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Energy Saving with Total Energy System for Cold Storage in Italy: Mathematical Modeling and Simulation, Exergetic and Economic Analysis

D. Friso

Department of Land, Environment, Agriculture and Forestry-TESAF
Research Unit: Agricultural and Food Engineering
University of Padova, Viale dell'Università 16, Legnaro 35020, Padova, Italy

Abstract

With the aim of energy saving in cold storage in Italy, an integrated system (Total Energy System – TES) was considered for production of cold air, obtained by assembling compression system, absorption system and cogeneration unit (CHP) fuelled with methane-gas, in two different plant solutions: 1) TES consisting of the CHP, mechanically coupled with the compression refrigerating machine and thermically with the absorption system both designed to cool the air in the store rooms; 2) TES* characterized by the CHP, mechanically coupled with the compression refrigerating machine, designed to cool the air in the store rooms, and thermically with the absorption system designed to cool the air in a pre-refrigeration plant. A mathematical modeling and a consequent computer simulation of the behavior of both integrated system (TES and TES*) and both related conventional systems (CES consisting of a compression machine to cool the air in the store rooms and CES* characterized by two compression machines, the first cooling air in the store rooms, the second cooling air in a pre-refrigeration plant) was carried out. As an overall results, an energy saving of 21÷16% was calculated. Considering the difficulty of comparing the different types of energy involved, an exergetic analysis was also carried-out, confirming a better exergetic efficiency of TES vs. CES. Finally, an economic evaluation was also conducted with a very attractive profitability index for TES and TES* systems fuelled with natural gas. The economic analysis showed that the use of bio-methane, in the TES system, is profitable up to a bio-methane cost of 0.46 €/m^3.

Keywords: Cold storage; Heat and power cogeneration; Total energy system; Energy saving; Exergetic analysis; Bio-methane.

1 Introduction

Of the overall world food production, 4,200 Mt/year, it is estimated that 40% consists of perishable goods and hence requires cold storage [1]. In Italy, where the statistics are approximate and rather old there are notable differences between areas, but as far as fruit and vegetables are concerned, about 2 Mt/year are kept in cold storage [1].

This corresponds, again referring to fruit and vegetables alone, to an annual energy consumption of around 0.5 Mtep [1]. This figure refers to the fuel consumed in thermo-electric power stations, given that almost all refrigerating machines are based on mechanical compression cycles using electric engines.

At the level of costs, it can be stated [1] that expenditure on energy represents around 12% of total costs for large refrigerated stores. This is a percentage that should certainly not be overlooked, as often occurs in planning the warehouses and relative refrigerating machines.

On the other hand, the natural gas network has been developed considerably in Italy, but is used mainly for heating purposes particularly in the winter months. This causes serious problems for the gas company which, with the gas pipelines from abroad being constantly open, is obliged to store the fuel in the summer at considerable cost. There is consequently a need to incentive gas consumption in summer, even by cutting the prices, and refrigeration could play an important role in this.

There is thus an interest in designing plants which would not only provide energy savings but would also use methane-gas directly for cold storage. In this sense we shall examine a Total Energy System (TES) consisting of a cogeneration unit, based on the internal combustion engine adapted for methane-gas power, mechanically coupled with a compression refrigerating machine and thermically with a absorption refrigerating system, using hot water at 95°C.

As an alternative, was also evaluated to feed the cogeneration engine with bio-methane, produced from organic waste [2 and 3] of fruits and vegetables processing and storage.

After choosing a refrigeration store for fruit and vegetable conservation and defining the size of the TES and a conventional compression system (CES) on the basis of project data, calculation was made of average monthly cooling loads and for an entire annual cycle. On the basis of these data, considering the energy flows imposed by interaction among the various components of the TES, an analysis was made of energy consumption, showing average monthly and overall savings from the TES with respect to the conventional system (CES).

Considering the difficulty of comparing the different types of energy involved, the valuation was completed with an exergetic analysis [4] and calculation was also made of the TES profitability index [5] to show whether it is economically viable.

Table 1 – List of symbols

Variable	**Description**	**Units**
C_o	Percentage of engine hourly consumption	%
COP	Coefficient of performance (cooling)	
E_a	Thermal power required by absorption system	kW
$E_{a\%}$	Percentage of maximum thermal power (abs. sys.)	%
E_c	Mechanical power required by compression system	kW
$E_{c\%}$	Percentage of maximum mechanical power (comp. sys.)	%
EX_a	Exergetic/transformation power required by absorption system	kW
EX_c	Exergetic/transformation power required by compr. system	kW
EXQ_a	Exergetic/transformation power delivered by absorption sys.	kW
EXQ_c	Exergetic/transformation power delivered by compression sys.	kW
EX_t	Exergetic/transformation power required by engine	kW
PNV	Present net value	€
Q_a	Cooling power delivered by absorption system	kW
$Q_{a\%}$	Percentage of cooling power delivered by absorption system	%
Q_c	Cooling power delivered by compression system	kW
$Q_{c\%}$	Percentage of cooling power delivered by compression system	%
Q_f	Total cooling load	kW
Q_t	Combustion (primary) thermal power required by engine	kW
R	Thermal energy/mechanical energy ratio delivered by the engine	
R_v	Profitability index	
T_{ap}	Approach temperature of cooling tower	°C
T_{db}	Dry bulb temperature	°C
T_{wb}	Wet bulb temperature	°C
T_c	Temperature of cold water entering the condenser	°C
T_e	Evaporation temperature of frigorific fluid	°C
T_p	Product temperature on entry to the cooling system	°C

2 Energetic and exergetic analysis formulation

In order to achieve a complete energetic and exergetic analysis, a mathematical modeling was carried-out. The equations obtained were the basis for setting up a computer programme simulating the behaviour of both the TES and the conventional system CES in relation to different conditions of operation imposed by variations in weather conditions and cooling loads in the course of the year.

With regard to the TES (fig.1), due to the cogeneration unit that presents the variation of the ratio of thermal power and mechanical power depending on the load, it was necessary to set up a simultaneous calculation simulation, using a suitable algorithm.

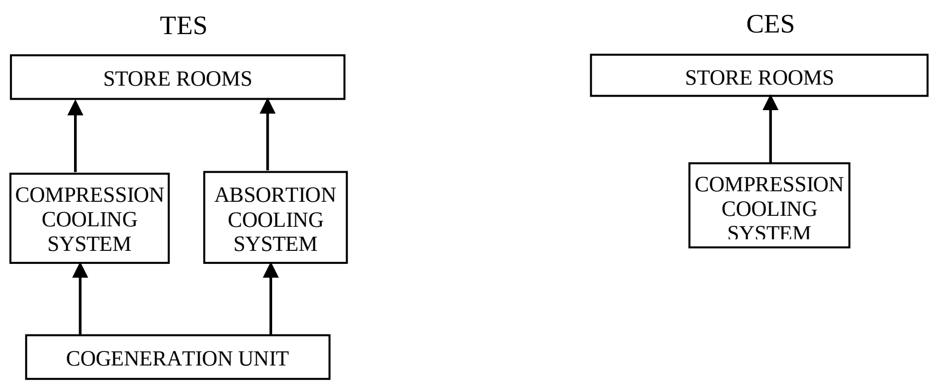

Figure 1- Scheme of Total Energy System TES (left) and Conventional Energy System CES (right).

Table 2 - Volumetric, thermal and functional features of the fruit and vegetable refrigeration plant [6].

<table>
<tr><td>Rooms Number (N.)</td><td>Rooms Vol. (m^3)</td><td>Walls Transmit. ($W/m^2°C$)</td><td>Ambient Temper. (°C)</td><td>Internal Temper. (°C)</td><td></td></tr>
<tr><td>25</td><td>1,250</td><td>0.46</td><td>38</td><td>2</td><td></td></tr>
<tr><td>Air Spare/h (N.)</td><td>Food Specific Heat (kJ/kg°C)</td><td>Package Specific Heat (kJ/kg°C)</td><td>Input Food Temper. (°C)</td><td>Respiration Heat (W/t)</td><td>Spare/day (%)</td></tr>
<tr><td>0.1</td><td>3.8</td><td>2.1</td><td>30</td><td>17.4</td><td>10</td></tr>
</table>

To begin it is necessary to define the size of the refrigerating machines with respect to the project conditions and then determining average monthly cooling loads, after choosing the volumetric, thermal and operative features of a cold store for fruit and vegetables. Tables 2 and 3 show the features of the warehouse and ambient weather conditions on which the planned and average monthly cooling loads were calculated. The planned cooling capacity was 1,250 kW and the mean monthly cooling power actually required from the plant is shown in Table 3.

Table 3 - Average monthly dry and wet bulb temperature (Northern Italy) and average monthly cooling loads [7].

Month	T_{db} (°C)	T_{wb} (°C)	Q_f (kW)
January	1.0	0.0	443
February	2.5	0.5	515
March	7.0	5.0	599
April	12.0	9.0	750
May	17.0	13.5	874
June	20.0	16.9	975
July	23.0	19.4	1,067
August	22.0	18.2	1,035
September	19.0	16.0	935
October	13.5	11.0	723
November	8.0	6.7	274
December	2.0	1.2	450

2.1 Mathematical modeling

As stated above, and with reference to Figure 2, the total energy system TES examined consists of a cogeneration unit, an internal combustion gas-powered engine producing a mechanical power to the flywheel of 300 kW, according to the project dimensions. The machine is not provided with an electric generator, as mechanical coupling directly with the compressor is used, thus eliminating two sources of energy loss. A planned thermal power of 540 kW is made available by means of exchangers on the cooling system and exhaust gas.

In order to set up the simulation program it was necessary to know the engine's efficiency pattern according to the load required, considering that constant speed was hypothesised. What was more important in fact was the relation between the engine's consumption per hour with respect to the maximum and the mechanical load as a fraction of that maximum; this relationship was in any case obtained from the efficiency which emerged as 0.31 under project conditions.

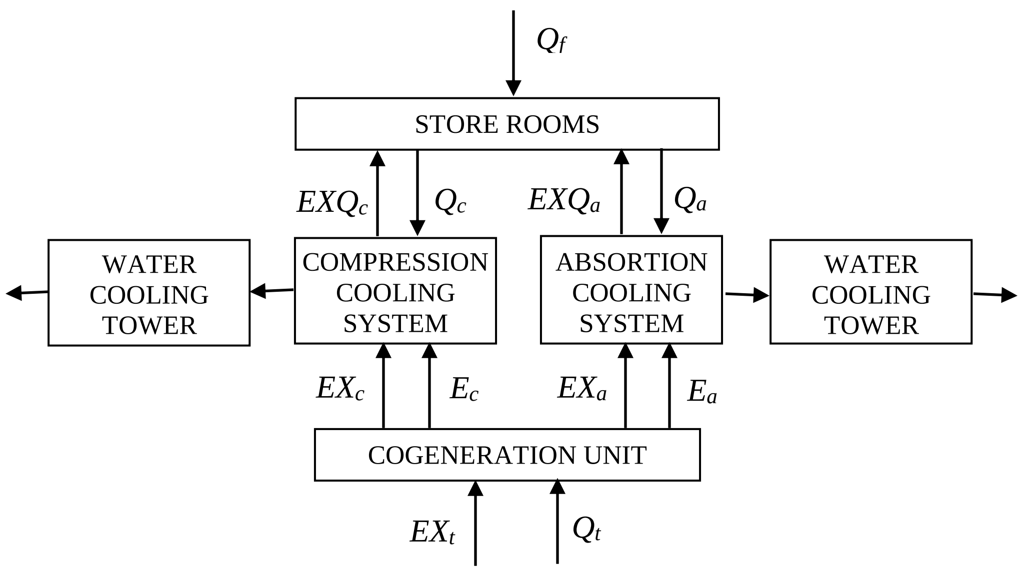

Figure 2- Block diagram of Total Energy System TES. Q_f is total cooling load; Q_c and Q_a are the cooling powers delivered by compression refrigerating machine and respectively by absorption system; EXQ_c and EXQ_a are, respectively, the exergetic/transformation powers delivered by compression machine and absorption system; E_c and E_a are the mechanical and thermal powers required, respectively, by compression refrigerating machine and absorption system; EX_c and EX_a are the exergetic/transformation powers required by compression refrigerating machine and absorption system; Q_t and EX_t are, respectively, the combustion (primary) thermal power and combustion exergetic/transformation power required by engine.

The transfer function appeared as:

$$C_o = 0.825 \cdot E_{c\%} + 17.5 \tag{1}$$

where: C_o is hourly consumption as a percentage of consumption at maximum load; $E_{c\%}$ is the percentage of the maximum mechanical load.

It follows that according to the mechanical load required, the ratio between thermal energy and mechanical energy supplied by the engine varies; under project conditions it was 1.8. The following equation, again obtained starting from engine performance [8], provides the value of this ratio *R*:

$$R = 5 - 0.075 \cdot E_{c\%} + 4.8 \cdot 10^{-4} \cdot E_{c\%}^2 - 5.76 \cdot 10^{-7} \cdot E_{c\%}^3 \quad (2)$$

The compression refrigerating machine was characterised by the fact that it operated at an evaporation temperature T_e of on average -13°C and condensation temperature varying according to ambient weather conditions - dry and wet bulb temperature - which affect the working of the water cooling tower necessary for cooling water for the condenser. Under project conditions the coefficient of performance (COP) - recalling the lack of the electric engine - was 3.3, while its efficiency in relation to the cooling load required and the temperature of cold water entering the condenser allowed us to identify the transfer function, using the minimum square method with experimental data [9]. This correlates the mechanical power required from the compressor $E_{c\%}$, again as a percentage of the maximum, with the percentage of the cooling load compared to the project load $Q_{c\%}$ and with the temperature of cold water entering the condenser T_c (°C):

$$\begin{aligned} E_{c\%} &= 6.956 + 0.475 \cdot Q_{c\%} + 2.457 \cdot 10^{-3} \cdot Q_{c\%}^2 + 3 \cdot 10^{-2} \cdot T_c + 1.047 \cdot 10^{-2} \cdot T_c^2 - \\ &- 1.968 \cdot 10^{-2} \cdot Q_{c\%} \cdot T_c + 2.5 \cdot 10^{-4} \cdot Q_{c\%}^2 \cdot T_c + 5.125 \cdot 10^{-4} \cdot Q_{c\%} \cdot T_c^2 - 5.871 \cdot 10^{-6} \cdot Q_{c\%}^2 \cdot T_c^2 \end{aligned} \quad (3)$$

Considering the planned mechanical power supplied by thermo-mechanical generating unit, equal to 300 kW, and the planned COP, equal to 3.3, a maximum cooling power of 1,000 kW emerges.
The absorption refrigerating system, operating at an evaporation temperature T_e of -13°C and condensation temperature varying with the conditions of the evaporation tower, is characterized by a response of absorbed thermal power ($E_{a\%}$), as a percentage fraction of the planned power, to the required cooling power ($Q_{a\%}$), again as a percentage of the planned power, according to the following equation obtained from data [9] with the minimum square method:

$$\begin{aligned} E_{a\%} &= 26.829 - 5.653 \cdot 10^{-2} \cdot Q_{a\%} + 6.998 \cdot 10^{-3} \cdot Q_{a\%}^2 - 2.172 \cdot T_c + 5.451 \cdot 10^{-2} \cdot T_c^2 + \\ &+ 2.834 \cdot 10^{-2} \cdot Q_{a\%} \cdot T_c - 1.63 \cdot 10^{-4} \cdot Q_{a\%}^2 \cdot T_c - 2.432 \cdot 10^{-4} \cdot Q_{a\%} \cdot T_c^2 + 1.136 \cdot 10^{-6} \cdot Q_{a\%}^2 \cdot T_c^2 \end{aligned} \quad (4)$$

Here the influence of the temperature of cold water entering the condenser, T_c, can

still be seen and is expressed in °C as in function (3).
Under project conditions this machine is characterised by a COP of 0.46 which, with the thermal power supplied by the cogeneration engine, gives the remaining 250 kW of cooling power.
With regard to the water cooling towers for the condensers and absorption system, an equation has been hypothesised giving the temperature of water entering the condensers, T_c:

$$T_c = T_{wb} + T_{ap} \quad (5)$$

where T_{wb} is the wet bulb temperature of ambient air and T_{ap} is the increase due to the tower's actual working. Only in ideal conditions can T_{ap} be hypothesised as zero, while in the simulation it was taken as constant at 5 °C. In reality the temperature T_c could be determined more precisely by using a thermal balance which would equate the heat to be disposed of at the tower, equal to the cooling load plus the work supplied to the compressor or plus the heat supplied to the absorber, with the increase in enthalpy experienced in the air. Taking all this into account and assuming air and water deliveries to be constant, it emerged that expression (5), though referred to project conditions alone, gives a fairly accurate picture of the behaviour of the towers, even with reduced loads corresponding to lower wet and dry bulb temperatures of ambient air.
As far as the conventional refrigerating machine (CES) is concerned, as this is a compression cycle the transfer function (3) may be used, adding the effect of the electric engine's efficiency which is assumed to be constant at 0.86.
Considering that there are few absorption refrigerating systems available that are able to operate at evaporation temperatures lower than 0°C, we decided to proceed with a second analysis, involving an absorption system using lithium bromide/water and which is more frequently marketed. In this case the evaporation temperature is necessarily higher than 0°C, given that the frigorific fluid is water. Here the only possibility is to use the cold produced as a source of pre-refrigeration, with secondary water refrigerated to 5°C coming out of the evaporator and used directly in the pre-refrigeration plant (fig. 3).

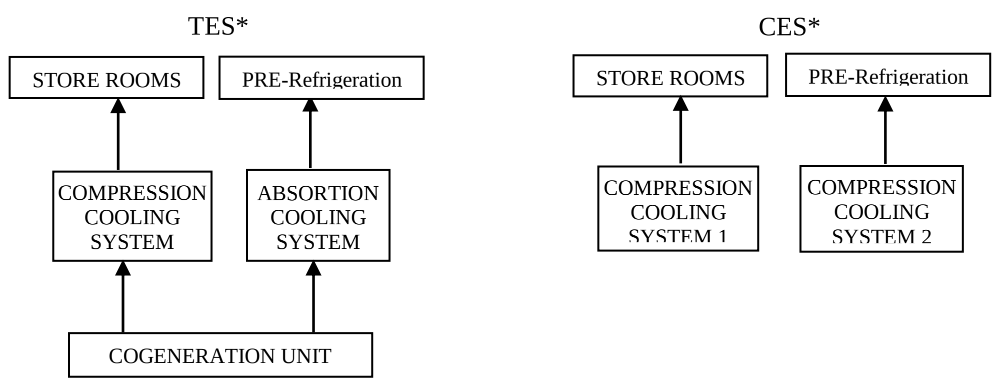

Figure 3- Scheme of TES* (left) and CES* (right).

In this situation the compressor machine in the total energy system (TES*) has to provide the entire refrigeration for the cold store rooms, hence with a planned power of 1250 kW. The planned mechanical power of the cogeneration unit thus rises to 380 kW and thermal power reaches 680 kW. The new absorption system operates with a planned COP of 0.74, corresponding to a cooling power of 500 kW, sufficient to provide pre-refrigeration daily for a quantity of fruit and vegetables equal to 10% of the goods stored in the refrigeration rooms, assuming product temperature T_p of 30°C on entry to the plant and 7 °C on exit. Two further considerations should be made regarding the closure of the pre-refrigeration plant in the winter months and the need, in order to make a fair comparison, to examine a conventional compression pre-refrigeration plant.

Therefore, the total energy system TES* was compared with a conventional system (CES*) consisting of a compression refrigerating machine for loads of cold storage and a second compression refrigerating machine for loads of pre-refrigeration (fig. 3).

2.2 Simulation program

The equations (1), (2), (3), (4) and (5) outlined above are the basis for setting up a computer programme in Visual Basic language simulating the two TES configurations presented along with the two conventional systems (CES) for comparison.

With regard to the first total energy system in which the absorption refrigerating system refrigerates the cold storage rooms, since the cogeneration unit is characterised by a variation in the relationship between thermal power and mechanical power according to the mechanical load, it was necessary to set up a simultaneous calculation simulation, using an algorithm based on a method identifying the zeros of a suitable function. The flow chart in Figure 4 illustrates the structure of this algorithm.

The flow chart is completed with blocks determining the COP of the two refrigerating machines comprising the TES and the conventional system (CES), as well as energy consumption for the TES and the electric power station supplying energy to the conventional machine (CES), assuming an overall efficiency from electricity production and transport of 0.36 [10].

A block has also been added to calculate exergetic flows on entry and exit from the various units, hence exergetic efficiency. In determining these flows we adopted Borel's approach, [11, 5 and 12], with the following hypotheses: ambient temperature variable and equal to average monthly dry bulb temperature (table 3); exergetic value of the fuel equal to its calorific value; exergetic/transformation powers on exit from the refrigerating machines calculated on the final fluid, i.e. on refrigerated air within the cold store rooms, or on the cold water in the case of pre-refrigeration.

In the case of the second total energy system (TES*), as the absorption system's function is basically independent of the compressor's function - i.e. to cool water for pre-refrigeration according to the latter's variable loads - a simpler programming model was used, involving so-called sequential calculation simulation.

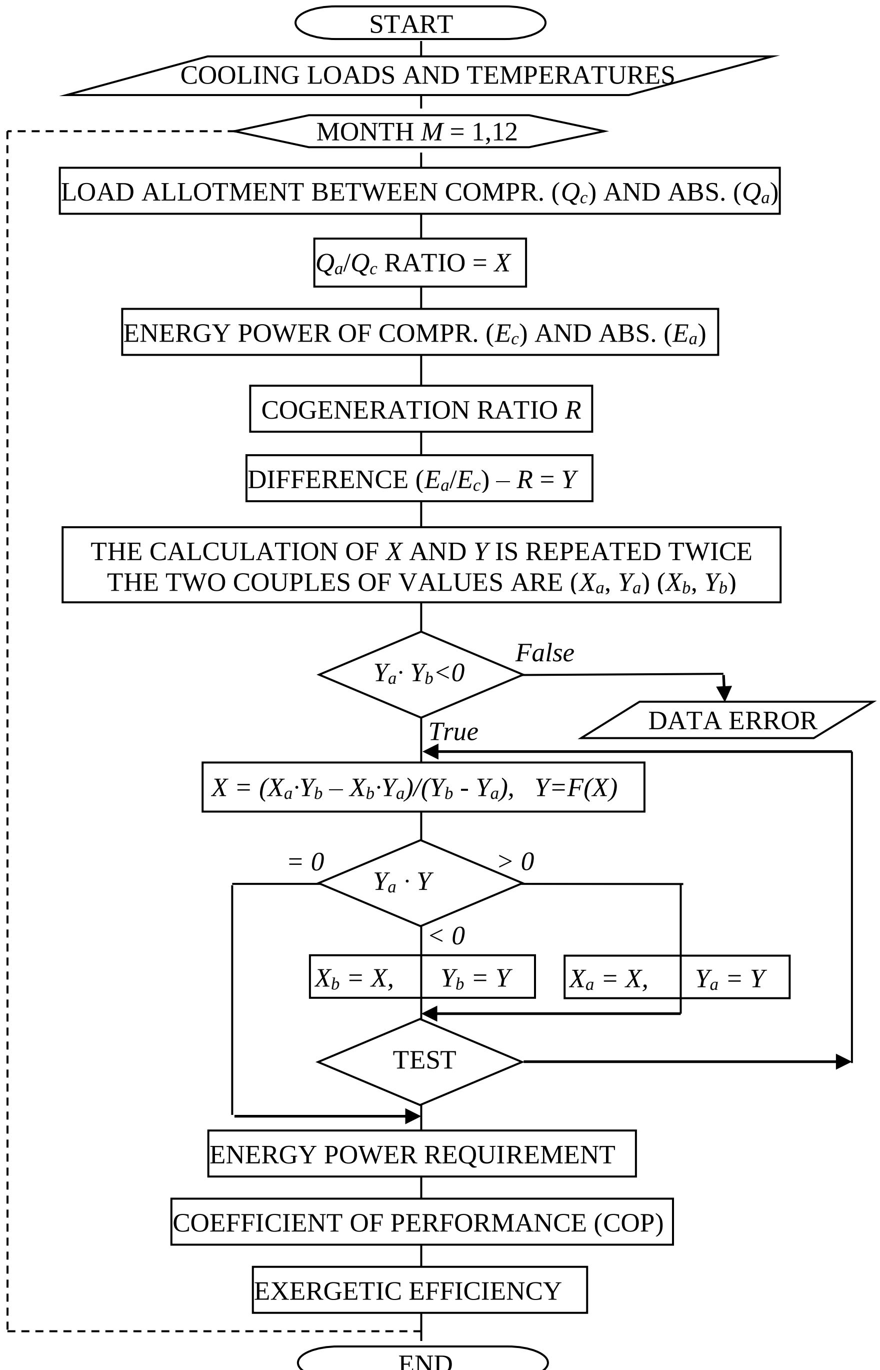

Figure 4 - Flow-chart of simulation program.

2.3 Economic analysis formulation

A cost-benefit analysis was undertaken, adopting the method of profitability index calculation (R_v) [5] for the two versions of the total energy system (TES and TES*), in relation to the cost of the substitute energy source (methane from natural gas or bio-methane).
This profitability index, the ratio between *PNV* - present net value - and the higher investment of the TES plant with respect to the conventional one (CES), was a function of the following parameters, partly derived from the previous energy analysis, the results of which are reported in the following section:

1) annual savings on electricity (cost 0.125 €/kWh [13]), TES: 256,000 €, TES*: 335,000 €;
2) annual methane consumption, TES: 472 km^3; TES*: 660 km^3;
3) higher annual cost of ordinary and extraordinary maintenance (7.5% of higher investment of plant), TES: 18,500 €, TES*: 22,700 €;
4) higher investment of plant, TES: 246,000 €; TES*: 303,000 €;
5) working life of all the refrigerating machines, on a cautionary basis, as 20 years; 10 for the cogeneration unit;
6) interest rate: 5%;
7) methane price: 0.33 €/m^3 [13].

3 Results

As far as the energetic analysis is concerned, the results of computer simulation of the total energy system (TES), with the cold produced by the absorption system being used directly in the store rooms, compared with the conventional system (CES), show (Figure 5) an undulatory pattern of energy consumption - required thermal power Q_t - with a maximum corresponding logically to the highest average monthly temperature of both the ambient and the products stored. With regard to the conventional system (CES), this consumption should be understood as the flow of thermal energy Q_t expended in the electric power station, calculated from the flow of electricity absorbed by the compressor, assuming a constant efficiency rate in electricity production and transport of 0.36 [10].

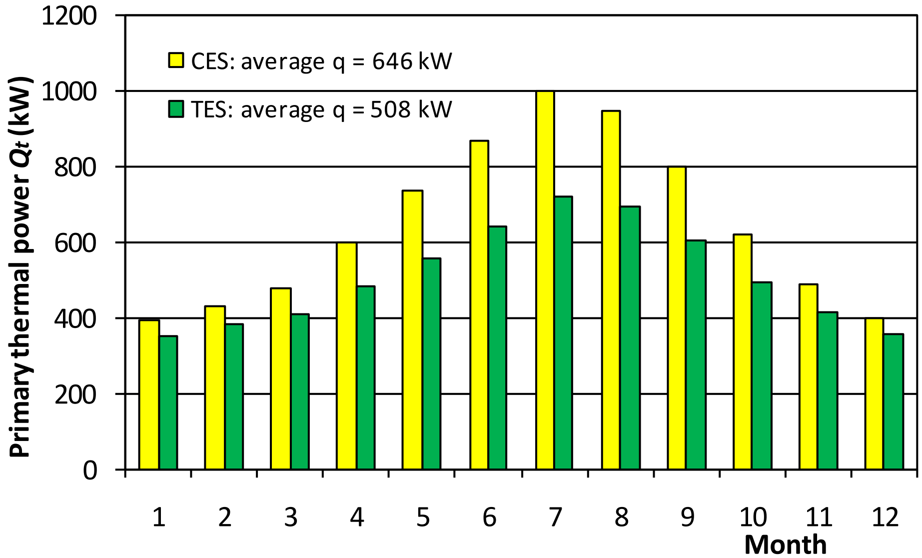

Figure 5 - Pattern of required primary thermal power Q_t, in the course of a complete year, of TES (Total Energy System) and conventional CES (Compression Energy System).

In the winter months the performance of the two systems is similar, as the TES engine, operating at lower loads, is less energy-efficient.

Again with regard to the energetic approach, Figure 6 shows the trends in coefficients of performance for the two refrigerating machines comprising the TES and for the conventional system (CES). Here too an undulatory pattern may be observed, though with double frequency, due on the one hand to the influence of the condensation temperature which tends to lower the COP in the summer months, and on the other hand to the lower values of the cooling loads in winter which also tend to reduce the COP.

The higher values found for the TES compression system, with respect to the conventional one (CES), are due to the lack of the electric engine powering the compressor. The overall average values of energy consumption, 646 and 508 kW for the conventional system (CES) and the TES respectively, show energy savings of 21% for the latter, corresponding to 0.105 MTep/year of energy saving expected for Italy.

With regard to exergetic analysis, Figure 7 shows overall exergetic efficiency for the TES and the conventional system CES respectively, with the latter including the electricity production and transportation processes.

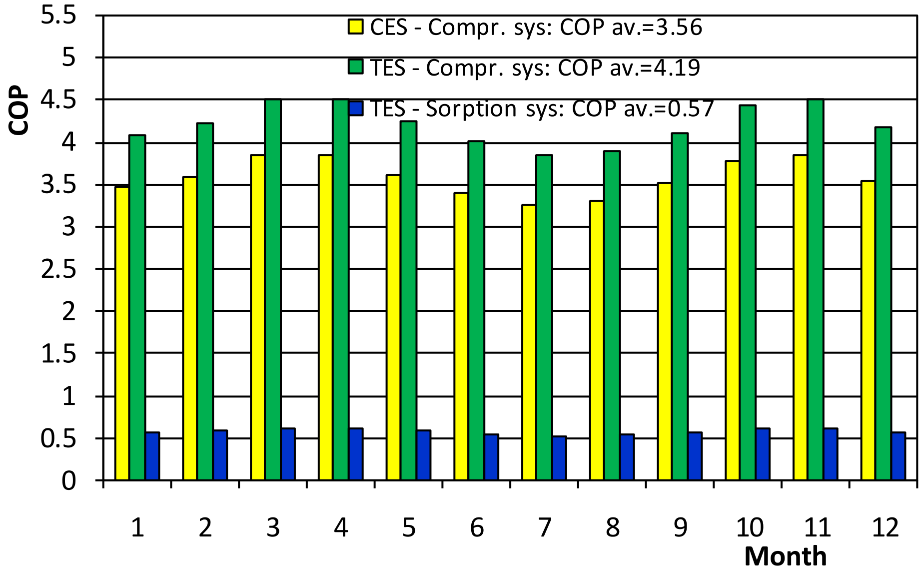

Figure 6 - Pattern of the coefficients of performance (COP) of the conventional compression system (CES), compression system of TES and absorption system of TES, during a complete year.

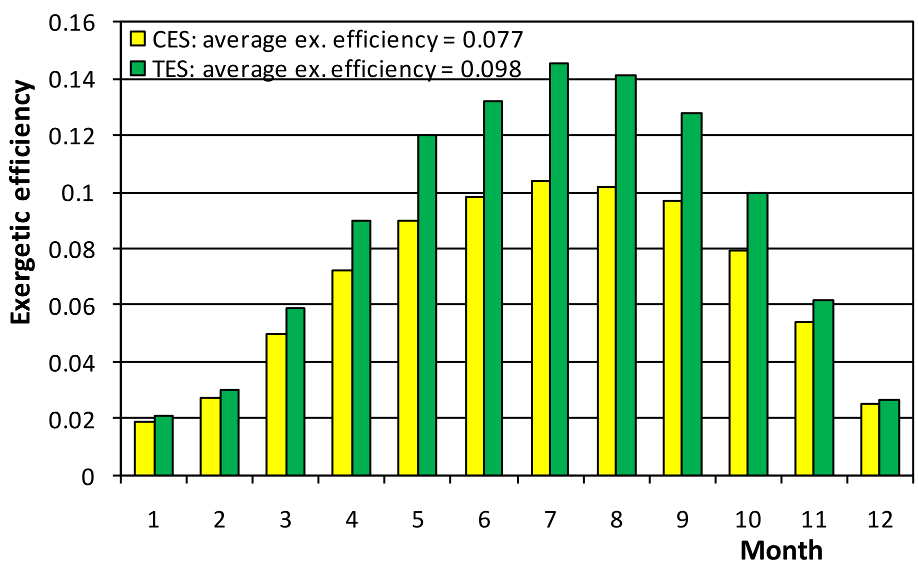

Figure 7 - Pattern of the exergetic efficiencies, during a complete year, of the TES and conventional CES, this including electric energy production and transport.

Average values are around 0.098 and 0.077 for the TES and the conventional system (CES), showing 21% greater efficiency for the former in using primary thermal energy; moreover, this figure is identical to the corresponding savings in energy consumption. This result is better than that obtained from [14] (about 10%). This is both for the different climatic conditions in the UK that for their lower efficiency of the cogeneration unit (CHP) that include electric generator and motor.
It is interesting to note that the exergetic efficiency of both systems is reduced about ten times then passing from the summer to the winter months, an indication that it is not advisable, in terms of the 2nd law of thermodynamics, to expend relatively expensive energy, such as high temperature thermal energy corresponding to combustion in an electric power station or an engine, in order to obtain, as a final effect, an air temperature in the refrigeration rooms which is not very different from the ambient air temperature.
As far as the economic valuation is concerned, it emerged that with the current discounted natural-gas price of 0.33 €/m^3 [13], the profitability index is around 3.15, corresponding to 3.7 years for pay-back.
The proposed total energy system is thus a valid solution from the exergetic point of view and is also economically advisable. The cost-benefit analysis illustrated in Figure 8 shows that the system is profitable up to a methane cost of 0.46 €/m^3. Therefore it is necessary that bio-methane has a cost lower than this limit.

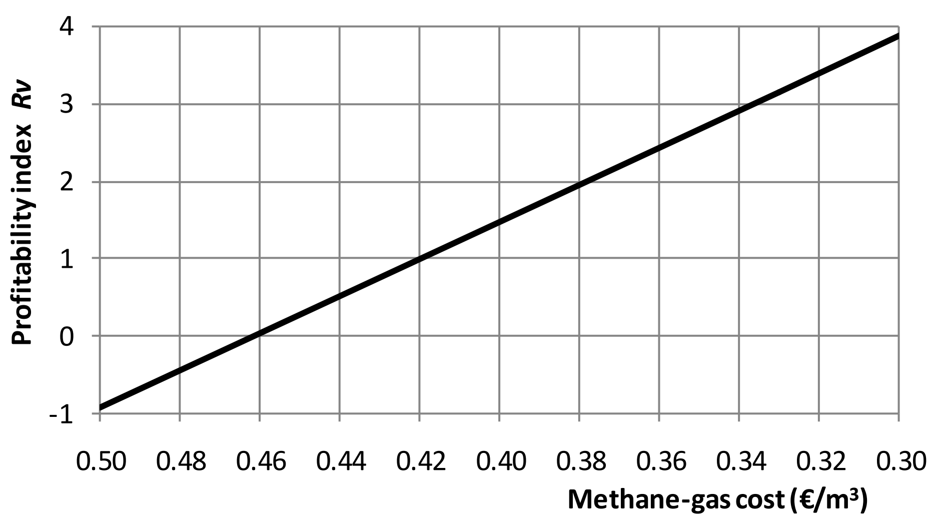

Figure 8 - Profitability index of TES under variable methane-gas price. For a natural gas price of 0.33 €/m^3, the profitability index results of 3.15. For a bio-methane cost of 0.46 €/m^3, the profitability index results zero.

The second system hypothesised, (TES*), with the absorption used for pre-refrigeration, shows a pattern of primary thermal power requirement which is illustrated in Figure 9.

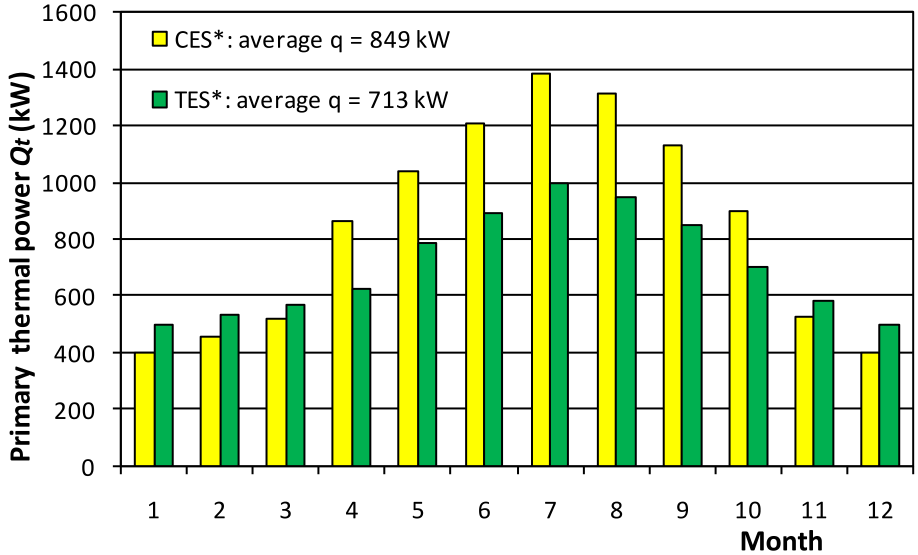

Figure 9 - Pattern of required primary thermal power Q_t, in the course of a complete year, of the second total energy system TES* and comparative conventional CES* (double compression system).

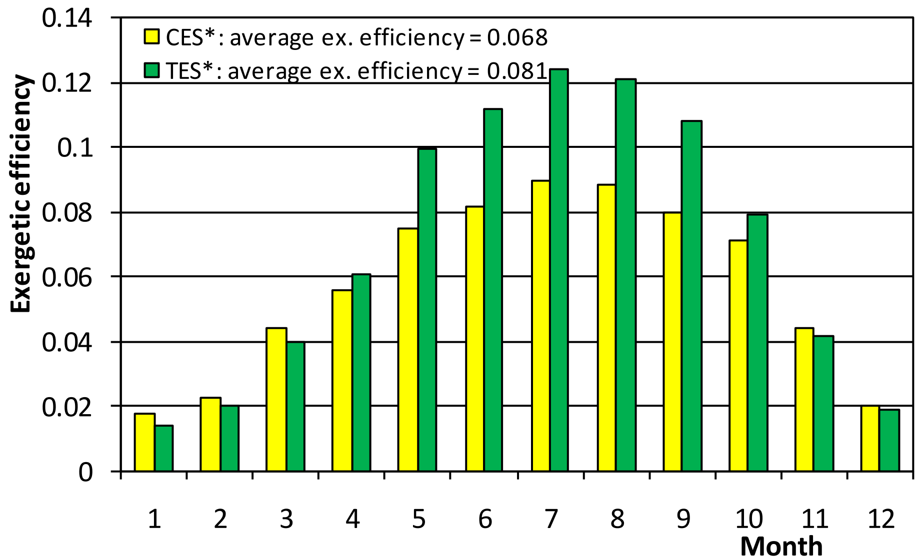

Figure 10 - Pattern of the exergetic efficiencies, during a complete year, of the second total energy system TES* and comparative conventional CES*, this including electric energy production and transport.

Comparison with the corresponding conventional system (double compression system CES*) shows greater consumption in the colder months when the absorption system in TES* is not operative, so that the thermal energy available in the cogeneration unit is dispersed through the relative evaporation tower, while the lower cooling loads required correspond to a significant reduction in the efficiency of the TES* engine. All in all energy saving is lower, with an annual average of around 16%. This latter value is also confirmed by exergetic analysis, the results of which are shown in Figure 10.
Again with regard to TES*, cost-benefit analysis shows a similar situation to TES, with a profitability index of 2.87, again with a discounted natural-gas price of 0.33 €/m^3.

4 Conclusions

Energetic analysis of the total energy system (TES), consisting of a methane-powered engine supplying mechanical power to a compression refrigerating machine and thermal power to an absorption system, both of which are used for cooling the refrigeration rooms, has shown a significant reduction in energy consumption of 21%, with respect to the conventional cooling system (CES). This figure was obtained by means of a comparison between primary energy consumption in the engine and the electric power station respectively.
In order to overcome the limitations of this traditional kind of energy calculation, based on the principle of metrological equivalence among different energy sources, an exergetic analysis was undertaken -based on the operative equivalence of energy sources- which not only confirmed the previous result, but also pointed out the behaviour of each single machine, showing that the greatest exergetic loss takes place in the refrigerating machines operating in winter.
The cost-benefit analysis, showing a profitability index of 3.15, demonstrated that this type of cooling system for fruit and vegetable storage is economically viable.
A similar analysis carried out on the second proposed system, (TES*), with absorption used for pre-refrigeration, showed energy savings of 16% and hence equally greater exergetic efficiency with respect to the corresponding conventional system (CES*). In economic terms, a profitability index of 2.87 emerged, similar to the former one.
To conclude, both systems are of interest in terms of energy consumption, though to different extents, as they would provide energy savings at Italian level up to 100 kTep if they were adopted on a wide scale, a fact which would be greatly encouraged by their unquestionable economic benefits. To ensure that the economic benefits will occur also fueling the engine with bio-methane, it is necessary that the bio-methane has a cost less than 0.46 €/m^3.

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