_{x_D} \) the exergy destroyed in the drying phenomenon is the irreversibility.

Being the generated energy obtained from an entropy balance for the same volume of control (4):

\[ \frac{W_{a}(E_{x_{1-2}})}{\Delta t} = \frac{Q_{evap}}{T_m} + m_a(s_1 - s_2) - \frac{Q_{loss}}{T_b} + S_{gen} \]  

(4)

Where \( \frac{W_{a}(E_{x_{1-2}})}{\Delta t} \) is the change in entropy of the habanero chili during the drying time, \( Q_{evap} \) is the heat required to evaporate the excess moisture in the habanero chili, \( T_m \) is the temperature at which is conducting evaporation, \( m_a(s_1 - s_2) \) is the entropy change of the drying air from the entry and exit of the column respectively, \( Q_{loss} \) is the heat that is lost to the surroundings, \( T_b \) the temperature when heat loss occurs and \( S_{gen} \) is the entropy generated in the control volume.

The exergy efficiency (\( \psi \)) is the main parameter for yield measure is obtained from the exergy balance, and it is the ratio between the required evaporation exergy (\( E_{x_{evap}} \)) and the drying air exergy of the next to the column (\( E_{x_{air1}} \)) is expressed as (5):

\[ \psi = \frac{E_{x_{evap}}}{\Delta E_{x_{air1-1}}} \]  

(5)
For the hybrid system, the above equations were used to determine the exergetic efficiency of the drying column, considering pre-treatment of the solid in the open sky solar dryer, with the new conditions of the system. $P = 1$ bar, $T_o = 308.15$ K, $T = 333.15$ K and RH = 70%.

### 3.2 Thermoeconomics

Thermoeconomics provides the system designer or operator, crucial information to the design and operation of a cost effective system [20]. The cost balance for a system is as follows (6):

$$\sum e \bar{C}_{e,k} + \bar{C}_{w,k} = \bar{C}_{q,k} + \sum e \bar{C}_{i,k} + Z_k$$

where: $Ce,k$ and $Ci,k$ are the average costs per unit of Exergy in US dollars per KWh ($\text{USD/KWh}$), $Cw,k$ and $Cq,k$ are the costs associated with a work and heat transfer respectively and $Zk$ is the cost rate associated with capital investment or operating and maintenance.

The thermoeconomic equation of studio system is as follows (7):

$$C_{i-a} + C_{w,e} + C_{f-ch} + C_{d-ch} = C_{o-a} + C_{irrev}$$

Where $C_{i-a}$ and $C_{o-a}$ is the cost of inlet air and output air into the system respectively, $C_{w,e}$ is the cost associated with electric energy, $C_{f-ch}$ and $C_{d-ch}$ is the cost of fresh and dry chilli and $C_{irrev}$ is the irreversibility cost of system. Being the cost of Air, electric energy and fresh chilli known values.

### 4. Results

#### 4.1 Initial system

The volume control and initial system was a pilot fluidized bed dryer operated with electric power. The system was characterized by the mass, energy and exergy balance. High power consumption and low energy and exergy efficiency were detected, due to the relationship among the transport phenomenon along the fluidized bed dryer column: momentum in the base to form the bed, heat and mass in the rest of the column to dry the wet solid (Fig 3).

These phenomenon were analyzed to identify the main low efficiency causes, such as:

1) The initial high moisture content of the solid at the dryer inlet (around 90%), affected the bed formation and the fluidization process by the solid wet loss due to the tendency to stick it on the column walls, also the velocity utilized was a greater value than the fluidization minimum velocity ($11 \text{m/s}$ instead of $2.4 \text{m/s}$).

2) The operating temperature was $T= 70^\circ \text{C}$, causing thermal degradation in the final product, also, there was a significant heat loss due to the lack of insulation along the column.

3) The mass transfer related to the postcritic period of moisture loss was not reached, because of the high velocity and temperature used even during long operation times, being the time an important factor during the post critic period.

Therefore, according the above analysis developed, velocity, temperature and time are the main variables associated to the transport phenomena developed in the fluidized bed dryer, they are related with the energy efficiency. The initial system had higher energetic efficiency with moisture content higher than 5% (Fig. 4) at $70^\circ \text{C}$ during 7.5 hrs, being this process inefficient in energetic and economical aspects therefore (efficiency 4%), the scaling process was not possible to realize under this conditions, so the initial system was analyzed through the exergy analysis and thermoeconomic optimization to determine the useful energy and the real cost of the final product in base of the air heating system.

Through a technical-economic analysis, the use of solar technology of low cost such as a flat plane collector and an open sky dryer to redesign the initial system were chosen. The first one was
coupled to the electric air heating existent into the fluidized bed dryer (Fig 3) and the open solar dryer was the pretreatment for habanero chili before the fluidized bed dryer, avoiding the stick effect.

4.2 Hybrid System Description

The system was conforming by 2 subsystems: an open sky solar dryer and a flat plane collector coupled a pilot fluidized bed dryer (Fig. 5)

Subsystem 1: Open sky solar dryer. The moisture content of the fresh chili was reduced from 90% to 30% in 6.8 hrs, this process required the major quantity of energy 70% (Table 1) in the drying process due to the biggest water lost takes place. (Fig.5a). this subsystem avoided the stick effect and reduced the time and electric energy used in the fluidized bed dryer.

Subsystem 2:.A flat plane solar collector coupled to a fluidized bed dryer (Fig. 5b): In this system the solid got into the fluidized bed dryer which has an electric blower, the leaving air from the blower exchanged heat with the hot water from the storage vessel from the flat plane collector, to obtain the required temperature (T=60°C) then, the drying process was completed and the final product had the moisture required of 5% in 3.5 hrs requiring 30% (Table 1) of the drying energy.

The initial drying process was developed at 70°C with thermal degradation of the product and great heat losses. The energy analysis indicated the energy efficiency was almost the same value at 50 and 60 °C (4.80 and 4.87% respectively) and considering the efficiency of the solar flat plane collector (80%) according to maximum daily solar radiation of 560 W/m² integrated existing in Yucatan, the optimum air temperature was established at 60°C. Therefore, improvements in the energy efficiency (11%) and in the drying time (3.5 hrs) into the fluidized bed dryer were obtained (Table 1).
4.3 Exergy Analysis

The initial and hybrid systems were analyzed through exergy analysis, to determine the useful energy (exergy) into the process, the real efficiency and the cost of the hybrid system. The results obtained of the energy analysis indicated that there was an increase in the process efficiency from 4 to 7%, also in the drying time (10.3 hrs.), but the heat loss is 92% (Q_loss) of the heat total, according the exergy analysis, the hybrid system reduced the input exergy around of 35%, (2.6 kW) that represents 0.52 kg of natural gas per kg of fresh habanero chili with the final moisture required of 5%.

The irreversibility in the hybrid system was reduced from 16% (3.47kW Fig 6a) to 2% (0.31 kW Fig 6b) respectively. Therefore, despite the fact the hybrid system increased the useful energy and reduced the irreversibility, the drying time was increased from 7.5 to 10.3 hrs (Fig 7), despite the fact of the drying time into the fluidized bed dryer was reduced 50%, the cost of the hybrid system need to be determined by the thermoeconomic analysis to establish the economic feasibility of the hybrid system.

Table 1: Heat streams involved in drying process.

<table>
<thead>
<tr>
<th>Process</th>
<th>Qinlet (kJ)</th>
<th>Qevap (KJ)</th>
<th>Qloss (KJ)</th>
<th>Time (h)</th>
<th>η Efficiency</th>
<th>Energy input (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Open Sky Solar Dryer</td>
<td>21,092.70</td>
<td>2137.173</td>
<td>18,955.53</td>
<td>6.8</td>
<td>10.13%</td>
<td>70</td>
</tr>
<tr>
<td>Solar - Electric Fluidized Bed Dryer</td>
<td>9,187.89</td>
<td>100.9553</td>
<td>9,086.93</td>
<td>3.5</td>
<td>11%</td>
<td>30</td>
</tr>
<tr>
<td>ALL PROCESS</td>
<td>30,280.59</td>
<td>2238.1283</td>
<td>28,042.46</td>
<td>10.3</td>
<td>7%</td>
<td>100</td>
</tr>
</tbody>
</table>

Fig.5. Hybrid drying process.
4.4 Thermoeconomic model

The real cost of the hybrid system developed was determined by the thermoeconomic model considering the main variables involved in the process: temperature, exergy cost of the air drying, time and exergetic efficiency.

The exergetic efficiency depends on the air cost and the temperature, through a canonical analysis considering the drying temperature from 50 to 70°C, maximum costs were obtained at the highest temperature (70°C) and lowest exergetic efficiency in the hybrid drying process. (Fig.8). The exergy efficiency and cost obtained between 50 and 60°C (black zone in Fig 8) are very close values, then the drying temperature of 60°C was selected to realize the drying into the fluidized bed dryer.

Thermoeconomic analysis was developed to determine the energy cost of the hybrid system, considering the environmental temperature air and the drying time into the fluidized bed dryer, due to in this equipment there was an electric consumption. The results obtained indicated that the drying time used in the fluidized bed dryer process (3.5 hours) had the biggest exergetic efficiency of 0.5% with a drying cost of 0.33 $ USD / KWh of heating (Fig. 9).
The air drying cost ($C_{Air}$) in the fluidized bed dryer, was obtained by the thermoeconomic model in function of temperature air feed and the exergetic efficiency of the system with the software NCSS2000 © with a lineal behavior with a $R^2 = 0.9999$.

$$C_{Air} = 6.567 - 1.0628 \times 10^{-2}(T) + 3.907 \left(E_{fi}\right)$$  \hspace{1cm} (8)

Where $C_{Air}$ is the air drying cost into the hybrid process in $ USD/KWh$, $T$ is the drying air temperature at °C and $E_{fi}$ is the global efficiency of the drying process.

The thermoeconomic model obtained was useful to establish the real cost of heating of the hybrid system to establish the dried habanero chili cost obtained by this process, being the cost reduced in 50% respect the commercial technology (Table 2).

Therefore, the hybrid system is a feasible and economic option for redesign the habanero chili drying at industrial scale.

<table>
<thead>
<tr>
<th>Cost per kg of dried chili ($ USD $)</th>
<th>Drying time (hours)</th>
<th>Energy source</th>
<th>Type of process</th>
<th>Final moisture of dried chili (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>15.4</td>
<td>12</td>
<td>Gas natural</td>
<td>Comercial</td>
<td>8</td>
</tr>
<tr>
<td>40.4</td>
<td>7.5</td>
<td>Electric Power</td>
<td>Fluidized bed dryer</td>
<td>5</td>
</tr>
<tr>
<td>7.7</td>
<td>10.3</td>
<td>Solar Electric</td>
<td>Hybrid process</td>
<td>5</td>
</tr>
</tbody>
</table>

5. Conclusions

The redesign process, distributes the total of the energy usage, being 70% of the energy used in the pretreatment of open sky solar dryer while 30% was used in the fluidized bed dryer. Being obtained an exergy saving of 2.6 KW that represents 0.52 kg of natural gas per kg of fresh habanero chili.

Habanero chili drying by a solar electric hybrid system is a real alternative to dehydrate this product, being comparable with other industrial dryers such as forced convection heat dryer and tray industrial dryer, improving the overall drying time, obtaining an efficient energy use and cost saving operation.
The Hybrid system developed is a feasible and economic option for redesign the habanero chilli drying at industrial scale. A new technology has been developed. The energy cost and the advantages of new technologies like hybrid systems only can be determined by the exergy analysis and thermoeconomic optimization in a real and feasible way.

Acknowledgements

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Nomenclature

$E_{fi}$ Overall efficiency of the drying process in function of the drying air cost.
$E_{ex_{evap}}$ Exergy due to evaporation of water in the solid, kW
$E_{ex_{loss}}$ Exergy loss to the environment, kW
$E_{x_{D}}$ Exergy destroyed in the drying phenomenon, kW
$h_1, h_2$ Enthalpy of the air inlet and outlet air of fluidized bed dryer, kJ/kg
$I$ Rate of Irreversibility, kW
$m_a$ Air stream fed to the fluidized bed dryer, kg/s
$Q$ Heat involved in the process, kJ
$Q_{evap}$ Heat required to evaporate the excess moisture in the habanero chili, kJ/s.
$S_{gen}$ Entropy generated into the column, kJ/K
$S_i, S_2$ Entropy of the air inlet and outlet air of fluidized bed dryer, kJ/(kg K)
$T_{m}, T_{o}, T_b$ Drying temperature, Reference state temperature, Evaporation temperature, K
$T$ The temperature of drying ire presented in the thermoeconomic model, °C
$W_{in}, W_{out}$ Work done inside and outside the system, kg/s

Greek symbols

$\text{Eff}$ Exergetic efficiency of the drying process
$\text{E}$ Energetic efficiency of the drying process
$\Delta E_{x_{m1-2}}$ Rate of exergy loss in the fluidized bed dryer
$\Delta E_{x_{a1-2}}$ Exergy difference between the air outlet and inlet to fluidized bed dryer

Subscripts and superscripts

$C_{e,k}, C_{i,k}$ The average costs per unit of Exergy, $USD/(kWh)$
$C_{w,k}$ Cost associated with work in the system for thermoeconomic equation, $ USD
$C_{q,k}$ Cost associated with work in the system for thermoeconomic equation, $USD
$C_{r,a}$ $C_{d-a}$ The cost of inlet and output air for the drying system, $USD
$C_{w,e}$ The cost associated with electric energy in system, $USD
$C_{f-ch}$ The cost of fresh chili, $USD.
$C_{d-ch}$ The cost of dry chili in the process at the end of the process, $USD

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$C_{irrev}$ The cost of system irreversibility, $\text{USD} / \text{(kWh)}$

$(mE)_{in}$ Mass flow input system, $\text{kg/s}$

$(mE)_{out}$ Mass flow output system, $\text{kg/s}$

$W_{(s_{in}-s_{out})}$ Entropy of the habanero chili during the drying time, $\text{kJ/K}$

$Z_k$ The cost for operating and maintenance of the fluidized bed dryer, $\text{USD}$

**References**


Thermoeconomic approach for the analysis of low temperature district heating systems

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Abstract:

In this paper a thermoeconomic analysis of district heating systems is performed. The analysis aims at comparing possible options to supply heat to the users, using low temperature networks. Thermoeconomic analysis consists a powerful tool to perform such analysis as it allows one to evaluate the possible options in terms of primary energy cost or economic costs. In the first case, the use of exergy as the quantity that is transported along the network makes it possible to properly consider the various qualities of energy that are used to supply heat to the network and to distribute it to the users. In the case of economic cost, the various cost contributions are considered: investment cost, cost of heat supplied to the network, pumping cost. A different cost can be calculated for the various users depending on their position and characteristics of the heating devices. This is a useful information in order to compare possible options for supply them heat.

Keywords:
District heating, low temperature heating, thermoeconomic analysis

1. Introduction

District heating is a rational and reliable way to supply heat to multiple users from a unique or few centralized thermal plants. Heat is mainly produced with systems as combined heat and power plants, biomass boilers and industrial processes heat recovery systems, integrated with high efficiency boilers to cover peak loads. One of the most interesting contributions of district heating networks to future energy systems is the opportunity to integrate heat productions from renewables [1].

The use of renewables in district heating may involve reduction of the operating temperatures. This idea is behind the concept of “low temperature district heating”. Low temperature district heating is typically characterized by supply temperatures between 75°C and 50 °C (even if lower temperatures may be considered) and return temperatures between 40 °C and 20 °C [2,3]. This allows the direct use of renewable energy sources as solar [4] and geothermal [5] or in combination with large-scale heat storages [6]. In addition, there is big potential for utilization of waste heat from cogeneration plants, waste-to-energy plants, heat pumps and industrial processes [7,8]. Low temperature networks allow one to increase the amount of heat recovery from exhausts and also to recover heat from low temperature processes.

The main issue on this kind of systems is referred to investment and operating costs. As any other system it needs to be more convenient than the alternatives. In addition, its energy sustainability should be carefully analyzed in order to ensure that the total primary energy required to supply heat to the users is smaller than possible alternatives.

To build district heating networks several years are usually necessary, with large expenses and discomfort to the community. For this reason, the system must be designed in its final structure, with few possibilities of making changes. In particular it is necessary to determine the possible users to be connected, the topology and the pipe diameter of each branch. Such a problem can be solved as a synthesis problem, i.e. an optimization where the system structure is not defined a priori.
In this way it is possible to define the optimal network that minimize (or maximize) an objective function, such as the minimum cost of heat or the maximum benefit.

This paper deals with the problem of district heating network (DHN) synthesis, i.e. the search for the optimal configuration of the network, which consists in the identification of users that should be connected to the network and those to which heat should be supplied through alternative systems. In particular, a low temperature district heating network fed with solar energy is considered. The analysis is conducted considering a supply temperature between 55 °C and 40 °C, while the return temperature is assumed 25 °C or 20 °C. Groundwater heat pumps run with solar photovoltaic are considered as the possible alternative, in order to obtain 100% renewable configurations.

A thermoeconomic approach [10] is applied to a small network by considering both monetary and energy cost as the objective functions in the optimization.

2. Thermoeconomic analysis

The optimal synthesis of energy systems is here approached by starting with a superstructure, which is a DHN involving all of the possible zones and thus all the users. The use of a superstructure is the most common approach to synthesis problems (see for example [11]). Once the superstructure is built, the synthesis problem can be solved as an optimization problem, provided that particular values of the variables associated with the components or with the internal flows correspond to the condition of absence of that component or flow. In the case of DHN when the optimal mass flow rate in a pipe is zero means that the pipe must be eliminated from the structure.

The procedure starts with the evaluation of the objective function in the initial configuration, corresponding to all the users connected with the network. The network is then reduced, through successive elimination of the users characterized by high costs and the corresponding pipes connecting these users with the rest of the network. The selection of the user to be disconnected to the network is operated using a probabilistic approach. The probability of a user to be disconnected increases with increasing unit cost of heat supplied to that user. As this procedure is not deterministic, it should be repeated several times in order to increase the possibility to find the true optimal configuration. The procedure is stopped when all the users are disconnected.

The details of the selected procedure are shown by considering the average primary energy consumption of heat provided to the users as the objective function to be minimized. This quantity is calculated as the exergetic unit cost of heat. The first step consists then in calculating the productive function:

\[ \bar{e} = \frac{C_{\text{net}}}{Q_u} = \frac{C_{\text{net}} + c_F \cdot Q_F + c_P \cdot L_P + c_a \cdot Q_a}{Q_u} \]  

(1)

The cost of network \( C_{\text{net}} \) is the amount of primary energy required to produce and install the insulated pipes. Components as heat exchangers, pumps, valves have been neglected in the analysis. Primary energy associated with excavation, installation and paving restoration has been also considered. \( C_{\text{net}} \) is an annual cost. Year is the best unit time to be used for thermoeconomic analysis of such system due to the production variation depending on the average external temperature and during the day. Cost functions used in this analysis are discussed in the annex.

The energy unit cost of heat has been calculated considering heat production from solar collectors only. The following expression for collector efficiency has been considered [12]:

\[ \eta = \eta_0 - a_1 \frac{\Delta T}{I_t} - a_2 \frac{\Delta T^2}{I_t} \]  

(2)

\( \Delta T \) is the difference between the average temperature of the fluid inside the collectors and the air temperature and \( I_t \) is the total radiation. Ambient temperature and solar radiation of Turin have been considered (see Table 1).
Table 1. Solar radiation and temperature for Turin

<table>
<thead>
<tr>
<th></th>
<th>Jan</th>
<th>Feb</th>
<th>Mar</th>
<th>Apr</th>
<th>May</th>
<th>Jun</th>
<th>Jul</th>
<th>Aug</th>
<th>Sep</th>
<th>Oct</th>
<th>Nov</th>
<th>Dec</th>
</tr>
</thead>
<tbody>
<tr>
<td>$I_t$ (MJ/month/m²)</td>
<td>155</td>
<td>218.4</td>
<td>378.2</td>
<td>340</td>
<td>607.6</td>
<td>645</td>
<td>728.5</td>
<td>573.5</td>
<td>405</td>
<td>288.3</td>
<td>165</td>
<td>145.7</td>
</tr>
<tr>
<td>$T$ (°C)</td>
<td>1.7</td>
<td>5.3</td>
<td>8.5</td>
<td>15.7</td>
<td>19.1</td>
<td>20.5</td>
<td>21.3</td>
<td>24</td>
<td>20.3</td>
<td>13.2</td>
<td>7.7</td>
<td>4.1</td>
</tr>
</tbody>
</table>

In the case of the available collectors: $\eta_0 = 0.718$, $a_1 = 0.974$ W/m²°C, $a_2 = 0.004$ W/m²°C². Excess heat produced when the heating demand is small is considered to be stored in a seasonal storage system. Efficiency is considered to be linearly dependent on the difference between the internal temperature and ground temperature. Efficiency is assumed 0.9 when the internal temperature is 90 °C and ground temperature is 13 °C [13].

Heat request by the users $Q_u$ and heat supplied by the thermal plant $Q_F$ differ because of heat losses $Q_L$. Heat losses have been calculated by considering each branch.

\[ Q_L = \pi \cdot D \cdot L \cdot k \cdot (T - T_g) \cdot t \]  

(3)

where $k$ is the overall heat transfer coefficient and $T$ is the average temperature between outgoing and return network, $T_g$ is the ground temperature and $t$ is time period (a year).

Last term on numerator of equation (1) accounts for the primary energy association with electricity required for pumping, being $c_p$ the exergetic unit cost of electricity and $L_p$ is the annual electricity consumption, calculated as:

\[ L_p = \frac{1}{\eta_p} \int G \cdot v \cdot \Delta p \cdot dt \]  

(4)

where $\eta_p$ is the average pump efficiency, $G$ is the water mass flow rate, $v$ is the water specific volume (constant) and $\Delta p$ the total pressure losses due to pipe friction and localized resistances.

The last term on the right hand side of equation (1) is the cost of heat supplied to the users fed with alternative systems. This is obtained as the product of the exergetic unit cost of the heat produced with these systems ($c_a$) times the annual heat supplied to the users not connected with the district heating ($Q_a$). In the initial network configuration this term is zero.

Terms in equation (1) depend on the thermal load supplied by the network and on its extension. The possible area to be heated by the thermal plant must be chosen. This area can be divided into zones, each including one or more buildings. The number of zones should be selected as trade-off between result accuracy (large number of zones) and time required for design and calculation (small number of zones). For each zone, the total volume of buildings is determined. The thermal barycentre can be easily located in the area by considering the position of buildings and their respective volume (the geometric barycentre can be used as well, especially when the building structure is sufficiently regular). At this point, the network connecting the thermal plant with TBs can be traced.

The annual heat load of each single zone $Q_z$ is calculated by considering, for the whole heating season, the daily difference between the internal temperature (20 °C) and the external temperature, the average thermal transmittance of buildings (through walls, windows, floor, etc.), the number of daily heating hours (hh). The thermal transmittance of building can be multiplied for a shape factor defined as the ratio of external surface and building volume; this quantity, here indicated as $r$, expresses the volumetric heat losses per unit temperature difference. This value has been measured for several buildings; an average value of 0.9 W/(m³K) can be assumed. The annual heat load for a zone, in kWh, is then calculated as

\[ Q_z = \frac{r \cdot DTD \cdot hh \cdot V_z}{1000} \]  

(5)

where $V_z$ is the total volume of buildings in the zone and DTD is the summation of daily difference between internal and external temperature, calculated for the whole heating season (degree day). The number of daily heating hours is considered to be the same as for buildings with individual
heating system, which is established by law, depending on DTD. In the specific case analyzed in this paper, this quantity is about 2730 °C, being the heating season from the middle of October to the middle of April, while the number of heating hours is 12 per day.

The total heat load is calculated as summation of the contributions of all zones. The network operates for longer time than specified, mainly due to four causes: 1) non contemporary request by the users, 2) presence of particular users, like hospitals, that requires heat for more than 14 hours per day and for an extended period, 3) domestic water demand, 4) presence of users that requires heat in summer for air conditioning through absorption chillers. For all these reason, the total load calculated through equation (5) has been considered as spread on 18 hours per day in the seasonal heating, moreover the thermal flow outside this period has been assumed non null, but calculated on the basis of the thermal losses.

The cost of the network is calculated by considering each single branch and depends on its diameter. Internal diameter of pipes is calculated by first determining the mass flow rate in each branch. The mass flow rate is imposed by the thermal requirement of each user downstream that branch:

\[
\Phi = G \cdot (h_y - h_r)
\]  

where \( \Phi \) is the thermal flow provided to the users (the maximum load is considered in design), \( G \) the water mass flow rate, \( h_y \) and \( h_r \) the enthalpies of fluid feeding and returning from the users. The diameter is determined by imposing the maximum velocity \( v_{\text{max}} \) allowed in the pipes. This value is mainly defined on the basis of economic criterion, since friction losses and thus pumping cost depend on the square of velocity. On the other hand, a too low velocity would determine a large pipe diameter, thus high investment costs. In this analysis a value of 1.5 m/s is considered. The water mass flow rate \( G \) is expressed as:

\[
G = \rho \frac{\pi D_{\text{int}}^2}{4} \cdot v_{\text{max}}
\]

A thermoeconomic analysis is then implemented for the designed network, where all the possible users are connected. In particular, a useful approach that can be adopted for this purpose is that proposed by Valero and co-workers in the eighties [14, 15]. One of its main characteristics is the matrix based approach, in particular the use of incidence matrix for expressing the equation of cost conservation. The only auxiliary equation to be applied is the assignment of the same unit cost to the flow exiting each bifurcation [16].

The unit cost of a flow \( c \) can then be calculated, by dividing the costs for the corresponding exergy flow:

\[
c = \frac{B}{\Psi} \cdot 3600
\]

Where \( B \) is the exergetic cost of a general flow and \( \Psi \) its exergy.

At this point, the unit cost for each user, can be calculated. This cost is not the same for all of them because of the different exergy destruction (mainly due to friction) and the pipe cost associated to the different paths joining the thermal plant with the users.

The network is then optimized using a probabilistic approach similar to simulated annealing [17]. The probability of users to be disconnected to the district heating network is assumed to be dependent on their unit cost. In the optimization procedure, users are progressively disconnected from the network. Each iteration the user to be disconnected is randomly selected from an ensemble where the number of samples for each user is proportional to its probability. The users disconnected with the network are considered to be heated through the alternative system, which is, in this case a solar photovoltaic driven groundwater heat pump. The average COP of the heat pump is assumed equal to 4 in the case of unperturbed groundwater temperature. In the case of multiple installations, possible interferences between heat pumps are considered, as discussed in [18]. A simplified
expression for the effects of the distance \( d \) between an upstream installation on a downstream installation is assumed:

\[
COP = \text{COP}_0 \cdot (0.9 + 0.0108 \cdot \ln(d))
\]  

(9)

Since a probabilistic approach has been considered, the complete optimization procedure has been repeated several times in order to increase the probability to find the true optimum.

The entire procedure is similar in the case of economic costs, the only difference is that unit costs are expressed in monetary units. Costs of insulated pipes have been considered as in [10], while the cost of solar collectors and storage system have been taken from [13]. No incentives have been considered for solar energy.

In the cases where minimum primary energy and minimum economic cost are competing, the optimization has been performed by imposing a variable constraint on the maximum acceptable cost of heat (i.e. the economic objective function), so that the problem can be treated as single objective optimization. Once an optimal point is found, the optimization is repeated by modifying the maximum acceptable cost of heat.

3. Application

Figure 1 shows a schematic of the district heating network that has been considered as the case study. It is a network located in a small town in the north west part of Italy. The maximum thermal request is about 7 MW [19]. Heat to this network is supplied by an internal combustion engine (about 3 MW) and gas boilers. This case study is considered since it is a reasonable size of network that can be fed with renewable energy and because there is availability of groundwater to feed groundwater heat pumps, that can be considered as potential alternatives to the district heating network.

![Fig. 1. Schematic of the District Heating Network.](image)

The network shown in the figure corresponds to all the users connected to the district heating system. This superstructure is progressively simplified in order to discover the optimal configuration. The analysis is conducted by considering various combinations of the supply temperature and return temperature, which are here assumed as parameters in the analysis instead of design variables. Therefore, several optimal curves are obtained for each couple of these parameters. These results are shown in figure 2 for the following cases: 40-20 °C, 45-20 °C, 50-20 °C, 55-20 °C and 55-25 °C.
Results show that the two objective functions are competing for the values of the supply and return temperatures here considered. The lowest exergetic unit cost of heat is obtained in the case of smallest supply and return temperature. This also corresponds to the highest economic cost. This configuration corresponds to 54% of the users connected to the district heating network. This percentage refers to the annual heat demand with respect to the total heat demand of the users in the urban area. Increasing the supply temperature, the number of users connected to the district heating network in this condition (i.e. minimum exergetic cost of heat) increases. It becomes 67% in the case of supply temperature of 45 °C, 80% in the case of supply temperature of 50 °C and 93% in the case of supply temperature of 55 °C. The reason of such behaviour can be analyzed by considering the diagram in figure 3.

Fig. 2. Optimization results.

Fig. 3. Effect of weighted distance from the thermal plant to the unit cost of heat in (kWh/kWh).
Figure 3 shows the exergetic cost of heat associated to the users when they are all connected to the network. In the figure this is represented as the function of $L^*$, which is the ratio between the distance of the user from the thermal plant and the mass flow rate required by the user. The graph shows that when the users are far from the plant, the efficient use of primary energy decreases. The only exception is represented by very small users (i.e. small mass flow rate required to satisfy the thermal request) located quite close to the thermal plant.

When the supply temperature is increased from 40 °C (crosses) to 55 °C (circles), the behaviour remains the same, but the exergetic costs increases of about 3%. This is due to the increase in the term due to heat production, which is basically associated to the efficiency of solar collectors, which decreases (of about 4.5%) because of the larger operating temperature.

Starting from the points in figure 2 corresponding with all users connected with the network (for each series, these are the points on the left part of the diagram), it is possible to reduce the economic unit cost of heat by disconnecting some users from the network (those characterized with larger economic unit cost of heat) and supplying them heat with groundwater heat pumps.

This can be observed by analyzing the exergetic and economic unit costs of heat as the users are disconnected to the network. This is analyzed in figure 4 in the case of supply temperature of 55 °C and return temperature of 20 °C.

![Fig. 4. Trends of exergetic and economic unit costs during a iterative network simplification.](image)

The reasons why the exergetic unit cost tends to increase as the users are disconnected is that solar district heating is more efficient than the alternative. In addition, there are interactions between the various heat pump installations that affect their efficiency, as discussed above. In contrast, the economic cost tends to decrease. The minimum economic cost is obtained with few users still connected with the district heating system (about 6-10% of the annual heat, depending on the combination of temperatures). This is due to the interferences between heat pumps, that cause a reduction in the COP of downstream installation and thus an increase in the primary energy consumption.

Also, it is interesting to compare the unit costs corresponding to a fixed amount of heat supplied to users connected with the district heating network, for the various supply temperatures and fixed return temperature (20 °C). The amount of heat supplied through district heating network is considered to be 55% of the total annual request. Figure 5 shows that an increase in the supply
temperature causes an increase in the exergetic unit cost but a decrease in the economic unit cost. The latter is due to the reduction of the investment costs associated with heat storage and pipe network. Nevertheless, the economic advantage obtained increasing the supply temperature tends to decrease with increasing temperature, in fact the distance between points at fixed increase in the supply temperature tends to reduce.

![Graph showing unit costs for fixed users connected to the network as the function of supply temperature.](image)

**Fig. 5. Unit costs for fixed users connected to the network as the function of supply temperature.**

In the case of higher return temperature (e.g. 25 °C), the Pareto front presents sudden increase in the economic unit cost of heat with decreasing exergetic cost. It should be also mentioned that in the case of high temperatures (65-40 °C, 70-35 °C, 75-30 °C...) no Pareto front takes place and the optimal system is obtained with most users heated through groundwater heat pumps.

### 4. Conclusions

In this paper the energy and economic optimization of a district heating network is conducted using a thermoeconomic based probabilistic procedure. The procedure is applied to a small low temperature district heating network. Groundwater heat pumps are considered as the possible alternative systems to supply heat to the users not connected to the district heating network. A multi-objective optimization is performed for various combinations of the supply and return temperatures. The analysis shows that supply and return temperatures play a crucial role in the optimal configuration. In particular a reduction of both temperatures allows one to achieve smaller cost of heat in terms of required primary energy, but causes an increase in the economic costs. An increase in the return temperature causes an increase in both costs, which conducts to non competing objective functions.

The most important terms that affect to optimal configuration are the efficiency of solar collectors and the possible thermal interferences between heat pump, and, from the economic viewpoint only, the investment cost due to the seasonal thermal storage and the pipe network.
Appendix: cost calculation

Purchase cost for pre-insulated pipes has been calculated through the following equation [20]:

\[ C_{\text{pipe}} = (a_0 + a_1 \cdot D + a_2 \cdot D^2) \cdot 1.25 \cdot 2 \cdot L \]

where \( D \) is the internal diameter and \( L \) the length of the pipe, 1.25 is a corrective factor used to include the cost of special components also determined through available data and 2 accounts for the double pipe. The values of polynomial coefficients have been updated with respect to those available in [20]: \( a_0 = 11.7 \text{ €}/\text{m}, a_1 = 133.7 \text{ €}/\text{m}^2, 1575 \text{ €}/\text{m}^3 \).

Installation costs include the excavation (5.2 €/m$^3$) and pavement restoring (10.3 €/m$^2$).

Concerning heat generation, the following specific equipment costs are considered: solar collectors 250 €/m$^2$, photovoltaic panels 2500 €/kW, heat pumps 500 €/kW [21], seasonal storage tank 80 €/m$^3$ [13]. Linear cost functions have been considered for these components.

References


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Thermo-economic assessment of a micro CHP system fuelled by geothermal and solar energy

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Abstract:
A micro combined heat and power (CHP) plant operating through an Organic Rankine Cycle (ORC) using renewable energy is analyzed. The reference system is designed to produce 50 kWe. The heat sources of the system are geothermal energy at low temperature (80-100°C) and solar energy. The system uses a solar field composed only by evacuated solar collectors, and work is produced by a single turbine. Different working fluids (e.g. R134a, R236fa, R245fa) are considered in the analysis. The aim of this paper is to assess the cost of the proposed CHP plant and to determine the most convenient working fluid through a thermo-economic analysis. The system is sized in base of the weather data of a city in the centre of Italy in three different months (January, March, July), and the main characteristics of the system (i.e. heat exchanger surface, solar collector area) are presented. The results of the thermo-economic analysis show that R245fa allows the lowest price of electricity production and the lowest overall cost of the CHP plant.

Keywords:
Organic Rankine Cycle, Combined Heat and Power, geothermal energy, solar energy, thermo-economic analysis.

1. Introduction

Organic Rankine Power Cycles are well proven and reliable technology for energy conversion, particularly for exploiting low-temperature heat source. The use of an organic vapour in place of water steam is very interesting for small and medium size power plants (50 to 5000 kW), with applications varying from heat recovery at gas turbine discharge [1-3] or internal combustion engines [4], to energy conversion from biomass [5-6], solar [7-9] and geothermal resources [10-12] (these last two cases with no significant market alternative, considering the low level of temperature of the resource). Today, Organic Rankine Cycles are increasing in popularity with several manufacturers of equipment available on the market [13-17].

In scientific literature concerning ORC system fuelled by low-temperature geothermal resources, much research has been dedicated to the selection of the optimal working fluid. The few works on the geothermal hybrid systems [6,18-20] investigated only the exploitation of geothermal resource at medium temperature [6,18,20] for electricity production (with power plant size ranging from 1 MWe to 550 Mwe). Only [19] considered a power plant for electricity production fuelled by geothermal at low-temperature and another renewable energy resource, i.e. biomass. In a previous paper [21], the authors presented an innovative CHP ORC system powered by low-temperature geothermal resource (i.e. 90°C) and solar energy captured by solar collectors. In this paper, the authors will investigate the innovative CHP ORC system through a thermo-economic analysis. The system proposed has a small size (i.e. 50 kWe) because it designed for small CHP applications.
2. CHP ORC power plant description

The reference case here considered has been previously presented in [21]. Figure 1 represents the layout of the power plant. The liquid organic fluid coming from the condenser is first pre-heated in a heat exchanger (Geothermal Heater) by a geothermal flow rate at low temperature (90 °C). Then, solar energy collected by an Evacuated Tube Collectors (ETC) field is used to heat the organic fluid up to the maximum temperature, set at 147°C (420 K). This temperature reflects the interest of using solar collectors without concentrators or with a limited concentration ratio, which have a low cost for installed unit surface. After producing mechanical work in the turbine, the organic fluid is still in superheated condition (high temperature and enthalpy values). This heat can be recovered in building space heating and domestic hot water production (CHP unit). From CHP point of view, the de-super-heater DSH, that provides heat at higher temperature, but with limited heat capacity, should be separated from the condenser, which instead provides heat at low and constant temperature, but with infinite heat capacity. The DSH cools the organic fluid at turbine outlet from superheated conditions down to saturated vapour conditions at variable temperature. Then, before being circulated to the Geothermal Heater by the pump to start the cycle again (point 1, Fig. 1), the saturated organic vapour that leaves the DSH is condensed at constant temperature.

![Diagram of Low temperature geothermal and solar CHP-ORC layout](image)

**Figure 1 – Low temperature geothermal and solar CHP-ORC layout**

The system was modelled through Engineering Equation Solver (EES® [22]), using data acquired from a long term local data source [23] for Florence, Italy. The CHP plant was designed in three different months: January, March and July. These months are selected to cover three different cases...
for air temperature and global radiation. In January, the global radiation is the lowest of the year, and this leads to the highest area of solar collectors; on the contrary, in July the yearly highest global radiation implies the lowest area of solar collectors. The design in March, indeed, is an intermediate situation between January and July. The design conditions in terms of air temperature and global radiation were taken at 13:00 of a reconstructed standard day for each of the three months (Table 1). Three different engineered refrigerants suitable for low-temperature energy conversion were selected: R134a, R236fa, R245fa [6,14]. The critical temperatures and pressures of these analyzed fluids are reported in Table 2. The different shape of the cycle in the T-s diagram is shown in Figure 2, referring only to fluids R134a (left) and R245fa (right). Table 3 reports the parameters assumed for the calculations. The collector efficiency is modeled by the following quadratic approximation [24]:

$$\eta_{coll} = c_0 - \left( c_1 + c_2 \cdot (T_{avcoll} - T_{amb}) \right) \cdot \left( \frac{T_{avcoll} - T_{amb}}{G_{tot}} \right)$$  \hspace{1cm} (1)$$

Where $T_{avcoll}$ is the average temperature of the collector thermal fluid and $T_{amb}$ is the air temperature. The operating data for the evacuated solar collector reported in Table 3 are taken from [24] for an ESTEC VR12 CPC®. The temperature of the collector thermal fluid at collector outlet is set 10 °C above the maximum temperature of the superheated organic vapour (point [6]). Heat exchanger surfaces are calculated through the NTU-effectiveness method [25], and each overall heat transfer coefficient $U$ is assumed to be the same for all the fluids for each different heat exchanger (Table 3). This assumption is justified by the fact that the overall heat transfer coefficient of the heat exchangers is mainly affected by the thermal conductivity of the water, which is at least four times higher than the thermal conductivity of the organic fluid. For all the three fluids, the temperature difference of the water in all the heat exchangers is quite the same. In addition, the variation of the thermal conductivity among the three organic fluids is very low, so this means that the $U$ of each heat exchanger is independent from the organic fluid. In order to contain the surface of the geothermal heat exchanger, the temperature difference $T_{geouut} - T_1$ is set at high value (15 °C).

![Figure 2 – ORC thermodynamic cycle for R134a (left) and R245fa (right)](image)

**Table 1: Ambient design conditions for each month [20]**

<table>
<thead>
<tr>
<th>Month</th>
<th>January</th>
<th>March</th>
<th>July</th>
</tr>
</thead>
<tbody>
<tr>
<td>Global Irradiation on the 30° tilted surface $G_{tot}$ [W/m²]</td>
<td>690</td>
<td>931</td>
<td>1011</td>
</tr>
<tr>
<td>Ambient temperature $T_{amb}$ [°C]</td>
<td>6</td>
<td>11</td>
<td>26</td>
</tr>
</tbody>
</table>

**Table 2: Fluids critical temperature and pressure**

<table>
<thead>
<tr>
<th>Fluid</th>
<th>R134a</th>
<th>R236fa</th>
<th>R245fa</th>
</tr>
</thead>
<tbody>
<tr>
<td>Critical pressure [bar]</td>
<td>40.6</td>
<td>32</td>
<td>36.5</td>
</tr>
<tr>
<td>Critical temperature $T_0$ [°C]</td>
<td>101</td>
<td>125</td>
<td>154</td>
</tr>
</tbody>
</table>
### Table 3: ORC system parameters

<table>
<thead>
<tr>
<th>CYCLE</th>
<th>SOLAR</th>
</tr>
</thead>
<tbody>
<tr>
<td>Turbine Power output $W_t$ [kW]</td>
<td>50</td>
</tr>
<tr>
<td>Condenser temperature $T_0$ [°C]</td>
<td>45</td>
</tr>
<tr>
<td>Maximum cycle temperature $T_6$ [°C]</td>
<td>147</td>
</tr>
<tr>
<td>Pump/Turbine isentropic efficiency</td>
<td>0.8</td>
</tr>
<tr>
<td>ECO - U [kW/(m²·°C)]</td>
<td>0.25</td>
</tr>
<tr>
<td>EVA - U [kW/(m²·°C)]</td>
<td>0.20</td>
</tr>
<tr>
<td>SH - U [kW/(m²·°C)]</td>
<td>0.125</td>
</tr>
</tbody>
</table>

### GEOTHERMAL

<table>
<thead>
<tr>
<th>Fluid</th>
</tr>
</thead>
<tbody>
<tr>
<td>Geothermal temperature ($T_{\text{geoin}}$) [°C]</td>
</tr>
<tr>
<td>Temperature difference $T_{\text{geoin}}$-$T_2$ [°C]</td>
</tr>
<tr>
<td>Temperature difference $T_{\text{geoin}}$-$T_1$ [°C]</td>
</tr>
<tr>
<td>GEO HX U [kW/(m²·°C)]</td>
</tr>
</tbody>
</table>

###CONDENSER

<table>
<thead>
<tr>
<th>Fluid</th>
</tr>
</thead>
<tbody>
<tr>
<td>Refrigerant inlet temperature [°C]</td>
</tr>
<tr>
<td>Temperature difference pinch point [°C]</td>
</tr>
<tr>
<td>DSH U [kW/(m²·°C)]</td>
</tr>
</tbody>
</table>

The main parameters of the CHP plant calculated for the value of $p[1]$ (upper cycle pressure) that maximizes the cycle efficiency are reported in Table 4.

### Table 4: Main parameters of the CHP plant

<table>
<thead>
<tr>
<th>Fluid</th>
<th>R134a</th>
<th>R236fa</th>
<th>R245fa</th>
</tr>
</thead>
<tbody>
<tr>
<td>Upper cycle pressure $p[1]$ [bar]</td>
<td>38</td>
<td>29.3</td>
<td>31</td>
</tr>
<tr>
<td>Condenser pressure $p[0]$ [bar]</td>
<td>11.6</td>
<td>5</td>
<td>2.92</td>
</tr>
<tr>
<td>Geothermal reinjection temperature [°C]</td>
<td>335</td>
<td>336</td>
<td>335</td>
</tr>
<tr>
<td>Temperature at DSH inlet $T[8]$ [°C]</td>
<td>98</td>
<td>93</td>
<td>62</td>
</tr>
<tr>
<td>Water temperature at DSH outlet [°C]</td>
<td>93</td>
<td>87</td>
<td>57</td>
</tr>
<tr>
<td>Organic fluid flow rate [kg/s]</td>
<td>1.77</td>
<td>1.84</td>
<td>1.33</td>
</tr>
<tr>
<td>Geothermal flow rate [kg/s]</td>
<td>0.95</td>
<td>0.87</td>
<td>0.60</td>
</tr>
<tr>
<td>Water flow rate at DSH [kg/s]</td>
<td>0.45</td>
<td>0.43</td>
<td>0.32</td>
</tr>
<tr>
<td>Water flow rate at Condenser [kg/s]</td>
<td>22.3</td>
<td>19.72</td>
<td>18.93</td>
</tr>
<tr>
<td>Geothermal power input $Q_{\text{geo}}$ [kW]</td>
<td>111</td>
<td>98.5</td>
<td>72.4</td>
</tr>
<tr>
<td>Solar power input $Q_{\text{solar}}$ [kW]</td>
<td>316</td>
<td>273.5</td>
<td>235</td>
</tr>
<tr>
<td>DSH heat recovered [kW]</td>
<td>102</td>
<td>82.5</td>
<td>23</td>
</tr>
<tr>
<td>Condenser heat recovered [kW]</td>
<td>280</td>
<td>247</td>
<td>237.5</td>
</tr>
</tbody>
</table>
3 CHP ORC thermo-economic analysis

The exergy analysis of the power plant was performed following the reference literature [26-28]. Since it is not a reactive system, the following classical definition of exergy was used:

\[ ex = h - h_0 - T_0 \cdot \left( s - s_0 \right) \quad (2) \]

where \( h_0 \) and \( s_0 \) are the specific enthalpy and the specific entropy of the element at the reference state, which was fixed at a pressure of 1 atm and a temperature given in Table 1 for each month.

The exergy inputs to the system come from (I) geothermal and (II) sun. The exergy from the sun is given by:

\[ Ex_{sun} = G_{sol} \cdot A_{coll} \cdot \left[ 1 - \frac{T_{amb}}{T_{sun}} \right] \quad (3) \]

where \( T_{sun} \) is taken as 75% of the equivalent black-body sun temperature, in agreement with [9, 29].

The thermo-economic analysis was developed applying the following thermo-economic balance for each component of the system:

\[ C_e + C_{Qe} + C_{We} = C_i + C_{Qi} + C_{Wi} + Z_{comp} \quad (4) \]

Where on the left side there are the costs per second (€/s) related to all the mass (\( C_e \)), heat (\( C_{Qe} \)) and work (\( C_{We} \)) fluxes that enter the component, on the right side there are the costs per second related to all the mass (\( C_i \)), heat (\( C_{Qi} \)) and work (\( C_{Wi} \)) fluxes that exit the component and also the cost per second of the component (\( Z_{comp} \)). \( C_e \), \( C_Q \) and \( C_W \) are calculated as follow:

\[ C = c \cdot \dot{E} \quad (5) \]

\[ C_Q = c_Q \cdot \dot{Q} \quad (6) \]

\[ C_W = c_W \cdot \dot{W} \quad (7) \]

Where \( c \), \( c_Q \) and \( c_W \) are the exergy specific cost respectively for mass flow rate, heat \( \dot{Q} \) and work \( \dot{W} \), while \( \dot{E} \) is defined as the product of the mass flow rate and the exergy:

\[ \dot{E} = m \cdot ex \quad (8) \]

The cost per second of the component \( Z_{comp} \) includes the cost of the component and costs for operation and maintenance (O&M). In the calculation, it was assumed that the system works for 15 years for 6000 hours per year. The following Table 5 reports the thermo-economic balance according to (4-8) for each component of the system. Differently from all the other points of the system, the exergy of the geothermal mass flow rate \( m_{geo} \) at the depth of 700 m - which is involved in the calculation of \( E_{well} \) through (8) - takes into account also the potential exergy \( -m_{geo}gz \), where \( g \) is the gravitational constant, \( z \) is the depth of the well and the sign minus indicates that level of zero potential exergy is set at the CHP plant level (i.e. sea level).
Table 5: Thermo-economic balance for each system component.

<table>
<thead>
<tr>
<th>COMPONENT</th>
<th>THERMO-ECONOMIC BALANCE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Geothermal wells</td>
<td>$E_{well} \cdot c_{well} = Z_{well}$</td>
</tr>
<tr>
<td>ORC Pump</td>
<td>$E_0 \cdot c_0 + \frac{W_p}{\eta_p} \cdot c_p + Z_p = E_1 \cdot c_1$</td>
</tr>
<tr>
<td>Geothermal pump</td>
<td>$E_{well} \cdot c_{well} = E_{geo} \cdot c_{geo} + W_{p,geo} \cdot c_p + Z_{p,geo}$</td>
</tr>
<tr>
<td>Geothermal heater</td>
<td>$E_{geo} \cdot c_{geo} + E_1 \cdot c_1 + Z_{geo} = E_{geoout} \cdot c_{geoout} + E_2 \cdot c_2$</td>
</tr>
<tr>
<td>Solar Collectors ETC</td>
<td>$E_{ain} \cdot c_{ain} = E_{aout} \cdot c_{aout} + Z_{coll} + E_{sun} \cdot c_{Qsol}$</td>
</tr>
<tr>
<td>ECO+EVA+SH</td>
<td>$E_{ain} \cdot c_{ain} + E_2 \cdot c_2 + Z_{solar} = E_{aout} \cdot c_{aout} + E_6 \cdot c_6$</td>
</tr>
<tr>
<td>Turbine</td>
<td>$Z_{t} + E_6 \cdot c_6 = W_t \cdot c_t + E_1 \cdot c_7$</td>
</tr>
<tr>
<td>Desuperheater</td>
<td>$Z_{dsh} + E_7 \cdot c_7 = E_Q \cdot c_Q + E_8 \cdot c_8$</td>
</tr>
<tr>
<td>Condenser</td>
<td>$E_8 \cdot c_8 + Z_{cond} = E_0 \cdot c_0 + E_{Q,cond} \cdot c_{Q,cond}$</td>
</tr>
</tbody>
</table>

Using the equations of Table 5 it is possible to calculate $c_1$, which is the cost of the energy produced by turbine, and $c_Q$ and $c_{Q,cond}$ which are respectively the cost of the heat released at the DSH and at the condenser. In order to solve the system of equations of Table 5 some auxiliary equations are needed. First of all, the following equations are added:

$c_6 = c_7 = c_8 = c_0$  \hspace{1cm} (9)

$c_{geoout} = c_{geo} = c_{pumpgeo}$  \hspace{1cm} (10)

$c_{aout} = c_{ain}$  \hspace{1cm} (11)

Then, the cost of the sun energy $c_{Qsol}$ is supposed to be zero, since it can be taken for free. However, the cost of the solar collectors $Z_{coll}$ is considered in the calculation. The power to run the ORC and geothermal pumps is supposed to be given by the electrical grid. Then, the cost of the electricity $c_p$ was taken from the Italian market database for 2011 [30] for each month studied. The $c_p$ values were fixed at: 76.1 €/MWh for January, 63.87 €/MWh for March, 87.26 €/MWh for July.

4. Results

The heat exchanger surfaces for each fluid are reported in Table 6: they are not influenced by the variation of the design month, since the operating temperature and pressure are fixed. Instead, the effective area of the solar collectors is affected by the design month, and it decreases as the solar radiation $G_{tot}$ increases (i.e. from January to July).

Table 6: Heat exchanger surface in each month studied (January, March, July).

<table>
<thead>
<tr>
<th>Fluid</th>
<th>R134a</th>
<th>R236fa</th>
<th>R245fa</th>
</tr>
</thead>
<tbody>
<tr>
<td>Geothermal Heater surface [m²]</td>
<td>79</td>
<td>53</td>
<td>31</td>
</tr>
<tr>
<td>ECO surface [m²]</td>
<td>35</td>
<td>31</td>
<td>14</td>
</tr>
<tr>
<td>EVA surface [m²]</td>
<td>37</td>
<td>32</td>
<td>58</td>
</tr>
<tr>
<td>SH surface [m²]</td>
<td>40</td>
<td>34</td>
<td>4</td>
</tr>
<tr>
<td>DSH surface [m²]</td>
<td>224</td>
<td>100</td>
<td>37</td>
</tr>
<tr>
<td>Condenser surface [m²]</td>
<td>146</td>
<td>129</td>
<td>124</td>
</tr>
</tbody>
</table>

326
Table 7: Solar Collectors effective area [m$^2$] for each fluid in each month (January, March, July).

<table>
<thead>
<tr>
<th>Fluid</th>
<th>January ETC area [m$^2$]</th>
<th>March ETC area [m$^2$]</th>
<th>July ETC area [m$^2$]</th>
</tr>
</thead>
<tbody>
<tr>
<td>R134a</td>
<td>555</td>
<td>482</td>
<td>413</td>
</tr>
<tr>
<td>R236fa</td>
<td>411</td>
<td>356</td>
<td>305</td>
</tr>
<tr>
<td>R245fa</td>
<td>378</td>
<td>328</td>
<td>281</td>
</tr>
</tbody>
</table>

The cost of the geothermal and ORC pumps were taken from commercial catalogue [33,34]. The two geothermal wells were assumed to be deep 700 m with a cost of 50 € per meter of perforation. Hence, $Z_{well}$ is the cost of the two wells, and it computes the cost for making the geothermal resource available to the use in the CHP plant. The geothermal pump, instead, consumes power in order to bring the geothermal resource from the 700 m of depth up to the CHP system.

For the cost of the ETC, we overlooked the high price (598 €/m$^2$) reported in [24]. The price was set at 187 €/m$^2$ which is an average value of the price reported in [35-37]. The O&M cost was fixed at 5% of the component cost for the heat exchangers and the pumps, and at 3% for the solar collectors. Table 8 contains all the costs of the plant components that are not influenced by the design month, while Table 9 reports the costs of the solar collectors for each fluid for the three design months. Table 10 summarizes the overall cost of the power plant for each fluids in each design month.

Table 8: ORC component cost [k€] for each fluid (constant for every month).

<table>
<thead>
<tr>
<th>Fluid</th>
<th>R134a</th>
<th>R236fa</th>
<th>R245fa</th>
</tr>
</thead>
<tbody>
<tr>
<td>Geothermal Heater cost [€]</td>
<td>13.170</td>
<td>8.830</td>
<td>5.100</td>
</tr>
<tr>
<td>ECO+EVA+SH cost [€]</td>
<td>24.640</td>
<td>20.810</td>
<td>16.645</td>
</tr>
<tr>
<td>DSH cost [€]</td>
<td>98.560</td>
<td>21.900</td>
<td>5.475</td>
</tr>
<tr>
<td>Condenser cost [€]</td>
<td>87.600</td>
<td>76.650</td>
<td>65.700</td>
</tr>
<tr>
<td>Geothermal wells [€]</td>
<td>70.000</td>
<td>70.000</td>
<td>70.000</td>
</tr>
<tr>
<td>ORC pump [€]</td>
<td>4.000</td>
<td>4.000</td>
<td>4.000</td>
</tr>
<tr>
<td>Turbine [€]</td>
<td>50.000</td>
<td>50.000</td>
<td>50.000</td>
</tr>
</tbody>
</table>

Table 9: Solar Collectors cost [k€] for each fluid in each month (January, March, July).

<table>
<thead>
<tr>
<th>Fluid</th>
<th>January cost [€]</th>
<th>March cost [€]</th>
<th>July cost [€]</th>
</tr>
</thead>
<tbody>
<tr>
<td>R134a</td>
<td>107.230</td>
<td>93.000</td>
<td>79.740</td>
</tr>
<tr>
<td>R236fa</td>
<td>79.320</td>
<td>68.780</td>
<td>58.980</td>
</tr>
<tr>
<td>R245fa</td>
<td>73.050</td>
<td>63.350</td>
<td>54.300</td>
</tr>
</tbody>
</table>

Table 10: ORC plant overall cost [k€] for each fluid in each design month (January, March, July).

<table>
<thead>
<tr>
<th>Month</th>
<th>January</th>
<th>March</th>
<th>July</th>
</tr>
</thead>
<tbody>
<tr>
<td>R245fa [€]</td>
<td>318.200</td>
<td>297.450</td>
<td>292.780</td>
</tr>
<tr>
<td>R236fa [€]</td>
<td>366.670</td>
<td>342.450</td>
<td>337.020</td>
</tr>
<tr>
<td>R134a [€]</td>
<td>476.660</td>
<td>448.760</td>
<td>442.510</td>
</tr>
</tbody>
</table>

The overall cost of the ORC plant with R134a is approximately 50% higher than the cost with R245fa, which has the lowest cost in each design month. This result is due to the higher heat exchangers surfaces needed by R134a in comparison to R245fa (see Table 6). The ORC plant overall cost diminishes when design month passes from January to July, and this result is due to the lower ETC surface (Table 7) that decreases the overall costs for solar collectors (Table 9).
Table 11 reports the cost in €/MWh of the work produced by the turbine $c_t$ for each fluid in each design month. R245fa is still the fluid with the lowest cost for MWh produced, while R134a presents the highest cost. These results are in agreement with the results showed in Table 10 for the overall cost of the ORC plant. Besides, the lowest value of $c_t$ is obtained in March for each fluid. This result is justified by the fact that at March there is the proper balance between ambient temperature and global solar radiation increase. High value of solar radiation allows to strongly decrease the ETC cost (Table 9), which diminishes of almost 30 k€ from January to March, but less than 8 k€ from March to July. At the same time, an ambient temperature of 11°C allows to control the exergy losses of the solar collectors and of the plant (Figure 3).

Table 11: specific cost of the work produced by the turbine [€/MWh] for each fluid in each design month (January, March, July).

<table>
<thead>
<tr>
<th>Month</th>
<th>January</th>
<th>March</th>
<th>July</th>
</tr>
</thead>
<tbody>
<tr>
<td>R245fa [€/MWh]</td>
<td>39</td>
<td>37</td>
<td>46</td>
</tr>
<tr>
<td>R236fa [€/MWh]</td>
<td>43</td>
<td>41</td>
<td>52</td>
</tr>
<tr>
<td>R134a [€/MWh]</td>
<td>41</td>
<td>39</td>
<td>49</td>
</tr>
</tbody>
</table>

Figure 3: Solar Collectors exergy losses for each fluid in each month (January, March, July).

Table 12 reports the cost in €/MWh of the heat released at the DSH, $c_Q$, for each fluid in each design month. R245fa has the lowest cost for MWh of heat produced, while R134a presents the highest cost. This result is given by the fact that for R245fa the DSH surface (Table 6), and consequently its cost (Table 8), is much lower in comparison to R134a. Instead, the values of $c_Q$ for R245fa and R236fa are similar because the slight higher cost of DSH for R236fa is almost compensated by the largest amount of heat recovered. Instead, at the condenser (Table 13), the highest specific cost is obtained with R245fa, because the quantity of heat recovered at the condenser with R245fa is lower than the heat recovered with R134a, while the costs are similar (Tables 4 and 8). Table 14 summarizes the specific cost of all the fluxes for each fluid in each design month involved in the thermo-economic analysis.

Table 12: specific cost of the heat released at the DSH [€/MWh] for each fluid in each design month (January, March, July).

<table>
<thead>
<tr>
<th>Month</th>
<th>January</th>
<th>March</th>
<th>July</th>
</tr>
</thead>
<tbody>
<tr>
<td>R245fa [€/MWh]</td>
<td>49</td>
<td>51</td>
<td>81</td>
</tr>
<tr>
<td>R236fa [€/MWh]</td>
<td>54</td>
<td>55</td>
<td>81</td>
</tr>
<tr>
<td>R134a [€/MWh]</td>
<td>99</td>
<td>106</td>
<td>154</td>
</tr>
</tbody>
</table>
Table 13: specific cost of the heat released at the condenser [€/MWh] for each fluid in each design month (January, March, July).

<table>
<thead>
<tr>
<th>Month</th>
<th>January</th>
<th>March</th>
<th>July</th>
</tr>
</thead>
<tbody>
<tr>
<td>R245fa [€/MWh]</td>
<td>49</td>
<td>51</td>
<td>78</td>
</tr>
<tr>
<td>R236fa [€/MWh]</td>
<td>45</td>
<td>46</td>
<td>60</td>
</tr>
<tr>
<td>R134a [€/MWh]</td>
<td>43</td>
<td>43</td>
<td>56</td>
</tr>
</tbody>
</table>

Table 14: specific cost $c_i$ [€/MWh] of all the fluxes involved in Table 3 for each fluid in each design month (January, March, July).

<table>
<thead>
<tr>
<th>Fluid</th>
<th>R134a</th>
<th>R236fa</th>
<th>R245fa</th>
</tr>
</thead>
<tbody>
<tr>
<td>Month</td>
<td>January</td>
<td>March</td>
<td>July</td>
</tr>
<tr>
<td>$c_1$ [€/MWh]</td>
<td>34</td>
<td>31</td>
<td>40</td>
</tr>
<tr>
<td>$c_2$ [€/MWh]</td>
<td>38</td>
<td>36</td>
<td>47</td>
</tr>
<tr>
<td>$c_6 = c_7 = c_8 = c_0$ [€/MWh]</td>
<td>27</td>
<td>26</td>
<td>34.5</td>
</tr>
<tr>
<td>$c_{ain} = c_{aout}$ [€/MWh]</td>
<td>13</td>
<td>10</td>
<td>10.6</td>
</tr>
<tr>
<td>$c_p$ [€/MWh]</td>
<td>76.1</td>
<td>67.8</td>
<td>87.3</td>
</tr>
<tr>
<td>$c_{geoin} = c_{geous}$ [€/MWh]</td>
<td>36</td>
<td>40</td>
<td>67</td>
</tr>
<tr>
<td>$c_{well}$ [€/MWh]</td>
<td>22</td>
<td>26</td>
<td>46</td>
</tr>
</tbody>
</table>

5. Conclusions

In this work, a thermo-economic analysis of a new micro CHP ORC system fuelled by two renewable energy resources (solar and low-temperature geothermal) was presented. The system was sized in base of the weather data of a city in the centre of Italy in three different months (January, March, July). The thermo-economic performance of three different working fluids (e.g. R134a, R236fa, R245fa) were compared. The results showed that the plant operating with R245fa is the less expensive, due to the fact that requires the lower surface of heat exchangers and the lower solar collector area. R245fa is also the most convenient working fluid in terms of cost of power produced by the system. In terms of heat recovered from the CHP system, the cost of the heat recovered at high temperature is less expansive for R245fa, while R134a present the lowest cost of the heat recovered at low temperature. Finally, the results also showed that the lowest cost of the CHP system for all the fluid is obtained in March, when there is the proper balance between the ambient temperature and the global solar radiation.

Acknowledgements

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Nomenclature

- $A_{coll}$: Solar collector area [m²]
- $C$: Cost [€/s]
- $c$: Exergy specific cost of the material streams [€/kJ]
- $c_Q$: Exergy specific cost for the transfer of heat [€/kJ]
cw: Exergy specific cost for the transfer of power [€/kJ]
E: Exergy [kJ/s]
ex: Specific Exergy [kJ/kg]
E_{sun}: Exergy from the sun [kW]
G_{tot}: Global radiation on the tilted surface
h: Enthalpy [kJ/kg]
m: Mass flow rate [kg/s]
p: Pressure [bar]
Q: Heat rate [kW]
Q_{solar}: Solar radiation incident to collector [kW]
s: Entropy [kJ/kgK]
T: Temperature [K]
T_{avcoll}: Average temperature of the collector thermal fluid [K]
T_{sun}: Temperature of the sun [K]
U: Global Heat exchange coefficient [kW/m^2-K]
x: Quality
W: Work [kW]
W_{t}: Power output of the cycle [kW]
Z: Component cost [€]
[1]...[n]: Thermodynamic point of the cycle
\eta_{coll}: Collector efficiency
\eta: Isentropic efficiency

Suffixes
ain: Collectors outlet
amb: Ambient
aout: Collectors inlet
coll: Collector
cond: Condenser
e: Component exit
goin: Geothermal inlet to the system
gout: Geothermal reinjection into the well
go: Geothermal
i: Component inlet
p: Pump
solarhx: Solar Heat exchanger, i.e. ECO+EVA+SH
t: Turbine
well: Geothermal well

Acronyms
CEPCI: Chemical Engineering’s Plant Cost Index
CHP: Combined Heat and Power
COND: Condenser
REFERENCES


[34] Danfoss Italia catalogue, available at: www.danfoss.com/Italy.


Thermo-Economic Evaluation and Optimization of the Thermo-Chemical Conversion of Biomass into Methanol

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Abstract:
In a carbon and resources constrained world, thermo-chemical conversion of lignocellulosic biomass into fuels and chemicals is regarded as a promising alternative to fossil resources derived products. Methanol is one potential product which can be used for the synthesis of various chemicals or as a fuel in fuel cells and internal combustion engines. This study focuses on the evaluation and optimization of the thermodynamic, economic and environmental performance of methanol production from biomass by applying process integration and optimization techniques. Results show the importance of the energy integration and in particular of the cogeneration of electricity for the improvement of the overall efficiency. The energy efficiencies of the evaluated scenarios reach up to 45% and chemical efficiencies up to 51%.

Keywords:

1. Introduction

Methanol was produced since the early 1800s through the distillation of wood to make ‘wood alcohol’. This method was replaced in the 1920s by large scale processes producing methanol from hydrogen and carbon oxides mixture obtained from the incomplete combustion and reforming of fossil fuels. Today, Methanol is produced mainly by reforming of natural gas, naphtha or refinery light gas [1]. Other ways to produce methanol, which are currently being investigated, include direct methane oxidation without the intermediate step of syngas production and reductive hydrogenative recycling of CO\textsubscript{2}, requiring hydrogen, but providing a way to use CO\textsubscript{2}.

The price of methanol is about 16 €\textsubscript{2011}/GJ\textsuperscript{1}, while as a reference, the OPEC basket price of crude oil is 107.46 US$/bbl, or approximately 14.7 €\textsubscript{2011}/GJ. The recent peaks in methanol prices at the end of 2007 and 2008 are mainly due to plants shutting down for scheduled maintenance but also to the increasing demand in the growing economies. High natural gas prices, especially in North America, have also driven the prices up but the subsequent slowdown of the world economy helped lowering the prices [3].

Methanol is mainly used as a feedstock for the synthesis of other products. Being a fuel with an octane number of 100, it can be blended with gasoline as an oxygenated additive or used directly in internal combustion engines with only minor modifications [3]. Furthermore, methanol can be used to produce electricity in direct methanol fuel cells (DMFC) or DME which can be used as a substitute to diesel fuel as well as household gas for cooking and heating, and gasoline (Methanol to Gasoline Synthesis) [4].

Previous studies have analyzed the technical and economic feasibility of the thermo-chemical conversion of biomass into methanol, exploring different scenarios. A comparison of the biomass

\textsuperscript{1} Values calculated considering 20.094 MJ/kg as the low heating value of methanol. Energy values are always expressed in terms of the LHV unless otherwise stated.
derived methanol production costs is carried out by Spath and Dayton [5] who reported values varying from 10 - 19.6 $/GJ\text{HHV} (28-54 €/GJ)^1 for the study of Wyman et al. [6] to 9 - 12 $/GJ\text{HHV} (18-23 €/GJ)^1 for the study of Hamelinck et al. [7]. Hamelinck et al. investigated promising conversion concepts and compared different types of gasifiers and gas cleaning steps obtaining overall HHV (High Heating Value) energy efficiency of 55%. More recently the models developed by Van Rens et al. [8] and Huisman et al. [9] addressed two scenarios: a present day design relying on proven technologies (though not on commercial scale for biomass applications) and a near future design studied within the CHRISGAS project [10] in particular for syngas cleaning and conditioning. Their results show, for the present day design relying on an oxygen/steam blown circulating fluidized bed gasifier, an energy efficiencies of 47.8% and a chemical efficiency of 50% (without considering the heat available for district heating). These evaluations focus on case-studies operating at fixed conditions, for which only limited process integration has been taken into consideration. Comparing the results of the various studies on a common basis is a very difficult task, because of the different technologies considered, the different assumptions made and the degree of process integration. The objective of the present work is to systematically investigate the thermo-chemical conversion of lignocellulosic biomass into methanol applying multi-objective optimization techniques and including a detailed heat integration model to evaluate the potential for heat recovery and valorization.

2. Methodology

The present work is based on a model superstructure that was previously developed to analyze and compare the thermo-chemical conversion of lignocellulosic biomass into syngas and liquid fuels (FT, MeOH, DME) [12,13]. The analysis is completed here by applying the thermo-economic process optimization methodology described by Gassner et al. [13], focusing on the gasification technology. An EF (Entrained Flow) gasifier and a FICFB (Fast Internally Circulating Fluidized Bed) gasifer are compared, as represented in Figure 1. The superstructure is built of single process-units thermo-economic models that can be assembled to systematically study different process configurations. The thermo-chemical models are developed using commercial flowsheet calculation software Belsim-Vali [14] providing the chemical transformation and the heat requirements of the process units. These models are coupled with the economic and the energy integration models. The energy integration model computes the minimum energy requirements of the process using the mass balance between the unit operations and the heat cascade as constraints. If the combustion of the waste streams is not sufficient to provide the heat requirement above the pinch, selected process streams may be used as fuels. Available excess heat can be recovered in a Rankine cycle producing electricity. The energy integration model is detailed in [13], [15]. The economic model evaluating the profitability of the plant is based on equipment sizing and costing taking into account the operating conditions. The superstructure approach allows a flexible and systematic analysis of different process configurations. Sets of optimal design solutions are generated by the simultaneous optimization of the process in terms of thermodynamic performance and economic performance as a function of the decision variables.

3. Process Description

The overall process of thermo-chemical conversion of biomass into liquid fuels consists of: feed preparation, gasification, gas cleaning and treatment, and fuel synthesis and purification. Figure 1 represents the unit operations of the process, as well as the energy integration options. It is focused on...
on two scenarios shown in the superstructure; the first, employing a FICFB gasifier and the second, an EF gasifier. These scenarios are referred to as FICFB and EF scenario.

Figure 1 Simplified process superstructure for MeOH production, including the energy integration options (full arrow: heat exchange, double arrow: steam and oxygen, broken arrow: possible streams that could be used as additional fuels).

3.1. Thermo-Economic Models

The thermo-economic models used in this study are based on the work of Tock et al. [16]. The models used for biomass pre-treatment, gasification and gas cleaning are identical to those previously developed by Gassner et al. [12,18].

3.1.1. Thermo-Chemical Conversion Models

The biomass supplied to the process (50% humidity) is dried in an air drying unit which is optimized in respect to the residual humidity $\Phi_{\text{d,wood}}$ and the inlet air temperature $T_{\text{d}}$. The dried biomass is then directly grinded for FICFB gasification or torrefied ($T_{\text{T, out}} = 260^\circ \text{C}$) in order to be pulverized as required by the EF gasification. The model of the torrefaction unit is based on simple conversion ratios [17].

The FICFB gasifier consists of an indirectly heated circulating fluidized bed where the heat required for gasification is provided by circulating the bed material between two physically separated combustion and gasification chambers. The model of the FICFB gasifier is described in detail by Gassner et al. [18]. In the combustion chamber ungasified char and fuels are oxidized with air to heat up the bed material which is transferred via a cyclone to a gasification chamber where steam reacts with the biomass feed to produce the syngas. The advantage of this gasification technology is that it produces an essentially nitrogen-free product gas without requiring air separation for the oxygen supply. The main disadvantages arise from the methane and tar content of the synthesised gas and from the high investments costs due to the complicated construction. A directly heated high temperature stage (HT stage) is introduced to reduce the methane and hydrocarbon content in the product gas through autothermal steam methane reforming. The heat for the endothermic reforming is thus satisfied by partial oxidation with pure oxygen.

In the EF gasifier, the pulverized feed is entrained with the reacting gases, solid particles and gases move at approximately the same velocity. Consequently, smaller particles are required making the torrefaction step necessary. In this case, gasification is carried out using both oxygen and steam, and heat is provided directly by the oxidation of the feed. Advantages of this technology are the high capacity per unit volume (especially for the pressurized reactors) and the simpler geometry [19] (relatively to a fluidized bed). Because of the high temperature ($1350^\circ \text{C}$), the product gas is almost
tar-free and a leach resistant molten slag is produced [20]. Disadvantages include the high oxygen consumption and a higher conversion of the energy of the feed into sensible heat [21].

The main operating conditions of the FICFB gasifier and the EF gasifier are summarized in Table 1. For both gasification scenarios the product gas is quenched with steam to a temperature of 800°C. In the gas cleaning step the product gas is cooled to 150°C before entering the filter and the scrubber where it is cooled to atmospheric temperature. The water gas shift (WGS) reactor and the acid gas removal (AGR) step are used to bring the synthesis gas to the specifications required for the synthesis of methanol, that is to a stoichiometric ratio \( s = (\text{H}_2 - \text{CO}_2)/(\text{CO} + \text{CO}_2) \) of 2. The exothermic WGS reaction produces extra \( \text{H}_2 \) and \( \text{CO}_2 \) at the expense of \( \text{CO} \) by the addition of steam. It has been shown that, for kinetic reasons and in order to control by-products, a value slightly greater than two is preferred [5]. The \( \text{CO}_2 \) concentration is typically adjusted to 4-8% for optimal activity and selectivity [5,22]. Furthermore, an excess of steam is required to allow an almost complete conversion of \( \text{CO} \) and to push the reaction away from solid carbon formation. The molar steam to carbon ratio is usually between 2-6, depending on the feedstock and reactor conditions [5]. In the model a steam to carbon (mainly \( \text{CO} \)) ratio of 2.5 was used. The absence of solid carbon at reactor conditions and at thermodynamic equilibrium is verified using the software Gemini [23], which calculates the equilibrium composition by minimization of the Gibbs energy of the system. The absence of solid carbon is verified at equilibrium but this does not guarantee that any carbon soot is produced. In order to maintain the steam to \( \text{CO} \) ratio and obtain the required gas compositions only part of the stream needs to be shifted. The WGS unit is modeled as a single intermediate temperature reactor optimized with regard to the water gas shift reaction temperature \( T_{\text{WGS}} \) and the inlet temperature \( (T_{\text{in,WGS}} = T_{\text{WGS}} - \Delta T_{\text{WGS}}) \). The AGR step is modeled taking into consideration the values for the energy integration and economic analysis for chemical absorption as described by Tock [24]. Methanol synthesis is modeled by a multistage reactor with four beds in series [12, 23, 24] and it is optimized in respect to the synthesis gas inlet temperature \( T_{\text{m,in}} \) and the reactors temperature and pressure \( (T_m, P_m) \). A fraction of the off-gases \( (R_m = 0.95) \) is recycled into the synthesis reactors, to increase methanol conversion. In order to increase the purity of the produced methanol a final purification step is required. Two distillation columns allow achieving a methanol purity of over 99% (Tock [24]). Oxygen is required for the EF gasifier and for the directly heated high temperature stage of the FICFB gasifier. Oxygen for gasification is conventionally produced by pressure swing adsorption or cryogenic distillation. An ASU (Air Separation Unit) is not included in the current model superstructure but it will be integrated in future studies. The energetic and economic costs of the oxygen supply have been considered. From an economic standpoint oxygen is considered as a utility, purchased at the price indicated by Kirschner [19]. The energetic cost of oxygen is taken into account considering an electricity consumption of 1080 kJ/kg\( \text{O}_2 \) for off-site oxygen production. Oxygen is delivered at standard ambient temperature and pressure. The decision variables relative to the integration of the steam network, for the heat integration model, include two steam production pressures \( (p_{sp1}, p_{sp2}) \) and one steam consumption temperature \( (T_{sc2}) \).

The reference scenario considered for the thermo-chemical conversion of biomass into methanol is a 20 MW\( _{th} \) sized plant. The main fixed operating conditions and the decision variables are summarized in Table 1.

### 3.1.2. Economic Evaluation

The economic performance is evaluated by the total production cost including investment and operating costs. The capital cost estimates provide a basis for the overall comparison by assessing the trends implied by the decision variables, rather than an accurate estimate of the project. The cost estimation approach follows the one adopted by Gassner et al. [11] and Tock et al. [12] relying on data available in the literature.
### Table 1 Main operating conditions and decision variables with their variation range

<table>
<thead>
<tr>
<th>Section</th>
<th>Description</th>
<th>Variable</th>
<th>Value/Range</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Dryer</strong></td>
<td>air dryer inlet T</td>
<td>$T_d$</td>
<td>[180, 240]</td>
<td>°C</td>
</tr>
<tr>
<td></td>
<td>Wood Φ at outlet</td>
<td>$F_{d,wood}$</td>
<td>[10, 35]</td>
<td>%</td>
</tr>
<tr>
<td><strong>Torrefaction</strong></td>
<td>Torrefaction T</td>
<td>$T_{T,out}$</td>
<td>260</td>
<td>°C</td>
</tr>
<tr>
<td><strong>Gasfier (FICFB)</strong></td>
<td>Steam to biomass ratio</td>
<td>$R_{sb}$</td>
<td>0.5</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>Gasification P</td>
<td>$p_g$</td>
<td>1.15</td>
<td>bar</td>
</tr>
<tr>
<td></td>
<td>Gasification T</td>
<td>$T_g$</td>
<td>847</td>
<td>°C</td>
</tr>
<tr>
<td></td>
<td>HT stage outlet T</td>
<td>$T_{HT}$</td>
<td>1350</td>
<td>°C</td>
</tr>
<tr>
<td></td>
<td>Steam preheating T</td>
<td>$T_{steam}$</td>
<td>400</td>
<td>°C</td>
</tr>
<tr>
<td><strong>Gasfier (EF)</strong></td>
<td>Steam to biomass ratio</td>
<td>$R_{sb}$</td>
<td>0.6</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>Gasification P</td>
<td>$p_g$</td>
<td>30.15</td>
<td>bar</td>
</tr>
<tr>
<td></td>
<td>Gasification T</td>
<td>$T_g$</td>
<td>1350</td>
<td>°C</td>
</tr>
<tr>
<td><strong>Water Gas Shift</strong></td>
<td>difference between unit inlet and reactor T</td>
<td>$\Delta T_{WGS}$</td>
<td>[0.1, 50]</td>
<td>°C</td>
</tr>
<tr>
<td></td>
<td>WGS react T</td>
<td>$T_{WGS}$</td>
<td>[250, 320]</td>
<td>°C</td>
</tr>
<tr>
<td></td>
<td>steam to CO for WGS</td>
<td>$R_{s/co}$</td>
<td>2.5</td>
<td>-</td>
</tr>
<tr>
<td><strong>Methanol Synthesis</strong></td>
<td>$S = (H_2-CO_2)/(CO+H_2)$</td>
<td>$S$</td>
<td>2.05</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>methanol synthesis inlet T</td>
<td>$T_{m,in}$</td>
<td>[227, 387]</td>
<td>°C</td>
</tr>
<tr>
<td></td>
<td>MeOH process P</td>
<td>$p_m$</td>
<td>[75, 90]</td>
<td>bar</td>
</tr>
<tr>
<td></td>
<td>MeOH synthesesys T</td>
<td>$T_m$</td>
<td>[252, 267]</td>
<td>°C</td>
</tr>
<tr>
<td></td>
<td>Recycled fraction</td>
<td>$R_m$</td>
<td>0.95</td>
<td>mol</td>
</tr>
<tr>
<td><strong>Steam Network</strong></td>
<td>Steam production P</td>
<td>$p_{sp1}$</td>
<td>[40, 120]</td>
<td>bar</td>
</tr>
<tr>
<td></td>
<td>Steam production P additional level</td>
<td>$p_{sp2}$</td>
<td>[40, 120]</td>
<td>bar</td>
</tr>
<tr>
<td></td>
<td>T of additional steam consumption level</td>
<td>$T_{sc2}$</td>
<td>[50, 250]</td>
<td>°C</td>
</tr>
<tr>
<td></td>
<td>Steam superheating T</td>
<td>$T_{SH}$</td>
<td>[350, 550]</td>
<td>°C</td>
</tr>
</tbody>
</table>

The currency exchange rates used are the yearly average exchange rates for 2010 [26] and all costs have been updated to year 2010 by using the Marshall and Swift Index. The investment costs are calculated on the basis of the methodology outlined in [26,27]. The major process equipments are roughly sized and their purchase cost is calculated and adjusted to account for specific process pressures and materials using correlations from literature. The *total investment cost* is then calculated using multiplication factors to take into account indirect expenses like labour, transportation, fees, contingencies and auxiliary facilities. The *operating costs* $[\text{€/GJ}_{\text{MeOH}}]$ take into account the cost of labour, maintenance (5% of the total investment), raw materials (biomass, oxygen) and utilities (electricity). The *production cost* $[\text{€/GJ}_{\text{MeOH}}]$ is the sum of the operating cost and the *depreciation cost*, the latter being the total investment cost divided by the present worth of annuity (1) (depending on the investment rate $ir$ and the economic lifetime $t$) and the yearly production of methanol.

$$\text{present worth of annuity} = \frac{(1+ir)^t-1}{ir \cdot (1+ir)^t}$$  \hspace{1cm} (1)

The main assumptions for the economic evaluation are summarized in Table 2.
Table 2. Economic evaluation assumptions

<table>
<thead>
<tr>
<th>Assumption</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Marshall and Swift Index (2010)</td>
<td>1473</td>
</tr>
<tr>
<td>US$ equivalency</td>
<td>1.3 US$/€</td>
</tr>
<tr>
<td>Expected lifetime (t)</td>
<td>15 y</td>
</tr>
<tr>
<td>Interest rate (ir)</td>
<td>6%</td>
</tr>
<tr>
<td>Plant availability</td>
<td>90%</td>
</tr>
<tr>
<td>Operators ²</td>
<td>4 p./shift [12]</td>
</tr>
<tr>
<td>Operator’s salary</td>
<td>66000 €/y</td>
</tr>
<tr>
<td>Biomass price b (Φ= 50%)</td>
<td>6 €/GJBM (21 €/MWhBM)</td>
</tr>
<tr>
<td>Electricity price b</td>
<td>43 €/GJe (155€/MWhe)</td>
</tr>
<tr>
<td>Oxygen price (1 – 10^5 m^3/h)[19]</td>
<td>0.03- 0.7 €/kg</td>
</tr>
</tbody>
</table>

² For a plant size of 20 MWa biomass input. For other production scales, an exponent of 0.7 with respect to plant capacity is used.

The prices of electricity and biomass are representative of the European market. The high price of electricity accounts for the production/consumption of “green” electricity.

3.2. Process Performance Indicators

To assess the process performance, thermodynamic, economic and environmental indicators can be defined. The considered indicators are:

- Energy efficiency:
  \[ \eta_{en} = \frac{LHV_{MeOH} \cdot \dot{m}_{MeOH} + \dot{E}^-}{LHV_{biomass} \cdot \dot{m}_{biomass} + \dot{E}^+} \]  

- Chemical Efficiency:
  \[ \eta_{chem} = \frac{LHV_{MeOH} \cdot \dot{m}_{MeOH}}{LHV_{biomass} \cdot \dot{m}_{biomass}} \]  

- Equivalent efficiency:
  \[ \eta_{eq} = \frac{LHV_{MeOH} \cdot \dot{m}_{MeOH} + \frac{1}{\eta_{CC}} \cdot (\Delta E^-)}{LHV_{biomass} \cdot \dot{m}_{biomass}} \]  

Where the superscripts – and + refer respectively to produced (output) and consumed (input) services. The equivalent energy conversion efficiency aims at correctly assessing the value of the produced or consumed by-products. In equation (4), contrary to definition (2), the consumed amount of power at the denominator is omitted and represented by the net overall output of electricity (\(\Delta E^\cdot\)) at the numerator [17]. The electrical power required is substituted by the equivalent amount of SNG (synthetic natural gas) which would be used for its generation in a CC (combined cycle).

The economic indicators are the total investment and the production cost previously described. The only environmental indicator taken into consideration is the yearly avoided CO₂ emissions [ktonCO₂/year] obtained by the substitution of conventionally produced methanol with the biomass derived methanol. The account of the CO₂ emissions assumes that the combustion of the biomass and derived methanol is carbon neutral, while it takes into account the emissions due to the harvesting and transport of biomass and the consumption of electricity which is dependent on the electricity mix. The CO₂ emissions relative to the use of fossil derived methanol take into account its production and combustion. The data used for environmental evaluation refers to the Swiss context and are taken from [29].
4. Optimization

The multi-objective optimization was carried out selecting as the objective functions, the minimization of the capital investment cost and the maximization of the equivalent efficiency (4). The generated pareto fronts for the two conversion scenarios employing the EF and the FICFB gasifiers are represented in Figure 2, in terms of specific investment cost (€/kWBM) and equivalent efficiency:

![Figure 2 Optimization results for the FICFB and the EF scenarios](image)

The designs A, B, and C are shown as representative of a high efficiency - investment cost, intermediate efficiency – investment and low efficiency – investment cost optimal designs for the FICFB scenario. The designs D and E are representative of a high efficiency – investment cost and low efficiency – investment cost optimal designs for the EF scenario. The performances of these designs are summarized in Table 3, and the composite curves of designs A, B, and C are represented in Figure 4.

<table>
<thead>
<tr>
<th></th>
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<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>FICFB</td>
<td>A</td>
<td>20</td>
<td>36.50</td>
<td>42.55</td>
<td>44.71</td>
<td>70.3</td>
<td>70.3 (15.6)</td>
<td>1736</td>
<td>29.6</td>
</tr>
<tr>
<td></td>
<td>B</td>
<td>20</td>
<td>33.31</td>
<td>44.78</td>
<td>49.18</td>
<td>37.2</td>
<td>98.4</td>
<td>1677</td>
<td>30.9</td>
</tr>
<tr>
<td></td>
<td>C</td>
<td>20</td>
<td>28.88</td>
<td>44.62</td>
<td>50.64</td>
<td>7.5</td>
<td>134.9</td>
<td>1632</td>
<td>32.9</td>
</tr>
<tr>
<td>EF</td>
<td>D</td>
<td>20</td>
<td>40.15</td>
<td>44.28</td>
<td>45.84</td>
<td>84.8</td>
<td>35.3 (38.5)</td>
<td>1241</td>
<td>24.1</td>
</tr>
<tr>
<td></td>
<td>E</td>
<td>20</td>
<td>37.30</td>
<td>43.12</td>
<td>45.25</td>
<td>68.7</td>
<td>49.2</td>
<td>1155</td>
<td>25.4</td>
</tr>
</tbody>
</table>

* The share of power required for oxygen production is reported in parenthesis. Design D results in a net exporter of electricity, if the electricity required for oxygen production is not accounted for. The electricity consumption due to oxygen production is similar across the FICFB and the EF scenarios. The oxygen requirement for a capacity of 20MWth is of 0.28 kg/s and 0.47 kg/s for the FICFB and the EF scenarios respectively.

4.1. Analysis of the optimization results

The comparison of the composite curves of designs A and D are represented in Figure 3. Design A, with a FICFB, presents a pinch at the gasification temperature, while in design D the pinch
disappears because for the EF gasifier the heat is directly provided by the partial oxidation of the feed.

Figure 3 Comparison of the composite curves of (a) design A, with a FICFB gasifier, and (b) design D with an EF gasifier.

The analysis of the optimization results of both scenarios suggests that the co-generation of electricity has a determining importance in the positioning of the designs on the pareto front for both gasifier options. Designs with the lowest efficiency are the ones with smaller units for heat recovery and electricity generation, which are also the cheapest ones. Moving from the lowest efficiency designs to the highest ones, the amount of electricity produced increases, the process remaining a net importer of electricity all along the pareto front. This can be visualized by the composite curves of the three designs A, B, and C represented in Figure 4.

Figure 4 Composite curves of designs A (a) B (b) and C (c) belonging to the FICFB scenario.

The role of the energy integration is also shown in Figure 5 where electric power and total electricity required by the process are represented as a function of the equivalent efficiency (4) for the FICFB and the EF scenario. The secondary axis indicates the chemical (2) and energy (3) efficiencies. These results show the importance of the heat integration and the cogeneration of electricity in improving the overall efficiency of the process. For both design solutions the highest equivalent efficiency processes are the ones displaying a higher fraction of the required electricity produced via a Rankine cycle. The overall equivalent energy efficiency results are higher for the EF designs than for the FICFB. This is in part due to the integration of the combined cycle which is able to provide a larger fraction of the electricity requirement. The EF gasifier, in fact, makes available a higher fraction of the feed as high temperature heat which may be converted into electricity. As explained before, an inconvenient of the FICFB gasifier is the presence of methane and tars in the produced gas. In this study tar removal and methane reforming is carried out by a HT stage following the FICFB gasifier. The temperature of the HT stage depends on technological constraints and the nature of the biomass resource. Its value greatly affects the performance of the
process, as higher temperatures impose larger oxygen consumption (and therefore energy requirements). For design A, for example, the equivalent efficiency $\eta_{\text{en,eq}}$ could be raised by 1.7% point to 38.2% ($\eta_{\text{en}} = 44.2\%$, $\eta_{\text{chem}} = 46.5\%$) if the HT stage temperature was reduced by 100°C to 1250°C. In a subsequent optimization study the influence of the temperature can be studied in more detail. Other technological options for the reduction of tars and the reforming of methane have been investigated in other studies, such as catalytic cracking and mechanism methods (i.e. scrubbers). A review of tar reduction and control technologies for biomass gasification is presented by Han et al. [30].

The energy efficiencies are similar for both scenarios and range between 42 and 45%. The chemical efficiencies range between 45 and 51%, corresponding to mass yields between 42 and 48%. These values are in the same range of the efficiencies reported in the literature but the comparison results difficult because of the different process options and operating conditions. The efficiency of the FICFB is slightly lower than what is reported by Hamelinck et al. [7] for a fluidized bed, this difference is mainly due to the presence of the HT stage in the present study.

![Figure 5 Effect of heat integration on the optimal conceptual designs for the FICFB scenario (a) and the EF scenario (b). The cogeneration power, total electricity requirement, the chemical and energy efficiency are represented as a function of the equivalent efficiencies. The x-axes do not have the same scale.](image1)

The investment cost build-up for the high efficiency designs of the FICFB scenario and the EF scenario are shown in Figure 6.

![Figure 6 Investment production cost build-up for the FICFB (design A) and the EF (design D) scenarios.](image2)

The cost of the gasifier represents a large fraction of the total investment cost. The higher cost for the FICFB is in large part due to the cost of the gasifier itself. The lower estimated cost of the EF gasifier is due to its simpler design and the possibility of building larger units. Furthermore for the FICFB scenario, two gasification units are required, while only one gasifier is required in the EF scenario. The production costs results are also lower for the EF gasifier mainly because of the impact of the lower investment costs. The oxygen required by the processes appears in the
production costs as it is purchased [19]. This is why design D results a slight net exporter of electricity. Nevertheless, for a more reliable comparison of the cost of the two technologies more information would be needed.

5. Sensitivity Analysis

The base price considered for biomass is 6 €/GJ and for electricity, 43 €/GJ (Table 2). For comparison, the price of electricity for industry in France is about 20 €/GJ [31], but in Italy and Switzerland about 40 €/GJ [30,31]. Furthermore, if the product is to be considered renewable, the imported electricity should also be provided by a renewable resource, which may result in a higher electricity price. The sensitivity analysis was carried out considering a price range for electricity between 20 and 50 €/GJ and for biomass between 3 and 12 €/GJ. The sensitivity of the production and operating costs of the electricity and biomass prices for the three previously described designs, belonging to the FICFB scenario is represented in Figure 7. The strong impact of the biomass cost highlights the importance of the chemical efficiency to obtain favourable methanol production costs.

![Figure 7 Sensitivity analysis for the FICFB scenario](image)

The reference scenario considered for this study is a 20 MWth sized plant. Figure 8 shows the variation of operating and production costs and the environmental impact in terms of yearly CO2 avoided emissions as a function of the plant size, for designs A and C.

![Figure 8 a) Production cost and b) avoided CO2 emissions sensitivity analysis on the plant capacity](image)

As expected, the economies of scale have non-linear impact on the production costs which decrease by about 15% from a plant capacity of 20 MWth of biomass input to about 200 MWth. The
production cost reduction between 200 MW _th_ and 400 MW _th_ is only about 2%. The avoided CO₂ emissions increase linearly with the plant capacity as they are proportional to the produced methanol. The distance at which biomass is sustainably available also affects the evaluation of the production cost and the CO₂ emissions in terms of the optimal plant capacity, but it hasn’t been the focus of this study.

6. Conclusions

The thermo-economic optimization of biomass thermo-chemical conversion into methanol was carried out considering as alternative scenarios, a FICFB gasifier and an EF gasifier. The performances of the optimized conceptual designs were evaluated consistently in terms of efficiency, costs and environmental performance. Results show the importance of the energy integration and in particular of the Rankine/cogeneration cycle for the improvement of the overall efficiency of the process. The EF scenario displays higher equivalent efficiencies in comparison to the FICFB scenario because the integration of a steam cycle allows for a larger production of electricity, satisfying part of the energy requirement of the plant. On the other hand, the overall energy efficiencies are similar for the two scenarios and result of about 43-45%. The chemical efficiencies range between 45-51%. From an economic standpoint, the production costs, ranging from 35 to 45 €/GJ remain well above the current price for natural gas derived methanol but the evaluation of the avoided CO₂ emissions highlights the potential of this biomass conversion route in a carbon constrained world.

Acknowledgments

Thanks are due to Geert Haarlemmer for helpful discussions.

Nomenclature

**Abbreviations**
- _AGR_ Acid Gas Removal
- _BM_ Biomass
- _DME_ Dimethyl ether
- _EF_ Entrained Flow
- _FT_ Fischer- Tropsch
- _FICFB_ Fast Internally Circulating Fluidized Bed
- _HHV_ Higher Heating Value, MJ/kg
- _LHV_ Lower Heating Value, MJ/kg
- _MeOH_ Methanol
- _WGS_ Water Gas Shift

**Roman Letters**
- _P_ Pressure, bar
- _T_ Temperature, °C
- _È_ Mechanical/electrical power, kW
- _m_ Mass flowrate, kg/s
Greek symbols

\( \Phi \)  Humidity, %
\( \eta_{en} \)  Energy Efficiency, %
\( \eta_{chem} \)  Chemical Efficiency, %
\( \eta_{en,eq} \)  Equivalent Efficiency, %

References


Thermoeconomic Fuel Impact Approach for Assessing Resources Savings in Industrial Symbiosis: Application to Kalundborg Eco-industrial Park

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Abstract:

Thermoeconomics is a powerful analysis tool for Industrial Symbiosis because: i) it provides a systemic approach and ii) the use of exergy allows the characterization of both energy and matter flows by using a common basis. Furthermore, techniques developed within Thermoeconomics for the analysis, synthesis, optimization and diagnosis of energy systems may be adapted for its application to Industrial Symbiosis. A general methodology for relating the savings of resources of a given eco-park with improvements in local efficiencies of components, introduction of integrations and variations of park production is presented. The approach is based on the fuel impact approach developed in Thermoeconomics for the diagnosis of the operation of energy systems, but introduces significant improvements: i) the introduction of integrations requires the modelling of discrete changes in the productive structure of the system and ii) the purpose of the comparison is to analyze effects of improvements instead of reduction of efficiency. The formulation is applied to the example of symbiosis in Kalundborg, where the present situation is compared to a reference state without symbiosis. Exchange flows considered include waste heat, steam, ash, waste gas and gypsum, which are shared among a power plant, a refinery, a district heating system, a plaster board industry, a cement plant, a fish farm and others. Application of the methodology points out how the integration of by-products causes local savings that, in turn, produce savings at eco-park level.

Keywords:
Industrial Symbiosis, Thermoeconomics, Kalundborg, Fuel Impact Formula

1. Introduction

The aim of Industrial Symbiosis is the transformation of conventional linear productive chain into material cycles by using wastes from industries as raw materials for others [1-3]. Accordingly, it is a core part of Industrial Ecology, which is a multidisciplinary area looking for achieving a more rational and balanced industrial organization, trying to imitate the structure and operation of natural ecosystems [1,4]. The ideas of Industrial Symbiosis are implemented in eco-industrial parks, which are defined as “a community of businesses that cooperate with each other and with the local community to efficiently share resources (information, materials, water, energy, infrastructure and natural habitat), leading to economic gains, gains in environmental quality, and equitable enhancement of human resources for the Business and local community” [5].

Several analysis tools are used in Industrial Symbiosis. The most widely used is Input-Output analysis [6,7]. This powerful methodology has a drawback: it uses different measurement units for the different flows. This disadvantage is overcome by using monetary costs, introducing subjectivity. The use of exergy can be seen as solution for this problem, because it allows one to
express all flows in the same physical units. Exergy has been used in several applications related to Industrial Ecology such as quantification of resource depletion [8] or life cycle assessment [9-11].

A first step towards the application of Thermoeconomics [12,13] to Industrial Symbiosis was the application of exergy to input-output analysis [14], while the actual application was developed in [15]. This research was continued in [16] where a methodology for cost analysis formation in eco-industrial parks was proposed.

Besides, one of the most interesting applications of Thermoeconomics developed during the last years is the diagnosis of energy systems [17-19]. This area of research aims at the detection of malfunctioning components in energy systems and the quantification of the additional fuel consumption caused by each one of them. In this paper, a methodology for quantifying resources savings in Industrial Symbiosis based on the fuel impact-approach developed for diagnosis is developed and applied. The goal of the approach is to compare two situations: with and without integration. Due to integration, the amount of resources consumed by the whole system is reduced. The method is able to quantify which part of these savings corresponds to: i) improvements in components (either by improving their efficiency or by using by-products), ii) reduction of the generation of wastes and iii) variation of the production of useful streams. As an example of application, a simplified version of the Kalundborg eco-industrial park is used.

2. Methodology

In this section, the methodology proposed is presented. In the first part, the main concepts of thermoeconomic analysis are reviewed while in the second, the method itself is described.

2.1. Review of thermoeconomic input-output analysis

Thermoeconomic input-output, also known as symbolic exergoeconomics, provides a general framework for thermoeconomic analysis based on matrix notation which can be easily implemented in computers [20]. In order to apply the approach, the physical structure of the system, where all physical flows appear, has to be substituted by the productive structure, where fuel and product flows are depicted. All components of this productive structure are numbered starting from 1, and the number 0 corresponds to the environment. $E_{ij}$ is the part of the product of component $i$ that becomes part of the fuel of component $j$. A generic component $i$ with its fuel and product is represented in Fig. 1.

\[
P_i = \sum_{j=0}^{n} E_{ij} \quad (1)
\]

\[
F_i = \sum_{j=0}^{n} E_{ij} \quad (2)
\]

Fig. 1. Fuel and product of a generic component $i$.

Accordingly, product ($P_i$) and fuel ($F_i$) of a given component $i$ are:

\[
P_i = \sum_{j=0}^{n} E_{ij}
\]

\[
F_i = \sum_{j=0}^{n} E_{ij}
\]
A table containing the values of all elements $E_{ij}$ as well as their summations by columns and rows is called fuel-product table.

The unit exergy consumption ($\kappa_{ij}$) is defined as the number of units of exergy that each component requires from the other components to obtain a unit of its product:

$$\kappa_{ij} = \frac{E_{ij}}{P_j} \quad (3)$$

The sum of all unit exergy consumptions ($k_j$) in a component is the inverse of the exergy efficiency of that component ($\eta_j$):

$$k_j = \sum_{i=1}^{n} \kappa_{ij} = \frac{E_j}{P_j} = \frac{1}{\eta_j} \quad (4)$$

Exergy cost of a flow $E_{ij}$ is the amount of exergy resources entering the system needed to produce that flow; it is represented as $E^*_{ij}$ and has units of exergy. The unit exergy cost ($k^*$) is the quotient between the exergy cost and the exergy of the flow, accordingly, it is non-dimensional:

$$k^*_{ij} = \frac{E^*_{ij}}{E_{ij}} \quad (5)$$

All flows of the product of a component are considered to have the same formation process and, accordingly, they have the same unit cost. This assumption constitutes a limitation of the method that can be overcome by a suitable definition of the productive structure. Thus, the following equation can be written:

$$k^*_{ij} = k^*_{P,i} \quad (6)$$

By applying the cost balance of all components, it can be demonstrated that the unit cost of the flows can be obtained by using the following equation:

$$k^*_{P} = (U_D - (KP)^\dagger)^{-1} \kappa_e \quad (7)$$

where $\kappa_e = (\kappa_{i1}, \ldots, \kappa_{in})$ is a $(n \times 1)$ vector whose elements contain the unit consumption of external resources and $(KP)$ is a $(n \times n)$ matrix whose elements are the unit exergy consumptions $\kappa_{ij}$. More details on thermoeconomic analysis can be seen in [20,21].

### 2.2. Application of thermoeconomic analysis to the quantification of resources savings

One possibility for quantifying resources savings is to apply an increment formulation of the exergy balance of the plant. Reduction of the resources consumed by the plant (PS) is, thus due to three effects: reduction of irreversibility in components ($-\Delta I_i$), reduction of plant production of useful products ($-\Delta P_r$), and reduction of amount of waste ($-\Delta P_w$):

$$PS = \sum_{i=1}^{n} (-\Delta I_i) - \Delta P_r - \Delta P_w \quad (8)$$

The problem of the previous formula appears because when savings occurs in a component, irreversibility usually varies also in other components in order to keep the plant production. For example, if there is a substitution of a turbine of a power plant that leads to higher efficiency, this turbine will need less steam in order to keep the power produced and, accordingly, irreversibility of the boiler will also decrease. For this reason, more detailed analysis is needed.

When the unit exergy consumption of a component decreases ($-\Delta \kappa_{ij}$), the irreversibility in this component also decreases leading to local savings (LS) in that component.

$$LS_{ij} = -\Delta \kappa_{ij} P_i \left( x_0 \right) \quad (9)$$
where \( P_i(x_0) \) is the product of component \( i \) in the reference situation (before the improvement). The total local savings in that component are:

\[
LS_j = -\Delta k_i P_i(x_0) = \sum_{j=0}^{n} LS_{ji}
\] (10)

This reduction of irreversibility is interesting because it is associated to the behavior of the component (it is proportional to the variation of its unit exergy consumption). However, it is not the variation in resources consumed by the plant. This correspondence is made through the unit exergy cost of fuel of the component. Accordingly, plant savings (PSk) are defined as the additional amount of fuel entering the plant (or park) caused by local savings, and it is calculated as:

\[
PS_{kj} = k^*_{P,j}(x)LS_{ji}
\] (11)

\[
PS_{ki} = \sum_{j=0}^{n} PS_{kj}
\] (12)

where \( k^*_{P,j}(x) \) is the unit cost of the product of \( j \) in the operation situation (after improvements). If the amount of product leaving the plant is reduced (\( \Delta P_s \)), it causes also plant savings (PSp) that can be calculated as:

\[
PS_{p,j} = -k^*_{P,j}(x)\Delta P_s
\] (13)

Besides, the reduction of the amount of waste leaving the plant (\( \Delta P_r \)) also causes plant savings (PSr) that can be calculated as:

\[
PS_{r,j} = -k^*_{P,j}(x)\Delta P_r
\] (14)

Finally, the total amount of plant savings fuel impact is a summation of plants savings due to variation in the components (PSk), plant savings due to reduction in plant production of useful products (PSp), and reduction of plant generation of wastes (PSr):

\[
PS = \sum_{i=1}^{n} PS_{ki} + \sum_{j=1}^{n} PSp_{ij} + \sum_{j=1}^{n} PSr_{ij}
\] (15)

3. Example of application

3.1. Description and exergy of flows

In order to demonstrate the applicability of the methodology proposed, an example based on Kalundborg eco-industrial park will be used. Since the application to the whole system is out of the scope of the present work and would be difficult because the lack of data, a simplified example has been defined considering the more representative flows of this eco-park. The working case is defined around the thermal power plant of Asnaes, which represents the core of the park, and includes the following symbiotic exchanges:

- Waste gas from the refinery is used in the boiler of the power plant, substituting part of the coal consumed in that plant.
- Process steam is bleed from the steam cycle of the power plant.
- Steam for district heating is bleed from steam cycle.
- Hot sea water at low temperature from power plant condenser is used in fish farm.
- Ash and slag from the power plant boiler are used as substitute for clinker in cement manufacturing.
- Gypsum produced in the desulfurization process of the power plant is used in a plasterboard industry.

In order to assess the effect that industrial symbiosis has on resources savings, the working case described is divided into two situations: with and without integration. The first case considers the
situation corresponding to industrial symbiosis with the six integrations considered before. The second case simulates a fictitious situation where no integration is present, in a way that symbiotic matter and energy flows are substituted by others produced by conventional processes. In other words: waste gas is flared thus causing an additional coal consumption in the boiler, integrations of steam and hot water are substituted by natural gas-fired boilers, ash and slag are replaced by clinker produced in a kiln and gypsum is imported from Spain by sea. Figure 2 represents a physical diagram corresponding to a superstructure able to represent both cases.

Exergy of all flows present in Fig. 2 are listed in Table 1, for both situations. Asterisks are used to indicate flows dissipated in the non-integrated case. Exergy values have been calculated using a data from [20-22], and details of calculations can be found in the first part of this paper¹.

3.2. Productive structure

The starting point for thermoeconomic analysis is the definition of the productive structure, which is represented in Fig. 3. This structure is the graphical representation of the Fuel, Product and Waste flows of the system considered. For this reason, the structure contains the Fuel-Product-Residue definition of each component. It should be noted that these flows of the productive structure are, in general, different from the actual flows of the physical structure; for instance, the product of the boiler going to steam turbine is the flow of live steam minus the flow of condensate water; accordingly, it is represented as a single flow in the productive structure.

The most interesting aspect of the productive structure defined is that components representing the considered integration do not operate in the same way in both cases considered. In other words, in the integrated situation they act as junctions directly connected to the power plant without consuming external resources, while in the example without integration they act as processes isolated from the power plant consuming external resources.

Fig. 2. Physical structure.

¹ Alicia Valero, Sergio Usón and Jorge Costa. Exergy Analysis of the Industrial Symbiosis Model in Kalundborg. Accepted for presentation in ECOS 2012.
Table 1. Yearly exergy of flows in the working example.

<table>
<thead>
<tr>
<th>Flow</th>
<th>Integration</th>
<th>No integration</th>
</tr>
</thead>
<tbody>
<tr>
<td>Coal</td>
<td>16.376</td>
<td>10.591</td>
</tr>
<tr>
<td>Waste gas*</td>
<td>841.7</td>
<td>841.7</td>
</tr>
<tr>
<td>Electricity power plant</td>
<td>3.968</td>
<td>3.968</td>
</tr>
<tr>
<td>Ash and slag*</td>
<td>21.79</td>
<td>22.24</td>
</tr>
<tr>
<td>Sulphur dioxide</td>
<td>25.98</td>
<td>26.51</td>
</tr>
<tr>
<td>Limestone for desulfurization</td>
<td>1.933</td>
<td>1.973</td>
</tr>
<tr>
<td>Water desulfurization</td>
<td>4.51</td>
<td>4.604</td>
</tr>
<tr>
<td>Gypsum from desulfurization*</td>
<td>2.036</td>
<td>2.078</td>
</tr>
<tr>
<td>Electricity for desulfurization</td>
<td>24.8</td>
<td>24.8</td>
</tr>
<tr>
<td>Process steam</td>
<td>193.4</td>
<td>193.4</td>
</tr>
<tr>
<td>District heating steam</td>
<td>63.8</td>
<td>63.8</td>
</tr>
<tr>
<td>Hot water for fish farm</td>
<td>14</td>
<td>14</td>
</tr>
<tr>
<td>Steam for turbines</td>
<td>4.487</td>
<td>4.487</td>
</tr>
<tr>
<td>Natural gas process</td>
<td>-</td>
<td>495.8</td>
</tr>
<tr>
<td>Natural gas district heating</td>
<td>-</td>
<td>307.7</td>
</tr>
<tr>
<td>Natural gas fish farm</td>
<td>-</td>
<td>254.4</td>
</tr>
<tr>
<td>Limestone clinker</td>
<td>-</td>
<td>16.26</td>
</tr>
<tr>
<td>Fuel-oil clinker</td>
<td>-</td>
<td>213.4</td>
</tr>
<tr>
<td>Clinker</td>
<td>-</td>
<td>59.51</td>
</tr>
<tr>
<td>Natural gypsum</td>
<td>-</td>
<td>2.036</td>
</tr>
<tr>
<td>Fuel-oil gypsum transport</td>
<td>-</td>
<td>9.703</td>
</tr>
</tbody>
</table>

Fig. 3. Productive structure
This type of structure able to properly representing both situations is a key innovative point of the work presented. It should be noted that the cost of waste flow leaving the flare is assigned to the boiler of the power plant. Once the productive structure has been defined, it is possible to calculate unit exergy costs for all flows and to apply the resources savings decomposition with the methodology presented above.

### 3.3. Unit exergy costs

As it was presented previously, the unit exergy cost is the amount of resources, measured in exergy, needed to obtain a unit of product of each component. Table 2 contains unit exergy costs of the products of each component for both situations.

**Table 2. Unit exergy costs of the products of the components**

<table>
<thead>
<tr>
<th>Component</th>
<th>Integration</th>
<th>No integration</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Separator + desulfurization</td>
<td>1.0054</td>
<td>1.0076</td>
</tr>
<tr>
<td>2. Boiler</td>
<td>2.3591</td>
<td>2.5550</td>
</tr>
<tr>
<td>3. Turbine</td>
<td>2.6676</td>
<td>2.8891</td>
</tr>
<tr>
<td>4. Process steam boiler</td>
<td>2.3591</td>
<td>2.5636</td>
</tr>
<tr>
<td>5. District heating boiler</td>
<td>2.3591</td>
<td>4.8229</td>
</tr>
<tr>
<td>6. Fish farm boiler</td>
<td>2.3591</td>
<td>18.1714</td>
</tr>
<tr>
<td>7. Clinker kiln</td>
<td>1.0054</td>
<td>3.8592</td>
</tr>
<tr>
<td>8. Gypsum transport</td>
<td>1.0054</td>
<td>5.7657</td>
</tr>
<tr>
<td>9. Flare</td>
<td>1.0000</td>
<td>1.0000</td>
</tr>
</tbody>
</table>

Table 2 shows how the unit exergy cost of integrated components is decreased, except the flare. These reductions appear because integrations, which act as junctions in the integrated case, are substituted by alternative processes consuming resources and having irreversibility. These reductions are analyzed in detail below.

- **Separator + desulfurization.** The variation is negligible and appears because in the case without integration flows of ash and gypsum entail irreversibilities due to the component. Accordingly, it was expected that the cost would remain almost constant, considering that processes of separation of ash and desulfurization keep their exergy efficiency constant.

- **Boiler and steam turbine.** The turbine keeps its exergy efficiency while the boiler has a small variation because irreversibility changes since, in the case without integration, water that was sent to the fish farm must be dissipated in the condenser. However, a small variation in the costs of the products of both components appears because of the asignation of a part of the cost of the flare. Accordingly, it should be highlighted that, in the non integrated case, the cost of the waste increases slightly the unit cost of the product of the boiler and, accordingly, the unit cost of electricity generated.

- **Process steam boiler.** The ancillary boiler is less efficient in exergy terms than the boiler of the power plant. However, in the example without integration they have similar unit cost. This fact is due because of the variation of unit cost of the boiler due to the dissipation of waste gas.

- **District heating boiler.** It can be seen how the unit cost of the district heating boiler is more than twice in the case without integration. This is due because of the low exergy efficiency of using natural gas for producing low temperature thermal energy for district heating.

- **Fish farm boiler.** This unit cost is the highest one, because the exergy efficiency of producing thermal energy at low temperature by using natural gas is very low.

- **Clinker kiln and gypsum transport.** It should be noted the high difference of unit cost of clinker kiln and gypsum transport, which are multiplied times 4 and 6, respectively. This is also due to
the consumption of external resources for producing or transporting these flows. Besides, it should be noted that the cost of ash and gypsum in the integrated example is roughly one, which means that every real process considered in the case without integration would entail an increment of the unit cost of these products.

3.4. Resources saved decomposition by using fuel impact approach

The analysis of unit exergy cost is very useful for assigning a cost to the product by using physical criteria. However, this parameter does not provide information about the resources savings caused by the integration of flows. Accordingly, to know these savings, resources impact approach has to be applied.

The total amount of resources savings is calculated at the difference between the consumption of resources in the non integrated case minus the consumption in the integrated situation. In this example, these savings are equal to 1514 GWh/year. However, the determination of resources savings is not enough, because it is needed to identify the causes that have generated it and to share this total amount into partial savings originated by each cause. A first idea to perform this decomposition may be to consider the variation of irreversibility, products and waste, as it can be seen in Table 3.

<table>
<thead>
<tr>
<th>Component</th>
<th>(\Delta I) [GWh/year]</th>
<th>(\Delta P_s) [GWh/year]</th>
<th>(\Delta P_r) [GWh/year]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Separator + desulfurization</td>
<td>25</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>2. Boiler</td>
<td>-357</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>3. Turbine</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>4. Process steam boiler</td>
<td>302</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>5. District heating boiler</td>
<td>244</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>6. Fish farm boiler</td>
<td>240</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>7. Clinker kiln</td>
<td>170</td>
<td>38</td>
<td>0</td>
</tr>
<tr>
<td>8. Gypsum transport</td>
<td>10</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>9. Flare</td>
<td>0</td>
<td>0</td>
<td>842</td>
</tr>
<tr>
<td><strong>TOTAL</strong></td>
<td><strong>635</strong></td>
<td><strong>38</strong></td>
<td><strong>842</strong></td>
</tr>
</tbody>
</table>

It should be noted that, for the decomposition of resources savings according to irreversibility, only exergy analysis is needed. However, this approach is not suitable for pointing out where savings are caused. For example, the boiler of the power plant can be considered. This component causes additional resources consumption (negative savings) because it has increased its production (and, thus, its irreversibility) but not because it has decreased its efficiency. In other words, it needs more fuel not because it works worse, but because its production is higher.

Accordingly, a more detailed methodology that considers the variation of efficiency of components, as well as changes in fuel used, is needed. This decomposition is shown in Table 4 and uses the concept of local savings and plant savings presented before.

Table 4 shows the decomposition of plant savings according to the methodology proposed. The total value is obviously the same as in Table 3, but the method is able to point out where savings are caused. The main contributions are due to the following components:

- **Ancillary boilers.** They show in terms of plant savings that the autonomous production of thermal energy is less efficient than the use of steam from the power plant. In other words, this shows the advantages of combined heat and power (CHP). Besides, it can be seen how the lower the temperature of thermal energy generated, the higher the impact on plant savings.

- **Clinker kiln.** This component has an impact on local savings caused mainly because of the substitution of clinker by another flow with lower exergy (ash), what originates a variation in its
product. Besides, there is also an impact caused by the consumption of external resources needed for producing clinker.

- **Flare.** This component has the highest impact, which appears because of the use in the system of a flow of high exergy (waste gas) that previously was wasted (flared).

### Table 4. Impact on savings based on local savings and plant savings

<table>
<thead>
<tr>
<th>Component</th>
<th>$LS$ [GWh/year]</th>
<th>$PSk$ [GWh/year]</th>
<th>$PSp$ [GWh/year]</th>
<th>$PSr$ [GWh/year]</th>
<th>$PS$ [GWh/year]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Separator + desulfurization</td>
<td>24</td>
<td>24</td>
<td>0</td>
<td>0</td>
<td>24</td>
</tr>
<tr>
<td>2. Boiler</td>
<td>9</td>
<td>16</td>
<td>0</td>
<td>0</td>
<td>16</td>
</tr>
<tr>
<td>3. Turbine</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>4. Process steam boiler</td>
<td>302</td>
<td>38</td>
<td>0</td>
<td>0</td>
<td>38</td>
</tr>
<tr>
<td>5. District heating boiler</td>
<td>244</td>
<td>157</td>
<td>0</td>
<td>0</td>
<td>157</td>
</tr>
<tr>
<td>6. Fish farm boiler</td>
<td>240</td>
<td>221</td>
<td>0</td>
<td>0</td>
<td>221</td>
</tr>
<tr>
<td>7. Clinker kiln</td>
<td>62</td>
<td>62</td>
<td>146</td>
<td>0</td>
<td>208</td>
</tr>
<tr>
<td>8. Gypsum transport</td>
<td>10</td>
<td>10</td>
<td>0</td>
<td>0</td>
<td>10</td>
</tr>
<tr>
<td>9. Flare</td>
<td>0</td>
<td>0</td>
<td>842</td>
<td>0</td>
<td>842</td>
</tr>
<tr>
<td>TOTAL</td>
<td>527</td>
<td>146</td>
<td>842</td>
<td>1514</td>
<td></td>
</tr>
</tbody>
</table>

Finally, it should be commented an interesting fact related to ancillary boilers: plant savings are smaller than local savings. This may be surprising because plants savings are calculated by multiplying local savings times the unit exergy cost of fuels, which is always higher than 1. This fact can be explained taking into account that plant savings of each component are calculated as a summation of several terms, and local savings in these boilers is produced by a positive term (less resources from the environment are consumed) and a negative term (more steam from the power plant is needed).

### 4. Conclusion

The analysis of integrations characterizing industrial symbiosis need the use of methodologies that should be systemic and also able to deal with the diverse nature of the flows involved. Thermoeconomic analysis accomplishes these premises because it uses a well structured matrix formulation and uses exergy as indicator for measuring.

In this paper, a methodology has been proposed for analyzing the savings in resources obtained in these integrations. The method compares the system with integration and an equivalent system without this integration. The comparison is performed by using an impact approach developed from the formulation used for diagnosis, and a suitable definition of the productive structure able to deal with both situations.

The methodology is applied to a simplified version of Kalundborg eco-industrial park. Although not all integrations present in the park are considered, the example is representative enough because it includes flows of energy (steam and hot water at several temperatures, as well as waste gas) and materials (ash and gypsum). Cost assessment shows how integration causes reduction of cost of different products (e.g. Unit exergy cost of steam for district heating decreases from 4.82 to 2.36, or unit exergy cost of gypsum is reduced from 5.77 to 1.01). Besides, the methodology is able to share total plant savings (1514 GWh/year) into different causes (e.g. 157 GWh/year for the integration of district heating, 62 GWh/year for the use of ash in the clinker or 842 GWh/year for the use of waste gas from the refinery).
Since the method is focused on energy and material resources savings during operation, capital, operation and maintenance costs are not considered. The extension of the methodology to consider also these important issues is an interesting research line.

Acknowledgments

Authors would like to acknowledge ARAID and IberCaja for its support within the project “Thermoeconomics and Industrial Ecology. Application to Teruel coalfield”, Young researchers program, 2010.

Nomenclature

\[ E \] exergy flow in a productive structure, kW
\[ E^* \] exergy cost of a flow in a productive structure, kW
\[ F \] fuel flux, kW
\[ I \] irreversibility, kW or GWh/year
\[ k^* \] unit exergy cost, –
\[ LS \] plant savings, kW or GWh/year
\[ P \] product flux, kW
\[ PS \] plant savings, kW or GWh/year
\[ U_p \] identity matrix \((n \times n)\)
\[ \langle KP \rangle \] matrix of unit exergy consumptions \((n \times n)\)
\[ x \] operation situation
\[ x_0 \] reference situation

Greek symbols

\[ \eta \] exergy efficiency
\[ \kappa \] unit exergy consumption
\[ \kappa_e \] vector of unit exergy consumption of external resources \((n \times 1)\)

Subscripts and superscripts

\[ P \] product
\[ r \] waste plant output
\[ s \] productive plant output
\[ em \] electromechanical
\[ ph \] physical

References


Thermoeconomics of a ground-based CAES plant for peak-load energy production system

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Abstract:
Compressed Air Energy Storage is recognized as a promising technology for applying energy storage to grids which are more and more challenged by the increasing contribution of renewable such as solar or wind energy.

The paper proposes a medium-size ground-based CAES system, based on pressurized vessels and on a multiple-stage arrangement of compression and expansion machinery; the system includes recovery of heat from the intercoolers, and its storage as sensible heat in two separate (hot/cold) water reservoirs, and regenerative reheat of the expansions.

The CAES plant parameters were adapted to the requirements of existing equipment (compressors, expanders and heat exchangers). A complete exergy analysis of the plant was performed. For all components, cost data were sought on the market or from reference bibliographic sources. The results allow to calculate the final cost of the electricity unit (kWh) which is made available under peak-load (production) mode, and to identify the contribution within the plant of capital costs and component inefficiencies.

Keywords:
CAES, exergy, thermoeconomics

1. Field of application of CAES
Compressed Air Energy Storage (CAES) and Pumped Hydro Energy Storage (PHES), air and water being the most inexpensive fluids, are likely to be the most reliable and technologically sound options for massive energy storage [1, 2].

PHES systems require two reservoirs at different elevation and a pump/turbine for storing/recovering energy in form of water head. Their high energy efficiency (60-78\% - [1]) has gained them a widespread application, but development of new PHES is often constrained by localization difficulties. In many situations where cost-effective and environmentally acceptable sites for PHES are unavailable, alternative technologies may be useful.

CAES systems replace water basins with underground storage volumes (caverns in salt or rock formations, porous rocks, depleted natural gas fields) which are used as reservoirs for pressurized air (60-70 bar). Normally, for a given capacity, a CAES uses less land surface than a PHES and doesn’t require an elevated reservoir. Natural storage volumes may be scarce or too distant from the energy grid, but artificial storage volumes can be developed in form of high-pressure vessels or underground pipes.

Artificial storage is obviously more expensive than a natural one. It has been shown [3] that an increase in the storage pressure (>100 bar) decreases the material cost of the vessels, although it makes the compression/expansion train more complex.
It is likely that artificial storage CAES would be preferable on a small scale (1-10 MW), whenever the convenience of a given site overcomes the lack of a natural reservoir. Incidentally, an artificial storage is easier to control in terms of air contaminants at turbine inlet.

Existing CAES plants are all gas fuelled, compressed air being heated in a combustion chamber ahead of the turbine during the expansion phase. Hence, these systems combine energy storage within a power plant. The first 290 MW plant built at Huntdorf (Germany) in 1978 is very simple [4]: no attempt was made to recover any thermal energy at turbine exhaust or at compressor exit, before storage in the cavern. The main purpose of this earlier plant was to transfer energy produced by nuclear or coal plants from low to peak demand hours.

A second plant, McIntosh – Alabama [5], has a recuperator between the hot turbine exhaust and the cold air stream coming from the cavern, in order to reduce the fuel consumption. Many refined schemes have been proposed [6, 7], with notable energy efficiency improvements or using alternative fuels [8].

A significant breakthrough is the so-called “Adiabatic CAES” (ACAES) [9, 10] that has been prompted by the increasing market share of renewable, intermittent energy sources. This configuration gets rid of the combustion chamber and takes full advantage of the thermal recovery from the hot, compressed air to raise the air temperature before expansion. If compression is performed in a common radial or axial machine, i.e. it is practically adiabatic, the recoverable thermal energy at compressor exit is of the same order of magnitude of compression work. Being compression and expansion performed in different phases, a Thermal Energy Storage (TES) is necessary. Expected benefits of this configuration are:

- improved energy efficiency,
- avoidance of a connection to the natural gas grid or any other fuel distribution,
- elimination of pollutant emissions
- lower turbine operating temperature.

The second point may be useful if the CAES is thought as an addendum to a wind farm or any other discontinuous, remote energy source. According to the economic analysis presented in [9], ACAES may be useful as:

- centralized plants – size around 300 MW – in countries with high spread between base and peak energy price;
- decentralized plants – size around 150 MW – near large windfarms in order to increase full load operation, peak price sales and increase utilization of the power lines;
- remote island solutions – size around 30 MW – integrated with wind power, aimed at increase of full load operation of wind turbines and savings of grid connection costs or fuel consumption.

When designing the heat recovery, storage and reclaim system, the simplest option is to introduce one or more heat exchangers on the compression and on the expansion train. Heat storage medium can be liquid or solid or phase change material. In principle, the optimum thermodynamic design should pursue a quasi-isothermal compression/expansion. In the limit of isothermal transformations, the work consumed per unit mass of compressed air would be minimum and the same amount would be returned during expansion, i.e. energy recovery would be complete. In this case, the TES would be ambient air itself and hence would have infinite thermal capacity at no cost. The energy recovery efficiency is hence expected to increase with the number of stages (compressor plus cooler or heater plus turbine) as demonstrated in [11].

In practice, when the compression/expansion is divided in a large number of stages in order to approximate an isothermal behaviour, concentrated losses in the connections between stages and heat exchangers become predominant. A compromise, accounting for system complexity and cost, must be pursued. In any case, increasing the number of stages reduces the maximum temperature of
the TES during compression and hence simplifies its design. For example, if the TES temperature is kept below 100°C, the storage medium can be liquid water at moderate pressure.

All components of a CAES plants are commercially available: compressors, heat exchangers, large volume vessels for high pressure (in case of artificial storage), radial expanders, insulated water reservoirs. Variable Inlet Guide Vanes (IGVs) may be used to adapt the compressors/expanders at the variable storage pressure. Variable configurations of the compression/expansion train have been shown in [3, 11] to extend the operating range. Reciprocating compressors may be integrated in the train for the last stages, when the storage pressure approaches its maximum.

Some caution must be used when comparing ACAES energy efficiency with gas fuelled CAES. For example the ACAES efficiency (energy output divided by input) should not be matched against the “round trip efficiency” of a non-adiabatic CAES [12], where a significant part of the energy is produced from a primary source at the time of peak demand, i.e. is not affected by a storage penalty.

1.1 - Mode of operation

The proposed CAES system (Figure 1) (from an original idea by “ENEL Ingegneria e Innovazione”), is designed as a typical backup unit for wind energy plants. It is based on a seven-stage intercooled compression train. The inlet air (25 °C, 1 bar, dry air) passes through the compression train (an air-based version of that described in [13]), driven by an electric motor (MD), and is stored in pressurized vessels with an overall volume of 6340 m$^3$. The compressed air storage reservoir (CAR) is built as a system of interconnected vessels; it is pre-pressurized at 77 bar, so that the compressor train starts operating at this pressure, and finishes its operating cycle when the discharge pressure reaches 125 bar. The compressors operated most of the time in off-design conditions (variable IGV setting helps maintaining a good efficiency at off-design). The intercoolers (IC) are water-cooled: the heat from the intercoolers is recovered and stored in a hot water reservoir (TES). Following the description of the air path, the plant is completed by a six-stage expansion train (TE), geared to the electric generator (EG). The expansion is reheated after each stage, recovering heat stored in the hot water tank in a water/air heat recovery heat exchanger (RH). The cold water flow at the RH discharge is recovered in the cold water reservoir (CWR), from which it is re-used in a closed loop for the next operating cycle. Before re-use, water is cooled down to the temperature of the environment by an external cooler.

![Fig. 1. Schematic of CAES plant](image-url)
2. CAES plant components

2.1 – Compressor train

The reference case is built around a typical multi-stage centrifugal compressor train using a GE O&G Nuovo Pignone SRL compressor (Figure 2), as it allows the modular selection of the required number of impellers optimizing the speed using different gear ratios between the High Speed Shafts and main shaft. This design allows also relatively simple inter-stage extraction, as is needed in the present case for intercooling (IC in Figure 1). The first three stages of the compressor train are equipped with variable-IGV control, which can adjust the IGV setting angle from +15° to -60°. It should be remarked that the compressor train is operated at constant mass flow (12.2 kg/s) and variable discharge pressure (fixed volume of the CAR): accordingly, the characteristic curve can be adjusted along the pressure range, as is shown in Figure 3.

![Fig. 2. SRL Compressor](image)

![Fig. 3. SRL Compression Train pressure-flow rate characteristic curve with variable IGV setting](image)

The variable-IGV operating mode allows also to maintain a good efficiency under off-design operating conditions; Figure 3 shows the efficiency achievable (obtained with the optimal IGV setting) under the main reference operating conditions, over the full pressure range considered in the CAES application.
Fig. 4. Compression Train calculated polytropic efficiency along the charging cycle (Varying IGV)

The GE internal tool CCS was used to select each impeller of the SRL compressor. CCS is based on an internal database of different types of centrifugal impellers which have already been designed for other applications and whose performance has been checked against test data. The tool has a number of inputs: \( P_{in} \), \( T_{in} \), \( m_{in} \), RPM, \( D_i \) for each stage; moreover, the inter-stage pressure drops, the casing size and rating, the size of flanges and volutes. The discharge pressure \( p_{out} \) is a code output, as well as the IGV pre-rotation angle and the BEP efficiency at design conditions.

The nominal design condition was assumed at a discharge pressure of 105 bar. The calculation of the operating point conditions was then repeated for off design, setting 5 discrete operating points: 125, 115, 95, 85 and 77 bar, and calculating the optimal IGV setting and the off-design efficiency. The power absorbed by the compression train ranges from 7.5 MW for \( p_{out} = 75 \) bar to 8.1 MW \( p_{out} = 125 \) bar. The SRL motor drive is an asynchronous electric motor having the following target data: \( W = 9600 \) kW; Nominal rotational speed = 1500 rpm; Nominal voltage \( V = 13.8 \) kV.

2.2 – Turbo-expander train

Fig. 5. Integrally-Geared Turbo-Expander

The turbo-expander is again of the integrally-geared type: an example is shown in Figure 5. In the present case, 6 stages were considered, geared on three shafts (rotating respectively at 26000, 14000
and 7300 rpm). Each stage is preceded by a Reheat Heat Exchanger (RH in Figure 1). Here again, each stage is equipped with variable-setting IGVs.

The turbo-expander mass flow rate was adjusted to maintain as far as possible a constant power output with variable inlet pressure: this operating condition results from the electric market day-ahead arbitrage pricing which is required by the Italian grid operator. In the present case, three reference conditions were assumed for calculating the design and off-design performance, which are summarized in Table 1:

### Table 1 Turbo-expander train reference operating conditions

<table>
<thead>
<tr>
<th></th>
<th>HP</th>
<th>IP</th>
<th>LP</th>
</tr>
</thead>
<tbody>
<tr>
<td>P&lt;sub&gt;CAR&lt;/sub&gt;, bar</td>
<td>125</td>
<td>98</td>
<td>75</td>
</tr>
<tr>
<td>m, kg/s</td>
<td>24.5</td>
<td>26.6</td>
<td>27.4</td>
</tr>
<tr>
<td>W, MW</td>
<td>8.60</td>
<td>8.62</td>
<td>8.34</td>
</tr>
</tbody>
</table>

The EG is an asynchronous electric generator having the following target data: W = 10500 kVA; Nominal rotational speed = 1500 rpm; Nominal voltage V = 13.8 kV;

### 2.3 – Heat Exchangers (IC, RH)

The heat exchangers were designed by an external provider after specifications by GE O&G Nuovo Pignone; a shell and tube arrangement was required; shell/tube arrangement between air and water were switched with variable pressure: in fact, air passes through shells (and water through the pipes) when the air pressure is low (first stages), while in the last stages air passes inside the pipes (and water through the shell side) in order to minimize heat exchanger capital costs. The shell side is provided with baffles which ensure a correct cross-flow arrangement. The water inlet/outlet conditions were specified as 20°C/90°C for the IC; and 89°C/80°C for the RH heat exchangers. Due attention was paid in not exceeding the boiling water temperature balancing the amount of water mass flow. Referring for example to coolers, the water mass flow rates were adjusted for each cooler under each operating condition in order to have a delivery temperature of 90°C, with a heat exchanger pinch temperature difference of 15 °C. The calculation was done by traditional HE sizing rules, assuming a constant overall heat transfer coefficient and surface. So, considering heat exchangers duty, it is possible, knowing the air inlet and outlet temperature, to calculate the water flow rates (data reported in Table 2).

### Table 2 Main data for CAES plant heat exchangers

<table>
<thead>
<tr>
<th>Unit</th>
<th>External Diameter [mm]</th>
<th>L [mm]</th>
<th>m&lt;sub&gt;w&lt;/sub&gt; [kg/s]</th>
<th>Q [kW]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Intercooler 1</td>
<td>900</td>
<td>4000</td>
<td>4.8</td>
<td>1414</td>
</tr>
<tr>
<td>Intercooler 2</td>
<td>900</td>
<td>5000</td>
<td>5</td>
<td>1500</td>
</tr>
<tr>
<td>Intercooler 3</td>
<td>900</td>
<td>4500</td>
<td>5.9</td>
<td>1800</td>
</tr>
<tr>
<td>Intercooler 4</td>
<td>650</td>
<td>6000</td>
<td>3.2</td>
<td>1000</td>
</tr>
<tr>
<td>Intercooler 5</td>
<td>600</td>
<td>6500</td>
<td>3.4</td>
<td>1000</td>
</tr>
<tr>
<td>Intercooler 6</td>
<td>580</td>
<td>7000</td>
<td>3.3</td>
<td>1000</td>
</tr>
<tr>
<td>Aftercooler</td>
<td>500</td>
<td>7500</td>
<td>3.5</td>
<td>1100</td>
</tr>
<tr>
<td>Heater 1</td>
<td>700</td>
<td>6000</td>
<td>12</td>
<td>2700</td>
</tr>
<tr>
<td>Heater 2</td>
<td>800</td>
<td>6000</td>
<td>9.5</td>
<td>1600</td>
</tr>
<tr>
<td>Heater 3</td>
<td>950</td>
<td>5000</td>
<td>9.3</td>
<td>1500</td>
</tr>
<tr>
<td>Heater 4</td>
<td>950</td>
<td>5000</td>
<td>9.3</td>
<td>1600</td>
</tr>
<tr>
<td>Heater 5</td>
<td>1000</td>
<td>5000</td>
<td>8.7</td>
<td>1500</td>
</tr>
<tr>
<td>Heater 6</td>
<td>1000</td>
<td>5000</td>
<td>8.9</td>
<td>1500</td>
</tr>
</tbody>
</table>
2.4 – Reservoirs (CAR, CWR, TES)

Air storage vessels (CAR) were selected considering an above ground storage ACAES plant, using 2115 TENARIS pressure vessels with a unit capacity of 3 m$^3$. The admissible work pressure for this vessel is 140 bar, tested following the 97/23/EC-PED standard. Each cylinder features 622mm external diameter and 12m length.

In the Thermal Energy Storage (TES), hot water reservoir works at ambient pressure, therefore water cannot exceed the boiling temperature. An average temperature of the hot tank of 89 °C was estimated, with an overall volume of water equal to 810 m$^3$, including 50 m$^3$ used as fixed quantity to make sure that the tank never empties.

The Cold Water Reservoir (CWR) has the same size; it starts its operation (storage mode) at 20 °C; at the end of the production mode, it is full of warm water at 47°C, which has been delivered from the TES passing through the HR network. The cooling load needed for reducing the CWR temperature from 47 to 20°C is provided by an external heat exchanger.

3. Exergy and Thermoeconomic Analysis

3.1 – Exergy analysis

Exergy Analysis was set in the classical reference form [14, 15]. Exergy is evaluated as an extensive property, so it can be transferred into or out of a control volume where steams of matter enter and exit. For each component, generally:

$$\frac{dE_{cv}}{dt} = \sum_{j} (1 - \frac{T_0}{T_j}) \dot{Q}_j - \left[ W_{cv} - p_0 \frac{dV_{cv}}{dt} \right] + \sum_{i} \dot{m}_i e_i - \sum_{e} \dot{m}_e e_e - E_D - E_L \tag{1}$$

The term $\dot{m}_i e_i$ accounts for the time rate of exergy transfer at the inlet $i$. Similarly, $\dot{m}_e e_e$ accounts for the time rate of exergy transfer at the outlet $e$.

The analyses considered in this work involves a slow evolution of steady-state operating conditions (as the pressure in the CAR is varied); for the analysis of most components, it is sufficient to consider the steady-state form of the exergy rate balance. At steady state, $\frac{dE_{cv}}{dt} = 0$ and $\frac{dV_{cv}}{dt} = 0$, so equation (1) reduces to

$$0 = \sum_{j} (1 - \frac{T_0}{T_j}) \dot{Q}_j - W_{cv} + \sum_{i} \dot{m}_i e_i - \sum_{e} \dot{m}_e e_e - E_D - E_L \tag{2}$$

This equation states that the rate at which exergy is transferred into the control volume exceeds the rate at which exergy is transferred out. The difference is the rate at which exergy is destroyed within the control volume due to irreversibilities. In compact form Eq. 2 reads:

$$0 = \sum_{j} \dot{E}_{a,j} - W_{cv} + \sum_{i} \dot{E}_i - \sum_{e} \dot{E}_e - E_D - E_L \tag{3}$$

$e_i$ and $e_e$ can be calculated from enthalpy and entropy referring to unit mass of the fluid:

$$e_i = (h_i - h_0) - T_0 (s_i - s_0) \tag{4}$$

In this study, the compressors were assumed to be working without any heat transfer to the external environment (Adiabatic conditions). Both Heat Exergy and the Exergy Loss terms disappear, leading to:

$$\dot{E}_{D,e} = - W_c + \dot{E}_i - \dot{E}_e \tag{5}$$

The compression power is easily found.
\[ W_c = m (h_t - h_w) \]  

The compressor work is negative, as the sign convention defines as negative an energy flow entering the system.

Heat exchangers are treated as steady-state adiabatic, no-work components; under these assumptions, \( \dot{E}_x \) can be calculated from:

\[ \dot{E}_{x,HE} = \dot{E}_{x,AV} + \dot{E}_{x,AR} - \dot{E}_{x,W} \]  

Exergy destruction within the piping (air and water sides) were evaluated calculating the friction losses by traditional correlations. The piping was considered adiabatic; the total pressure loss is converted into an entropy generation, and the exergy destruction is calculated consequently [14]:

\[ \dot{E}_D = T_0 \dot{S}_{gen} \]  

A closed-system balance was used to evaluate the total exergy \( E_{CAR} \) stored in the compressed air reservoir (vessels). Only physical exergy was considered, neglecting kinetic and potential contributions; air vessels were assumed, at the end of the charge, at a temperature of 25 °C (same as environment).

The specific physical exergy is

\[ e^{PH} = (u - u_0) + p_0(v - v_0) - T_0(s - s_0) \]  

In the case of an ideal gas with constant specific heat ratio, the specific physical exergy can be expressed as:

\[ \frac{e^{PH}}{c_p T_0} = \frac{T}{T_0} - 1 - \ln \frac{T}{T_0} + \frac{k - 1}{k} \ln \frac{p}{p_0} + \frac{T}{T_0} \left( \frac{p_0}{p} - 1 \right) \]  

From the assumptions made, air temperature is constant at ambient value, hence:

\[ e^{PH} = RT_0 \left[ \ln \frac{p}{p_0} + \left( \frac{p_0}{p} - 1 \right) \right] \]  

Note that in this case, the assumption of constant \( k \) is no longer necessary.

\( E_{CAR} \) over one operating cycle can be calculated by the difference between exergy stored at 125 bar and 77 bar, simply as:

\[ E_{CAR} = E_{125}^{PH} - E_{77}^{PH} \]  

The exergy loss corresponding to heat release to the environment to cool the water in the CWR at the end of the cycle can be estimated using the difference between two exergy levels, calculated by a closed system balance:

\[ E_{CWR,W} = (U_w - U_0) - T_0(S_w - S_0) \]  

\[ E_{CWR,C} = (U_c - U_0) - T_0(S_c - S_0) \]  

Where \( E_{CWR,W} \) and \( E_{CWR,C} \) are respectively the exergies of the CWR at the final temperature of 47 °C and at the starting value of 20 °C. Then the exergy loss can be estimated as:
3.2 – Thermoeconomic analysis

Thermoeconomics [14] is the branch of engineering that combines exergy analysis and economic principles to provide the system designer or operator with information not available through conventional energy analysis and economic evaluations, but crucial to the design and operation of a cost-effective system involving transformation of energy. At the base of a thermoeconomic analysis there is an economic analysis in order to detect and define each cost relative to the system. The system was modelled considering a sequence of steady states; all relevant entering and exiting material streams, as well as both heat and work interactions with the surroundings were included. Associated with these transfers of matter and energy are exergy transfers into and out of the system, and exergy destructions caused by the irreversibilities within the system. Since exergy measures the true thermodynamic value of such effects, and costs should only be assigned to commodities of value, it is meaningful to use exergy as a basis for assigning costs in thermal systems.

The cost balance applied to the \( k \)-th system component can be generally written as:

\[
\sum_{i} (c_{e,i} E_{e,i})_k + c_{w,i} W_k - c_{a,k} E_{a,k} + \sum_{i} (c_{r,i} E_{r,i})_k + Z_k
\] (16)

In the specific case of compressors, for each stage:

\[
c_{r,k} E_{r,k} = c_{l,k} E_{l,k} + c_{w} W + Z_k
\] (17)

No auxiliary relation is needed for compressor stages [14]. In the first compressor stage \( c_{l,1} \) was considered equal to zero (ambient air); in the other stages, \( c_{l,k} \) is equal to the cost stream exiting the preceding cooler. \( c_{w} \) was considered equal to 38 €/MWh, corresponding to use of off-peak electricity.

Each heat exchanger was evaluated referring to the component model shown in Figure 6, using the following relations:

\[
c_{2} E_{2} + c_{4} E_{4} = c_{1} E_{1} + c_{2} E_{2} + Z_k
\] (18)

Considering that the coolers work as a heat exchanger with double purpose (cooling + heat recovery), the thermo-economic assumption \( c_{2} = c_{4} \) was applied (equal cost of the two exit streams, which are both products of the component); additionally, the cost of cold water was assumed to be negligible \( c_{2} = 0 \) (the pumping cost is marginal, and anyway a calculation of the friction losses in the water loop was not performed, so that even pumping power was not calculated at this level of the analysis, which focuses on thermal and compressor/expander performance).

For heaters, the only purpose is that of heating air. \( c_{1} \) in this case is equal to the cost of the water stored in the hot tank (TES). As for any heat exchanger whose purpose is heating a cold stream [14], the auxiliary relation is \( c_{2} = c_{1} \). For the first RH, \( c_{3} \) is equal to the cost of air coming from the CAR; for all the others, it is given by the cost of the exit flow from each expansion stage.

![Fig.6 Schematic of heat exchanger](image)
The expander thermo-economic balance is similar to that of the compressor, considering now the work stream as an outlet:

$$c_{r,k} \dot{E}_{r,k} + c_{w,k} W_k = c_{i,k} \dot{E}_{i,k} + \dot{Z}_k$$  \hspace{1cm} (19)

The auxiliary relation in this case is [14]: $c_{r,k} = c_{i,k}$.

For the storage (CAR) reservoir (a component in which the capital cost is expected to be very high), the thermo-economic relation is applied over the complete lifetime considering the overall scheduled operating time in charge/discharge modes, and it reads:

$$\sum_t c_{r,CAR} \dot{E}_{r,CAR} t_{r,diach} = \sum_i c_{i,CAR} \dot{E}_{i,CAR} t_{i,ich} + Z_{CAR}$$  \hspace{1cm} (20)

The operating time in the different states of charge/discharge is considered in Equation 20, and the values calculated (based on the CAR size and on the operating lines of compressors and expanders) are presented in the following section.

4. Results - Exergy Analysis

4.1 Charge and discharge time

The application of the compressor characteristic curve to the ACAES plant gives the charge curve in Figure 7, which was calculated with reference to the CAR considered as a closed system, integrating the mass flow with a time step $\Delta t$ for each intermediate condition.

The CAR pressure time history is approximately a linear function of time. The result shows a theoretical time of 8.1 hours to fill completely the air storage vessels with a volume of 6340 m$^3$.

During the discharge process, the expander unit was simulated according to its characteristic curve with variable IGV setting. As the CAR pressure varies in a limited range (125-75 bar), the discharge function results approximately linear. The complete discharge time is 3.77 hrs.
4.2 – Results - Exergy analysis

The exergy balance was divided into CAES plant sections: compressors, intercoolers, expanders, reheaters. As an example, for the case $p_{\text{CAR}} = 125$ bar, the overall exergy destruction in compressors amounts to 1.3MW (for an absorbed power of 7.5 MW). The distribution of exergy destruction over the 7 stages is shown in Figure 8.

![Fig.8 Relative distribution of Compressors exergy destruction ($p_{\text{CAR}} = 125$ bar)](image)

The overall IC exergy destruction amounts to 517 kW, and its distribution is shown in Figure 9.

![Fig.9 Relative distribution of IC exergy destruction ($p_{\text{CAR}} = 125$ bar)](image)

The overall RH exergy destruction amounts to 625 kW, and its distribution is shown in Figure 10.

![Fig.10 Relative distribution of RH exergy destruction ($p_{\text{CAR}} = 125$ bar)](image)
The piping exergy destruction was very low (18 kW for all the IC connection pipes. The overall expander exergy destruction amounts to 1654 kW, and its distribution among the different stages is shown in Figure 11.

![Fig.11 Relative distribution of expander exergy destruction (p\text{\text{CAR}} = 125 \text{ bar})](image)

In practice, the operating conditions are continuously varying in time, as the delivery pressure in the CAR is increased from 75 to 125 bar. As an example, the trend of exergy destruction in time for the compressor train is shown in Figure 12. $\dot{E}_D$ was calculated for every compressor stage in each off-design condition considered (the minimum exergy destruction is achieved very close to the nominal design conditions). Interpolating the values with a polynomial function allows to define an approximate trend line, which was integrated in time to give the overall expected value over one charge cycle, $E_D = 10.1 \text{ MWh}$.

![Fig.12 Time history of calculated compressor exergy destruction](image)

A Sankey diagram showing the exergy destruction during the charge and discharge phases is shown in Figure 13.
5. Results – Thermoeconomic analysis - Conclusions

Detailed results about exergy analysis have been shown in Section 4, consequently the conclusions here reported represent only a synthesis of this case study. The cumulated exergy efficiency of the process (calculated through a time-resolved integration of the sequence of system operating conditions, along a complete charging/discharging cycle) is about 52%. Referring to the maximum pressure (125 bar), the power production distribution among the 6 expander stages is shown in Figure 14, and the cost of electricity produced by each stage, according to the thermo-economic analysis of the plant, is summarized in Figure 15.

The power-averaged cost of the electricity produced is calculated at 70 €/MWh, which corresponds to a marginal cost of \((70 - 38) = 32 \, \text{€/MWh}\) of the equivalent stored electrical energy; in practice, a 84% increase with respect to the base-load cost of electricity assumed (38 €/MWh). This is considered as a promising result for proposing ground-built ACAES systems as storage devices for the near future.

![Fig.13 Exergy flow Sankey diagram; (a) Charging (b) Discharging](image)

![Fig.14 Power produced by each expander stage \(p_{CAR} = 125 \, \text{bar}\)](image)
The distribution of the marginal cost buildup among the main plant components, in terms of cost of exergy destruction and capital+O&M costs, is shown in Figure 16.

It can be noticed that the work input amounts to 31% of the expenses; exergy destruction during plant operation (storage and production modes) respresents 29% of the marginal cost; about 40% can be accounted to capital expenses, with the largest share due to the pressure vessels (24%).

**Acknowledgments**

The Authors are grateful to GE O&G Nuovo Pignone SrL and to ENEL Ingegneria e Innovazione for having allowed publication of the present work, which was object of the placement/master thesis project work of one of the Authors (Simon Kemble) in the frame of an established University/Industry cooperation program.
Nomenclature

c  Cost of unit exergy, €/J

c_p  Constant-pressure specific heat, J/(kg K)
e  Exergy, J/kg
E  Overall system Exergy, J
h  Enthalpy, J/kg

\[ \dot{m} \]  Mass flow rate, kg/s
P  Pressure, bar
Q  Heat Rate, W
R  Gas Constant, J/(kg K)
s  Entropy, J/(kg K)
S  System Entropy, J/K
t  Time, s
T  Temperature, °C
u  Internal energy, J/kg
U  System Internal Energy, J
V  Volume, m³
W  Power, W
Z  capital cost, €

Acronyms

ACAES  Adiabatic CAES
BEP  Best Efficiency point
C  Compressor
CAES  Compressed Air Energy Storage
CAR  Compressed Air Reservoir
CWR  Cold Water Reservoir
EG  Electric Generator
HE  Heat Exchanger
TES  Thermal Energy Storage
IC  Intercooler (heat exchanger)
IGV  Inlet Guide Vanes
LPR  Low Pressure Reservoir
MD  Motor Drive (electric)
PHES  Pumped Hydro Energy Storage
RH  Reheater (heat exchanger)

Subscripts and superscripts

0  Reference state
air  Air
C  Compressor
ch  Charge
CV  Control Volume (delimiting component)
D  Destroyed
References


