Green Heating System:
Characteristics and Illustration with the Multi-Criteria Optimization of an Integrated Energy System

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Abstract

The characteristics of the 'Green Heating System', synonymous with 'Environmentally Friendly Heating System', is academically defined from the total energy systems' point of view, using the concept of 'reversible mode of heating' under the current and future technical, economic and environmental protection context. The exergy-based Specific Consumption Analysis Approach is used to quantitatively evaluate the influence of subsystems' exergy efficiencies on the overall performance of a heating system. Through a case study in the city of Beijing, it is shown that heating fuel specific consumption and thus the associated emissions can be dramatically reduced as a result of the implementation of a reversible mode of heating system. A multi-criteria optimization process based on a new evolutionary multi-objective algorithm is undertaken to investigate the trade-off between cost and environmental performances associated with such a system.

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1. Introduction

In the new millennium, with an increasing understanding of the importance of sustainable development, mitigating impacts on the global and local environment while meeting the increasing demand of energy supply is an issue of the highest importance in the development of urban areas. To meet this challenge, a more rational use of energy is urgently needed.

From the viewpoint of sustainability, fighting against irreversibility occurring in energy conversion processes under given technical, economic and environmental restrictions in order to reduce the entropy generation and exergy losses will guide a rational utilization of energy.

A break down of energy use shows that almost two thirds of primary energy is end-used in the form of thermal energy, and up to 5/6 of which is used as low-temperature heating below 100 °C. The heating first law energy efficiency of the current heating systems serving the general public varies over a wide range in China. It can be lower than 30% and can be even higher than 100% [11].

The so-called 'Green Heating System' is characterized as a heating system whose primary energy consumption as well as emission rates are significantly lower than those of currently implemented advanced heating systems. This term can only be defined dynamically as it is fundamentally related to the technical and economic evolution and environment legislation process.

In this paper, the case study of an integrated heating system in the city of Beijing, including Combined Heat and Power (CHP) and heat pump units, illustrates a typical 'Green Heating System' under the current context in China. Switching to natural gas is considered due to environmental legislation in urban areas. The exergy-based Specific Consumption Analysis Approach [11-13] and a multi-criteria optimization process based on a new evolutionary multi-objective algorithm are used to reveal the potential of energy conservation and of emission reduction[1], as well as the economic feasibility of such advanced systems implementation, which depends on the value of exogenous variables like the fuel cost or the investors pay-back time expectations.

2. Clarification of 'Green Heating System' using the exergy-based Specific Consumption Analysis Approach

2.1 Theoretical minimum heating specific consumption: the theoretical potential of the 'Green Heating System'

The theoretical minimum fuel/monetary specific consumption $b_{\min} / c_{\min}$ arises in a hypothetical ideal energy system, which is conceived as an entirely reversible energy system with an infinite life time and without management, tax or any other fixed cost. For heating systems, these values are:

$$b_{\min q} = 34.2(1 - T_0 / T_r) \text{ [kg c.e./GJ]}^2 \tag{1}$$

$$c_{\min q} = c_f b_{\min q} \text{ [US\$/GJ]} \tag{2}$$

As an example, with T_0 of 270 [K], T_r of 293 [K], and c_f of 0.03 [US$/kg c.e.], the theoretical minimum heating fuel specific consumption and heating specific total cost are of respectively 2.68 [kg c.e./GJ] and 0.08 [US$/GJ]. These values correspond to the theoretical limitation of improvements obtainable through energy conservation measures within a given environment.

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[1] The fuel specific consumption b [kg c.e./GJ] can directly be translated into a CO_2 emissions rate [kgCO_2/GJ] according to the carbon content of the fuel (when no CO_2 capture is considered), and is directly related to the specific total cost of energy conversion [US$/GJ]. SO_2 emissions is also of high concern in China due to a large utilization of coal and the specific fuel consumption can be translated into SO_2 emissions rate according to the coal sulfur content, taking into account usual exhaust gas post-treatment.

[2] LHV of 1 kg coal equivalent (c.e.) = 7000 kcal = 1/34.2 GJ

2.2 Reversible mode of heating, the way towards 'Green Heating System'

Although all heating systems are irreversible without exception, the extent of their irreversibility may differ greatly. They may be classified into two modes: conventional mode and reversible mode. The latter is also known as thermodynamic heating [1, 10] which is defined as a total energy system of heating fed by a non-thermal primary energy, in the energy conversion process of which at least one kind of non-thermal energies (e.g. electrical, mechanical or chemical energy) appears and at expense of this non-thermal energy the end product (i.e. the supplied heat) is generated by making use of a waste and/or ambient heat source. A heat pump system is typically a reversible mode of heating system. CHP heating systems, which make use of the waste heat by virtue of some power sacrifice due to the cogeneration heat can be viewed as thermodynamically equivalent to heat pump heating, and can also be considered a reversible mode of heating system. The sacrifice of a CHP plant in terms of power delivered due to heat generation is defined as the Equivalent Electricity Consumption Rate (EECR) [11]. The co-generating heat can thus be treated as generated by a virtual heat pump with the electricity consumption of EECR.

Boiler based heating and direct electric heating, on the contrary, are known as conventional or irreversible modes of heating. They do not fully satisfy the definition of reversible mode of heating and thus have much higher heating specific fuel consumption comparing CHP or heat pump heating as shown in reference [11]. The performance in terms of primary energy consumption of different kinds of heating systems can be quantitatively evaluated through the Specific Consumption Analysis Approach and the concept of 'Generalized Heating System'.

2.3 Specific Consumption Analysis of 'Generalized Heating System'

In the Specific Consumption Analysis Approach, both the fuel specific consumption b and the monetary specific consumption c are composed of: 1) the theoretical minimum specific consumption ($b_{\min}$ or $c_{\min}$) introduced above, and 2) the specific consumption accruals due to any irreversibility occurring in subsystems (b_I or c_I, where subscript I = 1,2,…,n is the order number of the subsystem in question).

$$b = b_{\min} + \sum_{0}^{n} b_I \tag{3}$$

$$c = (c_{\min} + \sum_{0}^{n} c_{FI})(1+\varepsilon) + \sum_{0}^{n} c_{ZI} \tag{4}$$

where, c_{ZI} denotes the cost accrual due to fixed charges for the unity end product attributed to subsystem I and symbol ε represents a coefficient taking into account an eventual pollution tax.

The 'Generalized Heating System' consists of six subsystems: 1) fuel energy delivery, 2) power generation, 3) power distribution, 4) heat generation, 5) heat distribution and 6) end-users subsystem, and can represent all types of heating systems. The first three subsystems listed above are also known as the 'hot end' and the last two are defined as the 'cold end'. For such a heating system, assuming that the exergy at the output of the Ith subsystem equals that at the input of the (I+1)th subsystem, the heating specific consumption accruals in subsystem I (b_{Iq} and c_{Iq}) and the heating fuel and monetary specific consumption (the later is also known as heating specific total cost) of the end product (b_q and c_q) can be respectively expressed as follows [11]:

$$b_{Iq} = (b_{\min q} / \prod_{J=I+1}^{n} \eta_J)(1/\eta_I - 1) \tag{5}$$

$$c_{Iq} = (c_f b_{\min q} / \prod_{J=I+1}^{n} \eta_J)(1/\eta_I - 1) \tag{6}$$

$$b_q = b_{0_q} + b_{\min q} / \prod_{I=1}^{n} \eta_I \tag{7}$$

$$c_q = (c_{F0_q} + c_{\min q} / \prod_{I=1}^{n} \eta_I)(1+\varepsilon) + \sum_{0}^{n} c_{ZIq} \tag{8}$$

where $\eta_J = E_J^{out} / E_J^{in}$ denotes the exergy efficiency/index of subsystem J, which consists of the exergy efficiencies of the fuel energy delivery subsystem η_b, of the power generation subsystem η_w, of the power distribution subsystem η_{WD}, of the heat distribution subsystem η_{QD}, of the end-user subsystem η_{user}, and the exergy index of the heat generation subsystem ε_Q. Both the 'hot end' and 'cold end' exergy efficiencies and the exergy index of the 'heat generation subsystem' play an important role in heating fuel specific consumption reduction.

Concerning reversible mode of heating system, CHP rather than heat pumps is often preferred due to its high exergy index. The association of heat pumps with cogeneration unit introduces however more design and operation flexibility regarding the heat to power ratio. The potential offered by such integrated energy systems regarding CO_2 emissions reductions and associated cost has been recently investigated through a case study in the city of Beijing [3]. The case study considered in this paper goes into more details regarding the influence of the subsystems designs and associated exergy efficiencies on the overall heating performance under different economic and environmental policy context.

3. Application to a district heating system supplying two building areas in the city of Beijing

The investigated case study considers two clusters of respectively residential and office buildings to be built in Beijing. Their main characteristics are given in Table 1. The two building areas, including small local heating distribution networks, are located along a primary linear district heating network supplied by a decentralized energy conversion plant. The residential and the office building areas are respectively located 4 km and 1 km away from the plant.

The superstructure of the energy conversion plant is given in fig. 1. The water of the return line enters the condenser of a two stage heat pump if the latter is chosen to be introduced. It recovers heat from the cold source materialized by a canal. The water mass-flow is then driven to the heat exchanger of a gas turbine cogeneration unit. An additional boiler complements the residual heating load in order to achieve the required supply temperature of the district heating network. The Gas turbine and the auxiliary boiler are natural gas fired. The district heating network supplies the two local substations through heat exchangers. Physical and costing parametric modeling of each one of these elements has been integrated into an overall modeling superstructure.

Transmission losses of heat and power are not considered in the specific consumption calculation. The proposed integrated system is then composed of the 'hot end', the heat generation subsystem, and the user subsystem.

When a gas turbine is selected, a virtual combined cycle based on the chosen gas turbine is introduced for the calculation of the EECR relative to the heat generation subsystem ($EECR_Q$). It defines a virtual 'hot end' and virtual 'heat generation subsystem' shown in fig. 2. This is required for the calculation of specific consumption accruals of the proposed integrated heating system which includes both reversible modes (heat pump and cogeneration unit) and an irreversible mode of heating (additional boiler). The $EECR_Q$ is the total power that would be produced by the virtual combined cycle. That is the sum of the power consumed by the heat pump (P_{HP}) and the network pump (P_{pump}), the power produced by the virtual bottoming cycle using the topping cycle exhaust gases ($EECR_{GT}$) and the power produced by the same virtual combined cycle with the amount of fuel consumed by the gas boiler ($EECR_{GB}$):

$$EECR_Q = (EECR_{GT} + P_{PUMP} + P_{HP} + EECR_{GB}) \tag{9}$$

The exergy index of the virtual heat generation subsystem is calculated as:

$$\varepsilon_{Qvirtual} = E_R / EECR_Q \tag{10}$$

Corresponding exergy efficiency of the 'hot end' is taken as the electrical efficiency of the combined cycle[3]. The exergy efficiency of the user's subsystem is calculated with their heating demand and required temperature level as well as the temperature located at the end of heat distribution subsystem.

The theoretical minimum heating specific fuel consumption b_{minq} is calculated according to the following equation:

$$b_{\min q} = 34.12 \cdot (Q_{user1} \cdot (1 - T_0 / \overline{T}_{r1}) + Q_{user2} \cdot (1 - T_0 / \overline{T}_{r2}))/(Q_{user1} + Q_{user2}) \tag{11}$$

The heating fuel specific consumption b_q and the virtual distribution of the heating fuel specific consumption accrual relative to the virtual hot end ($b_{HEvirtual}$), virtual heat generation subsystem ($b_{Qvirtual}$) and user subsystem (b_{user}) under different configurations and operation conditions can be calculated according to Eq. (7) and (5). The real heating fuel specific consumption accruals can then be derived as follows:

$$\begin{aligned} b_{HE} &= b_{HEvirtual} \cdot (1 - \varphi_{GB}) \\ b_Q &= b_{Qvirtual} + b_{HEvirtual} \cdot \varphi_{GB} \end{aligned} \tag{12}$$

where $\varphi_{GB} = Q_{GB} / Q_Q$ is the proportion between the heat provided by the gas boiler and the heat provided by the whole heat generation subsystem.

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[3] For simplification, the lower heating value of coal and natural gas are used as their exergy value (with an inaccuracy of 4%) for exergy efficiency calculations.

The virtual bottoming cycle efficiency influences the calculation of the EECR and the heating fuel specific consumption. It should ideally be derived through a sub-optimization process taking into account the chosen gas turbine and the optimum steam cycle (heat recovery steam generator and steam turbine). This is however very time consuming. A sensitivity study is undertaken instead in order to evaluate the influence of this parameter.

When no gas turbine is introduced, the power required is imported from the power grid. The exergy efficiency of the 'hot end' becomes the average electrical efficiency of the power grid (η_{e_grid}) taken as 31% in this case study. $EECR_Q$ is then calculated as:

$$EECR_Q = (P_{PUMP} + P_{HP} + Q_{GBfuel} \cdot \eta_{e\_grid}) \tag{13}$$

where Q_{GBfuel} is the heat content of the natural gas consumed by the gas boiler.

For a given improvement of the system considered, whether additional investment is compensated or not in practice by savings on operating cost depends on the value of exogenous variables (fuel cost, interest rate, level of pollution cost internalization…). An optimization process is therefore required for balancing specific fuel consumption reduction and higher investment cost.

Evolutionary algorithms have been proven to be robust and effective for the resolution of non-linear, non-continuous, and mixed real integer problems such as those encountered when dealing with integrated energy systems. The modeling and optimization of a heat pump based district heating system regarding technical, economic and environmental issues has been undertaken using genetic algorithms for the resolution of the so-called environomic formulation, which includes all criteria within a single objective aggregated function [6-8]. This formulation allows a minimization of the overall internalized cost of an energy system, accounting for design, installation, operation and also pollution through the introduction of pollution cost factors.

However, given the difficulty encountered sometimes when trying to express certain criteria in financial terms, a multi-criteria optimization may be preferred. For this purpose, a new clustering evolutionary multi-objective optimizer has been recently developed [9], and applied to the analysis of the trade-off between cost and environmental performance associated with the implementation of advanced integrated energy systems within urban areas [3, 4].

In this study, two objectives are simultaneously optimized: the heating specific total cost in [US$/GJ] and the heating fuel specific consumption in [kg c.e./GJ]. Pareto Optimal Frontiers (POFs) are derived, which define for a given technology, within its feasible domain, the minimum annual cost corresponding to any level of specific fuel consumption which is an indirect measure of the CO_2 and SO_2 specific emissions. Or inversely, they define the maximum fuel specific consumption reduction achievable for a given yearly budget.

The specific total cost of energy conversion is given by the sum of the annual capital cost and the annual operating cost, per unit of heat produced. The annual capital cost is a function of 1) the total investment cost, 2) the interest rate (6% in this case), and 3) the chosen period for the total capital cost amortization. The operating cost is related to the natural gas consumption and power importation from the grid. In the context of Beijing, the natural gas and electricity prices are respectively taken as 2 [UScts/kWh_{th}] and 6 [UScts/kWh] in most of the cases investigated. A sensitivity analysis regarding the natural gas price has been however undertaken. Revenues obtained through power exportation have to be subtracted to derive the net total cost of heating generation.

As shown above, the heating fuel specific consumption is the sum of the heating fuel specific consumption accruals associated with the different subsystems and the theoretical minimum heating fuel specific consumption. The specific fuel consumption associated with power importation from the grid considers an average value of 0.4 [kg c.e./kWh] [5].

The plant is operated during the space heating season, and is shut down the rest of the year. Daily fluctuations of the heating demand are considered to be managed by the network storage capacity. As mentioned above, the primary duty of the energy conversion system to be installed is the heating supply of the two building areas. The integrated energy system is then designed for the heating load, and if a cogeneration unit is introduced, its power is implicitly restrained at a maximum capacity which corresponds to an additional boiler converging to zero. It can be however decided to import all or part of the required power from the grid.

For a chosen winter supply temperature, and considering the heating demands and required level of temperature at the users, the returning temperature and the network mass-flow are derived. The natural gas boiler insures that the heat balance is respected, complementing the heat supplied by the heat pump and the heat recovered from the cogeneration unit.

Table 2 presents the decision variables of the optimization problem with their respective boundaries. In this problem, all variables are chosen to be of real type. However the introduction of constraints in the optimization problem induces discontinuities. As an example, in practice a maximum temperature is defined at the exit of a heat pump compressor for the lubricant not to be altered. Whenever the conditions are such that this constraint is not respected, the heat pump is not selected, which induces a structural change.

4. Results

Fig. 3 and Table 3 provide the obtained POFs and the super-configuration of typical solutions from optimization. Different scenarios are considered with respectively a 5 and 15 years amortization period[4], and with power selling prices of 3, 4, and 5 [UScts/kWh], which

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[4] The amortization period corresponds to the time allocated by investors for paying back investment plus interest.

corresponds to respectively 50 %, 67 % and 80 % of the price of power imported from the grid[5]. With respect to the virtual bottoming cycle efficiency considered in this case, which is 20%, none of the POFs ends up selecting a heat pump, since the exergy index of the gas turbine heat exchanger (around 55% in average) is higher than that of the heat pump (around 50% in average)[6]. For the scenario (AP5, ESP3), the cheapest but most emitting option (Ind.5.3.A) is a large gas boiler supplying most of the heat demand with a small gas turbine cogeneration unit used to generate the power required by the heating network pump. The introduction of a gas turbine with a higher capacity (from Ind.5.3.A to Ind.5.3.E) and a higher electrical efficiency (from Ind.5.3.E to Ind.5.3.L) allows a reduction of the heating fuel specific consumption. A lower specific consumption can be reached with about 19.8 [kg c.e./GJ][7] for the less emitting and most expensive solution at a heating specific total cost of about 10.6 [US$/GJ]. Correlatively, the overall system exergy index can reach 21 % -see fig. 5.

With an increase of either the power selling price or the amortization period, the starting point of each of the POFs appears at lower and lower heating fuel specific consumption levels, with lower heating specific total cost. Thus, the lowest level of total heating fuel specific consumption achievable with the given constraints, to which all POFs converge, makes the POFs narrower. The POF of scenario (AP5, ESP3) starts with the smallest gas turbine capacity considered and increases this capacity to reach the largest one for Ind.5.3.E (full supply of the heating demand), while its electrical efficiency remains at a low value along this segment (pressure ratio of 5). The other POFs start immediately with the largest gas turbine capacity but with gas turbine electrical

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[5] An abbreviation in the format of (APx, ESPy) is used in the later part of the paper to represent the specific scenario.

[6] These exergy indexes correspond to a 20 % bottoming cycle efficiency. A sensitivity analysis regarding the influence of this parameter is given further.

[7] With a two stages gas turbine heat exchanger design, in which the return water from the heating network is first dispatched between the heat pump condenser and a low temperature exhaust gases heat exchanger, and both flows being then mixed before entering a high temperature exhaust gases heat exchanger, a better heat recovery could be reached, resulting in a even lower heating fuel specific consumption.

efficiencies increased along with higher power selling price and shorter amortization periods. Under (AP15, ESP5) scenario, the lowest specific consumption of 19.8 [kg c.e./GJ] can be reached at the lowest heating specific total cost of 3.1 [US$/GJ]. The price of heat for natural gas based heating system is of 5.6 [US$/GJ] [2]. A large profit potential thus exists for such a cogeneration solution.

In fig. 4, POFs have been artificially prolonged for higher specific consumption levels by successive simulations using the values of the decision variables that obtained for the POF of (AP5, ESP3) scenario. A configuration based on a single gas boiler has also been artificially added as a reference (labeled as 'ref' in the figure) for analysis purposes. Heating specific total, operating and capital costs are provided as functions of the heating fuel specific consumption. Vertically, the capital cost curve remains the same as the amortization period is kept constant, while the operating cost curve is shifted down to lower values due to higher power exportation selling prices. The combination of these two curves results in the shape of the heating specific total cost curve. As the power exportation selling price increases, the minimum heating specific total cost occurs at progressively lower heating specific consumption levels (see ind.5.3.A, and ind.5.5.H). Horizontally, it is the specific operating cost curve that remains constant, but the shift of the specific capital cost curve due to an increase of the amortization period also results in the same trend for the minimum heating specific total cost. These points of minimum heating specific total cost correspond to the starting points of the POFs in fig. 3. It is easily understandable now why POFs start at lower specific fuel consumption levels when amortization period or electricity selling prices are increased. Beyond the point of minimum heating specific total cost, the specific fuel consumption criteria and the cost criteria are no longer competitive, which by definition correspond to the extremity of the POF.

For the (AP5, ESP3) scenario, the POF starts with a small capacity gas turbine (Ind.5.3.A) supplying the heating network pump but not with a boiler based configuration. As can be seen in fig. 4, this is due to a decrease of operating cost dominating the corresponding increase of capital cost. This is explained by 1) a higher performance in heating operation as can be seen by a decrease of the heating specific fuel consumption, and 2) a lower power generation specific cost (around 4.6 [UScts/kWh]) than the price of grid power importation. However, the latter remains higher than the power selling price and the gain of the higher heating performance reached by a higher capacity or efficiency gas turbine can not recover this expense as well as the increase of the investment cost. This gives no economic incentive but a heating efficiency incentive to install a larger and higher efficiency gas turbine that results in a lower heating fuel specific consumption with higher heating specific total cost.

Fig. 5 to 7 identify in detail the POF obtained for the (AP5, ESP3) scenario. Fig. 5 shows the improvement of the systems' heating performance along with an increase of the heating specific capital cost. From Ind.5.3.A to Ind.5.3.E, this improvement is obtained trough a progressive increase of the gas turbine capacity and correspondingly the reduction of the gas boiler share along the curve. The gas turbine reaches its maximal capacity for Ind. 5.3.E with 14.7 MW_{elec} according to the heat balance constraint to be respected by the plant with a similarly low electrical efficiency as that for Ind.5.3.A. There is no need anymore for an additional gas boiler. Such an evolution can be defined as a "mode of heating effect", i.e. an increase of share of the reversible mode of heating and correspondingly a decreasing share of irreversible mode of heating. This is confirmed by a look at the distribution of the specific fuel consumption accruals showing an increase of the heat generation subsystem's exergy index (fig. 7). The specific consumption accrual of the 'hot end' (fig. 7) and the increase of the heating specific capital cost (fig. 5) are due to the increasing gas turbine capacity. Increasing further this capacity with an electrical efficiency remaining unchanged would induce a network supply temperature violating the considered upper limit of 90 °C. The pressure ratio of the gas turbine has therefore to be increased (from Ind.5.3.E to Ind.5.3.L), which reduces the heat recovered from the exhaust gases due to a better electrical efficiency. Correspondingly, the exergy index of the 'hot end' and of the total heating system increases along the POF, which results in a lower heating fuel specific consumption, as shown in fig. 7. This can be seen as an 'efficiency effect'. The exergy index of the heat generation subsystem slightly varies along this segment of the POF due to the change of the heat recovery efficiency of the gas turbine cogeneration. From Ind.5.3.A to Ind.5.3.E, the gas turbine specific investment cost decreases due to economies of scale. Beyond this point, the design chosen for the gas turbine is of increasingly high quality, and the specific investment cost increases together with the pressure ratio and the electrical efficiency as shown in fig. 8. As can be clearly seen from fig. 3 to 7, the 'mode of heating effect' is much more effective than the 'efficiency effect'. This is not only due to the fact that changing the heating mode from 'irreversible' to 'reversible' can effectively reduce the heating fuel specific consumption, but also due to different trends of heating specific capital cost variation when increasing the capacity of gas turbine or when increasing the electrical efficiency. The 'Heating Cost Effectiveness' (HCE) can be measured through the reciprocal of the corresponding marginal cost. The average values of HCE related to the two different effects along the POF of (AP5, ESP3) are respectively 32.3 and 1.5 [kg c.e./US$]. A CO_2 tax of only 5.4 US$/ton will give economic incentive to implement Ind.5.3.E instead of Ind.5.3.A, allowing a 15 [kg c.e./GJ] heating fuel specific consumption reduction.

The heating exergy index of the total heating system increases along the POF together with the heating specific capital cost due to the increase of the gas turbine capacity or electrical efficiency.

The corresponding heating first law efficiency increases from about 0.9 to 1.75, which induces a decrease of the operating cost, shown in fig. 7.

The influence of the variation of economic conditions is examined with a case based on the same assumptions than those in fig. 3 for the (AP5, ESP3) scenario but considering a higher natural gas price of 4 [UScts/ kWh]. Fig. 9 and Table 4 shows the obtained POFs and the description of the configurations associated with several selected individuals when 1) the components can be chosen freely (lower curve), and 2) the configuration is forced not to consider a heat pump (upper curve). Under these two options, this time the cheapest configuration is composed of a large heat pump (34.6 MW_{th}[8]) fed by the power grid, and of a small additional gas boiler with an overall heating fuel specific consumption of 37.2 [kg c.e./GJ] (ind. A). This heating fuel specific consumption is lower than that of the gas boiler based heating plant in this case but remains at a high level due to the low grid electrical efficiency in Beijing (31%). A jump down of 41% with a value of 22.1 [kg c.e./GJ] is obtained with the introduction of a gas turbine (ind. B) at the expense of a 10% higher heating specific total cost. In such a case, the additional gas boiler is not required any more. The 'hot end' efficiency, which is the virtual combined cycle in this case, is much higher than that of the power grid, which results in a much lower heating fuel specific consumption. At the same time, the heating specific total cost is higher due to a gas turbine power generation cost than that of the electricity grid, which is mainly based on coal fired power stations with a price of coal about 4 times lower than the price of natural gas. Since the power selling price of 3 [UScts/kWh] appears to not allow the recovery of the power generating cost, there is no economic interest to introduce a larger capacity gas turbine than the one required by the heat pump and the heating network pump. A higher electrical efficiency of the gas turbine

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[8] This is the largest capacity available for the feasible domain of operating conditions in this case study.

(36.7%) is also chosen due to the higher natural gas price. Beyond Ind. B, further specific fuel consumption reduction is obtainable with an increase of the gas turbine capacity and simultaneously a decrease of the heat pump capacity. Reasons for such a reduction are similar than those in the precedent case. The cost increases significantly due to a high natural gas price.

When the heat pump is forced not to be considered, the gas boiler configuration (Ind.NoHP.A) becomes the cheapest option with a higher specific fuel consumption of 40.8 [kg c.e./GJ]. Comparing to the starting point (Ind.5.3.A) of the POF of (AP5, ESP3+) in fig. 3, gas turbine disappeared due to this high natural gas price, which make the power generating cost (about 6.9 [UScts/kWh]) even higher than the electricity price of the power grid. The introduction of a gas turbine unit (Ind.NoHP.B) reduces the heating specific fuel consumption at the expense of higher heating specific total cost. It is interesting to note here that under such a high natural gas price, either simple fuel switching (Ind.NoHP.A, natural gas fired boiler) or fuel switching through cogeneration (Ind.NoHP.B) are not competitive compared to the choice of a stand alone heat pump even though the latter is supplied by low efficiency coal-fired power plants.

It should be emphasized also that with a high natural gas price, the association of a heat pump with the cogeneration unit appears to be much more attractive than a cogeneration by itself since it allows a drastic cut of specific consumption from about 40.8 [kg c.e./kWh] (ind. NoHP.A) to about 22.1 [kg c.e/kWh] (ind. B) at a similar heating specific total cost. The difference between the two POFs progressively decreases due to an increasing performance of the heat generation subsystem exergy index (from Ind.NoHP.B to Ind.NoHP.C) or to an increase of electrical efficiency of gas turbine (from Ind.NoHP.C to Ind.NoHP.E) of the no heat pump configuration along the curve. Finally both curves converge as expected.

A sensitivity analysis of the bottoming cycle efficiency has been undertaken in order to examine its influences to the calculation of the EECR and the heating fuel specific consumption.

Fig. 10 and Table 5 shows the POFs and the description of the configurations associated with several selected individuals obtained with the same assumptions than those in fig. 3 for the (AP5, ESP3) scenario, but with a bottoming cycle efficiency of 20% and 25% respectively. In the latter case, the exergy index of the gas turbine exhaust gas heat exchanger is this time lower than that of the heat pump due to a higher power sacrifice for heating purposes. The trend of the POFs is similar up to Ind.25.E where a heat pump is introduced by contrast with the other case, allowing further specific fuel consumption reduction at the expense of a higher investment cost. The achievable minimum level of heating fuel specific consumption is higher than that of the other scenario in which a 20% bottoming cycle efficiency is considered. This is explained by a heat pump average exergy index of lower than that of the gas turbine cogeneration with a 20% bottoming cycle efficiency. The heat pump appears after the gas turbine electrical efficiency reaches 38.1%, which is lower than its highest value (38.6%), associated a lower electrical efficiency gas turbine (34.1%). This is due to the trade-off between the option of a higher electrical efficiency gas turbine and the heat pump option since both of them can reduce the heating fuel specific consumption when a 25% bottoming cycle efficiency is considered. A highest capacity heat pump (24.8MW_{th}) is finally reached associated with the highest electrical efficiency (38.6%) gas turbine at Ind.25.F, corresponding to the lowest heating fuel specific consumption (21.4 [kg c.e./GJ]) and the highest heating specific total cost (10.6 [US$/GJ]) in this case.

At a same heating specific investment cost (e.g. with the same super-configuration, like ind.25.B and ind.20.B), the heating fuel specific consumption is higher when a higher bottoming cycle efficiency is considered. This means that the heat pump option will be more and more interesting for reducing the heating fuel specific consumption with increasing efficiency of combine cycles for power generation. Correspondingly, cogeneration will be less preferred for this purpose.

The excess power generated by the cogeneration plant may not be able to be sold to the grid due to its small capacity. A case with respectively a 5, 10 and 15 years capital cost amortization period when power exportation is not allowed, with a 20% virtual bottoming cycle efficiency and natural gas and grid power prices similar to the case with power exportation is also optimized and analyzed. For the first two scenarios (AP5 and AP10), the cheapest but most emitting option is also a large gas boiler supplying most of the heat with a small gas turbine cogeneration unit mainly used to generate the power needed by the heating network pump, which is similar with the case when power exportation is allowed due to the same reasons. The introduction of a heat pump driven by a gas turbine cogeneration unit allows a reduction of the heating fuel specific consumption, and thus of the associated emissions, by up to 44 %. This reduction is obtained at the expense of an increase of the heating specific total cost by respectively 39% (AP5) and 7% (AP10). The cheapest solution to achieve the lowest heating fuel specific consumption in this case appears at a value of 6.6 [US$/GJ] when a 15 year amortization period is assumed. However, with a heating price of 5.6 [US$/GJ], there is no economic interests to make this solution survive. This is mainly due to the high investment cost of heat pump on the one hand, and the loss of opportunity for power selling on the other hand. A financial aid becomes necessary to help market penetration.

With 5 years amortization period, The POFs shows a similar general behavior than the previous case due to similar reasons with two distinct successive segments characterizing 1) a 'mode of heating effect' segment with a higher HCE (6.2 [kg c.e./US$] in average), and 2) an 'efficiency effect' segment, which allows further reductions but at lower HCE (4.6 [kg c.e./US$] in average).

It should be noticed that the driving force for the reduction of the heating fuel specific consumption is an increase of the gas turbine capacity and of the electrical efficiency due to the higher exergy index of gas turbine cogeneration (with the assuming 20% bottoming cycle efficiency) than that of the heat pump here. The specific investment of the heat pump is also high. However, the heat pump capacity still has to follow the increasing gas turbine capacity in order to prevent waste of power since exportation is not allowed. This appears to be penalizing the overall exergy index of the heat generation subsystem and results in the following effects comparing to the case when power exportation is allowed: the lowest heating fuel specific consumption which can be reached in this case is of 21.4 [kg c.e./GJ], to be compared with the 19.8 [kg c.e./GJ] obtained when exportation is allowed; correlatively, the overall system exergy efficiency can only reached 19 % against 21 % previously.

5. Conclusions

The characteristics of 'Green Heating System' has been academically defined and illustrated with the case study of a natural gas fired integrated heating system in the city of Beijing. With the help of the exergy-based Specific Consumption Analysis Approach and using a multi-criteria optimization process based on a new evolutionary multi-objective algorithm, the performances of the proposed heating system have been examined. The results show an achievable reduction of the specific fuel consumption down to respectively 19.8 and 21.4 [kg c.e./GJ] with a significant reduction of associated emissions, at a heating specific total cost of respectively 3.1 and 6.6 [US$/GJ] whether power exportation is allowed or not. It has been shown that significant improvements can be obtained at moderate cost with a switch to a higher share of 'reversible mode of heating'. Further reduction of the specific fuel consumption can be achieved with an improvement of the efficiency of the components but at higher marginal costs.

Although the CHP and heat pump heating are both reversible mode of heating, their respective heating performances vary with the different technical alternatives (e.g. different bottoming cycle efficiencies). Different economic context and environmental policies will also influence the decision regarding the choice between these two systems.

6. Acknowledgements

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Nomenclature

b	Fuel specific consumption accrual
c	Monetary specific consumption, i.e. specific total cost, Specific cost accrual, Local market price
c.e.	Coal equivalent, 1kg c.e.=7000 kcal =1/34.2 GJ
P	Power
Q	Heating demand; Heat
T	Temperature
$\overline{T}$	Mean temperature

Greek Letters

ε	A coefficient of specific cost taking into account the pollution tax
ε_Q	The exergy index of the heat generation subsystem
η	The exergy efficiency
μ	The first law energy efficiency
φ_{GB}	The proportion between the heat provided by the gas boiler and the heat provided by the whole heat generation subsystem.

Subscripts

0	Under the ambient condition; Item caused by factors not associated with individual subsystems
b	Fuel energy delivery subsystem
e_grid	Power grid
elec	Electrical
FI	Due to irreversibility of subsystem I
f	Fuel
$I=1, 2, \cdots, n$	The order number of the subsystems in question
$J=1, 2, \cdots, n$	The order number of the subsystems in question
min	Theoretical minimum value
Q	Heat generation subsystem; Heating
QD	Heat distribution subsystem
q	With respect to heat supply
R	Located at the end of heat distribution subsystem
r	Located at the user's side of the substation heat exchanger
th	Thermal
user	The end-users' subsystem
user1	The first end-users' subsystem in the case study: office buildings
user2	The second end-users' subsystem in the case study: residential buildings.
W	Power generation subsystem
WD	Power transmission and distribution subsystem
ZI	Due to fixed charge for the unity product attributed to subsystem I

Abbreviations

AP	Amortization Period [years]
BC	Bottoming Cycle
CC	Combined Cycle
CHP	Combined Heat and Power
EECR	Equivalent Electricity Consumption Rate, which means the sacrifice of a CHP plant in supplied power due to generating heat
ESP	Electricity Selling Price [UScts/kWh]
G	Generator
GB	Gas Boiler
GT	Gas Turbine
GTHX	Gas Turbine Heat eXchanger, i.e. gas turbine heat recovery device
HCE	Heating Cost Effectiveness
HP	Heat Pump
HX	Heat eXchanger
POF(s)	Pareto Optimal Frontier(s)

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FIGURE CAPTIONS

Fig. 1. Superstructure of the energy conversion plant and the heating network

Fig. 2. Illustration of the virtual exergy efficiency/index calculation (BC- Bottoming Cycle; GT-Gas turbine; GB-Gas Boiler; HP-Heat Pump; HX-Heat eXchanger; P-Power; G-Generator; users-the end user's subsystem)

Fig. 3. POFs obtained under different scenarios (AP: Amortization Period; ESP: Electricity Selling Price; the natural gas price is of 2 [UScts/kWh$_{th}$])

Fig. 4. Evolution of the heating fuel specific consumption and corresponding heating specific capital costs and operating costs under different scenarios

Fig. 5. Evolution of the overall exergy index and of the heating specific capital cost along the POF of scenario (AP5, ESP3)

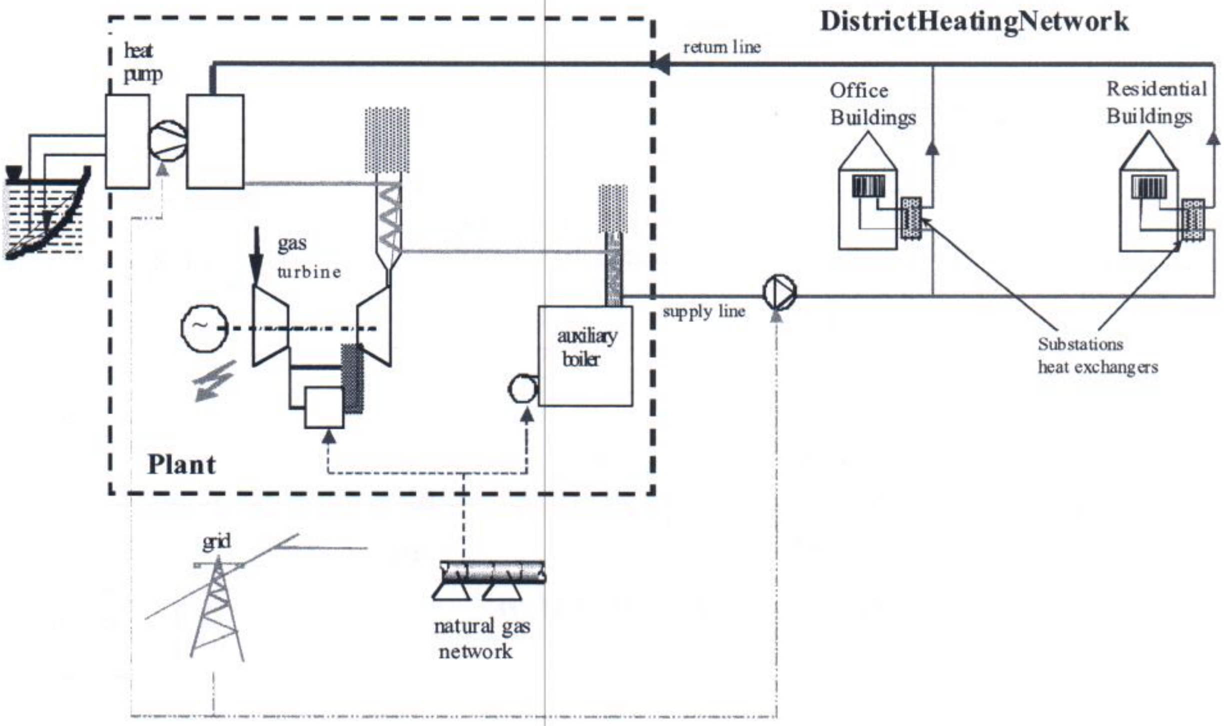
Fig. 6. Evolution of the heating first law energy efficiency and of the operating cost savings along the POF of scenario (AP5, ESP3)

Fig. 7. Share of each component in the total heating supply (right side) and distribution of the specific fuel consumption accruals (left side) along the POF of scenario (AP5, ESP3)

Fig. 8. Evolution of the pressure ratio, the electrical efficiency and the gas turbine specific electrical investment cost [US\$/kW$_{elec}$] in relative terms (individual 5.3.A as the reference) for scenario (AP5, ESP3)

Fig. 9. POFs obtained with the similar assumptions than those in fig. 3 for the (AP5, ESP3) scenario, for a natural gas price of 4 [UScts/kWh]. The upper curve corresponds to a situation where the introduction of a heat pump is intentionally forbidden

Fig. 10. POFs obtained with similar assumptions than those in fig. 3 for the (AP5, ESP3) scenario, with bottoming cycle efficiency of respectively 20% and 25%.



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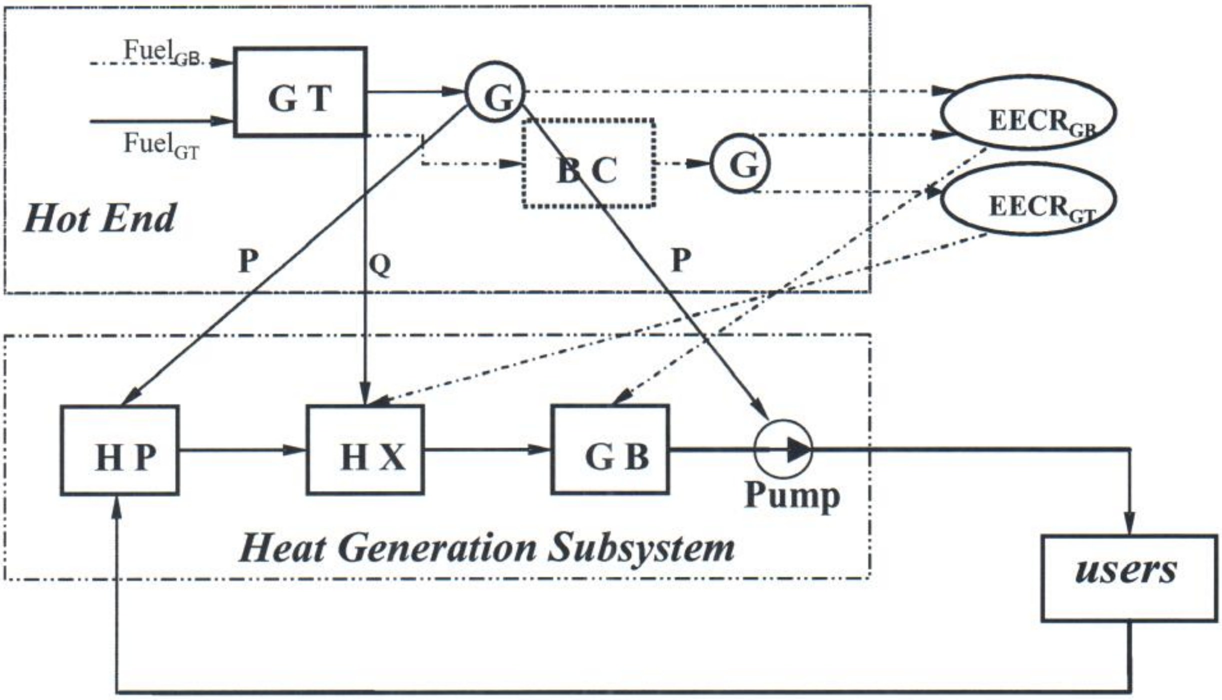

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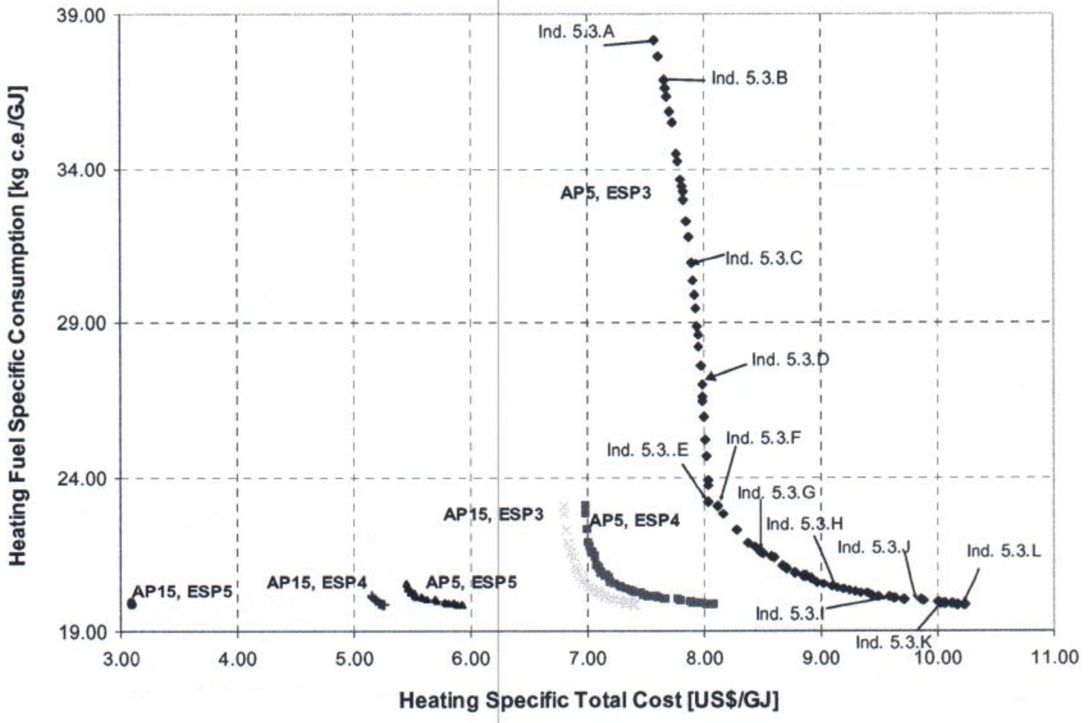

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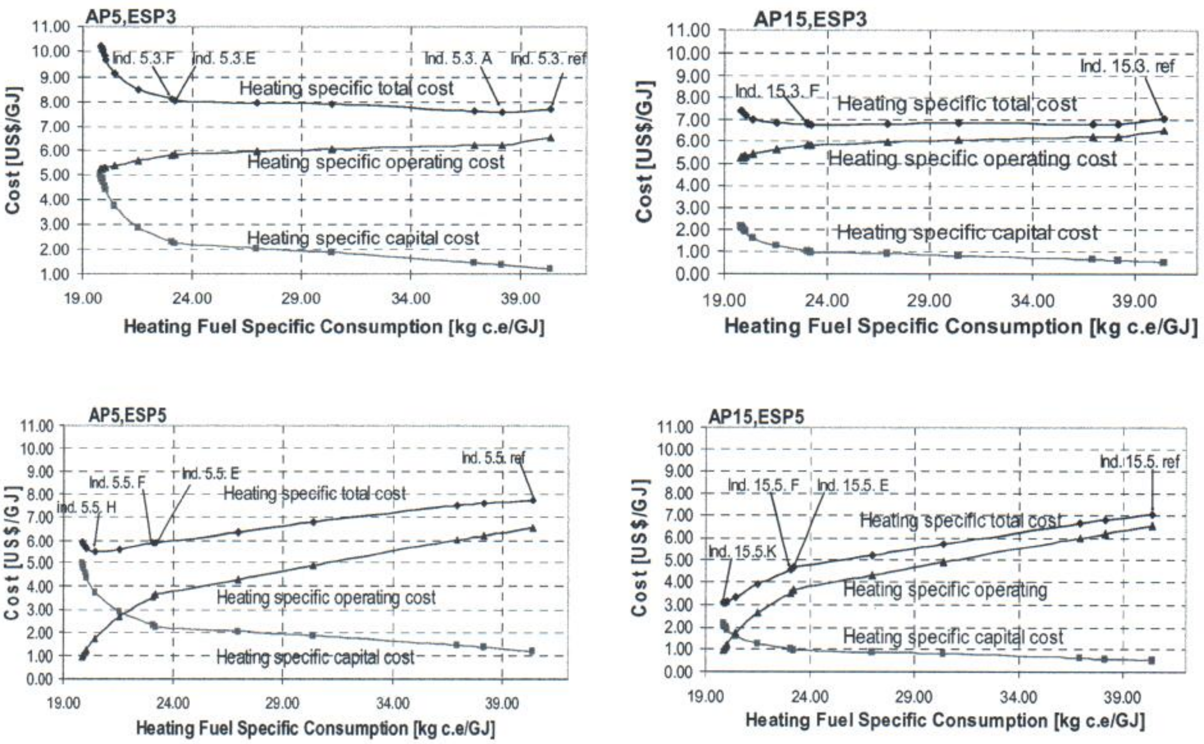

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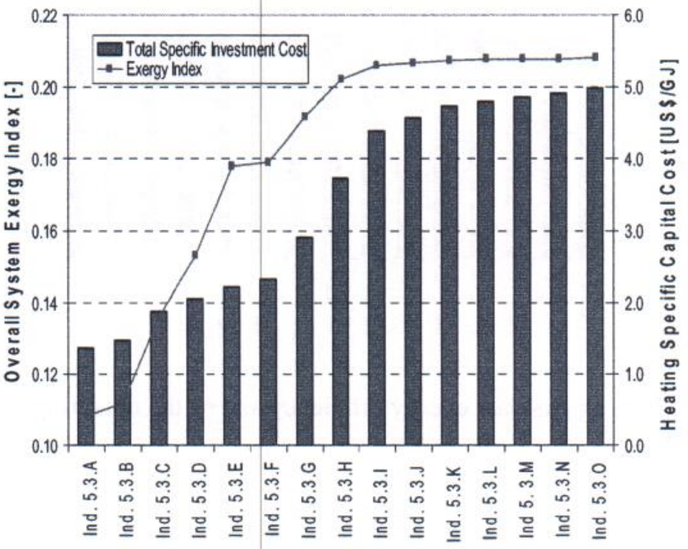

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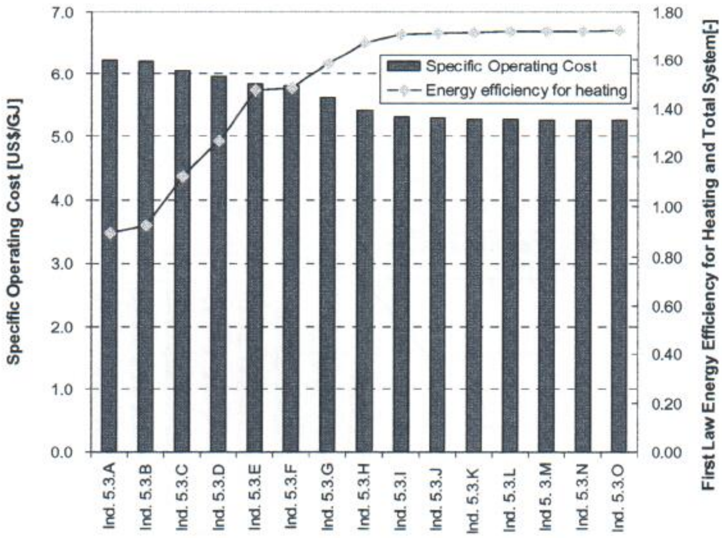


Fig. 6. Evolution of the heating first law energy efficiency and of the operating cost savings along the POF of scenario (AP5, ESP3)

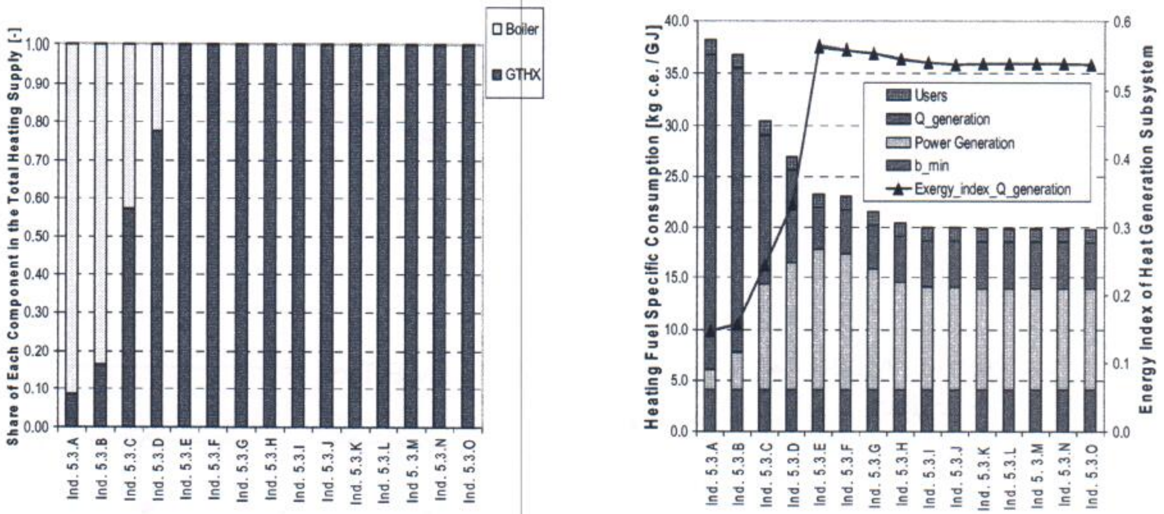
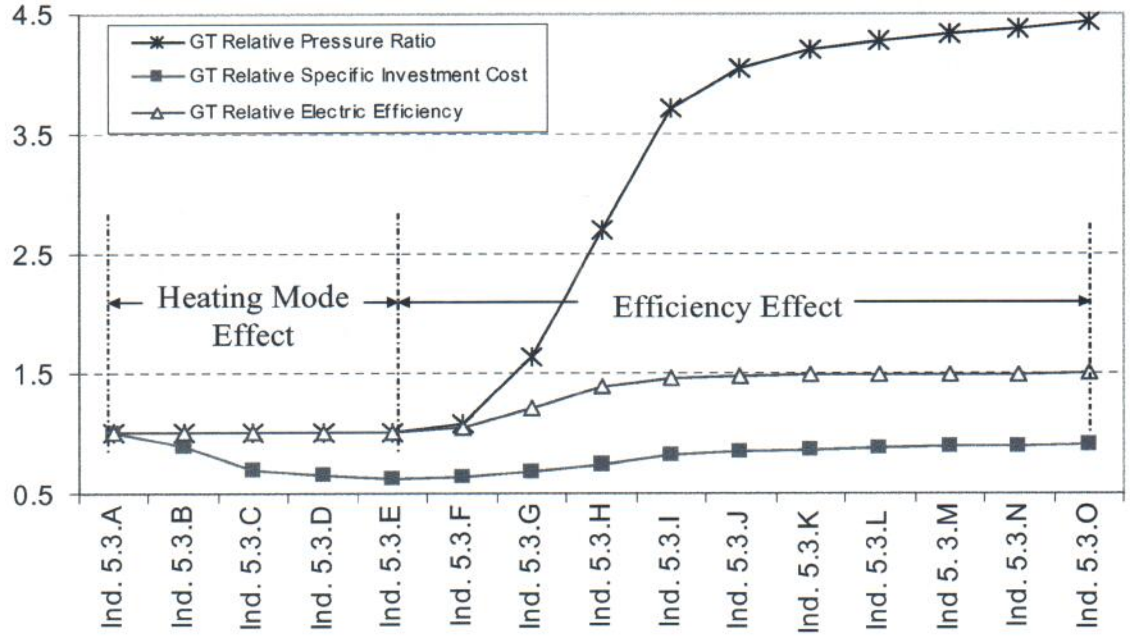


Fig. 7. Share of each component in the total heating supply (right side) and distribution of the specific fuel consumption accruals (left side) along the POF of scenario (AP5, ESP3)



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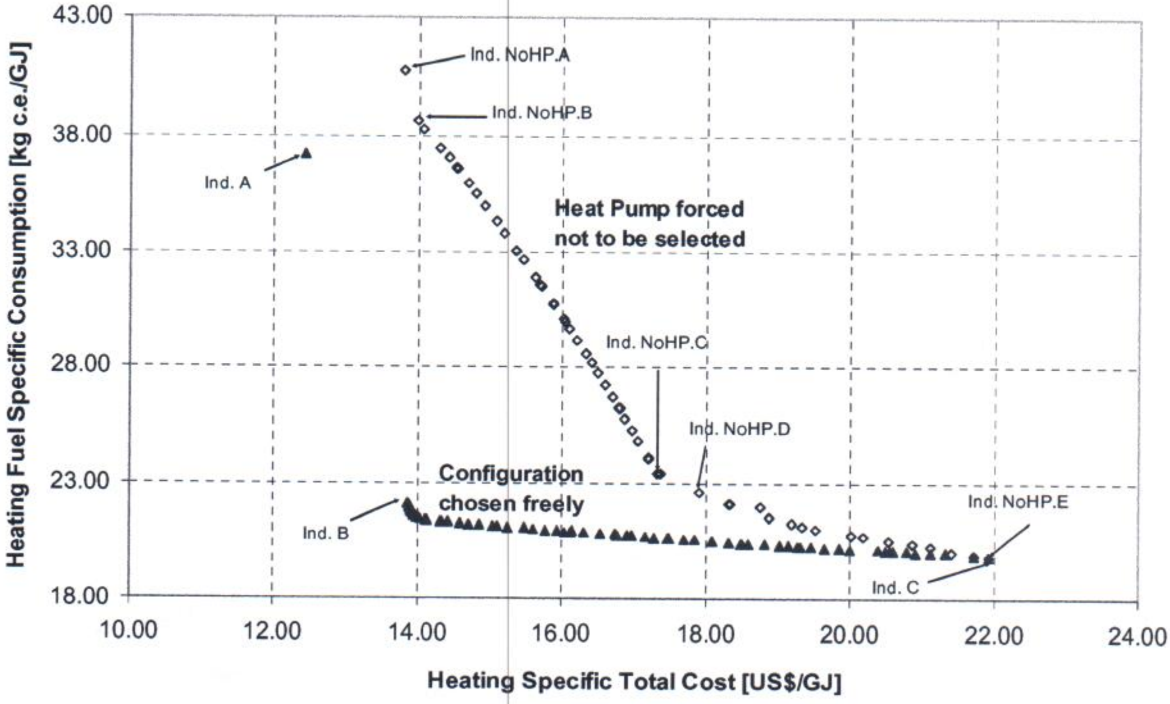

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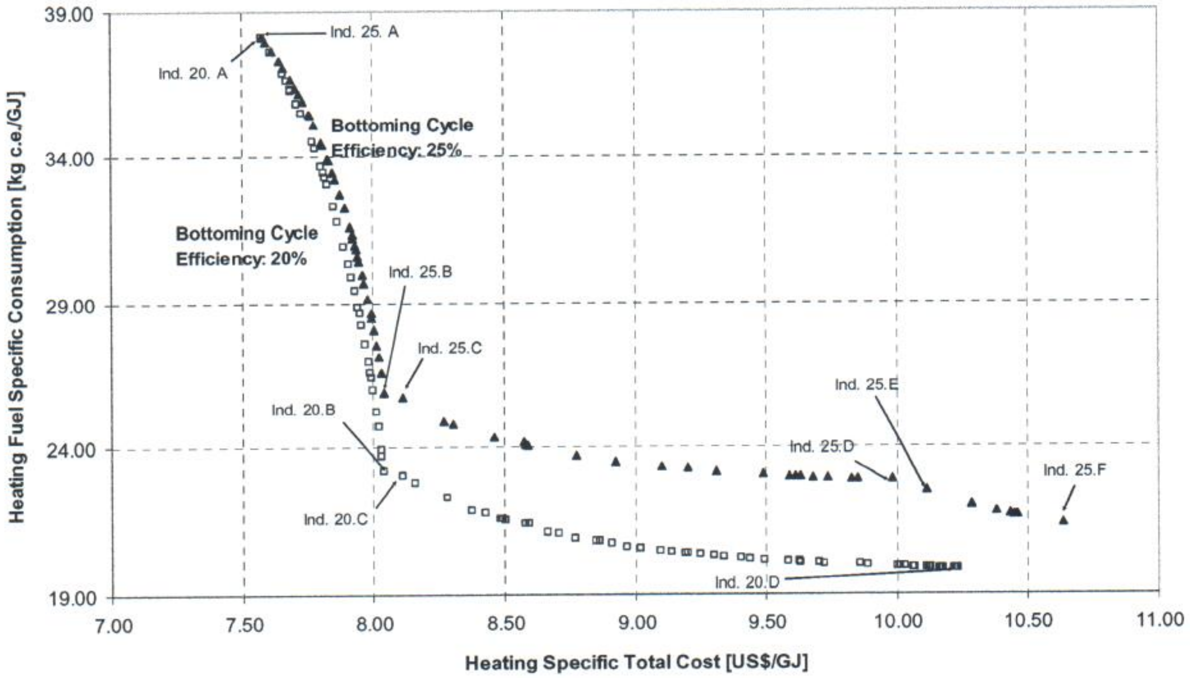

Fig. 10. POFs obtained with similar assumptions than those in fig. 3 for the (AP5, ESP3) scenario, with bottoming cycle efficiency of respectively 20% and 25%

TABLE CAPTIONS

Table 1. Characteristics of the two building areas considered

Table 2. The decision variables with their respective boundaries

Table 3. Description of the configurations associated with several selected individuals labelled in fig. 3

Table 4. Description of the configurations associated with several selected individuals in fig. 9.

Table 5. Description of the configurations associated with several selected individuals in fig. 10.

Table 1
Characteristics of the two building areas considered

	Space Heating Duration Time [days]	Floor Area [10^3 m^2]	Minimum Temp. Required in Winter [°C]	Heating Nominal Load [MW]
Office buildings	150	460	65	27.7
Residential buildings	150	136	80	6.9

Table 2

The decision variables with their respective boundaries

Decision variables	unit	lower bound	upper bound
Plant			
Heat Pump Nominal Power	MWth	0.0	40.0
Heat Pump Condenser Pinch	K	2.0	15.0
Heat Pump Evaporator Pinch	K	2.0	15.0
Gas Turbine Fuel Mass-flow	kg/sec	0.0	2.0
Gas Turbine Pressure Ratio	-	5.0	25.0
Gas Turbine Excess Air Ratio	-	2.0	3.0
Gas Turbine Inlet Temperature	K	1273.15	1575.15
GT Heat Recovery Device Pinch	K	5.0	20.0
Winter Supply Temperature	K	308.15	368.15
Office Buildings Area			
Substation Heat exchanger Pinch	K	2.0	15.0
Residential Buildings Area			
Substation Heat exchanger pinch	K	2.0	15.0

Table 3

Description of the configurations associated with several selected individuals labelled in fig. 3

Individuals	Gas Turbine Heat Generation [MWth]	Gas Turbine Electrical Efficiency [%]	Boiler Heat Generation [MWth]
Ind. 5.3.A	3.0	25.9	31.6
Ind. 5.3.E	34.6	26.0	0.0
Ind. 5.3.F	34.6	26.7	0.0
Ind. 5.3.L	34.6	38.6	0.0

Table 4

Description of the configurations associated with several selected individuals in fig. 9

Individuals	Heat Pump Heat Generation [MWth]	Gas Turbine Heat Generation [MWth]	Gas Turbine Electrical Efficiency [%]	Boiler Heat Generation [MWth]
Ind. NoHP.A	-	-	-	34.6
Ind. NoHP.B	-	3.0	25.8	31.6
Ind. NoHP.C	-	34.6	26.0	-
Ind. NoHP.D	-	34.6	28.3	-
Ind. NoHP.E	-	34.6	38.6	-
Ind. A	34.6	-	-	-
Ind. B	24.0	10.6	36.7	-
Ind. C	-	34.6	38.6	-

Table 5

Description of the configurations associated with several selected individuals in fig. 10.

Individuals	Gas Turbine Heat Generation [MWth]	Gas Turbine Electrical Efficiency [%]	Boiler Heat Generation [MWth]	Heat Pump Heat Generation [MWth]
Ind. 25.A	3.0	25.9	31.6	-
Ind. 25.B	34.6	26.0	-	-
Ind. 25.C	34.6	26.8	-	-
Ind. 25.D	34.6	38.1	-	-
Ind. 25.E	11.3	34.1	-	22.3
Ind. 25. F	9.8	38.6	-	24.8
Ind. 20.A	3.0	25.9	31.6	-
Ind. 20.B	34.6	26.0	-	-
Ind. 20.C	34.6	26.8	-	-
Ind. 20.D	34.6	38.6	-	-