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Exergonomic Optimization of an Air-Conditioning System

In this paper, exergonomic theory is applied to an air-conditioning system for optimization purposes. The investigation is addressed to an all-air system with air recirculation. The thermodynamic cycle includes a mixing plenum, a cooling and heating coil, chiller, and heater. The thermodynamic model is stated according to recent formulations of exergy for moist air streams, while the economic model is based on cost balance equations and real cost data for mechanical equipment. The objective function to minimize includes the following decision variables: fresh to total air rate, coefficient of performance for the chiller, inlet temperature of water for the cooling and the heating coils, temperature difference of the same streams. For the exergonomic optimization, the authors followed the approach proposed by Tsatsaronis (1984). The optimum configuration is obtained through an iterative procedure aimed at the design improvement. The results show that there is considerable room for improvement with respect to a system based on typical design parameters.

Introduction

Modern literature on thermodynamic aspects of energy systems is strongly oriented towards exergy (availability) analysis. Although a large number of published papers analyze energy conversion systems according to the second law, little attention is paid to humid air processes and air-conditioning devices. A clear feeling of it rises, for instance, from a report recently produced at the Twente University (Netherland) on exergy and related techniques (Cornelissen, 1995).

On the other hand, all the air treatment apparatuses as such (heating and cooling coils, humidifiers, air washers, etc.), and namely the assembly for air conditioning as a whole, are affected by very poor exergy efficiency. So they need improvements that should be suggested by the second law insight.

Fundamentals of availability analysis for humid air were stated in the seminal work by Wepfer et al. (1979). Since then, the subject was almost neglected, although reference was made to it in introductory books on exergy analysis, such as Moran (1989), Kotas (1995), Bejan (1988), and others.

The most recent literature on thermodynamic analysis of energy system records the introduction of exergonomic theory. It enjoyed a rather fast distribution, and, due to substantial contributions from many researchers (Valero, 1990; Tsatsaronis, 1984; Frangopoulos, 1990; Von Spakovsky, 1990), quite a high degree of formalization has already been achieved. So far, such a powerful methodology has been applied almost exclusively to power plants and chemical plants.

In this paper, we want to make use of exergonomics for optimizing a thermodynamic cycle for air conditioning. To this aim we will follow the approach proposed by Tsatsaronis (1990) fundamentally because, instead of minimizing rather automatically a predetermined objective function, it is based on an iterative procedure which stimulates the designer to exert a careful control of decision variables at every step.

The Thermodynamic Model

The air-conditioning apparatus under consideration is an all-air system with air recirculation. The thermodynamic cycle is given in Fig. 1, while the block diagram is reported in Fig. 2.

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The equipment arrangement is classical: psychrometric states for indoor and outdoor air are referred to as A and E, respectively (Fig. 1). The recirculated air (A) mixes adiabatically with the outdoor flow (E), resulting in the thermodynamic state M. Such a stream is conveyed to the cooling and dehumidifying section where it is treated down to R. This task is accomplished by a cooling coil (CC) fed by chilled water (streams 6 and 7 in Fig. 2) provided by an electrically driven chiller (CL). Condensate (labeled 15 in Fig. 2) is released to the environment. The air flow, leaving the CC, moves to the heating coil (HC) where, at constant specific humidity, it is heated up to I and made ready for entering the conditioned space. The heating coil is provided with hot water (labeled 8 and 9 in Fig. 2) produced in a boiler (BL), fed by conventional fuel and combustion air (streams 12 and 13). Exhaust gas from the boiler is, of course, released to the atmosphere (stream 14).

For modeling the physical system, it is necessary to know energy and exergy at the inlet and outlet section of every single component. We shall provide the computational tools in the following, by summarizing from a previous work (Cammarata et al., 1994).

Enthalpy for moist air is a function of temperature T_i and specific humidity x_i

$$h(T_i, x_i) = (C_{pa} + x_i C_{pv})T_i + x_i l \quad (1)$$

To give explicit formulas for exergy of humid air streams, it is necessary to introduce the definition of the *ultimate dead state:* it is that physical state in which the system attains its mechanical, thermal, *and chemical* equilibrium with the environment.

It can be demonstrated (Moran, 1989) that for the exergy of of humid air streams, the following expression holds:

$$e(T_i, x_i, p_i) = (C_{pa} + x_i C_{pv})\left(T_i - T_0 - T_0 \ln \frac{T_i}{T_0}\right) + (1 + \bar{x}_i)R_a T_0 \ln \frac{p_i}{p_0} + R_a T_0\left[(1 + \bar{x}_i) \ln \frac{1 + \bar{x}_0}{1 + \bar{x}_i} + \bar{x} \ln \frac{\bar{x}_i}{\bar{x}_0}\right] \quad (2)$$

Therefore, for any given mechanical component, at the ith section, energy and exergy flows can be expressed as follows:

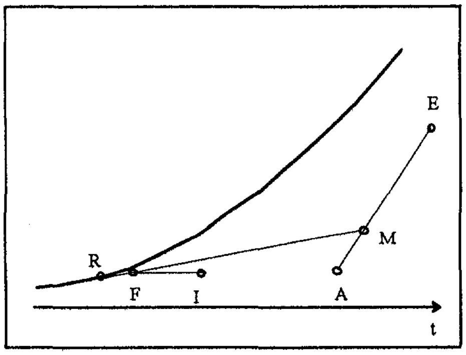

Fig. 1 Thermodynamic cycle in the psychrometric chart

$$En_i = \dot{m}_i h(T_i, x_i) \quad (3)$$

$$E_i = \dot{m}_i e(T_i, x_i, p_i) \quad (4)$$

For water feeding cooling and heating coil, specific enthalpy and exergy are given by

$$h(T_i) = C_{pw} T_i (°C) \quad (5)$$

$$e(T_i) = C_{pw}\left(T_i - T_0 - T_0 \ln \frac{T_i}{T_0}\right) \quad (6)$$

Finally, for the exhaust gas released by the boiler, we have

$$h(T_i) = C_{pg}(T_i - T_0) \quad (7)$$

$$e(T_i, x_i, p_i) = C_{pg}\left(T_i - T_0 - T_0 \ln \frac{T_i}{T_0}\right) + R_g T_0 \sum_j x_i^g \ln \frac{x_i^g}{x_0^0} + R_g T_0 \ln \frac{p_i}{p_0} \quad (8)$$

The Physical Model

The energy balance for the conditioned space (Q_s = sensible heat) provides the air mass flow rate conveyed to the room

$$\dot{m} = \frac{Q_s}{(C_{pa} + x_A C_{pv})(T_A - T_I)} \quad (9)$$

Assuming the mixing plenum is adiabatic, we have

$$T_M = (1 - \beta)T_A + \beta T_E$$

$$x_M = (1 - \beta)x_A + \beta x_E \quad (10)$$

The cooling coil (CC) has the following governing equations:

$$T_F = T_R + \text{BF}(T_M - T_R)$$

$$x_F = x_I = x_R + \text{BF}(x_M - x_R) \quad (11)$$

$$T_7 = T_6 + \Delta T_{67} \quad (12)$$

Taking into account the bypass factor (BF), the energy balance on the (CC) yields

$$\gamma = \frac{(1 - \text{BF})(h_M - h_R)}{C_{pw}(T_7 - T_6)} \quad (13)$$

For the heating coil (HC), the following relationships hold

$$T_9 = T_8 - \Delta T_{89} \quad (14)$$

$$\alpha = \frac{(C_{pa} + x_F C_{pv})(T_I - T_F)}{C_{pw}(T_8 - T_9)} \quad (15)$$

The electric power requirement for chiller operation can be assessed as follows:

$$P = \frac{\gamma \dot{m} C_{pw}(T_{2f} - T_{1f})}{\text{COP}} \quad (16)$$

Reference Design Data

The reference design data are reported in Table 1. These are assumed to be typical values for an air-conditioning system of the type under consideration and represent the starting point for the optimization procedure.

It is worth mentioning that such a plant is aimed at extracting from the conditioned space the sensible heat Q_s = 30 kW with the following restriction: $T_I \geq 18$°C, due to thermal comfort requirements.

By applying the previously stated formulas to the reference design data, it is possible to obtain the results shown in columns three and four of Table 2.

Premises to the Economic Model: "Fuel" and "Product"

In order to carry out the economic model and the exergonomic optimization, it is necessary to state the concept of "prod-

Nomenclature

BF = bypass factor
BL = boiler
c = exergetic cost ($/GJ)
CC = cooling coil
c_f = fuel cost ($/GJ)
c_{el} = electricity cost ($/GJ)
CL = chiller
$\dot{C}_T$ = global cost ($/h)(objective function)
COP = coefficient of performance
C_p = specific heat (J/(Kg K))
$\dot{D}$ = exergetic cost rate ($/h)
e = specific exergy (kJ/kg)
$\dot{E}$ = exergy flow (kW)
En = specific energy (kJ/Kg)
f = exergonomic factor
h = entalpy (kJ/kg)
HC = heating coil
i = interest rate (percent)
I = net capital cost ($)
l = latent heat of vaporization (kJ/Kg)
$\dot{m}$ = mass flow rate (Kg/s)
MX = mixing plenum
N = no. of years
P = electric power (kW) for chiller operation
p = pressure (Pa)
r = relative cost difference
T = temperature (°C)
Z = capital cost ($)
$\dot{Z}$ = capital cost rate ($/h)
x = specific humidity
$\bar{x} = x/0.622$
$\alpha = \dot{m}_h/\dot{m}$
$\beta = \dot{m}_E/\dot{m}$
ϵ = exergetic efficiency
$\gamma = \dot{m}_c/\dot{m}$
σ = maintenance factor
τ = annual time of plant operation

Subscripts

a = air
c = cool water through CL and CC
D = exergy destroyed
f = fuel
h = hot water through BL and HC
i = ith section
in = inlet
k = kth plant component
L = exergy lost to environment
out = outlet
P = product
v = water vapor
w = water (saturated)

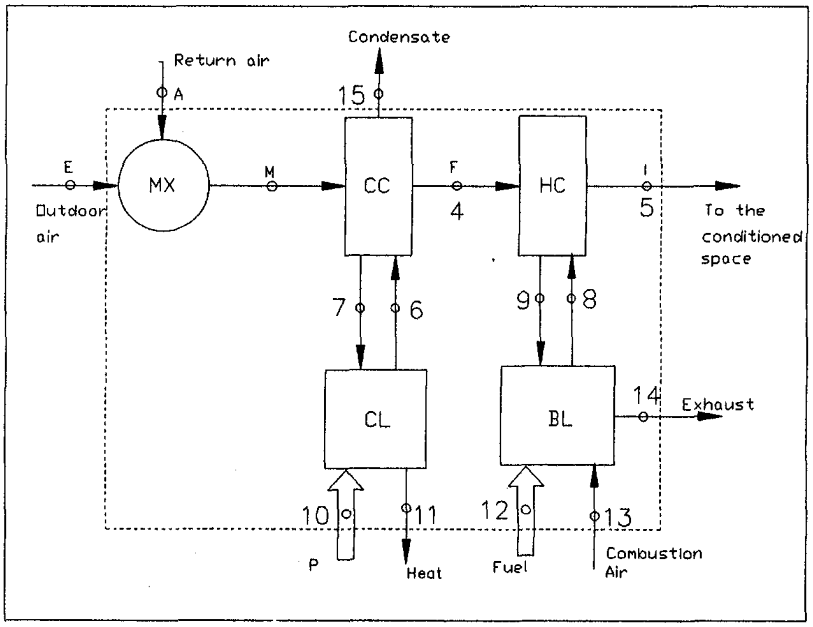

Fig. 2 Block diagram for the HVAC

Table 1 Reference design data

T_E	32	°C	φ_E	70%	
T_A	25	°C	φ_A	50%	
T_0	32	°C	BF	10%	
T_R	10	°C	β	30%	
T_6	5	°C	p_E	101.325	kPa
ΔT_{67}	3	°C	C_{pa}	1.000	kJ/(kg K)
T_8	60	°C	C_{pv}	1.870	kJ/(kg K)
ΔT_{89}	8	°C	C_{pw}	4.186	kJ/(kg K)
COP	2.5		ρ_a	1.2	kg/m^3
			Q_s	30	kW

Table 2 Mass flow rate ($\dot{m}$), exergy ($\dot{E}$), specific exergetic cost (c), and cost flow ($\dot{D}$) for streams in Fig. 2 at reference design conditions

N. Stream Fig. 2	Stream	$\dot{m}$ kg / s	$\dot{E}$ kW	c $/MJ	$\dot{D}$ $/h
1	Air E	1.266	0	0	0
2	Air A	2.954	1.810	1.512	9.852
3	Air M	4.220	1.189	2.302	9.852
4	Air F	4.220	6.112	1.004	9.817
5	Air I	4.220	4.506	1.657	11.945
6	H_20 T_6	5.831	26.453	0.314	13.285
7	H_20 T_7	5.831	16.734	0.314	8.404
8	H_20 T_8	0.805	4.081	0.212	1.382
9	H_20 T_9	0.805	2.116	0.212	0.717
10	P_{CL}	-	48.822	0.046	3.5542
11	Q	-	11.976	0	0
12	P_{BL}	0.001	23.859	0.015	0.563
13	Comb. Air	0.013	0.000	0	0
14	Exhaust	0.013	0.520	0	0
15	Condens.	0	0	0	0

uct'' and ''fuel'' for any system component. We define ''product'' as the useful product or commodity produced by every system unit and ''fuel'' as the resource provided to this aim. Related to product and fuel, there is an exergy flow. For example, as far as the heating and cooling coils are concerned, the ''product'' ($\dot{E}_P$) will be the *net* exergy flow associated with the treated air stream (the commodity); the ''fuel'' will be the *net* exergy flow ($\dot{E}_F$) associated with the supply water stream.

Exergy may also be lost ($\dot{E}_L$), e.g., released to the environment as waste, or destroyed ($\dot{E}_D$) by irreversibilities (also referred to as *anergy*). On the basis of the previous definitions, it is possible to state the exergy balance and the exergy efficiency, respectively, as follows:

$$\dot{E}_F = \dot{E}_P + \dot{E}_L + \dot{E}_D \quad \epsilon = \dot{E}_P/\dot{E}_F \tag{17}$$

Consequently

$$\frac{\dot{E}_D + \dot{E}_L}{\dot{E}_P} = \frac{1 - \epsilon}{\epsilon} \tag{18}$$

The most meaningful data for the case under study are collected in Table 3, while a pictorial representation of exergy and anergy is given in Fig. 3.

The Economic Model

Modeling the economic aspects is the most crucial part of this and similar studies. Cost information is required for every system unit. Ideally, cost data are obtained from in-field investi-

Table 3 Exergy of fuel ($\dot{E}_F$) and products ($\dot{E}_P$) for various components at reference design

	E_F (kW)	E_P (kW)	ε (%)
CC	9.7189	4.9235	50.659
CL	48.8217	9.7189	19.907
HC	1.9643	1.6058	81.752
BL	23.8589	1.9643	8.233
System	72.6806	2.6963	3.710

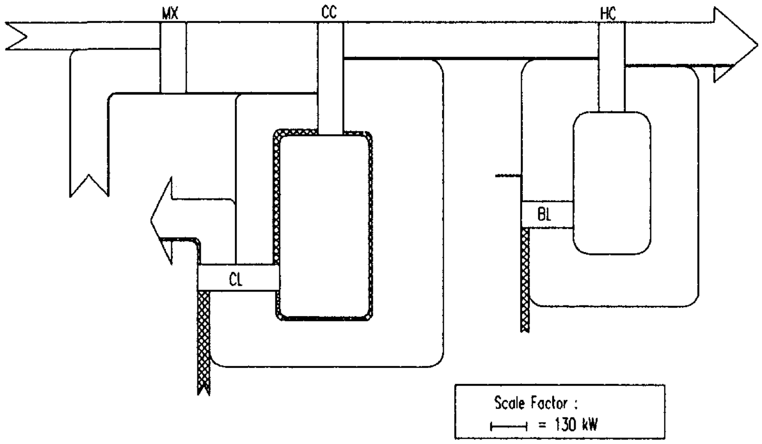

Fig. 3 Exergy and anergy flows in air handling units (light streams: anergy; shaded streams: exergy)

gation; in reality, such data are not always available or reliable and may not be in the required form for optimization.

In Figs. 4 and 5 are reported data obtained from an italian producer for boilers and chillers, respectively. Dots represent real data, while continuous lines plot the best-fit equations, given by:

- For chillers

$$Z_{GF} = (c_1 + c_2\text{COP}) + (c_3 + c_4\text{COP})P_{GF} + (c_5 + c_6\text{COP})P_{GF}^2 + (c_7 + c_8\text{COP})P_{GF}^3 \quad (19)$$

with

$c_1 = 95351.76$ \$ $\quad c_5 = 13.96386$ \$/KW^2

$c_2 = -11734.5$ \$ $\quad c_6 = -1.89039$ \$/KW^2

$c_3 = -1899.73$ \$/KW $\quad c_7 = -0.03221$ \$/KW^3

$c_4 = 284.6479$ \$/KW $\quad c_8 = 0.004538$ \$/KW^3

COP = 3 refers to an air-cooled condenser and COP = 4 to a water-cooled condenser. The last one includes the cost of cooling tower and circulation pump.

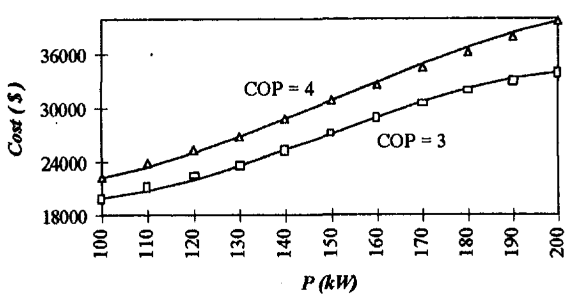

Fig. 4 Cost data for chillers

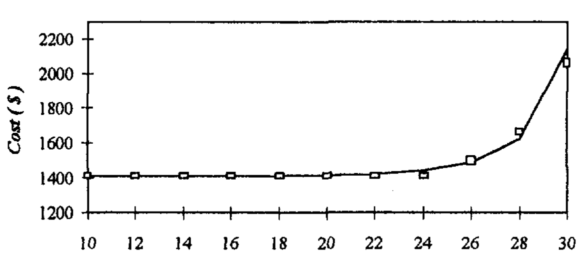

Fig. 5 Cost data for boilers

- For boilers

$$Z_{\text{BL}} = c_1 + c_2\,\text{Exp}(c_3\,\text{Exp}(c_4 P)) \quad (20)$$

with

$$c_1 = 1406.25\ \$ \quad c_2 = 6.25\ 10^{-4}\ \$$$

$$c_3 = 0.4055 \quad c_4 = 9.531\ 10^{-2}\ \text{kW}^{-1}$$

According to the Tsatsaronis (Tsatsaronis and Pisa, 1994) method of optimization, for each component, it is necessary to adopt a cost equation of the following type:

$$I = B\left(\frac{\epsilon}{1-\epsilon}\right)^n (\dot{E}_{pr})^m \quad (21)$$

Such a relationship means that the net capital cost I is proportional to the exergy of "product" or commodity ($\dot{E}_P$) and to the term $\epsilon/(1-\epsilon)$, which expresses the ratio of exergy of the produced commodity to the globally lost exergy (see Eq. (18)). Therefore, I is proportional to the size of the component and to its thermodynamic quality.

By using regression techniques, it was possible to derive the following values of B, m, n for various components (Table 4):

- The capital annual costs (excluding fuel costs) are obtained by adding the capital, operating, and maintenance costs as follows:

$$Z = (\text{CRF} + \sigma)I + \omega\tau E_P + R \quad (22)$$

where $\text{CRF} = i/(1-(1+i)^{-N})$ is the capital recovery factor; σ and ω are two coefficients that account for that part of the operating and maintenance cost, respectively, depending on I and on the plant size; τ is the annual time of plant operation at nominal capacity and R the remaining costs. As a result, for

Table 4 Constants to be used in Eq. (21)

Comp.	B \$/kW^m	n	m
CC	861.84	0.35	0.750
CL	3598	0.181	0.001
HC	307.73	0.20	0.820
BL	2154	0.024	0.0067

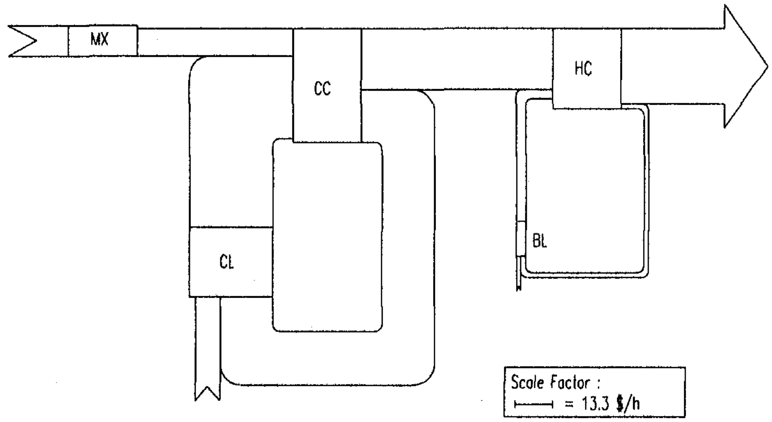

Fig. 6 Cost flows in the air handling units

any kth component, the operating and maintenance cost rate (excluding fuel) will be

$$\dot{Z}_k = Z_k/\tau \tag{23}$$

• Thermoeconomic optimization is aimed at minimizing the *total* cost, which is obtained by adding to the previously stated cost rate the fuel consumption cost. Thus, for the whole system we have

$$\dot{C}_T = \dot{Z}_{CC} + \dot{Z}_{CL} + \dot{Z}_{HC} + \dot{Z}_{BL} + c_f P_{BL} + c_{el} P_{CL} \tag{24}$$

referred also to as "objective function" (OF).

Such a function can be minimized through either analytical (such as Lagrange multipliers) or numerical methods (search methods). In any case, it is necessary to define the decision variables. These are as follows:

$$\beta, \text{COP}, \text{BF}, T_6, \Delta T_{67}, T_g, \Delta T_{89} \tag{25}$$

The method used to minimize Eq. (24) is outlined in the following.

Exergy Cost Equations

Exergy cost equations can be stated very much like mass, energy, and exergy balances for each component. They are cost balances stated by assigning a cost value to the exergy of each material and energy stream through the component. Let c_i ($/kJ) and $\dot{D}_i$ ($/h) be, respectively, the cost per exergy unit and the cost flow rate associated with the ith stream. Thus

$$\dot{D}_i = c_i \dot{E}_i \tag{26}$$

The *cost balance* for each unit can be stated as follows:

$$\dot{Z} + \sum \dot{D}_{i,\text{in}} = \sum \dot{D}_{i,\text{out}} \tag{27}$$

Since the air-conditioning apparatus is described in terms of 5 components (MX, CC, HC, CL, BL) and 14 streams (see Fig. 1), nine additional equations are required to calculate the cost flow rates for all streams in the system. These can be based on the following assumptions:

(i) The exergetic cost of material flows released to the environment by the component is zero (e.g., exhaust gas).

(ii) Every output flows that is not an exhausted stream has the same exergy cost as the input (e.g., water streams through CC or HC).

Since there exist three kinds of exergy (thermal, mechanical, and chemical exergy), costs should be associated to each kind of exergy. In the following, superscript Ch means chemical exergy, M mechanical, PH physical (i.e., mechanical + thermal) exergy. The exergy cost equations are reported as follows:

Mixing plenum (MX):

$$\dot{D}_1^{PH} + \dot{D}_2^{PH} - \dot{D}_3^{PH} = 0 \tag{28}$$

$$\dot{D}_1^{PH} = 0 \tag{29}$$

Cooling coil (CC):

$$\dot{D}_3^{PH} - \dot{D}_4^{PH} + \dot{D}_6^{PH} - \dot{D}_7^{PH} - \dot{D}_{15}^{PH} = -\dot{Z}_{CC} \tag{30}$$

$$c_6^{PH} = c_7^{PH} \tag{31}$$

$$\dot{D}_{15}^{PH} = 0 \tag{32}$$

Heating coil (HC):

$$\dot{D}_4^{PH} - \dot{D}_5^{PH} + \dot{D}_8^{PH} - \dot{D}_9^{PH} = -\dot{Z}_{HC} \tag{33}$$

$$c_8^{PH} = c_9^{PH} \tag{34}$$

$$c_2^{PH} = c_5^{PH} \tag{35}$$

Chiller (CL):

$$\dot{D}_{10}^{M} - \dot{D}_{11}^{T} + \dot{D}_7^{PH} - \dot{D}_6^{PH} = -\dot{Z}_{CL} \tag{36}$$

$$\dot{D}_{10}^{M} = c_{el} \dot{E}_{CL} \tag{37}$$

$$\dot{D}_{11}^{T} = 0 \tag{38}$$

Boiler (BL):

$$\dot{D}_9^{PH} - \dot{D}_8^{PH} + \dot{D}_{12}^{Ch} + \dot{D}_{13}^{PH} - \dot{D}_{14}^{PH} = -\dot{Z}_{BL} \tag{39}$$

$$\dot{D}_{12}^{Ch} = c_f \dot{E}_{BL}^{Ch} \tag{40}$$

$$\dot{D}_{13}^{PH} = 0 \tag{41}$$

$$\dot{D}_{14}^{PH} = 0 \tag{42}$$

By solving the previous set of balance equations, it possible to determine the exergetic costs c_i and the cost flows $D_i = c_i E_i$ for each stream. The results for the reference design are included in Table 2, while a flow diagram is given in Fig. 6.

Exergonomic Optimization

After calculating the cost rate D_i associated with each stream, it is possible to proceed to the assessment of the cost effectiveness through the exergonomic variables listed in the forthcoming.

Let us mention first the average cost per exergy unit of fuel ($c_{F,k}$) and the average cost per exergy unit of product ($c_{P,k}$), respectively, defined as

Table 5 Relevant exergonomic variables

Comp.	c_F \$/MJ	c_P \$/MJ	r %	f %	$\dot{D}_D$ \$/h	$\dot{Z}$ \$/h
CC	0.31389	0.69035	1.19934	5.42743	5.41882	0.31098
CL	0.0455	0.31389	5.89873	40.1874	4.44336	2.98545
HC	0.21162	0.82814	2.91331	19.5245	0.27308	0.06625
BL	0.01475	0.21162	13.3471	16.8226	1.13501	0.22955

$$c_{F,k} = \frac{\dot{D}_{F,k}}{\dot{E}_{F,k}} \quad c_{P,k} = \frac{\dot{D}_{P,k}}{\dot{E}_{P,k}} \tag{43}$$

$\dot{E}_{F,k}$ and $\dot{E}_{P,k}$ appear in Table 3, while $\dot{D}_{F,k}$ and $\dot{D}_{P,k}$ are the *net* cost flow, respectively, for the "fuel" stream and the "product" stream. For instance, taking data from Table 2, for the CC is $D_P = D_{(F)} - D_{(M)} = 22.088 - 9.852 = 12.236$ (\$/h). From Table 3, $E_P = 4.9$ (kW); thus, $C_P = D_P/E_P = 0.69$ (\$/MJ). Further, from Table 2, $D_F = (26.453 - 16.734)\ 0.314\ 3.6 = 10.98$ (\$/h); $E_F = 9.7$ (kW); thus, $C_F = D_F/E_F = 0.314$ (\$/MJ). This last result could be read directly from Table 2, since the entering and exiting fuel flows through the CC have the same exergy cost.

Another useful exergonomic variable is the cost flow rate associated with the destroyed exergy $\dot{D}_{D,k}$, which can be assessed by either

$$\dot{D}_{L,k} = c_{F,k}\dot{E}_{L,k} \quad \text{if} \quad \dot{E}_{P,k} \quad \text{is imposed}$$

or

$$\dot{D}_{D,k} = c_{P,k}\dot{E}_{D,d} \quad \text{if} \quad \dot{E}_{F,k} \quad \text{is imposed} \tag{44}$$

Finally, the relative cost difference (r_k) and the exergetic factor (f_k) are defined as follows:

$$r_k = \frac{c_{P,k} - c_{F,k}}{c_{F,k}}; \quad f_k = \frac{\dot{Z}_k}{\dot{Z}_k + \dot{D}_{D,k}} \tag{45}$$

All the relevant exergonomic data are reported in Table 5; namely, the higher the values of r_k, f_k and $D_{D,k}$, the larger the room for improvement. Hence, in our case, the most critical items seem to be the CC and the CL.

Now, let us consider the *exergonomic balance*

$$\dot{D}_P = \dot{D}_F + \dot{Z}$$

Thus

$$(c_p - c_F)\dot{E}_p = c_F(\dot{E}_D + \dot{E}_L) + \dot{Z}$$

which, combined with Eq. (45) and taking into account Eq. (18), results in

$$r_k = \frac{1 - \epsilon_k}{\epsilon_k} + \frac{\dot{Z}_k}{c_F\dot{E}_{P,k}} \tag{46}$$

Such a relationship shows that r_k identifies the real cost sources in the component, i.e., the cost rate of exergy destruction $D_{D,k}$ and the cost rate associated with the investment cost Z_k. Further, by combining Eqs. (46), (21), and (22), yields (dropping the subscript k)

$$r = \frac{1 - \epsilon}{\epsilon} + \frac{\dfrac{(\mathrm{CRF} + \sigma)}{\tau}\left(\dfrac{1 - \epsilon}{\epsilon}\right)^{-n}\dot{E}_P^m + \dot{E}_P\omega + \dfrac{R}{\tau}}{c_F\dot{E}_P} \tag{47}$$

which, considered as a function of $(1 - \epsilon)/\epsilon$, shows a minimum corresponding to

$$\epsilon_k^{\mathrm{OPT}} = \frac{1}{1 + F_k}; \quad r_k^{\mathrm{OPT}} = \frac{(n_k + 1)F_k}{n_k} \tag{48}$$

$$D_{D,k}^{\mathrm{OPT}} = c_{F,k}\dot{E}_{P,k}F_k; \quad F_k = \sqrt[n_k+1]{\frac{(\mathrm{CRF} + \sigma)B_k n_k}{\tau c_{P,k}\dot{E}_{P,k}^{1-m_k}}} \tag{49}$$

The variable F_k expresses the *exergonomic similarity,* originally stated by Szargut (1971).

It is then possible to adopt an optimization strategy which could not aim at calculating the global optimum of a predetermined objective function by means of direct mathematical methods, as conventional approaches do, but tries to find a "good" solution for the overall system design. The exergonomic optimization can thus be carried out by means of an iterative procedure in which the variables r_k^{OPT}, $\epsilon_k^{\mathrm{OPT}}$, $\dot{D}_{D,k}$ are used to determine the changes in the component's design. The criterion is aimed at improving the cost effectiveness of the overall design, i.e., at reducing the costs of the final products. Engineering judgments and critical evaluations must be employed when deciding on the changes to adopt from one iteration step to the next.

A substantial help in this direction is offered by the functions

Table 6

Variable	Case 1		Case 2		Case 3		Case 4		Case 5		Case 6		Case 7	
β	30%		25%		25%		20%		20%		20%		20%	
BF	10%		10%		12%		12%		12%		12%		12%	
T_6 (°C)	7		7		5		5		5		5		5	
ΔT_{67} (°C)	5		5		8		6		6		6		5	
T_8 (°C)	60		60		70		75		80		82		85	
ΔT_{89} (°C)	8		8		8		8		8		8		8	
COP	2.5		2.5		2.8		3.0		3.2		3.6		4.0	
Comp.	Δr %	Δε %	Δr %	Δε %	Δr %	Δε %	Δr %	Δε %	Δr %	Δε %	Δr %	Δε %	Δr %	Δε %
CC	13	-42	13	-41	18	-42	23	-43	25	-43	26	-42	29	-43
CL	38	-63	39	-62	46	-54	50	-48	52	-45	56	-38	59	-30
HC	8.4	-14	8.4	-14	15	-37	18	-44	22	-49	23	-51	25	-53
BL	23	-89	23	-89	19	-85	19	-84	19	-83	19	-82	19	-81
O.F. (\$/h)	13.180		12.862		11.822		11.237		10.919		10.333		9.780	

Table 7

Variable	Case 2		ΔBF		ΔT_6		$\Delta(\Delta T_{67})$		ΔT_8		$\Delta(\Delta T_{89})$		ΔCOP	
β	25%		25%		25%		25%		25%		25%		25%	
BF	10%		12%		10%		10%		10%		10%		10%	
T_6 (°C)	7		7		5		7		7		7		7	
ΔT_{67} (°C)	5		5		5		8		5		5		5	
T_8 (°C)	60		60		60		60		70		60		60	
ΔT_{89} (°C)	8		8		8		8		8		25		8	
COP	2.5		2.5		2.5		2.5		2.5		2.5		2.8	
Comp.	Δr %	Δε %	Δr %	Δε %	Δr %	Δε %	Δr %	Δε %	Δr %	Δε %	Δr %	Δε %	Δr %	Δε %
CC	12.7	-41	12.9	-41	14.9	-45	10.9	-37	17	-41	5.73	-41	12.7	-40
CL	39.2	-62	39.8	-62	42.4	-58	38.5	-63	39.2	-62	39.2	-62	43.4	-57
HC	8.39	-14	8.28	-15	8.39	-14	8.39	-14	15.2	-36	1.55	1.57	8.39	-14
BL	22.7	-89	20.3	-89	22.7	-89	22.7	-89	21.8	-86	23.4	-92	22.7	-89
O.F. ($/h)	12.862		12.608		12.669		12.880		12.848		12.903		12.227	

$$\Delta r_k = \frac{r_k - r_k^{\mathrm{OPT}}}{r_k^{\mathrm{OPT}}} 100 \quad \Delta\epsilon_k = \frac{\epsilon_k - \epsilon_k^{\mathrm{OPT}}}{\epsilon_k^{\mathrm{OPT}}} 100 \qquad (50)$$

Optimum design for any *single component* generally corresponds to the minimum value of Δr_k and $\Delta\epsilon_k$, while the overall system design is not obtained when *all components* operate at the lowest possible value of Δr_k and $\Delta\epsilon_k$: the effects of any change in system variables must always be checked on the global cost function (the objective function, Eq. (24)).

If the change of a decision variable is accompanied by a positive effect on the objective function, this variable is candidate for a similar change in the next iteration step; otherwise, the variable remains unchanged.

When a change in a decision variable results in opposite trends in the optimization of two or more components, the contradictory effects can be corrected by adjusting the step size of that variable. Anyway, the impact on the objective function must be considered before deciding on further changes.

The Results

Table 6 summarizes the results for the optimization of the case under study. The column referred to as Case 1 contains the reference design data and related exergonomic variables; the following columns relate to further optimization steps. Going from Cases 1 through 7, the objective function (OF) is monotonically decreasing, thus showing a continuous improvement in the design. The passage from one given configuration to an improved one is suggested by numerical values assumed by the exergonomic variables. For instance, Case 2 is obtained from Case 1 simply by reducing β from 30 percent to 25 percent. The deviations in $\Delta\epsilon$ and Δr are not great, but the OF reduces significantly; thus, the change can be accepted for future steps. To proceed with the optimization, it is necessary to study the sensitivity of the whole system to each decision variable. To this aim the variables of Case 2 were submitted to slight changes one at a time, with the results summarized in Table 7.

Table 8

Variable	Case 3		Δβ		ΔT_6		$\Delta(\Delta T_{67})$		ΔT_8		$\Delta(\Delta T_{89})$		ΔCOP	
β	25%		20%		25%		25%		25%		25%		25%	
BF	12%		12%		12%		12%		12%		12%		12%	
T_6 (°C)	5		5		6		5		5		5		5	
ΔT_{67} (°C)	8		8		8		6		8		8		8	
T_8 (°C)	70		70		70		70		75		70		70	
ΔT_{89} (°C)	8		8		8		8		8		20		8	
COP	2.8		2.8		2.8		2.8		2.8		2.8		3.0	
Comp.	Δr %	Δε %	Δr %	Δε %	Δr %	Δε %	Δr %	Δε %	Δr %	Δε %	Δr %	Δε %	Δr %	Δε %
CC	17.8	-42	18.9	-41	16.6	-40	18.9	-45	20.1	-42	8.35	-42	17.8	-42
CL	46.1	-54	47.1	-53	44.6	-57	46.6	-53	46.1	-54	46.1	-54	48.4	-51
HC	15	-37	15	-37	15	-37	15	-37	18.3	-44	2.76	-25	15	-37
BL	19.5	-85	19.8	-85	19.5	-85	19.5	-85	19.1	-84	20	-87	19.5	-85
O.F. ($/h)	11.822		11.586		11.906		11.813		11.820		11.828		11.468	

Those new values inducing positive trends to the OF will be accepted (namely BF, T_6, ΔT_{67}, T_8, COP), and the others rejected (ΔT_{89}). This new asset gives raise to design configuration, referred to as Case 3 (Table 8), for which the same procedure can be repeated. This time the adopted new values belong to β, ΔT_{67}, T_8, COP, being the other unchanged. With these data, one can build up Case 4 and proceed further.

The optimization process ends when further trials do not produce significant design improvement. For the case under study, that happens with the variable set of Case 7.

The comparison between the OF of the base case (Case 1) and that of the final design (Case 7) shows how large is the improvement achieved and how beneficial the optimization.

Conclusions

The exergonomic method is a valuable tool for optimizing the design of complex systems. In this paper, an attempt was made to apply this approach to a typical air-conditioning unit.

The results show how far is the improved design from the reference design, although based on typical data.

The optimization was achieved through an iterative procedure rather than through the search of a the global optimum of a predetermined function by means of direct mathematical methods. This requires engineering judgments and critical evaluations at every step of the optimization process, but allows the designer to carry out an energy-conscious design.

Further investigations are in progress, aimed at the comparison with other optimization procedures such as Lagrangian methods, search methods, genetic algorithms, and so on. The results will be reported in a future paper.

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