Cultura e Tecnica per Energia Uomo e Ambiente

EUROPEAN DIRECTIVE 2009/28/EU:
AiCARR's Position about
Italian Decree 28/11
FOR ASPECTS RELATED TO
HEAT RENEWABLE SOURCES

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AiCARR'S POSITION PAPERS

AiCARR is a cultural nonprofit association dedicated to the creation and promotion of culture and technology for sustainable comfort.
The Association was founded in 1960 and has always been involved in issues related to the rational use of energy and natural resources. AiCARR is focused on innovation for energy infrastructures, on the field of energy production systems and on the building sector.
AiCARR's main aims are to develop, support and disseminate the concept of sustainable comfort, to promote professional training and development for operators in the sector and to enhance their professional qualifications. AiCARR contributes to setting the standards for the sector, to discussions on how to improve legislations, speaks for the profession and is consulted as authoritative partner by other associations and governmental bodies, both in Italy and in other Countries. AiCARR's core topics are the design of building-plant system, the development and dissemination of technical standards, the innovation of plant and building technologies to achieve energy savings, maintenance of the plants, energy requalification of existing buildings, the use of renewable energy sources.
AiCARR has more than 2600 members (designers, machinery producers, installers, maintenance operators, university professors, researchers, students, officials of bodies and governmental agencies, of scientific and operational national and international institutions). AiCARR's members are high level protagonists in the world of air conditioning and energy saving. Against this background, AiCARR's positions about the energy sector represent the summary of opinions and evaluations by independent experts.
AiCARR's position papers are the fruit of the experts' joint efforts and represent the official position of the Association on topics of particular interest for the energy sector.

Michele Vio
AiCARR President

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1 - Foreword

AiCARR is a cultural nonprofit association dedicated to the creation and promotion of culture and technology for sustainable comfort.
The Association was founded in 1960 and has always been involved in issues related to the rational use of energy and natural resources. AiCARR is focused on innovation for energy infrastructures, on the field of energy production systems and on the building sector.
Most AiCARR's members are high level eminent protagonists in the field of air-conditioning and energy saving: from university professors to designers, from installers to producers of machines and components for the plants.
AiCARR's cultural vocation is shown by the fact that members have other associations to represent their work claims: designers have, by way of example, a professional body, or producers are members of industrial employers' confederations, such as ANIMA.
AiCARR' positions in the energy sector are, therefore, the highlights of points of view of impartial and independent experts.
AiCARR feels bound to intervene and express its opinion about Italian Decree 28/11, transposing the European Directive 2009/28/EU, since it considers that the Italian Decree is fundamental for the future evolution of the concept, design and construction of air-conditioning plants, and AiCARR is a highly authoritative stakeholder in Italy on this subject.
Some considerations contained in this document may appear to be trivial; as a matter of fact, they are the fruit of an intense exchange between the competences on many different disciplines that coexist at AiCARR and the spirit of the present document is to be constructive. Moreover, it should not be forgotten that most of AiCARR's remarks refer directly to the European Directive 2009/28/EU.

2 - Purpose and structure of the document

AiCARR's objective is that real energy savings and actual support to the technologies that can help meet this target, with no exceptions, are achieved in the sector when Italian Decree 28/11 comes into force through its implementing acts.
This is the reason why AiCARR has felt it was necessary to illustrate and critically review problems and difficulties that a designer may run into in some cases, if he adopts a misleading interpretation of the Italian Decree, and assist him with suggestions on how to operate, obviously in compliance with the provisions of European Directive 2009/28/EU on the use of energy from renewable sources, transposed into Italian law by Italian Decree 28/11.
The document is divided up into 3 parts:
- Summary of AiCARR's position;
- Part 1: Critical aspects of the Italian Decree and proposals by AiCARR;
- Part 2: Analysis of the problems linked to the use of seasonal averages and indications on methods to be used;

and also has some annexes.
All suggestions are supported by numerical examples to draw a clear distinction between the data obtained with the method suggested by the Italian Decree and the data obtained with AiCARR's method. Anyhow, to gain a clear picture of the content of the document, one needs to read it as a whole, paying particular attention to numerical examples.

OVERVIEW ON AiCARR'S POSITION

Italian Decree 28/11 transposing European Directive 28/09, usually indicated as Renewable Energy Sources (RES) Directive, has two different aims: defining "*the instruments, the mechanisms, the incentives and the institutional, financial and legal framework that are needed to reach by 2020 the targets on the overall share of renewable energy in gross final energy consumption"* and dictating *"standards related to the transmission of statistical data between member states"*.
In line with the technical-scientific approach on which the Italian Decree is based and considering the importance of a right mix in energy sources, in the wake of events in Europe after the tsunami hit Japan, and considering the particular needs of the countries in the Mediterranean area, AiCARR believes it has to take a stance on the first of the two aims. As a matter of fact, AiCARR doesn't consider it its duty to review and comment on statistical accounting methods between states, but is keen on underlining that these methods cannot and should not interfere with the correct design of energy production systems needed for heating, cooling and hot water production to reach the objectives laid down in the Italian Decree, which are, by the way, much more ambitious than the ones contained in the 20-20-20 agreement.
By the same token, AiCARR, in its quality of cultural association, doesn't want to be involved in the way incentives for the systems that produce energy from renewable sources should be earmarked, which is a task performed by other trade associations. AiCARR would like to make this concept very clear and avoid any possible misuse of the contents of the present document, which is aimed at making its position clear and putting forward some proposals to overcome the critical aspects previously mentioned.
Some of the proposals by AiCARR may appear difficult to be implemented at European level, even though they are not in contrast with the spirit of European Directive 28/09. This document is not intended to change agreements that have been reached at European level. Rather, each proposal is aimed to help reaching the 50% objective, established for 2017 by the Italian Decree that is much higher than the lower limit imposed by the European Directive, thanks to systems that use the lowest possible amount of primary energy. This is particularly important in the case of plants that are installed in non-residential buildings, where the need for cooling is equal or higher than the need for heating and hot water. Unless action is consistent, there is a risk that the number of cases where "*The technical impossibility to comply, **completely or partially,** with back-up obligations…"*, laid down in paragraph 6 of annex 3 of the Italian Decree, may rise out of all proportion and hinder reaching the objectives that were set out by the Decree itself.
This document also is intended to promote the review of Parts 1, 2 and 3 of technical specifications UNI/TS 11300, that is under way at the moment of this writing, and the drawing up of the definitive version of Part 4, into which a public survey has just been completed. Indeed, AiCARR believes that UNI/TS specifications have necessarily to be harmonized with the objectives of the Italian Decree.

Critical Aspects Contained in Italian Decree 28/11

The energy share from renewable sources, that is expected to reach 50% when the system will be implemented at full rate in 2017, is much higher than the share established by European Directive 28/09. Moreover, this target is difficult, if not impossible to be reached for many plants which have summer needs that exceed winter needs.
AiCARR would like to stress that the Italian Decree contains some critical aspects that, if not overcome, would entail a risk: many designers would appeal to the technical impediment clause and the objectives of the Decree would be not be met. In the following, an overview is given of all the critical issues of the Italian Decree and of AiCARR's suggestions to reach full compliance with the Italian Decree.

Point 1: Calculation of the amount of energy from renewable sources used by heat pump system.

The calculation method proposed by the Italian Decree has the advantage of being very simple and straightforward. Yet, the calculation method appears to be conceived for the residential sector, in particular for single apartments, which entails four critical aspects in the case of complex medium and large size plants, because:

1) **it leads to inaccurate results**, not necessarily consistent with the objectives contained in the 20-20-20 agreement on the 20% increase in energy efficiency and primary energy saving, because it could promote production systems that consume a higher amount of primary energy;
2) **it promotes heat pumps with poor efficiency rather than the more efficient ones**;
3) **it doesn't make distinctions between energy production systems**, because it doesn't allow for type of back-up system that is chosen (electrical resistance, condensing boiler, or any other);
4) **it creates a disadvantage for the non-electric heat pumps**, such as absorption heat pumps or gas engine heat pumps

Moreover, the calculation method is applied according to a seasonal average, which leads to a further critical aspect because:

5) **the renewable energy share calculated by this method is inconsistent**. As a matter of fact, the calculation takes simultaneously into account very different conditions: when renewable energy sources are not used, because the instantaneous efficiency of the heat pump is too low, and when renewable energy sources are used.

***Suggestions* by AiCARR**

To overcome these first four critical aspects, AiCARR suggests a consistent calculation method, in line with the approach of standards and legislations on energy efficiency that are in force, with the aim of comparing the different generation systems that use heat pumps, and select the ones that are more virtuous in terms of energy efficiency.
To overcome the fifth critical aspect, AiCARR suggests to base the calculation on seasonal averages only in simple cases, such as individual heating systems of apartments in the residential sector. In all other cases, the calculation has to be performed based on values referred to periods of one hour of operation, and obtained with a bin method or with more refined models.

Point 2: Share of renewable energy sources: peculiarities of the Mediterranean climate

The Italian Decree establishes that, in using heat pumps, energy from renewable sources can only be used to produce hot water and winter heating, while the renewable share is calculated over the energy consumption of the whole year, which also accounts for summer cooling. Thus, to the purpose of the calculation, the energy needed for summer cooling systems can be compensated, partially or completely, by the surplus of energy obtained from renewable sources obtained in the winter. In absolute terms, the greater the surplus, the greater the winter energy consumption, *ceteris paribus*. Therefore, the following critical aspect may occur:

1) **any intervention aimed at reducing heating demand in winter may be discouraged**, because it reduces the surplus of energy from renewable sources available for summer cooling systems in absolute terms. There are two different kinds of intervention, both more effective in the winter than in the summer:
   - better insulation by building envelopes;
   - installation of heat recovery systems.

Moreover, heat pumps can be optimized in different ways, for winter or summer operation. The calculation method established by the Italian Decree favors heat pumps that are optimized for winter operations, without taking into account whether energy consumption is greater in the winter (residential sector and hotel industry, in northern Italy) or in the summer (tertiary sector, commercial buildings, etc.). This can result in another critical aspect, because:

2) **the Italian Decree could promote systems that consume a greater amount of energy when the summer consumption is greater than the winter one.**

   AiCARR's suggestions

To overcome the first critical aspect, AiCARR believes it appropriate to promote all systems that can lead to energy consumption reductions, especially in summer operation. Thus, AiCARR suggests to:

a) consider heat recovered from exhaust air as a renewable energy source, both for summer and winter operations;
b) consider direct and indirect Free-Cooling as a renewable energy source, should it come from aerothermal, hydrothermal or geothermal source;
c) consider production of cooling energy as a waste of the production of thermal energy in the operation of heat pumps and not vice versa, in all cases when concurrent opposite loads are supplied with energy by total recovery systems;
d) promote the heat pumps that have high efficiency ratings in summer operations, whenever the energy requirements of the system are greater in the summer season;
e) promote energy systems that use geothermal energy whenever, operating on a yearly basis, they could make use of the rejected heat from the condenser to regenerate the ground in summer.

Point 3: District Heating (DH) and Cogeneration/Combined Heat and Power (CHP)

Italian Decree 28/11 actually considers energy from DH systems as a renewable source, without specifying if thermal energy comes from systems operating in CHP mode (thermal energy production as a by-product of electricity production).
If this were not the case, if boiler based DH systems were considered a renewable source of energy, there would be a critical aspect, because:

1) **systems that have nothing to do with energy from renewable sources would be promoted.**

   AiCARR's suggestions

To overcome this critical aspect, AiCARR suggests to:

a) consider DH systems by the same standards of the others, executing the calculations mentioned in point 1 (calculation of the amount of energy from renewable sources produced with heat pump systems) ;
b) promote cogeneration and trigeneration systems, at least for winter operation, especially if supplemented by heat pumps.

Point 4: Energy used for fluid pumping operations

Italian Decree 28/11 does not specify whether and when the energy usage for fluid pumping operations has to be

taken into account. In particular, for thermal solar power, the Italian Decree considers that all the energy produced has to be considered as renewable energy, despite the electricity needed for pumping (equivalent to zero only for natural circulation systems). This may result in the following critical aspect:

1) **consider systems that consume different amounts of electrical power for pumping fluids as equivalent systems**.

AiCARR's Suggestions

To overcome the critical aspect, AiCARR suggests to:

a) promote solar thermal systems, that have a low electricity consumption for pumping;
b) promote systems that have low electricity consumption, for both pumps and fans.

Point 5: Biofuels

Italian Decree 28/11 considers all the thermal energy produced as renewable energy, despite the energy cost for its production. This can lead to the following critical aspect:

1) **promote biofuel excessively and favor speculation of their production.**

AiCARR's Suggestions

To overcome this critical aspect, AiCARR suggests to:

a) in the energy balance, to account for the amount of energy consumption needed for the production and transport of biofuel.
b) in the energy balance, allow for the possibility to include the share of biomethane input into the grid.

PART 1

PART 1

CRITICAL ASPECTS IN THE ITALIAN DECREE

AND PROPOSALS BY AiCARR

The Italian Decree has two main aims: define "*the instruments, the mechanisms, the incentives and the institutional, financial and legal framework that are needed to reach the objectives by 2020 on the overall share of renewable energy in gross final Energy consumption"* and dictate *"standards related to statistical transfers between member states"*. Indeed, AiCARR doesn't deem it to be its duty to review and comment on methods of statistical accounting among the states, but it is keen on underlining that these methods cannot and should not interfere with the correct design of energy production systems energy needed for heating, cooling and hot water production to reach the objectives laid down in the Italian Decree, which are, by the way, much higher than the ones contained in the 20-20-20 agreement.

1.1 - Energy produced from renewable source and saving of primary energy.

Italian Decree 28/11 is the transposition of the European Directive 28/09, which, in turn, is included in a wider context, that does not only consist of air conditioning systems of inhabited environment, but also of the whole process of production and use of energy. Final energy is usually the concept that is taken as benchmark, meaning the energy consumed by the end user. The production of thermal energy from renewable sources, obtained both with heat pumps and with thermal solar power, is calculated based on this approach that has the undoubted advantage of being simple and user friendly to anyone who is not familiar with energy consumption issues.
In other fields, the approach is different: in particular, when it comes to energy efficiency in buildings, the stakeholder is a technician, who is mainly focused on primary energy savings, and aim at addressing the building envelope, the HVAC system and, in general, the building as a whole, that has to consume less energy "at the source" (i.e. primary energy) and not at "the point of delivery".
This very different approach may result in the critical aspects of the Italian Decree, which is not an independent act within the legislative context in Italy, but rather it has to be in line with what is established elsewhere, in particular in the whole range of laws and standards related to energy efficiency in the building sector. Unfortunately, though, in the present situation, the legislative acts may reach a short-circuit, because one law may lay down the use of a certain production system that is not included in the Italian Decree, as shown in the following part of this document.
This risk is linked to the different approach by the Italian Decree, based on final energy, and to the legislation on energy efficiency, based on primary energy savings. Just to mentioned some examples, the Italian Decree has to comply both with technical specifications UNI/TS 11300, mentioned in Italian Decree 59/09, and also in the specific case of cogeneration, with Italian Decree of 5th September 2011 on incentives to high efficiency cogeneration.
This risk can be overcome only combining energy production from renewable sources and primary energy savings. This is what AiCARR puts forward, which, in substance, transforms the primary energy saving into final energy, to adapt what laid down in the legislation on energy efficiency to the provisions of the Italian Decree, so that the overall approach becomes consistent. In so doing, one undoubted advantage is obtained: the more virtuous energy production systems in terms of primary energy would coincide with the ones that also produce the greater amount of energy from renewable sources, unlike what may happen if the Italian Decree is implemented as it is now.

1.2 - Energy production from renewable sources with heat pumps

Energy production from renewable sources with heat pumps is one of the trickiest points in Italian Decree 28/11, since it plays an important role for many choices, not only technical, but also political choices.
Through this document, AiCARR intends to underline that a lot of common sense is needed to give the right interpretation to the content of the European Directive and reach the ultimate goal, which is to save energy.
For the calculation of energy from renewable sources obtained with a heat pump, Italian Decree 28/11 lays down the use of the equation reported in the European Directive:

$$E_{RES} = E_{HP}\left(1 - \frac{1}{SPF_{HP}}\right) \tag{1.1}$$

where E_{HP} is the seasonal thermal energy delivered by a heat pump and SPF_{HP} is the Seasonal Performance Factor for that heat pump, defined as:

$$SPF_{HP} = \frac{E_{HP}}{E_{abs}} \tag{1.2}$$

where E_{abs} is the energy input required to drive the heat pump considered as a whole, including auxiliary devices; obviously, in the case of an electric heat pump, it means electrical energy.

The Italian Decree also prescribes a minimum level below which no more energy from renewable sources is produced. The calculation is based on the following ratio:

$$SPF_{Min} > \frac{1{,}15}{\eta} \tag{1.3}$$

where, for heat pumps operating on electricity, η is the efficiency for the transformation of primary energy into electrical energy, calculated on the basis of the lower calorific value, that takes into account also distribution to the electricity grid, and 1,15 is an upper bound established by the Directive. Conventionally, all over Europe, electricity production coefficient is considered equal to 0,4, which implies that SPF_{Min}= 2,875; obviously, if the efficiency value changed, also the threshold value would change. For non electric heat pumps η is set to 1.

The calculation from Equation (1.1) must be considered as conventional, because it is based on a simplified statistical method, the details of which will be explained in more depth later on in this document, and which is anyway different from the approach used in standards and directives that deal with energy efficiency. As a matter of fact:

1) the Equation only takes into account the heat pump, disregarding the generation system as a whole, recognizing as equal low efficiency back-up generators, such as the ones with electrical resistance, and high efficiency ones, such as condensing boilers;
2) the Equation offers a seasonal average calculation, instead of considering on a case by case basis when and if energy is actually produced using a renewable source. The seasonal average calculation entails serious mistakes, since, using a simplified method, the quantity of energy produced from renewable energy sources is sometimes undervalued or even considered to be inexistent, while in other cases, it is overestimated, as shown in Numerical Example 2. The issue is very relevant and is dealt with separately in 2.1.

The conventional calculation approach based on (1.1) and (1.3) is almost invariably not compliant with energy efficiency standards and may lead to foster systems that consume higher quantities of primary energy. In particular, this approach results in a handicap for the non-electric heat pumps. A description of the different kinds of heat pumps is reported in Annex A.

The conventional approach moreover creates a disadvantage for all condensing boilers, that, despite not being able to use energy from renewable sources, can nevertheless contribute to reduce energy consumption whenever it is difficult to adopt heat pump systems because of adverse conditions. Generally speaking, it should not be neglected that, especially for heating units, the use of fossil fuels, such as methane, enables to also use the quantity of latent heat that is otherwise lost when the fuel itself is used in power plants for the production of electricity. Therefore, as will be better described below in the document, the comparison between different possible solutions has to be made on the basis of the same primary energy.

Argument on the equation adopted by Italian Decree 28/11

As mentioned, Equation (1.1) is the one reported in the European Directive on RES, that assumes a typically statistical approach. Undoubtedly, the Equation allows to perform a quick calculation of the amount of energy produced by each member State adopting as unique reference for the calculation *SPF* values of heat pump driven units, but without putting them on an equal footing in terms of primary energy. AiCARR is not interested in expounding on the genesis of the Equation, that was more the consequence of a political compromise than the result of thermodynamic assessments. But what AiCARR is very keen on highlighting that it is not possible to implement Equation (1.1) the way it is laid down in the Italian Decree, one of the reasons why being the very high energy share estimated for 2017 (50% energy from renewable sources).

Equation (1.1) enables to meet the objectives of the Italian Decree with regard to summer air-conditioning only in a few cases and when the heating requirements are much greater than the cooling ones: the Equation enables to produce a share exceeding 50% of the overall energy from renewable sources in the winter and also to cover a part of the summer production, or even all of it. Numerical Example 1 helps to better understand this issue.

NUMERICAL EXAMPLE 1

Let it be given an energy requirement for heating and domestic hot water of 100.000 kWh: with a heat pump that has SPF_{HP} = 4, the share of energy produced from renewable sources, according to Equation (1.1), is 75%, which gives a total of 75.000 kWh. Thus:

$$E_{RES} = E_{HP}\left(1 - \frac{1}{SPF_{HP}}\right) = 100.000\left(1 - \frac{1}{4}\right) = 75.000 \quad \text{kWh}$$

Therefore, since the amount of energy needed to meet 50% of the demand for heating and domestic hot water, as established by the Italian Decree, is equal to 50.000 kWh, 25.000 kWh of energy from renewable sources are left over, and they can be used to cover 50.000 kWh of the cooling requirements.

From this point of view, Equation (1.1) is appreciable and very useful, with respect to the residential sector: having simple methods, immediately understandable, also allows to simplify the calculation of possible economic incentives for the so called thermal renewable sources, just as it has been proposed by other Italian Associations. AiCARR shares the spirit of this approach, but would like to highlight some critical aspects that could occur if the conventional value of Equation (1.1) is misunderstood, particularly so when the demand for cooling is similar or higher than that for heating. Paragraph 1.2.2 contains the criteria put forward by AiCARR to avoid such critical aspects, without any prejudice to the Equation that remains unchanged.

1.2.1 - Calculation of the share of energy from renewable sources delivered by a heat pump: critical aspects and proposals to address them

European Directive 28/09 on RES is based on final energy, while other Directives and the Italian legislation on energy efficiency are based on primary energy. It is not AiCARR's intention to expound on the two different approaches, yet it would like to stress that the calculation of the renewable energy share delivered by a heat pump, when exclusively based on SPF_{HP}, may lead to serious mistakes in evaluation. To avoid these mistakes and make the text of the Italian Decree consistent with the provisions in the Directive and with Italian laws on energy efficiency, AiCARR has worked out a method whereby the Seasonal Primary Energy Ratio, PER_S, is also included in the calculation. PER_S is the ratio of the heat energy delivered by the heat pump to the primary energy consumed to produce this heat. The method, that does not interfere with the present approach, but at the same time allows for the evaluation of the incidence of the Primary Energy Ratio, is described in paragraph 1.2.2.
The relationship between PER_S and SPF_{HP}, is:

$$PER_S = \eta SPF_{HP} \tag{1.4}$$

for electrically driven compression heat pumps, while non-electric heat pumps, the relationship is as follows:

$$PER_S = SPF_{HP} \tag{1.5}$$

In compliance with European Directive 28/09, and therefore the upper bound coefficient introduced in Equation (1.3) that takes into account several factors, such as the fact that the efficiency η for the production and distribution of electricity is calculated based on the lower calorific value, the minimum level threshold for PER_S, below which no more renewable energy is produced, is equal to 1,15.
When the value of the electricity conversion factor is equal to 0,4 and, as shown in Figure 1.1, SPF_{HP} = 2,875 - minimum level imposed by Italian Decree 28/11 - PER_S is equal to 1,15, that corresponds to the upper bound coefficient that is present in Equation (1.3).
In Annex B, some concepts related to the performance of heat pumps are explained in detail.

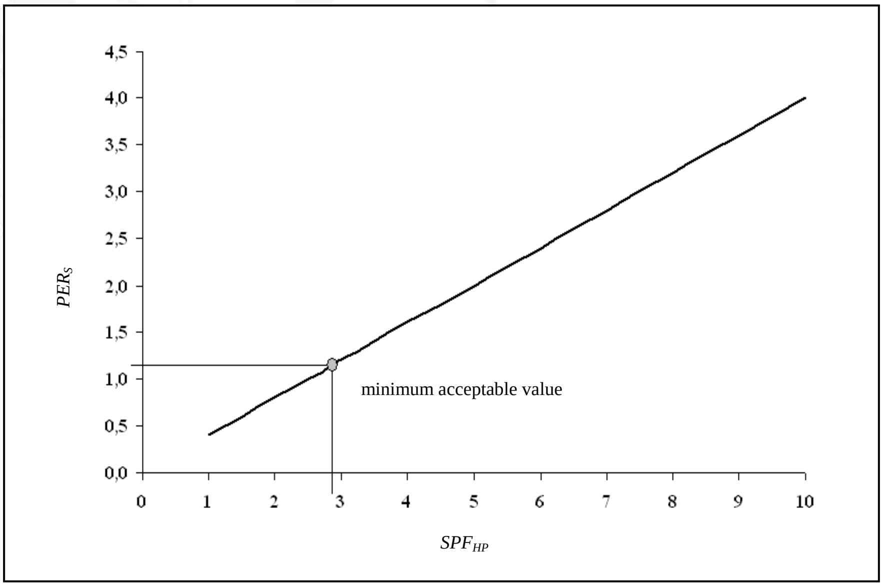


Fig. 1.1 - PER_S values in function of SPF_{HP} with η = 0,4.

1.2.1.1 - Flattening in the renewable energy share for high *SPF* values

The first consequence of the use of Equation (1.1) is shown in Figure 1.2 and is about the renewable energy share for high SPF_{HP} values.

Operating heat pumps at different values of SPF_{HP} (3, 4 or 5) entails important differences, both from the technological and from the economic points of view. For example, against the background of the climate of a city in central Italy, value 3 is reachable with a low quality air heat pumps, value 4 with a high quality air heat pump, or with a good geothermal heat pump, value 5 with an excellent geothermal heat pump characterized coupled to a large borehole heat exchangers field. In economic terms, the second and third solutions are much more expensive than the first.

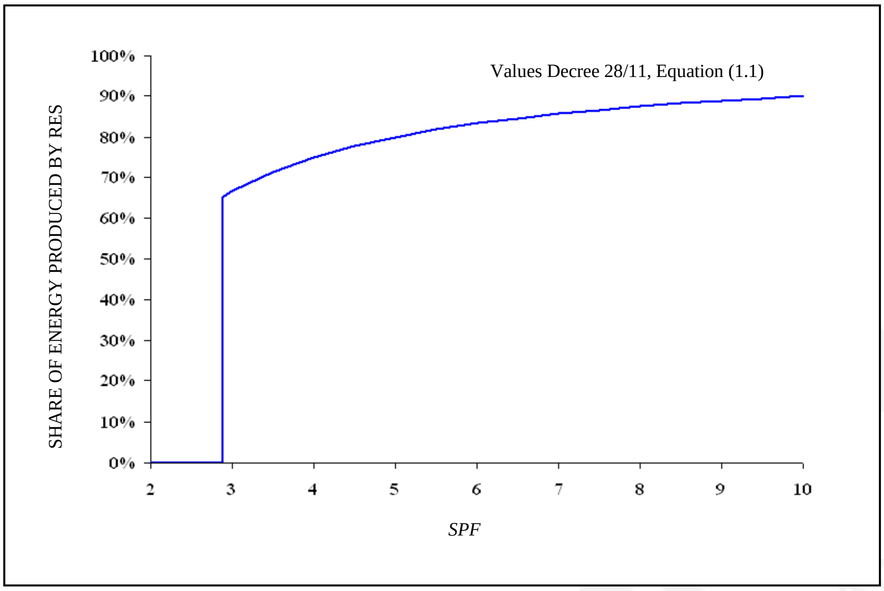


Fig. 1.2 - Values of the renewable energy share calculated with Equation (1.1) in compliance with Italian Decree 28/11.

Before going over to the description of AiCARR's method, critical aspects linked to the use of Equation (1.1) have to be reviewed.
Table 1.1 reports the variations in the share of energy from renewable sources in relation to different values for SPF_{HP}. A thorough review of the values reported in the Table clearly shows that by using Equation (1.1), as stated in the Italian Decree, the difference between value SPF_{HP} = 3 and value SPF_{HP} = 4 is translated into an increase in the renewable energy share that amounts to only 13%, while this percentage goes up to 20% between value SPF_{HP} = 3 and SPF_{HP} = 5. According to thermodynamics, in terms of the savings achieved in primary energy, though, a heat pump with SPF_{HP} = 5 would save much more energy than a heat pump with SPF_{HP} = 3, as it is reported in the last row of Table 1.1.
The question that should be asked is: faced with such reduced additional shares of energy delivered by renewable sources, who is going to invest huge amounts of money to install very high efficiency systems? The reply is obvious: no one. Unless there is a method that is also capable of promoting primary energy savings, besides the share of energy produced from renewable sources, everyone is going to select the cheapest solutions, and therefore Equation (1.1) promotes mediocrity, instead of rewarding excellence. More specifically, the topic is dealt with in paragraph 2.2.2.2.

Tab. 1.1 - Variations in the percentage of renewable energy share in connection to different SPF_{HP} values in accordance with Equation (1.1,) laid down in Italian Decree 28/11.

	SPF_{HP} = 3	**SPF_{HP} = 4**	**SPF_{HP} = 5**
Percentage of renewable sources calculated with Equation (1.1)	67%	75%	80%
Additional share in comparison to *SPF* 3	-	13%	20%
Primary energy savings in comparison to SPF_{HP}= 3	-	33%	66%

1.2.1.2 - Miscalculation of the renewable energy share sources due to the equation adopted by Italian Decree 28/11

Equation (1.1) in the Italian Decree may be misleading and lead to mistakes in the evaluation of the amount of energy, because it is based on the concept of seasonal average, and not on a case by case detailed calculation. The worse the mistake is, the greater the share of energy produced by the heat pump in conditions of low efficiency.
Mistakes may be due to two main reasons:

1) The calculation based on Equation (1.1) is not progressive: as soon as the lower limit value of 2,875 fixed by the Italian Decree for SPF_{HP} is exceeded, the share of energy produced from renewable sources, given by the term between brackets in the second side of the Equation, shoots up to 65%, giving rise to the stepwise discontinuity visible in Figure 1.2.

2) using Equation (1.1) two simultaneous conditions are allowed for: when energy from renewable sources is produced, and when this is not the case. This means that mistakenly the renewable energy share may turn out to be either overestimated or, more dangerously, underestimated.

Numerical Example 2 helps to better understand the matter.

NUMERICAL EXAMPLE 2

Let it be given an electric heat pump that delivers 1.000 kWh with COP = 2 (electricity consumption 500 kWh) and 1.000 kWh with COP = 4 (electricity consumption 250 kWh), that in real terms may be the case of a water to water heat pump that operates using two water sources, one at a higher temperature, not always available, and the second one at a lower temperature.

SPF_{HP} corresponds to the average COP value and is equal to 2,66 (ratio of the energy produced by the heat pump to the electrical energy absorbed by the compressors: 2.000/750).

If the right calculation is used, that is when detailed calculations on a case by case basis are performed, for COP = 2, the PER value is 0,8, which means that there is no production of energy from renewable sources, while for COP = 4, the amount of energy from renewable sources calculated applying Equation (1.1) is equal to 750 kWh, which corresponds to 37,5% of the total energy production.

If the calculation, instead, is performed based on the seasonal average, in accordance with Italian Decree 28/11, the amount of energy from renewable sources turns out to be equal to 0 kWh, because SPF_{HP} = 2,66 is lower than the lower limit of 2,875. It is clear that this is a serious mistake, because the risk is that a possible solution to cover a large share of the energy requirements from renewable sources is discarded, even though this is exactly what the Italian Decree would like to achieve for the heating period.

The seasonal average approach may also lead to an overestimation in the share of renewable energy. If only the amount of energy delivered by the water to water heat pump changed slightly, for example if the energy produced at COP value 4 were increased to 1250 kWh, and accordingly, the energy produced at COP value 2 were decreased to 750 kWh, SPF_{HP} would become equal to 2,91 (2.000 kWh produced divided by 687,5 kWh absorbed by the compressors), the lower limit of 2,875 would be slightly exceeded and the amount of energy produced from renewable sources would be 1.313 kWh, that is 65,6% of the total, if the present approach in the Italian Decree is followed. This percentage is obviously wrong, because out of the 1.250 kWh produced at COP value 4, only 937,5 kWh, equal to a percentage of 46,8% of the overall total requirement, would derive from a renewable source of energy.

The approach taken by the Italian Decree has two negative effects, that are described below, since it places at a disadvantage non electric heat pumps, and doesn't take into account the influence of a possible back-up with other generators.

1.2.1.2.1 - The handicap imposed on non-electric heat pumps

Heat pumps can be also fired with energy sources that are different from electricity. As a matter of fact, there are both gas engine heat pumps (Total Energy system), and heat pumps that operate with different cycles (absorption heat pumps). Both technologies are described in Annex A.

These kinds of heat pumps are in a strong disadvantage, in that they are characterized by the direct use of Primary Energy and therefore their SPF_{HP} values, if directly included in Equation (1.1), provide E_{RES} values that are lower than those of electrical heat pumps. This happens because, as already mentioned, Equation (1.1) doesn't allow for the Primary Energy Ratio, but rather only includes final energy.

To understand to what extent this approach is wrong, consider a refrigeration cycle capable of having a seasonal average *COP* value 4, with a compressor connected in one case to the electricity grid, and in another case to an engine with 30% electrical efficiency and 50% thermal efficiency. In the second case, all the thermal energy of the engine cooling is recovered and used for the heating. Figure 1.3 illustrates the diagram of the energy balances. As can be seen, when it comes to *PER* values, the pump driven by the endothermic engine has a better behavior, because it collects heat from the engine, yet, it is placed at a disadvantage by Italian Decree 28/11, since the amount of energy from renewable sources is calculated based on Equation (1.1).

Numerical example 3 highlights this critical aspect.

NUMERICAL EXAMPLE 3

Let it be given a yearly production of 100.000 kWh obtained with the heat pumps in Figure 1.3. For electrical heat pumps (PER = 1,6) the equation is as follows:

$$E_{RES} = E_{HP}\left(1 - \frac{1}{SPF_{HP}}\right) = 100.000\left(1 - \frac{1}{4}\right) = 75.000 \quad \text{kWh}$$

For endothermic engine heat pumps (PER = 1,7), where it is not necessary to comply with the lower limit of equation (1.3) because the system is not driven by electricity, the equation is as follows:

$$E_{RES} = E_{HP}\left(1 - \frac{1}{SPF_{HP}}\right) = E_{HP}\left(1 - \frac{1}{PER}\right) = 100.000\left(1 - \frac{1}{1{,}7}\right) = 41.175 \quad \text{kWh}$$

As can be seen, the endothermic engine heat pump is placed at a disadvantage since it produces a smaller amount of energy from renewable sources, even though its primary energy consumption is lower (PER = 1,7 versus PER = 1,6).

It is obvious that this contradiction has to be overcome.
AiCARR's proposal is reported in paragraph 1.2.2.

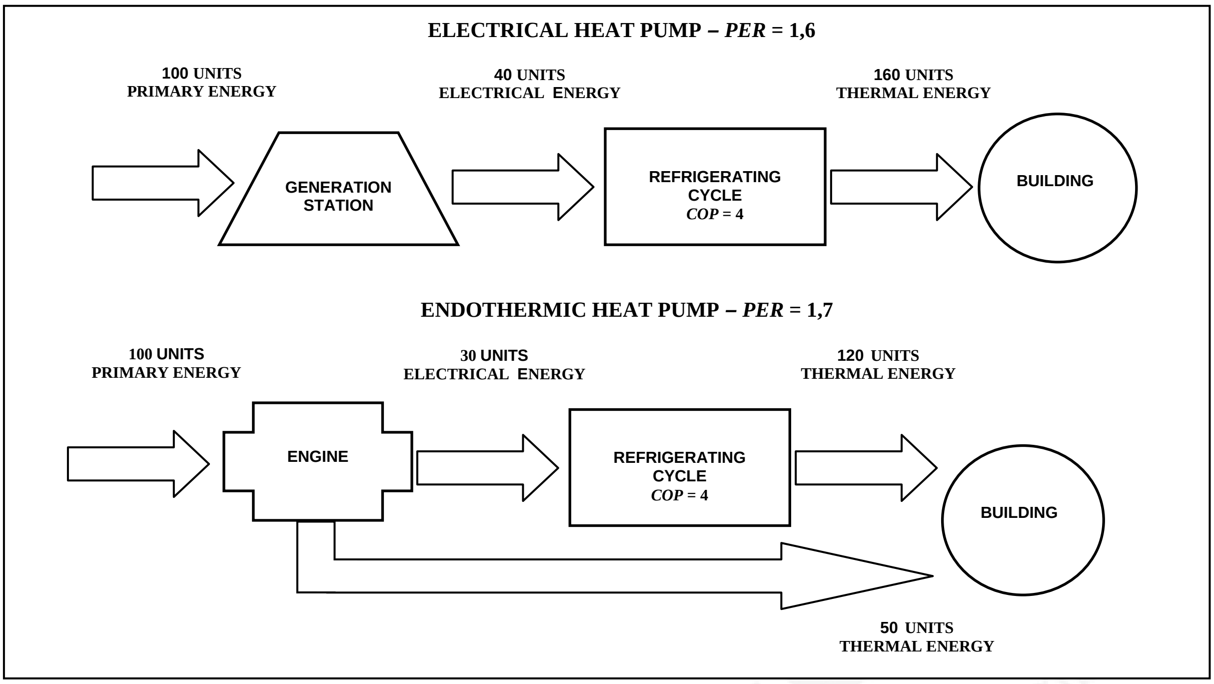


Fig. 1.3 - Energy Balance on heat pumps with electrical compressor and with compressor fit together with an endothermic engine. The *COP* value for the refrigeration cycle is the same in both cases.

1.2.1.2.2 - The problem of mixed systems (heat pump backed-up with other generators)

Heat pumps can or must be backed up with other generators, since their capacity may turn out to be insufficient for example due to a deteriorated performance linked to climatic conditions or because it is desirable to optimize the system that drives the pump. In these cases, Equation (1.1) is not sufficient to evaluate the amount of energy from renewable sources and must be corrected. Since these are multi-fuel systems, primary energy is the main concept to be taken into account.

To better understand the issue at stake, assume to have an electrical heat pump with a *COP* value that varies in relation to the temperature values of outdoor air as shown in Figure 1.4 that reports the state of the *COP* of an air source heat pump with a low-medium energy class and depicts the critical aspects that can take place in borderline cases and strengthens the concept that the Italian Decree is likely to promote mediocrity and discard excellence. The heat pump starts delivering energy from renewable sources when *PER* is equal to 1,15, calculated on the lower calorific value; therefore, in this case, for η = 0,4 - which means *COP* = 2,875 – it corresponds to a value of the outside temperature equal to around 2,5°C.

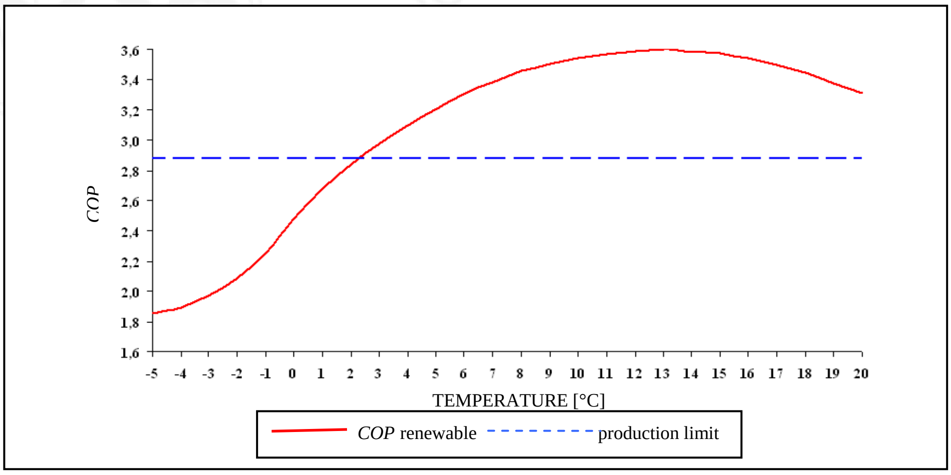


Fig. 1.4 - *COP* of an electrical heat pump in function of the values of outside air temperature (approximate values valid for an air source heat pump with medium-low energy efficiency class with twin-scroll compressors that operate in a plant at medium-high temperature ranges with adjustment for climatic conditions).

To get around this difficulty, the heat pump can be backed up with other generators, driven both with electricity and other fuels. To make a comparison, consider the following 4 cases:

Case 1: heat pump backed up with electrical resistance;
Case 2: heat pump backed up with natural gas condensing boiler with 105% efficiency on the lower calorific value;
Case 3: heat pump backed up with the boiler of case 2. The boiler operates in a stand-alone mode until heat pump *COP* value drops below 2,6, which corresponds to *PER* = 1,05 for the condensing boiler (the heat pump is disconnected when its primary energy consumption becomes higher than the boiler's one);
Case 4: heat pump in a stand-alone mode (in this case, the size is larger, because the stand-alone pump has to meet the maximum required capacity: for simplicity, the profile of the *COP* value is the same of the other three cases).

Given that the *PER* of the electrical resistance is equal to 0,4 and the one of the natural gas boiler to 1,05, in all the 4 cases, the production from renewable sources only happens when a certain temperature value is exceeded, according to the findings in Figure 1.5, which shows that below the cut-off point at which the heat pump starts operating alone, the various systems have different behaviors, fact that goes completely unobserved in the case that only Equation (1.1) is applied.

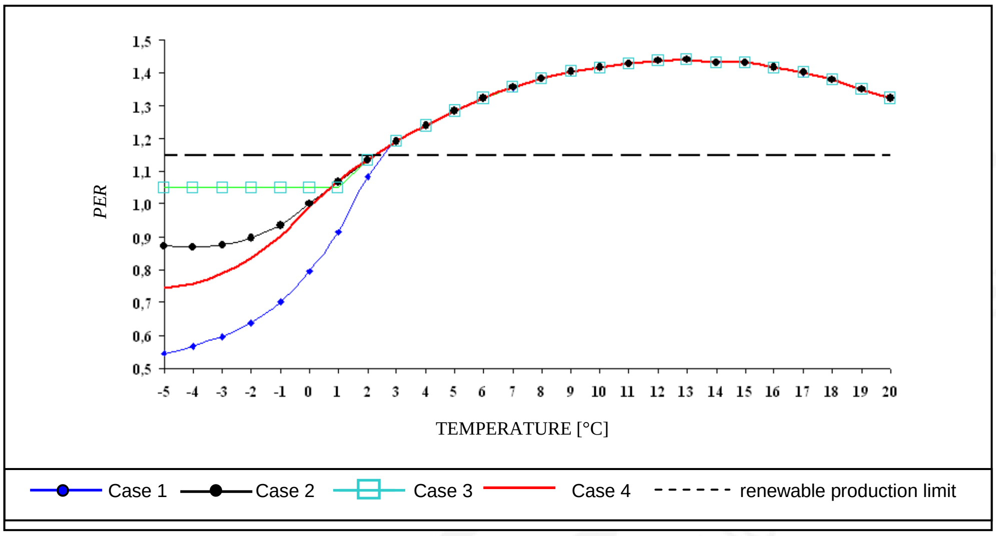


Fig. 1.5 - *PER* of a mixed system in function of outside air temperature values.

The analysis of the curves reported in Figure 1.5 shows that the amount of energy produced from renewable sources is essentially the same for all systems. The systems, though, present fairly important variations when it comes to PER_S. In particular, the curves that represent the Primary Energy Ratio coincide when values stay above 1,15, set point for the production of renewable energy, with the exception of the curve related to case 1 (back-up with electrical resistance), that presents a slight deviation at temperature values between 2 °C and 3 °C. Moreover, the four cases provide different results at low temperature values, when no energy from renewable sources is produced. By using Equation (1.1), conversely, values for energy production from renewable energy sources are subject to huge variations as shown in *Numerical Ex*ample 4, depending upon the case that is being considered.

NUMERICAL EXAMPLE 4

Let it be given a heat pump with a COP value equal to the one in Figure 1.4 for a building with the same required heating capacity as the one reported in Table 1.2.
Suppose that in cases 1 to 3 the heat pump has a maximum capacity of 72,4 kW at minimum temperature conditions, and that in case 4 the heat pump is sized to supply all of the needed capacity also at -5°C (therefore it has to provide a capacity of 125 kW, nearly twice as much as the others: for simplicity's sake, suppose that the COP profile is the same). The characteristics related to the operation are described in Table 1.3, from which it can be derived that for low outside air temperatures, the worst solution with regard to PER is represented by case 1 (back-up with electrical resistance), the best by case 3 (boiler operating with heat pump COP lower than 2,6 and heat pump operation is terminated).
The PER_S values are:
Case 1: PER_S = 1,022
Case 2: PER_S = 1,163
Case 3: PER_S = 1,206
Case 4: PER_S = 1,140
All this is completely disregarded if the calculation is performed following Equation (1.1), whereby the SPF_{HP} values are weighted only with the energy amounts supplied by the heat pump. Table 1.4 reports the SPF_{HP} values together with the data on the amount of energy supplied with renewable sources for cases from 1 to 4. Comparing PER_S values and SPF_{HP} values for case 4 (heat pump on its own) and case 1 (back up with electrical resistance), it turns out that the first, despite its better PER_S value (1,140 versus 1,022), has a lower SPF_{HP} (2,85 versus 2,92), even lower than the set point, and it turns out that the system doesn't produce any energy from renewable sources. This is a paradox, that can be explained by the fact that most of the production of heat energy supplied

by the heat pump at low air temperature leads to a reduction in SPF_{HP} values. In the calculation executed with Equation (1.1) the back-up heating is not considered, even though it takes place at worse performance levels (the COP value of an electrical resistance is 1, lower than the heat pump's one).

Tab. 1.2 – Required heating capacity for the building considered in Numerical Example 4.

Temperature [°C]	Number of hours	Required capacity [kW]	HP capacity [kW]	*COP*	Required capacity [kWh]
-5	35	125	72,4	1,86	4.375
-4	60	120	75,5	1,89	7.200
-3	85	115	76,7	1,97	9.775
-2	145	110	78,9	2,09	15.950
-1	194	105	81,0	2,25	20.370
0	224	100	83,2	2,48	22.400
1	261	95	85,4	2,68	24.795
2	271	90	87,5	2,84	24.390
3	285	85	85	2,98	24.225
4	297	80	80	3,10	23.760
5	302	75	75	3,21	22.650
6	280	70	70	3,30	19.600
7	275	65	65	3,39	17.875
8	270	60	60	3,45	16.200
9	256	55	55	3,51	14.080
10	251	50	50	3,54	12.550
11	225	45	45	3,57	10.125
12	211	40	40	3,59	8.440
13	187	35	35	3,60	6.545
14	153	30	30	3,58	4.590
15	130	25	25	3,58	3.250
16	113	20	20	3,54	2.260
17	101	15	15	3,50	1.515
18	88	10	10	3,45	880
19	89	5	5	3,38	445
20	77	0	0	3,31	0
	4.865				**318.245**

Likewise, case 3 gives a higher SPF_{HP}, but the amount of energy supplied by the heat pump is lower than the one in cases 1 and 2 (161.349 kWh versus 193.657 kWh, as shown in Table 1.4) because the heat pump operation is terminated when its PER is lower than the boiler's one and supplies less energy even though with a higher SPF_{HP} value and, therefore, according to Equation (1.1), the amount of energy supplied by renewable sources is smaller.

Tab. 1.3 - Energy produced by the heat pump and by back-up generators (electrical resistance or boiler) and PER values of the system

T [°C]	Case 1: Back up with electrical resistance			Case 2: Back up with boiler			Case 3: boiler in stand-alone production when *COP* <2,6			Case 4: HP on its own	
	HP [kWh]	Elettrical resistance [kWh]	*PER*	HP [kWh]	boiler [kWh]	*PER*	HP [kWh]	boiler [kWh]	*PER*	HP [kWh]	*PER*
-5	2.532	1.843	0,55	2.532	1.843	0,87	0	4.375	1,05	4.375	0,74
-4	4.472	2.728	0,57	4.472	2.728	0,87	0	7.200	1,05	7.200	0,76
-3	6.519	3.256	0,60	6.519	3.256	0,87	0	9.775	1,05	9.775	0,79
-2	11.436	4.514	0,64	11.436	4.514	0,90	0	15.950	1,05	15.950	0,83
-1	15.721	4.649	0,70	15.721	4.649	0,94	0	20.370	1,05	20.370	0,90
0	18.639	3.761	0,79	18.639	3.761	1,00	0	22.400	1,05	22.400	0,99
1	22.284	2.511	0,92	22.284	2.511	1,07	22.284	2.511	1,05	24.795	1,07
2	23.726	664	1,08	23.726	664	1,13	23.726	664	1,13	24.390	1,14
3	24.225	0	1,19	24.225	0	1,19	24.225	0	1,19	24.225	1,19
4	23.760	0	1,24	23.760	0	1,24	23.760	0	1,24	23.760	1,24
5	22.650	0	1,28	22.650	0	1,28	22.650	0	1,28	22.650	1,28
6	19.600	0	1,32	19.600	0	1,32	19.600	0	1,32	19.600	1,32
7	17.875	0	1,35	17.875	0	1,35	17.875	0	1,35	17.875	1,35
8	16.200	0	1,38	16.200	0	1,38	16.200	0	1,38	16.200	1,38
9	14.080	0	1,40	14.080	0	1,40	14.080	0	1,40	14.080	1,40
10	12.550	0	1,42	12.550	0	1,42	12.550	0	1,42	12.550	1,42

(%)

Tab. 1.3 -

T [°C]	Case 1: Back up with electrical resistance			Case 2: Back up with boiler			Case 3: boiler in stand-alone production when *COP* <2,6			Case 4: HP on its own	
	HP [kWh]	**Electrical resistance [kWh]**	***PER***	**HP [kWh]**	**boiler [kWh]**	***PER***	**HP [kWh]**	**boiler [kWh]**	***PER***	**HP [kWh]**	***PER***
11	10.125	0	1,43	10.125	0	1,43	10.125	0	1,43	10.125	1,43
12	8.440	0	1,44	8.440	0	1,44	8.440	0	1,44	8.440	1,44
13	6.545	0	1,44	6.545	0	1,44	6.545	0	1,44	6.545	1,44
14	4.590	0	1,43	4.590	0	1,43	4.590	0	1,43	4.590	1,43
15	3.250	0	1,43	3.250	0	1,43	3.250	0	1,43	3.250	1,43
16	2.260	0	1,42	2.260	0	1,42	2.260	0	1,42	2.260	1,42
17	1.515	0	1,40	1.515	0	1,40	1.515	0	1,40	1,515	1,40
18	880	0	1,38	880	0	1,38	880	0	1,38	880	1,38
19	445	0	1,35	445	0	1,35	445	0	1,35	445	1,35
20	0	0	0,00	0	0	0,00	0	0	0,00	0	132
	294.320	**23.925**		**294.320**	**23.925**		**235.000**	**83.245**		**318.245**	

Tab. 1.4 - Overview of the main data in the four considered cases. The energy amount supplied by renewable sources is calculated according to Equation (1.1)

	SPF_{HP}	PER_{Syst}	**Energy supplied by renewable source [kWh]**	**Percentage on total**
Case 1	2,92	1,022	193.657	60,9%
Case 2	2,92	1,163	193.657	60,9%
Case 3	3,19	1,206	161.349	50,7%
Case 4	2,85	1,140	0	0,0%

It is evident that Numerical Example 4 clearly shows a critical aspect that has to be addressed. AiCARR puts forward one possible solution which is described in paragraph 1.2.2.

1.2.2 - AiCARR's suggestions to overcome critical aspects

Some preliminary remarks. AiCARR believes that the calculation method laid down by the Italian Decree presents two fundamental problems: one is connected to Equation (1.1), the other to the use of seasonal average values of the coefficients. In the following, some suggestions that address the solutions to the critical points linked to the use of Equation (1.1). Aspects related to the use of seasonal average values are addressed in part 2 of the present document, in particular in paragraphs 2.1 and 2.2.
In paragraph 1.2.2.1, AiCARR would like to submit a method that is in line with the provisions laid down by the European Directive, overcoming critical aspects and aligning the Italian Decree to legislations and standards on energy efficiency, for example promoting the solutions that lead to real primary energy savings.
However, to improve the accuracy of the calculation, as mentioned in paragraph 2.2, the sum of hourly values, and not seasonal average values should be considered.

1.2.2.1 - AiCARR's Method

AiCARR submits a method whereby *SPF* values continue to be used, systems that actually provide energy savings, that are characterized by high PER_S values, are promoted, and heat pumps operating with different fuels are compared.
AiCARR's Method is based on Equation (1.6), where, though, the *SPF* value is not the same value of a heat pump, but rather the conventional *SPF* value of the whole system, which is obtained dividing the PER_S value of the whole system, $PER_{S,Syst}$, by the efficiency of production and distribution of the electricity η:

$$E_{RES} = E_{Tot}\left(1-\frac{1}{SPF_{C,Syst}}\right) = E_{Tot}\left(1-\frac{\eta}{PER_{S,Syst}}\right) \qquad (1.6)$$

where:

E_{Tot} = total energy supplied by the whole system;
$SPF_{C,Syst}$ = conventional *SPF* for the whole system of energy production; it corresponds to the *SPF* value of the heat pump when it operates on its own in the stand-alone mode;
$PER_{S,Syst}$ = Seasonal Primary Energy Ratio of the whole system of energy production.

The lower limit value for $SPF_{C,Syst}$ proposed by AiCARR is 2,5, which corresponds to $PER_{S,Syst}$= 1. This value is lower than the 1,15 limit value, above which there is production of energy from renewable energy sources, but it is worth being reminded of the following:

- in a seasonal average calculation, two different situations are allowed for at the same time: when there is production of energy from renewable sources, and when there isn't. Therefore, even with $PER_{S,Syst}$ values below 1,15, there can be production of energy from renewable sources, as already shown by Numerical Example 2 and as will be better demonstrated by the cases 1 and 4 of Numerical Example 6;
- The $PER_{S,Syst}$ = 1 value corresponds to the value of a condensing boiler of acceptable quality. Consequently, if this value is set as a lower limit level, the energy that is supplied by any system that, in its entirety, has a primary energy consumption higher than the one used by a condensing boiler, will never be considered as energy that is supplied by a renewable source. If this were not the case, either the system would be wrong, or its size would be wrong.

Rationale behind Equation (1.6)

Equation (1.6) reveals what is described in paragraph 1.1: the primary energy, expressed in terms of $PER_{S,Syst}$, is transformed into final energy to comply with the approach of the European Directive. Using AiCARR's Method, all energy production systems are compared and the ones that consume less are promoted, since they are characterized by a higher $PER_{S,Syst}$ value.

As already mentioned, Equation (1.6) can be used to compare electricity driven heat pumps with the non-electric ones, or with systems equipped with back-up generators, electrical and/or fired with other fuels. Numerical Examples 5 and 6 illustrate respectively these two cases.

NUMERICAL EXAMPLE 5

In Numerical Example 3 the heat pump with endothermic engine has PER_S = 1,7, higher than the value of an electricity driven heat pump, equal to 1,6, and yet, in accordance with Equation (1.1), it produces a much smaller amount of energy from renewable sources.

If, instead, AiCARR's Method is applied:

- *for an electrical heat pump*

$$E_{RES} = E_{Tot}\left(1 - \frac{1}{SPF_{C,Syst}}\right) = E_{Tot}\left(1 - \frac{\eta}{PER_{S,Syst}}\right) = 100.000\left(1 - \frac{1}{4}\right) = 75.000 \quad \text{kWh}$$

which is the same amount of energy from renewable sources calculated according to Equation(1.1).

- *for an endothermic engine heat pump:*

$$E_{RES} = E_{Tot}\left(1 - \frac{1}{SPF_{C,Syst}}\right) = E_{Tot}\left(1 - \frac{\eta}{PER_{S,Syst}}\right) = 100.000\left(1 - \frac{1}{1{,}7}\right) = 100.000\left(1 - \frac{1}{4{,}25}\right) = 76.470 \quad \text{kWh}$$

A higher value than the one obtained for an electrical heat pump, which is exactly rightly so, since the $PER_{S,Syst}$ in this case is higher.

NUMERICAL EXAMPLE 6

Consider Numerical Example 4, where the value of the total energy requirement is equal to 318.245 kWh, as reported in Table 1.2. If Equation (1.6) is applied to the same case, the results reported in Table 1.5 are obtained, that clearly demonstrate that with AiCARR's Method, the systems that are promoted are the ones with a higher PER_S.

Tab. 1.5 - Overview of the main data related to the four cases examined by Numerical Example 4

	Energy requirement [kWh]	PER_S	$SPF_{C\ Syst}$	AiCARR's Method		Italian Decree 28/11	
				Energy from Renewable sources [kWh]	% on total	Energy from renewable sources [kWh]	% on total
Case 1	318.245	1,022	2,55	193.657	60,9%	193.657	60,9%
Case 2	318.245	1,163	2,91	208.812	65,6%	193.657	60,9%
Case 3	318.245	1,206	3,02	212.716	66,8%	161.349	50,7%
Case 4	318.245	1,140	2,85	206.580	64,9%	0	0,0%

The two considered examples clearly show that AiCARR's Method promotes the heat pump driving systems that consume less primary energy in percentage of the energy produced from renewable sources.

In particular, Numerical Example 6 shows that AiCARR's Method addresses one of the most serious mistakes in the approach of the Italian Decree. As a matter of fact, Figure 1.5 on page 15 shows that the amount of energy produced from renewable sources is similar in all four cases, but that case 4 is placed at a disadvantage by the use of Equation (1.1) because the SPF_{HP} value drops below the lower limit value given that the heat pump, if compared to the other three cases, supplies a higher amount of energy at the low temperature ranges; the element that makes the difference is the $PER_{S,Syst}$ value, because when the back-up sets in after cut in point, primary

energy consumption differs from one case to the other, a factor that is not allowed for by Equation (1.1), while AiCARR's Method is successfully addressing and solving it.

1.3 - Mediterranean countries: the importance of summer air conditioning

In the countries of the Mediterranean area, the energy demand for summer air conditioning is a relevant fraction of the yearly total amount for air conditioning, therefore an important issue is the strong demand for energy produced from renewable sources in summer.
The European Directive, on which the Italian Decree is based, follows the typical Northern European approach, whereby the weight of winter heating is definitely greater than the one of summer air conditioning. In these conditions, a heat pump system for the production of heating and domestic hot water is in itself sufficient to cover 50% of the requirement, this being the limit level fixed by the Italian Decree. As a matter of fact, thanks to the share of energy supplied by "conventional" renewable sources, following Equation (1.1) of the Italian Decree, in winter operation an energy surplus is delivered by renewable sources that can be used in the summertime for air conditioning. As was shown in Numerical Example 1, a heat pump with SPF_{HP} = 4 is capable of meeting the 50% limit level requirement in the case when the energy demand of the summertime represents 50% of the winter one.
In the Mediterranean countries, this can be achieved only in the residential sector, while in all the other cases, the weight of the summer demand for energy is definitely greater, due to internal heat loads caused by the presence of people, by lighting needs and the electric equipment. Figure 1.6 illustrates with a diagram how the percentages split up between the two different sectors, clearly showing that in the buildings of the tertiary sector, a remarkable amount of energy is required when heating and cooling loads overlap, and there is a simultaneous need for both spaces heating and cooling this problem, addressed in Annex D, needs customized solutions.
In our latitudes, the role played by the thermal insulation of the structures is strong in the winter and poor in the summer, when the load due to solar radiation passing through the transparent surfaces of the envelope has to be reduced, for example with appropriate shading devices; this could lead to too little illumination and a reduction in visual comfort, and consequently, a possible increase in electricity consumption for lighting and more energy required for air conditioning. On the other hand, even in environments that are completely adiabatic, more cooling energy would be needed to compensate internal loads.
The minimum requirements imposed by the Italian Decree (50% in 2017) are very ambitious and it is absolutely impossible to meet them based on the present approach laid down by the Italian Decree. The risk is that in most cases the designer is going to invoke and apply the technical impediment clause. AiCARR's suggestions, reported in paragraphs from 1.4 to 1.10, are aimed at mitigating this risk.

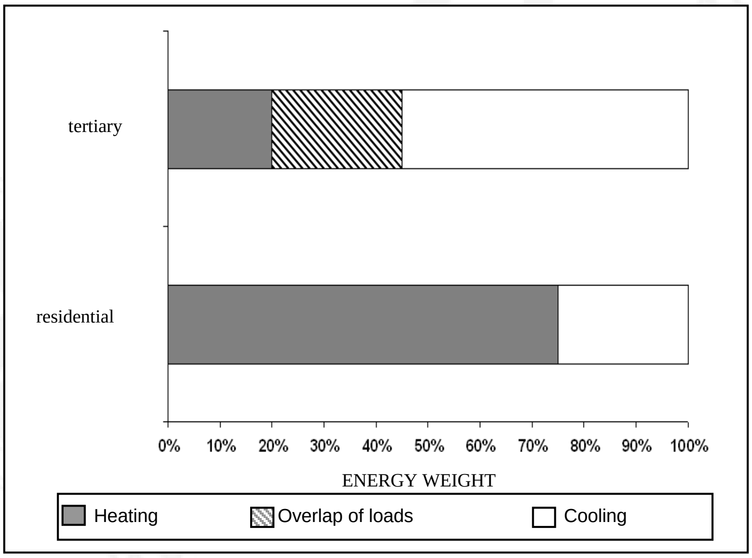


Fig. 1.6 - Energy share required for the heating and cooling in the residential and tertiary sectors.

1.3.1 - Percent shares of renewable energy and absolute energy consumption: risks connected to the selected approach

The Italian Decree lays down a percent share target for the energy amount that has to be supplied by renewable sources. Unfortunately, though, energy consumption data must be expressed in absolute terms and the provisions contained in the Italian Decree are not always suitable to meet energy saving needs.
To meet the target imposed by the Italian Decree, and taking into account that in winter operation energy supplies from renewable sources are relatively easy but that in summer operation this kind of energy production is less easy, the simplest solution is exploiting the surplus of winter energy supplies using heat pumps with high SPF_{HP} values. In following paragraphs, the problems deriving from all this are closely examined. This part aims at clarifying that one possible solution – avoiding to introduce thermal insulation, so that winter consumption levels are increased, with a surplus of E_{RES} and coverage guaranteed for the whole year – is appropriate to meet the percent share target provision of the Italian Decree, but completely wrong from the technical point of view. Indeed, thermal

insulation has a great influence on energy consumption levels in winter, but in summer it is not relevant at all. Numerical Example 7 clarifies the concept more effectively.

NUMERICAL EXAMPLE 7

Let it be given a building that has an energy requirement of 100.000 kWh, needed for the heating and production of domestic hot water, and of 50.000 kWh needed for the cooling system.
A heat pump system that operates with SPF_{HP} = 4 meets the 50% percent share target imposed by the Italian Decree, because the total energy supplied from renewable energy sources is equal to 75.000 kWh, which represents 75% of the energy input required in winter operation, and is 50% of the 150.000 kWh which represent the yearly energy requirement amount.
Suppose that it is possible to insulate the envelope and that this intervention reduces energy requirements in winter by 28.000 kWh and by 5.000 kWh in summer. In accordance with the Italian Decree, this intervention would not be effective to meet the percent share target: as a matter of fact, the total yearly requirement drops to 117.000 kWh and the amount of energy supplied by renewable sources is reduced to 54.000 kWh (75% of the 72.000 kWh needed in winter), which represents 46% of the total energy needs. Therefore, at least 4.500 kWh designers have to achieve that at least 4.500 kWh, needed for the summer air conditioning, are produced from renewable energy sources.

In the example illustrated above, the best solution in terms of energy consumption is not promoted, and cases like this one are critical points to be addressed.
If some technologies used in summer are not considered as renewable sources of energy, the risk is that designers may adopt systems that in winter consume more energy, because their main concern is the absolute value of the energy supplied by renewable sources, which they don't want to reduce.

1.4 - Why should heat recovery from exhaust air be considered as a renewable energy source.

NOTE: In this paragraph and in the following ones AiCARR provides some suggestions to foster the systems with higher energy efficiency that have to be adopted in buildings independently of the choice made on the calculation method by the Italian Decree, which means independently of whether seasonal averages are used, as laid down by the Italian Decree, but this is not the right approach.
In the second part of the present document, AiCARR explains the reason why the sum of accurate estimates on a case by case basis has to be selected to achieve correct results.

Based on the contents of paragraph 1.3.1, heat recovery from exhaust air has to be considered as a renewable source of energy, which is obvious if it is considered that the air extracted from the environment contains "aeraulic energy", is in general in favorable conditions if compared to outside air and is an inexhaustible source, at least as long as the plant is working.
The problem that may arise is linked to the calculation method of the amount of energy that is supplied by a renewable source. AiCARR proposes to use a criterion that is similar to the one that is used in the case of heat pumps:

$$E_{RES} = E_{Rec}\left(1-\frac{1}{SPF_{Rec}}\right) \tag{1.7}$$

where:
E_{Rec} = energy recovered from the system, for each the summer, and the winter operation;
SPF_{Rec} = Seasonal Performance Factor of the exhaust air recovery system, defined by:

$$SPF_{Rec} = \frac{E_{Rec}}{E_{abs,Rec}} \tag{1.8}$$

where $E_{abs,Rec}$ is the electrical energy that the recovery system absorbs to offset the load losses of its components. In the case of dynamic recovery units, $E_{ass,Rec}$ has to take into account the electrical energy that is absorbed by compressors.

Rationale behind Equations (1.7) e (1.8)
These equations are used to put heat recovery on an equal footing with a heat pump. The Seasonal Performance Factor, SPF_{Rec} in this case, is equal to the ratio of the energy that is consumed to the electricity that is consumed. In this way, the systems that achieve the best energy savings are promoted.
Numerical Example 8 helps to better understand the mistakes that would be made if exhaust air were not considered as a renewable energy source.

NUMERICAL EXAMPLE 8

Let it be given, for sake of simplicity, an all-air installation working at a 1.000 kg/h mass flow of air throughout the year. The air has to be conditioned at ambient temperatures. In a city in Northern Italy, Milano for example, the total thermal energy demand would be equal 20.850 kWh in winter operation and 9.409 kWh in summer opera-

tion. Assume that the energy is supplied by a heat pump with SPF_{HP} = 4 both in summertime and in wintertime. It is possible to reduce consumptions by introducing sensible heat recovery systems (e.g. cross-flow static recuperator), with efficiency levels that are respectively equal to 50% and 75%: in the first case, the electrical energy that is absorbed by the recovery system, $E_{abs,Rec}$, is equal to *973 kWh, in the second to 1.582 kWh (the higher the efficiency, the greater the pressure drop of the recovery system).*
Tables 1.6 and 1.7 illustrate the results that are obtained considering the energy that is recovered respectively reducing the heat requirements or using AiCARR's Method.

The results reported in Table 1.6, even though referred to a specific case, are meaningful: heat recovery has a much greater effect in winter, when the difference in temperature and enthalpy values between inside and outside air is higher, which also entails that winter requirements are reduced due to the higher efficiency of the heat recovery system. With this approach, though, following the Italian Decree definition, the renewable energy share drops as well, reaching the paradox whereby the use of recovery systems with an efficiency rate equal to 75%, although leading to a 30% reduction in the electricity consumption of the system, supplies only 19% of the required energy with renewable sources, versus a percentage of 52% obtained with the system without heat recovery.
The results are completely different if AiCARR's Method is applied. As can be seen when examining the values reported in Table 1.7, considering exhaust air as a renewable source is correct not only formally, but also from the energy balance point of view. As a matter of fact, all in all, a larger share of energy from renewable sources is substantially in line with the energy consumption reduction targets. The introduction of a heat recovery unit with a 50% efficiency leads to a 21% reduction in energy consumption and to a 16% increase in the amount of energy produced from renewable sources. Similarly, the introduction of a heat recovery unit with an efficiency of 75% leads to a 30% reduction in energy consumption and to a 24% increase in the amount of energy produced from renewable sources.

Tab. 1.6 - Example of the results that can be obtained considering the amount of energy that is recovered thanks to heat requirement reductions, in accordance with the provisions laid down by the Italian Decree. For the data, please refer to Numerical Example 7

		Without recovery	**Recovery 50%**	**Recovery 75%**
Thermal energy requirement [kWh]	Winter	20.850	20.850	20.850
	Summer	9.409	9.409	9.409
	Total	**30.258**	**30.258**	**30.258**
Thermal energy recovery, E_{Rec} [kWh]	Winter	0	8.864	13.295
	Summer	0	1.331	1.997
	Total	**0**	**10.195**	**15.292**
Thermal energy produced by HP, E_{HP} [kWh]	Winter	20.850	11.986	7.554
	SPF_{Pdc}	4,00	4,00	4,00
	% RES	75%	75%	75%
	Summer	9.409	8.077	7.412
Electric energy consumed, $E_{abs,Rec}$ [kWh]	Winter HP	5.212	2.997	1.889
	Summer HP	2.352	2.019	1.853
	Recovery	0	973	1.582
	Total	**7.565**	**5.989**	**5.323**
	Saving	**-**	**21%**	**30%**
Renewable energy produced, E_{RES}	**[kWh]**	**15.637**	**8.990**	**5.666**
	%	**52%**	**30%**	**19%**

Tab. 1.7 - Examples of the results that can be obtained with AiCARR's Method. For the data, please refer to Numerical Example 7

		Without recovery	**Recovery 50%**	**Recovery 75%**
Thermal energy requirement [kWh]	Winter	20.850	20.850	20.850
	Summer	9.409	9.409	9.409
	Total	**30.258**	**30.258**	**30.258**
Thermal energy requirement, E_{Rec} [kWh]	Winter	0	8.864	13.295
	Summer	0	1.331	1.997
	Total	**0**	**10.195**	**15.292**
	SPF_{Rec}	0	10,5	9, 7
	% renewables	0%	90%	90%

(%)

Tab. 1.7 -

		Without recovery	Recovery 50%	Recovery 75%
Thermal Energy produced by HP, E_{HP} [kWh]	Winter	20.850	11.986	7.554
	SPF_{HP}	4	4	4
	% renewables	75%	75%	75%
	Summer	9.409	8.077	7.412
Electrical energy consumed, $E_{ass,Rec}$ [kWh]	Winter HP	5.212	2.997	1.889
	Summer HP	2.352	2.019	1.853
	Recovery	0	973	1.582
	Total	**7.565**	**5.989**	**5.323**
Renewable Energy produced, E_{RES} [kWh]	HP	15.637	8.990	5.666
	Recovery	0	9.221	13.711
	Totale	**15.637**	**18.211**	**19.376**
	%	**52%**	**60%**	**64%**

1.4.1 - Further reasons why heat recovery has to be considered as a renewable energy source

A further reason to follow AiCARR's Method is that in so doing one also takes into account the electricity needed to operate the recovery system, which is not allowed for in the Italian Decree. Heat recovery, as a matter of fact, is evaluated by the Italian Decree as a reduction in heating and cooling loads, capable of reducing energy requirements, but ignoring the electricity consumption that is needed for the heat recovery.
For example, it has to be underlined that the provisions laid down in the Italian Decree are not in line with some European standards, one of which is EN 13779:2008, that provides an energy classification for fans based on the *SFP* ratings (Specific Fan Power), in [W·s·m^{-3}], defined as the capacity needed to move 1 m^3/s of air in an air conditioning system. In the heat recovery process, ignoring the weight of the energy needed for ventilation in the heat recovery is a mistake that can lead to results that are opposite to those that are wished for by the legislator.

1.5 - Why should Free-Cooling be considered as a renewable energy source

Also direct and indirect Free-Cooling systems that are analyzed in Annex C, have to be considered as a renewable energy source and not as a heating requirement reduction system.
Direct Free-Cooling systems use an inexhaustible aerothermal source, outdoor air supply, which is in better energy conditions than indoor air; indirect Free-Cooling systems use an inexhaustible source - geothermal, hydrothermal or aerothermal - to pre-cool or to totally cool the water that has to be conveyed to the cooling plant. Hence, there is no reason why Free-Cooling should not be considered a system that exploits energy from renewable sources.
In this case again, AiCARR' Method proceeds by analogy to the reasoning made in relation to heat pumps, considering that the energy from renewable sources is equal to:

$$E_{RES} = E_{FC}\left(1 - \frac{1}{SPF_{FC}}\right) \tag{1.9}$$

where E_{FC} is the energy needed for the cooling produced with the Free-Cooling system and SPF_{FC} is the Seasonal Performance Factor of the Free-Cooling system, defined by:

$$SPF_{FC} = \frac{E_{FC}}{E_{abs,FC}} \tag{1.10}$$

where $E_{abs,FC}$ is the surplus of electrical energy that the Free-Cooling system may need to operate. This surplus is not always present, since it depends on possible supplemental heat exchangers or airflow rates and on the possible involvement of other pumps in the hydrothermal or geothermal systems. If this surplus is not available, the SPF_{FC} value would be equal to ∞ and all the energy produced by the Free-Cooling system should be considered renewable.

Rationale behind the Equations (1.9) and (1.10)

The calculation compares a Free-Cooling system to a heat pump. The SPF_{FC} efficiency is equal to the ratio of the energy that is produced for free with the Free-Cooling to the possible surplus of electrical energy.
Numerical Example 9 clarifies the implications of considering Free-Cooling systems as a renewable source of energy.

NUMERICAL EXAMPLE 9

Let it be given an HVAC plant, working all year long, with a winter energy requirement of 500.000 kWh and a summer energy requirement of 1.000.000 kWh. The total energy requirement of the plant is equal to 1.500.000

kWh/year and the renewable energy share to fulfill the provisions of Italian Decree 28/11 (as of 2017) is equal to 750.000 kWh.
Suppose that a Free-Cooling system with SPF_{FC} = 10 is capable of producing 500.000 kWh.
Suppose in short that the production of energy from renewable sources (heat pump in the winter period) is equal to 350.000 kWh.
If the Free-Cooling share is not accounted as renewable, the 500.000 kWh produced with the Free-Cooling represent a reduction in the total energy requirement, whose value is thus converted to 1.000.000 kW, while the amount of energy from renewable sources that has to be produced becomes 500.000 kWh. Out of this quantity, 350.000 kWh are already produced in winter, so that the remaining 150.000 kWh have to be produced during summer.
If, conversely, the Free-Cooling system is accounted for as a renewable source, in line with AiCARR's proposal, the 350.000 kWh produced in the winter are supplemented by the ones obtained with Equation (1.10) in the summer:

$$E_{RES} = E_{FC}\left(1 - \frac{1}{SPF_{FC}}\right) = 500.000\left(1 - \frac{1}{10}\right) = 450.000 \quad \text{kWh}$$

Therefore, the total sum gives an amount of 800.000 kWh, well above the lower bound of 750.000 kWh requested by Italian Decree 28/11.

Therefore, Free-Cooling systems should be indeed considered as contributing to renewable energy share, and it is fair that systems that adopt a Free-Cooling unit are promoted.

1.6 - Heat recovery from refrigerators in summer operation

The recovery of condensation heat from refrigerators is useful when there is an overlap of loads in a plant, in other words when there is a simultaneous request of heating and cooling, as shown in Figure 1.7.
The overlap of loads can take place because there is a simultaneous demand of heating in one area of the building, and of cooling in another, as shown in Figure 1.7, or alternatively because there is a simultaneous demand of cooling and domestic hot water production and/or heating of fresh air. The overlapping of heat loads is analyzed and discussed in Annex D.
There are different ways to recover heat from refrigeration cycles, as illustrated in detail in Annex E. When the refrigeration system works with the so-called total heat recovery, one should consider that its main goal is to produce heating energy and that refrigeration energy production is a byproduct, which, for a building, has to be seen as an energy requirement reduction for the summertime. In layman terms, the refrigeration system keeps on working as a heat pump, producing heat and recovering cold. If, on the other hand, the refrigeration system works as a refrigerator, recovering heat and reducing the thermal energy requirement of a building, the risk is that designers may adopt energy-consuming solutions, that are far from being in line with the spirit of the Italian Decree. Numerical Example 10 helps to understand the problem.

NUMERICAL EXAMPLE 10

Let it be given a plant that constantly requires both heating energy and cooling energy, for a total of 100.000 kWh "for heating purposes" and 75.000 kWh "for cooling purposes".
The requirement can be fulfilled in two different ways: using two separate generators, one heat pump and one refrigeration unit (there could possibly be one single generator functioning alternatively in the two modes), or rather a total recovery system, illustrated in Annex E. Suppose that the Seasonal Performance Factor values are the same in the two cases: EER = 3 and COP = EER + 1 = 4 (the meaning of EER is examined thoroughly in Annex B).

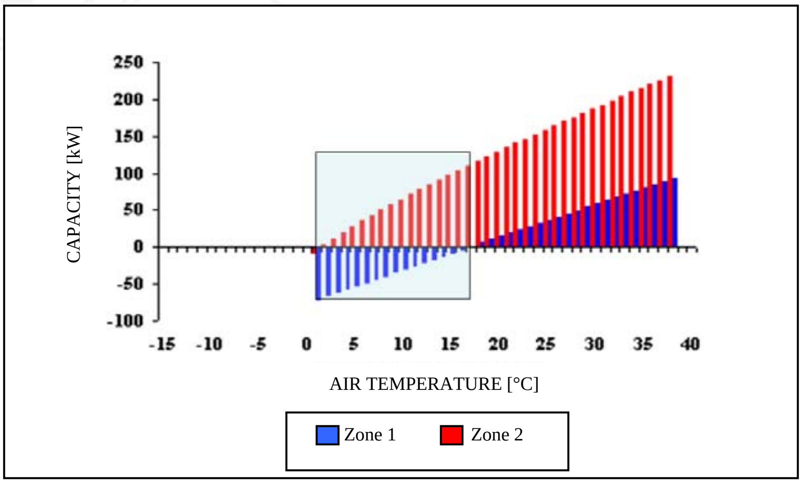


Fig. 1.7 - Overlap of loads in a building (negative values = heating; positive values = cooling).

In the first case the energy balance is shown in Figure 1.8. The electricity consumption is equal to 50.000 kWh; the dual-fuel system with double generation shall be as follows:
- *Heating requirement = 100.000 kWh*
- *Cooling requirement = 75.000 kWh*
- *Electricity consumption: (100.000/4)+(75.000/3)= 50.000 kWh*
- *Total thermal requirement = 175.000 kWh*
- *SPF = 4*
- *Energy produced from RES = 75.000 kWh*
- *Share of energy produced from RES in the total thermal requirement = 43%*
- *Amount of energy that has to be produced from other RES = 12.500 kWh*

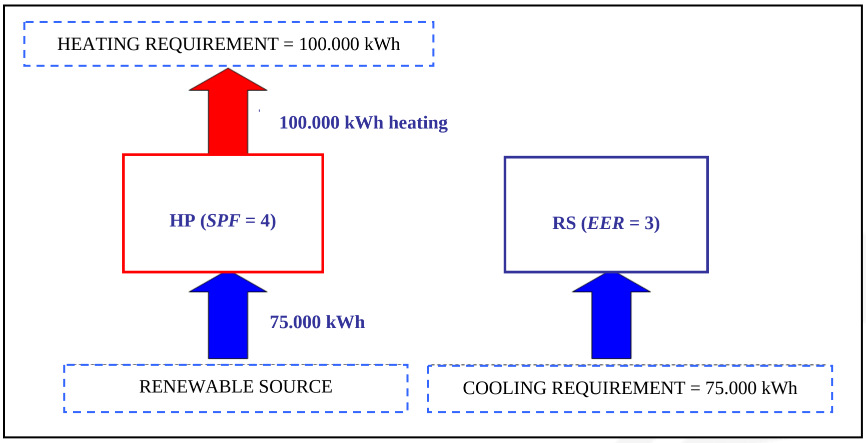


Fig. 1.8 - Energy Balance of a dual-fuel system.

In the second case the energy balance is shown in Figure 1.9. The electricity consumption is equal to 25.000 kWh. As a matter of fact, in the case of the total heat recovery system, if thermal energy is considered to be heat recovery energy, it is as follows:
- *Heating requirement = 0 kWh (all recovered)*
- *Cooling requirement = 75.000 kWh*
- *Total thermal requirement = 75.000 kWh*
- *Energy produced from RES = 0 kWh*
- *RES share in the total thermal requirement = 0%*
- *Amount of energy that has to be produced from other RES = 37.500 kWh*
- *Electricity Consumption= (100.000/4) = 25.000 kWh*

If however the refrigeration energy is considered to be recovery energy it is as follows:
- *Heating requirement = 100.000 kWh*
- *Cooling requirement = 0 kWh (all recovered)*
- *Total thermal requirement = 100.000 kWh*
- *SPF = 4*
- *Energy produced from RES = 75.000 kWh*
- *Share of energy produced from RES compared to total thermal requirement = 75%*
- *Amount of energy that has to be produced from other RES = 0 kWh*

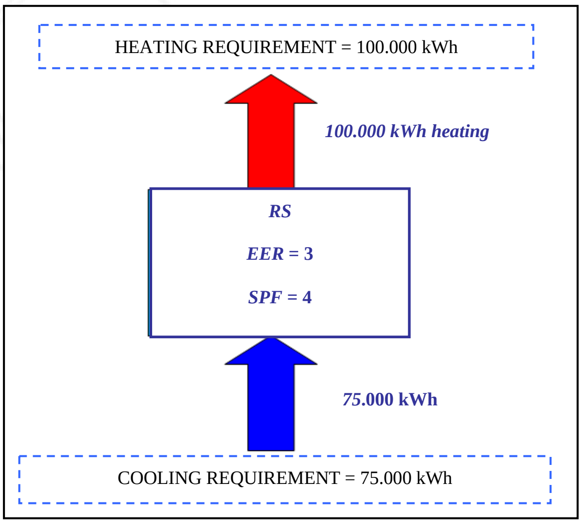


Fig. 1.9 - Energy balance of a total heat recovery system.

It clearly appears that the second solution is remarkably better than the first one, because it cuts to half the energy consumption and therefore it has to be promoted in accordance with Italian Decree 28/11. This can take place only by assuming that the refrigeration energy is the byproduct and the thermal requirement is the load to satisfy and not the opposite.

Considering heat recovery on thermal energy as a reduction of the load entails the paradox that the system that consumes an amount of electrical energy equal to half the one consumed by another system requires, unlike the first system, a more than double supplementation of the amount of energy from renewable source. Moreover, at this point, it should be made clear which energy sources can be considered to be renewable, because, as soon as there is a back-up with, for example, some Solar Cooling, the amount of thermal energy recovered by the refrigeration unit would diminish and the amount of energy from renewable sources should be further increased to comply to the Directive's requirement.
Instead, not only should the heat production be considered as the main one, but the cooling energy recovered has to be considered both a cooling requirement reduction and a renewable energy source.
In summary, AiCARR's Method is based on the assumption that also some parts of the plant should be considered as a renewable energy source, and in some operating modes they replace conventional renewable sources, for example hydraulic circuits when heating and cooling are produced simultaneously (the so called Water Loop Heat Pump systems, WLHP), or else the total heat recovery of exhaust air.

1.7 - Optimized production of refrigeration energy.

Heat pumps can be optimized towards winter operation or summer operation, for reasons that are explained in Annex F. Obviously, the optimization should be executed in winter operation if the maximum requirement is requested in this season, in summer operation if the opposite situation takes place.
In accordance with the provisions of Italian Decree 28/11, the risk is that of giving special attention to winter requirements in each and every case. AiCARR's Method starts from the assumption that the heat pumps that should be promoted are the ones that function well also when they operate as refrigerators, allowing for a summer requirement reduction $E_{R,Sum}$, equal to:

$$E_{R,Sum} = E_{HP,Sum} \frac{1}{2} \left[\frac{EER_S - (SPF_{HP} - 1)}{SPF_{HP}} \right] \tag{1.11}$$

where:
$E_{HP,Sum}$ = refrigeration energy produced by the heat pump in summer operation;
EER_S = energy efficiency ratio of the heat pump in summer operation.
Equation (1.11) has to be applied only when the summer requirement met by heat pumps is higher than the winter one and when the $E_{R,Sum}$ value turns out to be greater than 0.
If the idea to focus the evaluation on *PER* were accepted, it would be enough to multiply SPF_{HP} values by the efficiency η for electricity production and distribution to reach the correct result.

NOTA:	Equation (1.11) remains unchanged also when a calculation based on the sum of accurate results is imposed, as laid down in paragraph 2.1. Actually, the calculation of $E_{R,Sum}$ can be executed only based on the seasonal average.

Rationale behind the Equation (1.11)
The Equation was been written to promote heat pumps that have a higher energy efficiency rating in summer operation than for winter operation, in the case when the highest energy input requirement is for the cooling operation and not for the heating one.
It has to be reminded that, ceteris paribus, winter efficiency is roughly equal to summer efficiency plus one unit, because in winter operation, the compressor is involved in achieving the desired target. Therefore, for a heat pump that is optimized in summer operation, the cooling operation is as follows:

$$SPF_{HP} < EER_S + 1 \tag{1.12}$$

while for a heat pump optimized in winter operation, the heating operation mode is as follows:

$$SPF_{HP} \geq EER_S + 1 \tag{1.13}$$

which confirms that the value of Equation (1.11) is positive only if the heat pump is optimized in summer operation.
Numerical Example 11 clearly explains the advantages of this kind of approach.

NUMERICAL EXAMPLE 11

Let it be given an HVAC plant that requires 100.000 kWh in winter operation and 200.000 kWh in summer operation. The total requirement of the plant therefore is equal to 300.000 kWh.
Suppose to use two different heat pumps, one with SPF = 4 and EER_S = 3 and the other one with SPF = 3 and EER_S = 4. Also suppose that for technical obstacles that are provided for in Italian Decree 28/11, other energy renewable sources cannot be used.

In accordance with the approach by the Italian Decree, in the first case, it can be shown that:
- *Electrical energy consumption in the winter = 25.000 kWh*
- *Electrical energy consumption in the summer = 66.666 kWh*
- *Total consumption of electrical energy = 91.666 kWh*
- *Energy produced from RES = 75.000 kWh*
- *RES share over the total thermal requirement = 25% (75.000/300.000)*

And in the second case:
- *Electrical energy consumption in the winter = 33.333 kWh*
- *Electrical energy consumption in the summer = 50.000 kWh*
- *Total electrical energy consumption = 83.333 kWh (-11% compared to the first case)*
- *Energy produced from RES = 66.666 kWh*
- *RES share over the total thermal requirement = 22,2% (66.666/300.000) (-12% compared to the first case)*

The second case cheaper in primary energy consumption because it saves 11% of electrical power, but it is placed at a disadvantage by Italian Decree 28/11, because the share of energy from the renewable source that is exploited is lower by 12% (it should be higher by 11% instead).
If AiCARR's Method is used instead, in the first case, the deduction in summer requirement is equal to 0 kWh, while in the second case, it is equal to:

$$R_E = E_E \frac{1}{2}\left[\frac{EER_M - (SPF - 1)}{SPF}\right] = 200.000\,\frac{1}{2}\left[\frac{4 - (3 - 1)}{3}\right] = 66.666 \quad \text{kWh}$$

Hence, it can be shown that:
- *Summer requirement = 133.333 kWh (200.000 – 66.666)*
- *Winter requirement = 100.000 kWh*
- *Total requirement = 233.333 kWh*
- *Energy produced from RES = 66.666 kWh*
- *Share on the total = 28% (66.666/233.333) (12% more if compared to the first case).*

AiCARR's Method provides a calculation that goes along with thermodynamics and real energy savings, overcoming the critical aspect and thus rewarding the system that consumes less energy.

1.8 – Regeneration of geothermal sources

As is well known, a heat pump extracts thermal energy from the cold source and delivers it to the hot one. In winter operation, the heat source is the plant and the cold source is represented by air, ground, surface or deep water, which implies, in general, by a renewable energy source. In summer, it switches: the cold source is the plant, while the heat source is the air, the ground, etc.
A heat transfer in summer is not considered as energy from a renewable source, because it is a non-reusable thermal energy. This can be correct in the case of an air thermal source, because it is an open cycle: the heat energy that is removed from the plant is conveyed to the outdoor air.
This is not the case, though, when thermal sources are ground and water, both surface and deep. In this case, the borders of the system are not clearly identifiable: the physical mechanism that regenerates thermal energy removed from the ground during the winter is thermal conduction or a system that combines convection and conduction. This process requires a certain time scale, depending upon the operation strategy of the heat pump, the intensity of the specific heat flow from the source and the physical properties of the ground. The ground conditions tend to reach a "long term equilibrium" after a certain number of years of operation of the heat pump, usually after at least 8 years. The reduction in ground temperature values results, under identical efficiency conditions, in a lower thermodynamic efficiency of the device, and hence a lower amount of heat extracted from the ground and more heat conducted from the surrounding ground, which explains the achievement of an equilibrium. Yet, at the point of equilibrium, referred to ground temperature values, heat pump devices may not be in the best conditions to function properly, and the expression used to describe this situation is "thermal drift", which means a reduction in efficiency that sets in when temperature values start going down.
If the system is operational in the summertime as well, regeneration of the source is possible thanks to the condensation heat. A possible thermal drift with increase in ground temperature values results in an improved winter efficiency, which, in any case, has to be considered as a positive factor.
AiCARR's Method starts from the assumption that also a share of the energy that is re-supplied to the ground during the summer season should be considered as renewable energy source, and the equation is as follows:

$$E_{RES,reg} = \frac{1}{y} E_{g,Sum} = \frac{1}{y} E_{HP,Sum}\left(1 + \frac{1}{EER_S}\right) \quad \text{if} \quad E_{HP,Sum}\left(1 + \frac{1}{EER_S}\right) < E_{g,Win}$$

$$E_{RES,reg} = \frac{1}{y} E_{g,Win} \quad \text{if} \quad E_{g,Sum} > E_{g,Win} \tag{1.14}$$

where:

$E_{RES,reg}$ = energy from renewable source derived from regeneration of the source;
$E_{HP,Sum}$ = refrigeration energy produced by the geothermal heat pump;

$E_{g,Sum}$ = energy supplied to the ground in summer operation;
EER_S = average seasonal energy efficiency of the heat pump in summer operation;
$E_{g,Win}$ = energy extracted from the geothermal source in the winter period;
y = recharge coefficient: it corresponds to the number of years necessary for the ground regeneration (it can be considered equal to 10).

Rationale behind the Equation (1.14)
The Equation was conceived to calculate a fraction of the waste heat energy produced during the winter operation, during regeneration of the geothermal source.
The problem is the drop in performance of the heat pump when the long-term, sustainable new equilibrium point is reached, and the heat pump is only used for winter operation. When the heat pump is also used for summer operation, an amount of thermal energy, equivalent to the sum of the whole amount of the refrigeration energy output and the amount of energy required to run the compressor, is re-supplied to the ground source. AiCARR proposes to consider a fraction of such total amount as renewable energy source, in function of the number of years needed for regeneration.
Numerical Example 12 helps to better understand.

NUMERICAL EXAMPLE 12

Calculate the additional amount of energy output by regeneration from renewable energy sources using a geothermal heat pump that produces 100.000 kWh with SPF_{HP} = 4, assuming that in summer operation, it has EER_S = 4 and that the summer energy usage is $E_{HP,Sum}$ = 50.000 kWh in the first case and 300.000 kWh in the second case.
The energy extracted from the ground in winter operation is equal to 75.000 kWh, equivalent to the ratio of the energy produced in the winter to SPF_{HP} (100.000(1-1/4) = 75.000).
Therefore, the additional amount of energy from renewable source is as follows:
*Case 1: $E_{,Sum}$ = 62500 kWh = 50.000+50000*1/4) kWh (less than $E_{g,Win}$= 75.000 kWh)*

$$E_{RES,reg} = \frac{1}{y} E_{HP,Sum}\left(1+\frac{1}{EER_S}\right) = \frac{1}{10} 50.000\left(1+\frac{1}{4}\right) = 6.250 \quad \text{kWh}$$

*Case 2: $E_{g,Sum}$ = 375000 kWh = 300.000+300.000*1/4 kWh (greater than $E_{g,Win}$= 75.000 kWh)*

$$E_{RES,reg} = \frac{E_{g,Win}}{y} \frac{75.000}{10} = 7.500 \quad \text{kWh}$$

NOTA: Equation (1.14) remains unchanged even when a calculation based on the sum of actual results is prescribed, as indicated in paragraph 2.1. Indeed the calculation of the $E_{RES,reg}$ can be executed on seasonal averages only.

1.9 – District Heating and cogeneration/trigeneration systems fired by fossile fuels

District Heating has to be considered as a system that is capable of exploiting renewable energy sources only when connected to cogeneration and trigeneration systems. District Heating systems that are powered exclusively by boilers with heat generators should absolutely be ignored.
The rationale is always the same: only systems with low energy usage levels, that have high PER_S values, should be promoted.
In the case of cogeneration and trigeneration systems, it is even more important to always refer to the concept primary energy. This is the approach on high efficiency cogeneration that was adopted in the financial support scheme introduced by an Italian Decree issued on 5th September 2011 by the Department for Economic Development.
The effect of cogeneration and trigeneration systems has to be considered as follows:
1) when these systems are connected to electrically driven heat pumps, the heat energy share that is recovered to produce the electrical power to be used for the operation of the heat pumps themselves (refer to paragraph 2.3.1) has to be treated under the same conditions that have been applied to endothermic engine heat pumps;
2) in cogeneration systems without heat pumps and in other systems, which have a remaining share of the electrical power used by other devices, different from electrically driven heat pumps, the heat recovery of the engine has to be considered as a heat load reduction;
3) in trigeneration systems, the production of refrigeration energy has to be considered as an energy usage reduction for the summer cooling operation.

Numerical Example 13 helps to better understand the three points that have just been listed.

NUMERICAL EXAMPLE 13

A trigeneration system has a seasonal average electrical efficiency of 30% and a seasonal average thermal efficiency of 50%. The absorption refrigeration unit used in the summer has COP_S = 0,7.
The system is in service in a building with an energy usage of 1.100.000 thermal kWh in winter operation and of 1.000.000 thermal kWh in summer operation.
The system has an output of 300.000 electrical kWh in winter operation and of other 300.000 electrical kWh in summer operation.
Coverage of winter energy usage is guaranteed:

- *by a heat pump with SPF_{HP} = 4 (electricity consumption 150.000 kWh) for 600.000 kWh;*
- *by the trigeneration system for 500.000 kWh.*

Therefore in winter operation the trigeneration system produces 300.000 electrical kWh, of which 150.000 kWh are used to operate the heat pump.
In winter, the heat energy output obtained from the heat recovery of the engine is equal to 500.000 kWh (value that is obtained by dividing the value of 300.000 electrical kWh by the electrical efficiency - equal to 0,3 - and multiplying the result by the heat efficiency - equal to 0,5), of which 250.000 kWh are produced as heat recovery from the electrical energy output that is used to operate the heat pump.
In summary, the situation is as follows:

Winter operation – point 1
The system behaves like an endothermic engine heat pump, for the share of electrical power output that is equivalent to the energy usage needed to operate a heat pump, that functions with PER_S = 1,7 (sum of the product of the electrical efficiency's value multiplied by SPF_{HP} plus the thermal efficiency value (0,3 · 4 + 0,5 = 1,7), as shown in Figure 1.3 on page 13. Accordingly, the result obtained from Equation (1.6) is $SPF_{C,Syst}$ = 1,7/0,4 = 4,25 and the share of energy from renewable sources is equal to 76,4%.
Therefore, the renewable energy share that is produced by the heat pump – trigeneration system is equal to 650.000 kWh; as a matter of fact, 600.000 kWh are produced by the heat pump, 250.000 kWh are recovered as waste heat recovered from the engine, and their sum, equal to 850.000, has to be multiplied by the share of energy from renewable sources calculated based on Equation (1.6), that is equal to 0,764.

Winter operation – point 2
Heat recovery from the engine in winter operation is equal to 500.000 kW, of which 250.000 kWh are calculated as share of the heat pump - trigeneration system, while the remaining 250.000 kWh have to be considered as heat requirement reduction.

Summer operation – point 3
In summer, the thermal energy output from the waste heat recovered from the engine is equal to 500.000 thermal kWh that are conveyed to an absorption refrigeration unit and that are used to produce 350.000 refrigeration kWh, that have to be considered as a summer heat requirement reduction.

Final summary:
- *Winter thermal requirement: 850.000 kWh, given by the difference between the thermal requirement and the heat recovered from the engine, equal to 1.100.000 – 250.000, as in point 2*
- *Summer thermal requirement: 650.000 kWh, given by the difference between the thermal requirement and the output of the absorption refrigeration unit fired by the waste heat recovered from the engine, as in point 3*
- *Total thermal requirement: 1.500.000 kWh, equal to the sum of the summer plus winter requirements*
- *$PER_{S,Syst}$= 1,7*
- *$SPF_{C,Syst}$= 4,25*
- *Energy produced from renewable sources in accordance with (1.6) = 650.000 kWh*
- *Renewable energy share on the total requirement: 43,3%*
- *Additional energy to be produced with renewable sources in the summer: 100.000 kWh.*

1.10 - Solar Thermal Energy

According to AiCARR's Method, also in the case of thermal solar energy, the energy output for domestic hot water and for heating and cooling is based on *SPF*, considering the electrical power needed for auxiliary devices:

$$E_{RES,Sol} = E_{Sol}\left(1 - \frac{1}{SPF_{Sol}}\right) = E_{Sol}\left(1 - \frac{EE_{abs,Sol}}{E_{Sol}}\right) \qquad (1.15)$$

where:

$E_{RES,\ Sol}$	=	energy from renewable sources;
E_{Sol}	=	energy produced by thermal solar system;
$EE_{abs.Sol}$	=	electrical energy input for the operation of system auxiliaries.

Rationale behind the Equation (1.15)
AiCARR's Method puts the thermal solar system on the same ground with a heat pump. The Seasonal Performance Factor, SPF_{Sol}, is equal to the ratio of the value of the energy amount produced by the thermal solar system to the value of the amount of the electrical energy needed to the operation of auxiliary devices, as shown in Figure 1.10.

NUMERICAL EXAMPLE 14

Let it be given two domestic hot water production systems that use a solar thermal system and compare them: one natural circulation system, that is capable of producing 10.000 kWh on a yearly basis, and the other forced circulation system, that is capable of producing 12.000 kWh in thermal energy that uses an electrical energy of 1.000 kWh for the pumping operations.
In the first case thus:

$$SPF_{Sol} = \frac{E_{Sol}}{EE_{abs,Sol}} = \frac{10.000}{0} = \infty$$

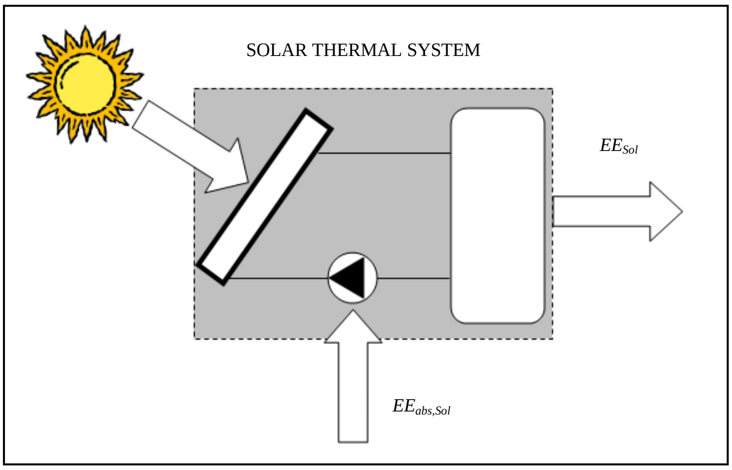


Fig. 1.10 - Balance on a thermal solar system in accordance with Equation (1.15).

where the whole amount of the energy output has to be considered as renewable source energy.
In the second case, conversely, the following formula appears:

$$SPF_{Sol} = \frac{E_{Sol}}{EE_{abs,Sol}} = \frac{12.000}{1.000} = 12$$

The amount of energy from renewable sources is as follows:

$$E_{RES,Sol} = E_{Sol}\left(1 - \frac{1}{SPF_{Sol}}\right) = 12.000\left(1 - \frac{1}{12}\right) = 11.000 \quad \text{kWh}$$

In the case of Solar Cooling, that is the usage of the thermal energy output obtained by the solar thermal system for the production of refrigeration energy through absorption type refrigeration systems or through chemical de-humidifiers, the SPF_{Sol} factor contained in Equation (1.15) has to be changed following the equation:

$$SPF_{Sol,cool} = \frac{E_{RS}}{EE_{abs,Sol} + EE_{abs,RS}} = \frac{\varepsilon_{RS} E_{Sol}}{EE_{abs,Sol} + EE_{abs,RS}} \tag{1.16}$$

where:

E_{GF} = refrigeration energy produced by absorption refrigeration systems or chemical dehumidifiers;
ε_{GF} = efficiency of absorption refrigeration systems or chemical dehumidifiers;
$EE_{abs,RS}$ = electrical energy consumed for the operation of the absorption refrigeration systems or chemical dehumidifiers and their auxiliary devices, for example evaporative cooling towers, as shown in Figure 1.11.

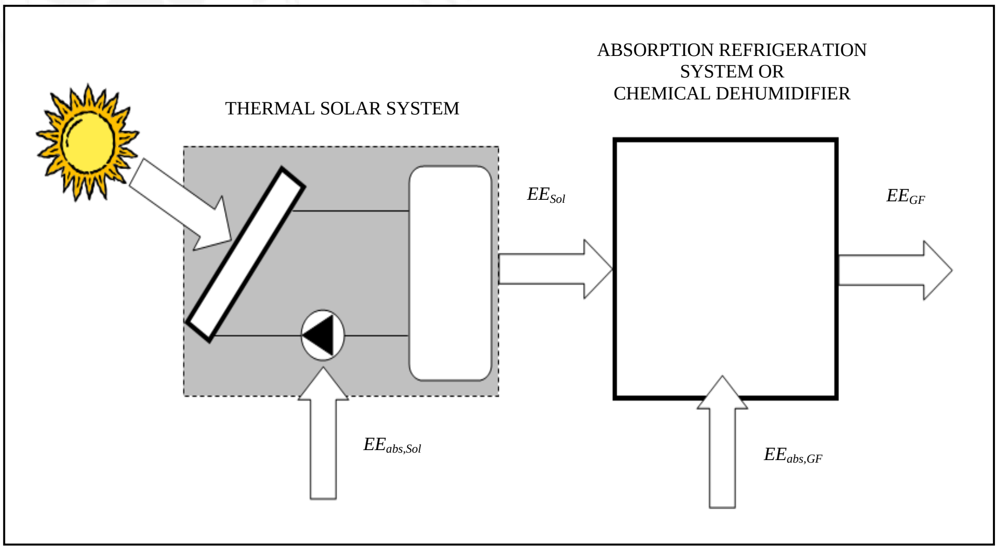


Fig. 1.11 - Balance on a thermal solar system in accordance with Equation (1.15).

The amount of energy from renewable sources that is exploited for the cooling becomes:

$$E_{RES,Sol} = \varepsilon_{RS} E_{Sol} \left(1 - \frac{1}{SPF_{Sol}}\right) \tag{1.17}$$

Rationale behind the Equation (1.17)

The Equation considers that the thermal energy produced by the thermal solar system is converted to refrigeration energy in amounts that are proportional to the efficiency of the absorption refrigeration system, ε_{GF}, or of the chemical dehumidifier; this value may also be higher than 1, for example in the case of double-effect absorption refrigeration systems.

Nevertheless, not all of the refrigeration energy that is produced by the thermal solar system can be considered as being renewable, because the system anyhow requires electrical energy to work, in particular when circulation pumps or fans for evaporative cooling towers are involved.

NUMERICAL EXAMPLE 15

In the summer season, a solar thermal system produces an amount of thermal energy of 1.000.000 kWh and uses 50.000 kWh of electrical energy for the pumping operation. The thermal energy is conveyed to an absorption refrigeration system which has an efficiency of 0,7 and an electric consumption for auxiliary devices (evaporative cooling tower and relative pumping system) equal to 80.000 kWh.

The $SPF_{Sol,Cool}$ value is as follows:

$$SPF_{Sol,Cool} = \frac{E_{RS}}{EE_{abs,Sol} + EE_{abs,RS}} = \frac{\varepsilon_{RS} E_{abs,Sol}}{EE_{abs,Sol} + EE_{abs,RS}} = \frac{0{,}7 \cdot 1.000.000}{50.000 + 80.000} = \frac{700.000}{130.000} = 5{,}38$$

The amount of refrigeration energy from renewable sources produced by Solar Cooling is:

$$E_{RES,Sol} = \varepsilon_{RS} E_{Sol} \left(1 - \frac{1}{SPF_{Sol}}\right) = 700.000\left(1 - \frac{1}{5{,}38}\right) = 569.888 \text{ kWh}$$

Thermal solar systems can also be used as a thermal source to supplement heat pumps in raise the heat level, as outlined in Figure 1.12. In this case, the heat energy produced by the thermal solar system, E_{Sol}, should not be considered as a renewable energy source, because it is only involved in raising the SPF_{HP} value of the heat pump. Yet, the electrical energy usage, $EE_{abs,Sol}$, that has to be added on top of the one of the heat pump and its auxiliary devices, should be allowed for.

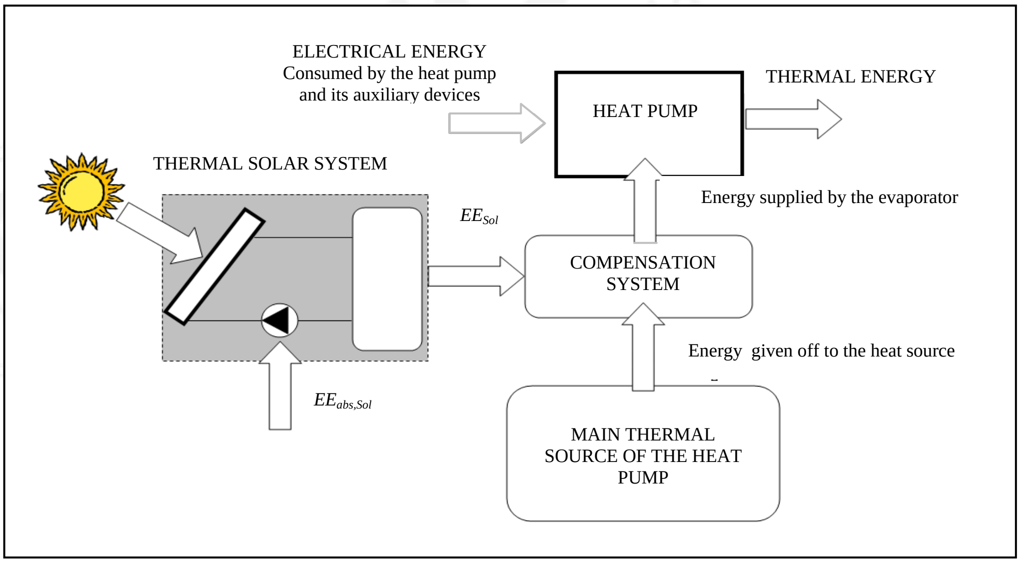


Fig. 1.12 - Balance on a heat pump run on solar power.

NUMERICAL EXAMPLE 16

A ground source heat pump supplies an amount of thermal energy of 100.000 kWh with an electric energy usage of 25.000 kWh, of which 20.000 kWh for the compressors and 5.000 kWh for the auxiliary devices. The SPF_{HP} value is equal to 4,00.

It is decided to supplement the thermal source with a solar thermal system, as is depicted in Figure 1.12. This option allows to lower the amount of electricity used for the compressors to 16.000 kWh, even though the total value is thereby increased by 2.000 kWh, necessary for the pumping device of the solar thermal system.

The amount of thermal energy that is supplied by the heat pump remains unchanged, while the value of the electricity usage drops down to 23.000 kWh (16.000 for the compressors + 5.000 for auxiliary devices + 2.000 for the solar system). As a consequence, the SPF_{Syst} goes up and reaches a value of 4,35 (100.000/23.000). The

amount of thermal energy that is produced by using renewable energy sources in consequence goes up too, in accordance with Equations (1.1) or (1.6), depending upon the selected calculation approach.

In short, the solar thermal system can be used for regeneration of a geothermal source of a heat pump in summer. This case has to be approached based on criteria that are similar to the ones laid down in paragraph 1.7, bearing in mind that the electrical energy usage of the solar thermal system $EE_{abs,Sol}$ has to be added up. Therefore, the following formula appears:

$$E_{RES,reg,Sol} = \frac{1}{y} E_{Sol} \left(1 - \frac{1}{SPF_{Sol}}\right) = \frac{1}{y} E_{Sol} \left(1 - \frac{EE_{abs,Sol}}{E_{Sol}}\right) \tag{1.18}$$

where:

$E_{RES,reg,Sol}$	=	energy from renewable source originating from regeneration of a geothermal source through thermal solar system;
E_{Sol}	=	energy supplied to the source by the thermal solar system;
SPF_{Sol}	=	*SPF* of the thermal solar system;
$EE_{abs,Sol}$	=	electrical energy used by the thermal solar system during regeneration of the ground;
y	=	recharge coefficient: it corresponds to the number of years necessary for the regeneration of the ground (it can be considered equal to 10).

Numerical Example 17 helps to better understand.

NUMERICAL EXAMPLE 17

Calculate the additional amount of energy produced from renewable energy sources by regeneration of a solar thermal system that produces 50.000 kWh in the summer and has an electrical energy usage of 4.000 kWh. If Equation (15) is applied, the following formula is obtained:

$$E_{RES,reg,Sol} = \frac{1}{y} E_{Sol} \left(1 - \frac{EE_{abs,Sol}}{E_{Sol}}\right) = \frac{1}{10} 50.000 \left(1 - \frac{4.000}{50.000}\right) = 4.600 \quad \text{kWh}$$

NOTA: Equations (1.15), (1.16) and (1.18) can remain unchanged even when a calculation based on the sum of real results is prescribed, as indicated in paragraph 2.1. As a matter of fact, the mistakes made because of the seasonal average are slight. The mistake becomes relevant only when the thermal solar system is used as supplementation in increasing the temperature value of the cold source of a heat pump.

1.11 - Energy usage for the pumping of vector fluids

Whatever method is selected, in the calculation of the *SPF* rating, the energy usage for vector fluids, be it air or water, always has to be taken into account, because their energy weight influences the seasonal balance. This is another way to promote the systems that have lower total/primary energy usages.

1.12 - Biofuel and biomass

Italian Decree 28/11 considers that biomass energy systems are a renewable energy source. A clear distinction should be made between the biomass that is extracted from processing waste and the biomass that is derived from the remains of forests or directly from crops, because the transportation factor has to be allowed for. As a matter of fact, one thing is using wood chips derived from the remains of forests on the spot or in nearby areas, but their transportation over long distances gives a completely different picture. In the first case, biomass can be considered almost a 100% renewable energy source, while, in the second case, the energy used for transportation should be calculated by means of a reduction coefficient and be deducted from the amount of renewable source energy that has been produced.

Direct agricultural crops, tough, can never be considered to be 100% renewable, because some energy is used both to produce fertilizers, and to power other machines (tractors, attachments, etc.) and also for pumping water and for irrigation water. A reduction coefficient that allows for these energy usages should be introduced. Only the less energy-consuming crops would be promoted in this way.

The problem is very serious, because an indiscriminate use of biomass to produce energy, prompted by speculation, can result in taking away potential farmland from food production, both for man and animals, with the consequence that a great national wealth is destroyed, that our country is obliged to become dependent on foreign markets for food supplies, that at present are all produced in Italy, and face a huge increase in agricultural produce prices.

Situations such as these have to be avoided thanks to appropriate policies.

It has to be highlighted that the expected and progressive introduction of biomethane in the distribution network that is obtained from wastes, animal excrements and vegetal and animal origin waste can give a relevant contribution to the use of renewable energy sources in winter operation of systems and potentially also in summer operation if biogas is used in a thermally activated refrigeration system. It would be advisable to consider the possibility that the amount of biomethane used can be considered as a renewable energy contribution produced by a system, as long as the supply is tied up to an appropriate contract of multiannual supplies.

PART 2

SUGGESTIONS TO IMPROVE THE ACCURACY OF CALCULATIONS

2.1 - Foreword

One of the most evident shortcomings of the European Directive and, in consequence, of Italian Decree 28/11, is that these two legislative acts appear to refer only to the residential sector, in particular single-family residential houses.
When addressing more complex situations, very important factors come into play, such as the management strategies of the whole system of energy production. One example is about shopping centers, where more systems could be adopted, both the conventional ones and the ones that are capable of exploiting renewable energy sources, from Free-Cooling to heat pumps, from Solar Cooling to sophisticated forms of thermal storage of refrigerated water. In these cases, it is absolutely necessary to better define the results that are entailed by design options, by making more accurate energy assessments, based on 1-hour time intervals. Another example, referred to the residential sector, is the one of apartment blocks, where thermal storage could be used to concentrate the whole production of domestic hot water on daytime, when air temperature values are higher, and improve the production efficiency of air source heat pump systems.
In summary, if a designer is supposed to optimize the performance of a given system, the prescription that detailed calculations should be carried out is legitimate and necessary.

2.2 - Seasonal average values and calculated values

In the first part, it was repeatedly stressed that calculations based on seasonal averages can lead to serious mistakes. This is true for heat pumps, but also for the heat recovery from exhaust air, for Free-Cooling and for the heat recovery from the condensation of refrigeration systems.

2.2.1 - Shortcomings of seasonal averages

As was discussed in paragraph 1.1.1.2, the application of Equation (1.1) adopted by Italian Decree 28/11 for the assessment of energy from renewable source:

$$E_{RES} = E_{HP}\left(1 - \frac{1}{SPF_{HP}}\right) \tag{1.1}$$

implies mistakes because, since it is based on an average SPF_{HP} value, it simultaneously considers very different situations, also comparing them, where sometimes renewable energy sources are involved and other times where they are not. As a matter of fact, the average SPF_{HP} value is calculated by way of the ratio of the thermal energy that is produced to the electrical energy that is used by the heat pump all year round, without any distinction between the different operating conditions.
All equations suggested by AiCARR in the first part, as specified in the note at the beginning of paragraph 1.3, follow the approach of the Italian Decree, and are therefore referred to the average SPF_{HP} value. This choice was made to create clarity, meaning that the critical aspects deriving from the implementation of the Italian Decree have both a conceptual nature and also they are due to the seasonal average approach: both concepts derive from the European Directive. In the first part, the inaccuracies in some concepts have been underlined, while the second part of the document addresses the mistakes made because of the approach on the seasonal average, mistakes that involve the entire range of situations described in the first part, with a few exceptions that have been identified in each individual paragraph.

2.2.2 - The correct calculation based on detailed values

As a matter of fact, for the system as a whole, the renewable energy share should be calculated hour by hour, using the following equation in replacement of Equation (1.6):

$$E_{RES} = \sum_h E_{RES,h} = \sum_h \left[E_{Tot,h}\left(1 - \frac{1}{SPF_{C,Syst,h}}\right)\right] \tag{2.1}$$

where:
$E_{RES,h} = 0$ if $SPF_{C,Syst,h} < SPF_{Min}$
with:
$SPF_{C,Syst,h}$ = conventional *SPF* of the whole system at hour *h*; it corresponds to the accurate *COP* of the heat pump in the case when the heat pump is the only generator.

If the production and distribution efficiency of the electrical energy is considered to be equal to 0,4, SPF_{Min} has to be equivalent to 2,875, because this is the condition where the accurate *PER* value is equal to 1,15.

Rationale behind the Equation (2.1)
The Equation is similar to Equation (1.6), with the only difference that the calculation is executed for each single hour of operation, discarding the conditions where *PER* is too low. In this way, the energy output that has been obtained without renewable energy sources is not allowed for in the calculation.
Numerical Example 18 will help readers better understand the above statement.

NUMERICAL EXAMPLE 18

Based on Numerical Examples 4 and 6, calculate the amount of energy produced from renewable sources with Equation (2.1) and compare the results with the ones obtained respectively with Equations (1.1) and (1.6).
Table 2.1 illustrates the results of the detailed calculation on a case by case basis executed with Equation (2.1), while Table 2.2 compares the results obtained by: the method contained in the Italian Decree, AiCARR's Method, and AiCARR's Method with detailed calculations.

Tab. 2.1 - Results of the detailed calculation

Temperature	**Case 1**		**Case 2**		**Case 3**		**Case 4**	
	PER	E_{RES} **(2.1)**	***PER***	E_{RES} **(2.1)**	***PER***	E_{RES} **(2.1)**	***PER***	E_{RES} **(2.1)**
-5	0,55	0	0,87	0	1,05	0	0,74	0
-4	0,57	0	0,87	0	1,05	0	0,76	0
-3	0,60	0	0,87	0	1,05	0	0,79	0
-2	0,64	0	0,90	0	1,05	0	0,83	0
-1	0,70	0	0,94	0	1,05	0	0,90	0
0	0,79	0	1,00	0	1,05	0	0,99	0
1	0,92	0	1,07	0	1,05	0	1,07	0
2	1,08	0	1,13	0	1,13	0	1,14	0
3	1,19	16.096	1,19	16.096	1,192	16.096	1,192	16.096
4	1,24	16.094	1,24	16.094	1,24	16.094	1,24	16.094
5	1,28	15.589	1,28	15.589	1,28	15.589	1,28	15.589
6	1,32	13.668	1,32	13.668	1,32	13.668	1,32	13.668
7	1,35	12.597	1,35	12.597	1,35	12.597	1,35	12.597
8	1,38	11.510	1,38	11.510	1,38	11.510	1,38	11.510
9	1,40	10.064	1,40	10.064	1,40	10.064	1,40	10.064
10	1,42	9.006	1,42	9.006	1,42	9.006	1,42	9.006
11	1,43	7.289	1,43	7.289	1,43	7.289	1,43	7.289
12	1,44	6.090	1,44	6.090	1,44	6.090	1,44	6.090
13	1,44	4.726	1,44	4.726	1,44	4.726	1,44	4.726
14	1,43	3.308	1,43	3.308	1,43	3.308	1,43	3.308
15	1,43	2.342	1,43	2.342	1,43	2.342	1,43	2.342
16	1,42	1.622	1,42	1.622	1,42	1.622	1,42	1.622
17	1,40	1.082	1,40	1.082	1,40	1.082	1,40	1.082
18	1,38	625	1,38	625	1,38	625	1,38	625
19	1,35	313	1,35	313	1,35	313	1,35	313
20	0.00	0	0.00	0	0.00	0	1,32	0
		132.020		**132.020**		**132.020**		**132.020**

Tab. 2.2 - Results comparing the different formulas

	Case 1	**Case 2**	**Case 3**	**Case 4**
Energy produced from renewable sources calculated with (1.1) [kWh]	193.657	193.657	161.349	0
Share on total	60,9%	60,9%	50,7%	0,0%
Energy produced from renewable sources calculated with (1.6) [kWh]	193.657	208.812	212.716	206.580
Share on total	60,9%	65,6%	66,8%	64,9%
Energy produced from renewable sources calculated with (2.1) [kWh]	132.020	132.020	132.020	132.020
Share on total	41,5%	41,5%	41,5%	41,5%

The results of Numerical Example 18 reported in Table 2.2 show once again what the problems involved in the use of Equation (1.1) are like: all the four cases provide the same renewable energy share. If the calculation is executed with the equation indicated in the Italian Decree, the results are different.
A further important aspect is that the share of energy produced from renewable sources obtained with Equation (2.1) is much lower than the share that would be obtained using Equation (1.1). This is due to what is accounted for in paragraph 1.2.1.2.2: the heat pump at the lower end of the efficiency range was used on purpose to demonstrate that the Italian Decree promotes mediocrity and not excellence.
Numerical Example 19 shows that designers can play a role to improve performance in heat pump systems.

NUMERICAL EXAMPLE 19

Start from case 3 of Numerical Example 18: heat pump backed up by a condensing boiler, that completely stops working when COP is lower than 2,6, or rather when PER drops below the level of the condensing boiler's PER value. If the objective is reaching a share of energy produced from renewable sources that exceeds 60 %, in accordance with Equation (2.1), the efficiency of the heat pump has to be enhanced by 20% and its capacity has to go up from 72 kW to 105 kW, when the outside air temperature value is equal to -5 °C. In this case COP = 1,15 is reached as soon as outside temperature values reach -0 °C.
Table 2.3 reports the comparison between what happens in this Numerical Example and what happens in Numerical Example 18: thanks to these expedients, the amount of energy produced from renewable source goes up to 201.259 kWh, equal to 63% of the overall energy usage.

Tab. 2.3 - Comparison between the results of case 3 before and after efficiency enhancement measures

	Case 3: previous version			**Case 3: enhanced version**				
	COP	***PER***	**E_{RES} [kWh]**	***COP***	***PER***	**Energy from Boiler [kWh]**	**Energy from Heat Pump [kWh]**	**E_{RES} [kWh]**
-5	1,86	1,05	0	2,24	1,05	4.375	0	0
-4	1,89	1,05	0	2,28	1,05	7.200	0	0
-3	1,97	1,05	0	2,37	1,05	9.775	0	0
-2	2,09	1,05	0	2,51	1,05	15.950	0	0
-1	2,25	1,05	0	2,72	1,09	0	20.370	12.870
0	2,48	1,05	0	2,99	1,19	0	22.400	14.902
1	2,68	1,05	0	3,23	1,29	0	24.795	17.110
2	2,84	1,13	0	3,42	1,37	0	24.390	17.259
3	2,98	1,19	16.096	3,59	1,44	0	24.225	17.479
4	3,1	1,24	16.094	3,73	1,49	0	23.760	17.398
5	3,21	1,28	15.589	3,87	1,55	0	22.650	16.790
6	3,3	1,32	13.668	3,98	1,59	0	19.600	14.677
7	3,39	1,35	12.597	4,08	1,63	0	17.875	13.495
8	3,45	1,38	11.510	4,16	1,67	0	16.200	12.308
9	3,51	1,4	10.064	4,22	1,69	0	14.080	10.747
10	3,54	1,42	9.006	4,27	1,71	0	12.550	9.609
11	3,57	1,43	7.289	4,30	1,72	0	10.125	7.771
12	3,59	1,44	6.090	4,33	1,73	0	8.440	6.490
13	3,6	1,44	4.726	4,34	1,73	0	6.545	5.035
14	3,58	1,43	3.308	4,31	1,73	0	4,590	3.526
15	3,58	1,43	2.342	4,31	1,72	0	3.250	2.496
16	3,54	1,42	1.622	4,27	1,71	0	2.260	1.730
17	3,5	1,4	1.082	4,22	1,69	0	1.515	1.156
18	3,45	1,38	625	4,15	1,66	0	880	668
19	3,38	1,35	313	4,07	1,63	0	445	336
20	3,31	0	0	3,99	1,60	0	0	0
			132.020 41,5%					**203.853 64,1%**

Numerical Examples 18 and 19 show that only detailed calculations are accurate and enable designers to make the right choice to reach the target results.
Obviously the same considerations can be true also for all the other systems (Free-Cooling, heat recovery and others): the calculation of the different *SPF* values should always be executed on an hour by hour basis, excluding the conditions where PER_h < 1,15.

2.2.2.1 - The disadvantage for PER_h <1

The calculation method adopted in the previous paragraph enables to determine the accurate energy amount produced from renewable sources by each individual system. However, this method is not capable of identifying how energy is produced below the minimum level fixed for PER_h = 1,15. As a matter of fact, if it is considered that a condensing boiler of a fairly good quality has PER_h= 1, it is evident that if the value drops below the minimum level, it means that energy is produced in an inefficient manner.
Therefore, AiCARR suggests that in the cases when PER_h< 1, a value equal to the following equation should be deducted from the amount of renewable energy calculated with Equation (2.1) a value should be deducted equal to:

$$E_{REs,dtr} = \sum_h E_{RES,dtr-h} = \sum_h E_{Tot,h}\left(1 - PER_h\right) \qquad (2.2)$$

Thanks to this correction, the total energy produced from renewable sources is thus equal to:

$$E_{RES,Tot} = E_{RES} - E_{RES,dtr} \qquad (2.3)$$

where E_{RES} is calculated in accordance with Equation (2.1).
In this way, the designer is prompted to use heat pumps in an efficient way, avoiding to push them to extreme performances, and supplementing them with systems that have the same high rating of efficiency. Moreover, this measure hinders the excessive use of supplementation with electrical resistance that is very cheap, but extremely harmful.
Numerical Example 20 shows an application.

NUMERICAL EXAMPLE 20

Apply Equations (2.2) and (2.3) to the cases of Numerical Example 18. Table 2.4 shows the results.

Tab. 2.4 – Disadvantage created for PER_h < 1

T [°C]	**Energy produced [kWh]**	**Case 1**		**Case 2**		**Case 3**		**Case 4**	
		E_{RES} [kWh]	**$E_{RES,dtr}$ [kWh]**	**E_{RES} [kWh]**	**$E_{RES,dtr}$ [kWh]**	**E_{RES} [kWh]**	**$E_{RES,dtr}$ [kWh]**	**E_{RES} [kWh]**	**$E_{RES,dtr}$ [kWh]**
-5	4.375	0	1.988	0	560	0	0	0	1.127
-4	7.200	0	3.127	0	949	0	0	0	1.748
-3	9.775	0	3.955	0	1.223	0	0	0	2.077
-2	15.950	0	5.768	0	1.664	0	0	0	2.636
-1	20.370	0	6.091	0	1.316	0	0	0	2.006
0	22.400	0	4.606	0	0	0	0	0	186
1	24.795	0	2.095	0	0	0	0	0	0
2	24.390	0	0	0	0	0	0	0	0
3	24.225	16.096	0	16.096	0	16.096	0	16.096	0
4	23.760	16.094	0	16.094	0	16.094	0	16.094	0
5	22.650	15.589	0	15.589	0	15.589	0	15.589	0
6	19.600	13.668	0	13.668	0	13.668	0	13.668	0
7	17.875	12.597	0	12.597	0	12.597	0	12.597	0
8	16.200	11.510	0	11.510	0	11.510	0	11.510	0
9	14.080	10.064	0	10.064	0	10.064	0	10.064	0
10	12.550	9.006	0	9.006	0	9.006	0	9.006	0
11	10.125	7.289	0	7.289	0	7.289	0	7.289	0
12	8.440	6.090	0	6.090	0	6.090	0	6.090	0
13	6.545	4.726	0	4.726	0	4.726	0	4.726	0
14	4.590	3.308	0	3.308	0	3.308	0	3.308	0
15	3.250	2.342	0	2.342	0	2.342	0	2.342	0
16	2.260	1.622	0	1.622	0	1.622	0	1.622	0
17	1.515	1.082	0	1.082	0	1.082	0	1.082	0
18	880	625	0	625	0	625	0	625	0
19	445	313	0	313	0	313	0	313	0
20	0	0	0	0	0	0	0	0	0
Totals [kWh]	**318.245**	**132.020**	**27.629**	**132.020**	**5.712**	**132.020**	**0**	**132.020**	**9.780**
$E_{RES,Tot}$ [kWh]		**104.391**		**126.309**		**132.020**		**122.241**	
Share on total		**32,8%**		**39,7%**		**41,5%**		**38,4%**	

2.2.2.2 - How to overcome the flattening of the renewable energy share for high *SPF* values

The flattening of the share of energy produced from renewable sources in the case of high SPF_{HP} values is reported in paragraph 1.2.1.1 and is one of the main critical aspects present in the Italian Decree, because mediocrity is promoted, and excellence is not promoted. Based on what has been discussed until now, it is evident that this critical aspect can be overcome adopting the precise calculation method and at the same time, taking steps as suggested to place at a disadvantage the systems that operate with *PER* < 1.
Numerical Example 21 clarifies this concept.

NUMERICAL EXAMPLE 21

Start from case 4 of Numerical Examples 4, 6 and 18 and replace the heat pump SPF_{HP} = 2,86, first with a heat pump SPF_{HP} = 3 and later with another heat pump SPF_{HP} = 5. For these three different cases, Table 2.5 reports the amount of energy obtained from renewable sources produced in accordance with the Italian Decree, using Equation (1.1), and in accordance with AiCARR's Method that is based on detailed calculations and that introduces a disadvantage for systems operating with PER <1, using Equation (2.3).

Tab. 2.5 - Comparison between the Method of the Italian Decree and AiCARR's Method with detailed calculations and disadvantage for PER < 1 systems

	Italian Decree 28/11 Equation (1.1)		AiCARR's Method with correct detailed calculation - Equation (2.3)	
	E_{RES} [kWh]	% on total	E_{RES} [kWh]	% on total
SPF_{HP}= 2,86	0	0,0%	122.241	38,4%
SPF_{HP}= 3	212.163	66,7%	143.636	45,1%
SPF_{HP}= 5	254.596	80,0%	254.596	80,0%

It should be noted that, with AiCARR's Method, from SPF_{HP} = 3 to SPF_{HP} = 5 there is a 77% improvement in the share of energy that is produced by the heat pump from renewable sources, and excellence is promoted. Conversely, as already reported in paragraph 1.2.1.1, with Equation (1.1), there only is a 20% improvement, which means that mediocrity is promoted. This result is obtained because the detailed calculation does not allow for all conditions where PER < 1,15 and no energy is produced from renewable sources, and places at a disadvantage all conditions where PER <1, as shown in Table 2.6.

Tab. 2.6 - Details of the calculations reported in Table 2.5

T [°C]	Energy produced [kWh]	SPF_{HP} = 2,86				SPF_{HP} = 3				SPF_{HP} = 5			
		PER	$SPF_{C, Syst,h}$	E_{RES} [kWh]	$E_{RES\ dtr}$ [kWh]	*PER*	$SPF_{C,Syst,h}$	E_{RES} [kWh]	$E_{RES\ dtr}$ [kWh]	PER	$SPF_{C\ Syst\ h}$	E_{RES} [kWh]	$E_{RES\ dtr}$ [kWh]
-5	4.375	0,74	1,86	0	1.127	0,78	1,95	0	968	1,30	3,24	3.027	0
-4	7.200	0,76	1,89	0	1.748	0,79	1,99	0	1.481	1,32	3,31	5.024	0
-3	9.775	0,79	1,97	0	2.077	0,83	2,07	0	1.700	1,38	3,44	6.941	0
-2	15.950	0,83	2,09	0	2.636	0,88	2,19	0	1.984	1,46	3,65	11.588	0
-1	20.370	0,90	2,25	0	2.006	0,95	2,36	0	1.107	1,58	3,94	15.214	0
0	22.400	0,99	2,48	0	186	1,04	2,60	0	0	1,73	4,33	17.247	0
1	24.795	1,07	2,68	0	0	1,12	2,81	0	0	1,87	4,68	19.515	0
2	24.390	1,14	2,84	0	0	1,19	2,98	16.198	0	1,98	4,96	19.492	0
3	24.225	1,19	2,98	16.096	0	1,25	3,13	16.476	0	2,08	5,21	19.592	0
4	23.760	1,24	3,10	16.094	0	1,30	3,25	16.452	0	2,17	5,42	19.392	0
5	22.650	1,28	3,21	15.589	0	1,35	3,36	15.919	0	2,24	5,61	18.628	0
6	19.600	1,32	3,30	13.668	0	1,39	3,47	13.945	0	2,31	5,78	16.221	0
7	17.875	1,35	3,39	12.597	0	1,42	3,55	12.843	0	2,37	5,92	14.869	0
8	16.200	1,38	3,45	11.510	0	1,45	3,62	11.729	0	2,42	6,04	13.529	0
9	14.080	1,40	3,51	10.064	0	1,47	3,68	10.252	0	2,45	6,13	11.793	0
10	12.550	1,42	3,54	9.006	0	1,49	3,71	9.171	0	2,48	6,19	10.532	0
11	10.125	1,43	3,57	7.289	0	1,50	3,74	7.421	0	2,50	6,24	8.510	0
12	8.440	1,44	3,59	6.090	0	1,51	3,77	6.199	0	2,51	6,28	7.102	0
13	6.545	1,44	3,60	4.726	0	1,51	3,77	4.811	0	2,52	6,29	5.509	0
14	4.590	1,43	3,58	3.308	0	1,50	3,76	3.368	0	2,50	6,26	3.860	0
15	3.250	1,43	3,58	2.342	0	1,50	3,75	2.384	0	2,50	6,25	2.733	0
16	2.260	1,42	3,54	1.622	0	1,49	3,72	1.652	0	2,48	6,19	1.897	0
17	1.515	1,40	3,50	1.082	0	1,47	3,67	1.102	0	2,45	6,12	1.268	0

(%)

Tab. 2.6 -

T [°C]	Energy produced [kWh]	SPF_{HP} = 2,86				SPF_{HP} = 3				SPF_{HP} = 5			
		PER	$SPF_{C, Syst,h}$	E_{RES} [kWh]	$E_{RES\ dtr}$ [kWh]	*PER*	$SPF_{C,Syst,h}$	E_{RES} [kWh]	$E_{RES\ dtr}$ [kWh]	PER	$SPF_{C\ Syst\ h}$	E_{RES} [kWh]	$E_{RES\ dtr}$ [kWh]
18	880	1,38	3,45	625	0	1,45	3,62	637	0	2,41	6,03	735	0
19	445	1,35	3,38	313	0	1,42	3,54	319	0	2,36	5,91	370	0
20	0	1,32	3,31	0	0	1,39		0	0	2,31	5,79	0	0
	318.245			**122.241**				**143.636**				**254.596**	

2.3 - Problems related to the calculation of energy requirements

The greater the size and complexity of buildings, the stronger the need to perform calculations of energy usage on an hourly basis. As a matter of fact, in this specific field more than in any other, using too simplified calculation methods means the risk to obtain results that are distinct from the expected targets.

2.3.1 - General elements on the calculations of energy usage

It has to be clarified that no matter how accurate the mathematical model that is used, the calculation of the energy requirement and relative energy usage is always statistical, for the reasons that are reported below.
Under the same indoor climatic conditions and hours when the building is being used, the energy requirement of a building is given by the following:

$$E_{Tot} = E_T + E_V + E_{int} \qquad (2.4)$$

where:
E_T = energy requirements for transmission;
E_V = energy requirements linked to ventilation;
E_{int} = endogenous loads.
The energy requirements for transmission are those that are due to the heat exchanges through the envelope of the building and depend upon outside air temperature values, on solar radiation, that obviously plays an important role on transparent structures and less so on the opaque ones, upon the possible presence of shieldings and shadings, on the stratigraphy of the walls. Obviously, energy requirements of opposed operation modes, heating and cooling, that are simultaneously generated in different environments, do not cancel each other, but rather, in the balance, they have to be treated separately.
The energy requirements linked to ventilation depend upon the conditions of the outside air, in terms of temperature and relative humidity.
Internal loads depend upon the presence of people and other thermal energy sources, such as lights and electric equipment, and can easily be assessed, in a different way depending upon for which use the building is meant. For example, the assessment is much simpler in an office building, where there always are people inside, but becomes more difficult in a hotel, where the occupancy rate in the rooms is dependent on many factors, and can be very different according to the location of the hotel, either in a city, or in a tourist site. Even more complex is the assessment in the residential sector, because the occupancy can vary very much, based both on the type of tenants (single people, couples without children with both married partners who work, or families with many members and children who stay at home all day long), and on the use of the apartment, that can be vacant, or else used as second home, or used every now and then for business reasons. At this moment, the number of unsold houses is very high, and this aspect cannot be neglected, because the trend in heating requirement can dramatically change, with implications also on the load of the generators and, in consequence, on their operation.
All in all, the endogenous loads are always fruit of statistical models, more or less assessable in accordance with the type of use.
All the other requirements depend upon the climate and in consequence on average values of weather variables. As a matter of fact, their calculation is based on something that doesn't exist in reality: the average climatic year that can never coincide with the actual year that develops in real conditions.
Lastly, it is very difficult to estimate the use of the system itself. For example, in the residential sector, the use of summer cooling is not constant, nor is it foreseeable, because it depends on the habits of the people and on the site where the building is located.
Everything that has been said is very important, because it means that actual conditions can never correspond to calculation results, if the benchmark is one single year of operation, because the actual year is always different from the average year. As a consequence, final users may realize that there are deviations between real consumption data and estimated consumption data that may be either overestimated or underestimated. Since they are in trouble understanding that these differences are only due to climatic variations, what finally happens is that final users may misunderstand the efficiency of the plant. The comparison between actual data and data based on calculations can take place only allowing for many years of operation, in the range of 10 to 15 years that represent the actual life cycle of the system. In the shorter time frames, it might always be desirable to provide sensitivity curves that take into account the variations of energy and economic data in relation to weather variations.
From the engineering point of view, the severity of the mistake that is being made is not so much dependent upon percentages, but rather it is linked to the order of magnitude. A 50% mistake on the parameter that has a 5% incidence on energy consumption means an error of 2,5%. Instead, a 10% mistake on a parameter that has an incidence to the extent of 50%, means an error of 5%. Therefore, it is absolutely useless to try and reach absolute

accuracy for parameters that are not really important (such as the behavior of opaque walls with extremely low transmittance), while overestimating the heat gain due to endogenous loads or the actual amount of the inlet of outside air, or efficiency variations of the generators due to their sequence operation can lead to often markedly more serious mistakes.

2.3.2 - Aspects of the energy balance

Independently of topics dealt with in following paragraphs, when an energy balance is performed for a given building, a lot of attention should be paid to time intervals and spaces. Consider the balance that is expressed in (2.4), apply it to a building, assuming that the benchmark time interval is one day, and ignore the orientation of the different rooms: the mistake here is very serious, because in this way loads that occur at different times are summed up in environments that are not interconnected. In summary, one gets to the point where the assumption is that with solar energy coming from the south at midday one provides the heating in the north at midnight. All this obviously doesn't make any sense, even though sometimes energy balances are made this way and often taken for likely. To avoid mistakes, the balance expressed by Equation (2.4) has to be split up in many time intervals (one hour maximum frequency) and in many spaces (at least for each single orientation, or better, for each single room, obviously bearing in mind the use it is conceived for and occupancy profiles). In this way, Equation (2.4) becomes more complicated, and develops into:

$$E_{Tot} = \sum_{h} \sum_{Or} \left(E_T + E_V + E_{int} \right)_{h-Or} \tag{2.5}$$

where the subscripts *h* and *Or* stand for respectively one single hour and the specific orientation of the portion of building that is taken into account.

It is evident that, if this approach is followed, monthly average data on air temperature and relative humidity cannot be used, because during the day, the greater the change in the data, the greater the thermal range. Also solar radiation may be subject to marked variations according to orientations, with peak values that take place at very different times of the day. Figure 2.1 shows the trends in temperature data and relative humidity data on an average day in the month of February in Milano.

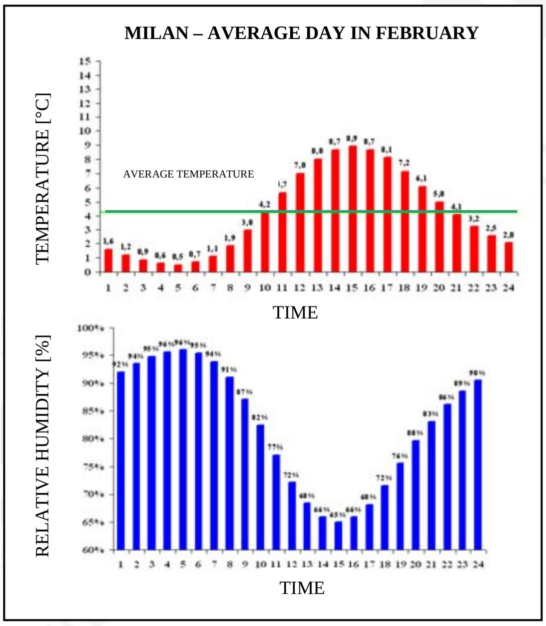


Fig. 2.1 - Example of temperature data and relative humidity data on an average day in February in a city in Northern Italy.

2.4 - The influence of the climate: the Bin Method

To allow for the influence of climate, the more widely spread model is the Bin Method, which consists of constructing hourly frequency distribution curves for temperature and, desirably, also for air relative humidity.

Following the methodology described in annex G, it is relatively simple to construct curves for the Bin Method starting from monthly average data.

The Bin Method presents a series of limitations, illustrated below, that have to be allowed for and overcome, if and when possible. Whenever this is not possible, it is unavoidable to adopt calculation methods that consider average year trends based on 365 days a year.

There are five main limitations in the Bin Method:

1) the same temperature value can occur in very different conditions;
2) under identical temperature conditions, relative humidity conditions can be very different one from the other;
3) under identical temperature conditions, thermal loads can be very different one from the other;
4) the evaluation of the operation percentage for generators is very approximate;
5) it is not possible to evaluate thermal loads when they are densely grouped in certain time scales.

2.4.1 - Limitations of the Bin Method and possible ways to overcome them

First limitation of the Bin Method

It depends on the fact that the same temperature value may be recorded in conditions that are totally different one from the other. If this fact is not allowed for, the risk is putting together conditions that have nothing to do with one another, such as values obtained in the middle of the night mixed with values in broad daylight. Figure 2.2 reports a clear example: in Milano, a temperature value of 6°C may be recorded at 3.00 pm on 15th December but at 5 am on 9th March.

This limitation may be overcome quite easily by dividing hourly frequency distribution curves at least between day and night, as shown in Figure 2.3. In this way, it is at least possible to also have to avoid considering weather data related to night hours for plants that work only in the daytime, as is the case for office buildings. Under equal conditions, an air source heat pump with evaporator that works in an office building has a higher SPF_{HP} value than the SPF_{HP} of the same machine installed in a hotel or in a residential sector building, because in the daytime, weather conditions are milder. By the same token, summer average efficiency is worse because the higher temperatures in the daytime make the operation of the system more difficult.

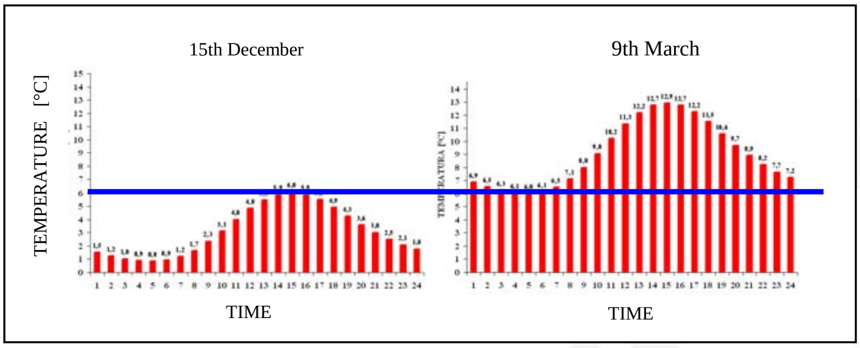


Fig. 2.2 - First limitation of the Bin Method: the same temperature data can take place during day and night. Milano's case.

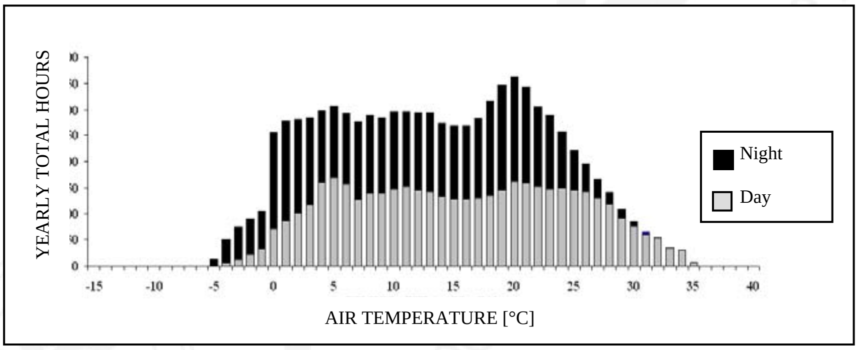


Fig. 2.3 - Overcoming the first limitation of the Bin Method: construction of hourly frequency distribution curves, during day and night in a Northern Italy city.

Second limitation of the Bin Method

It is the consequence of the first one: since identical temperature values can be recorded in very different conditions, relative humidity data may also be very different. Figure 2.4 shows the humidity conditions in Milano at 3.00 pm on 15th December and at 5.00 pm on 9th March, when temperature values are 6 °C: relative humidity in the first case is equal to 77%, in the second to 95%. This aspect can be very important when the plant systems are strongly influenced by relative humidity of outdoor air, such as heat pumps, heat recovery systems from exhaust air and Free-Cooling systems.

In some cases, it is possible to overcome the second limitation executing more calculations on the basis of the overall temperature frequency divided in relative humidity periods. One example on how frequency is divided is reported in Figure 2.5.

Third limitation of the Bin Method

It is a consequence of the first two: under identical conditions of outdoor temperature, the heat requirement expressed as capacity may be different in function of humidity and outdoor radiation conditions and of endogenous load profiles. Heat requirement may also vary in any of the areas in which the building has been split up in accordance with the implementation of (2.3). In consequence, the relationship between heat requirement and outdoor temperature is by no means straightforward, as shown in Figure 2.6, that for summer and winter depicts the areas that are detailed of heat requirement data and the curves that represent average requirement data.

It is not easy to overcome the third limitation, and it is only feasible when the area representing heat requirement data is not too high. It is in any case necessary to divide the requirements between daytime and nighttime, as shown in Figure 2.6, especially for building areas with a different orientation.

Realistically, it is also feasible to construct curves for the Bin Method that are differentiated for each single month, here again following the indications reported in Annex G.

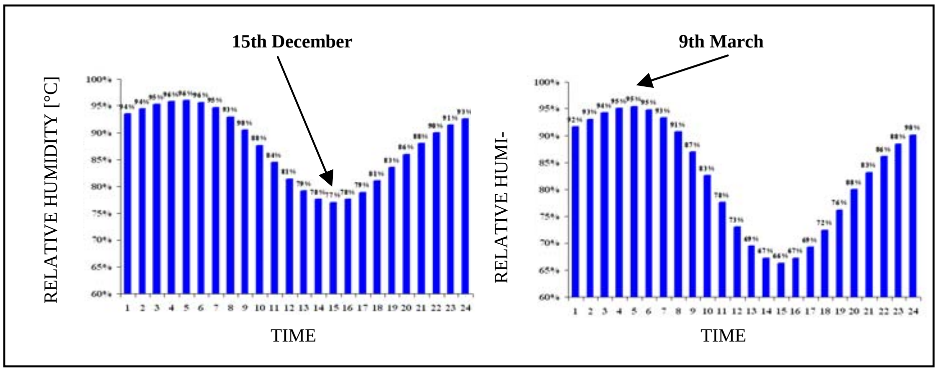


Fig. 2.4 - Second limitation of the Bin Method: with identical temperature values, relative humidity values can be very different.

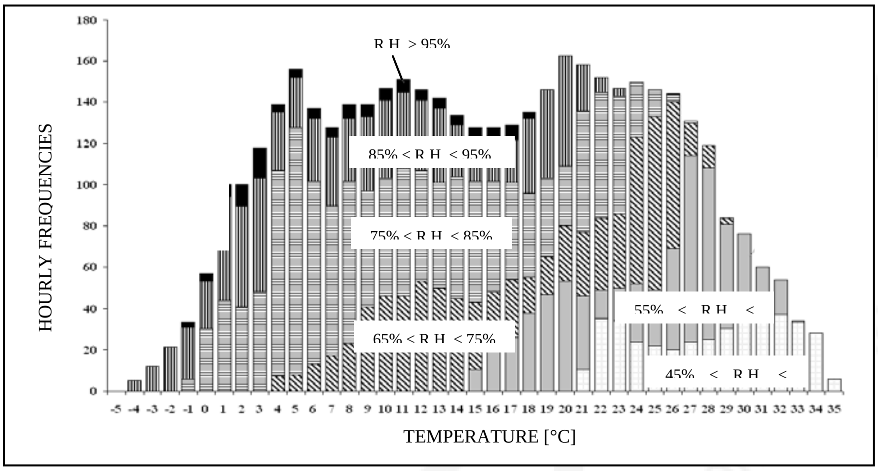


Fig. 2.5 - Overcoming of the second limitation of the Bin Method: hourly frequency split on more relative humidity value ranges for a given city in Northern Italy, daytime.

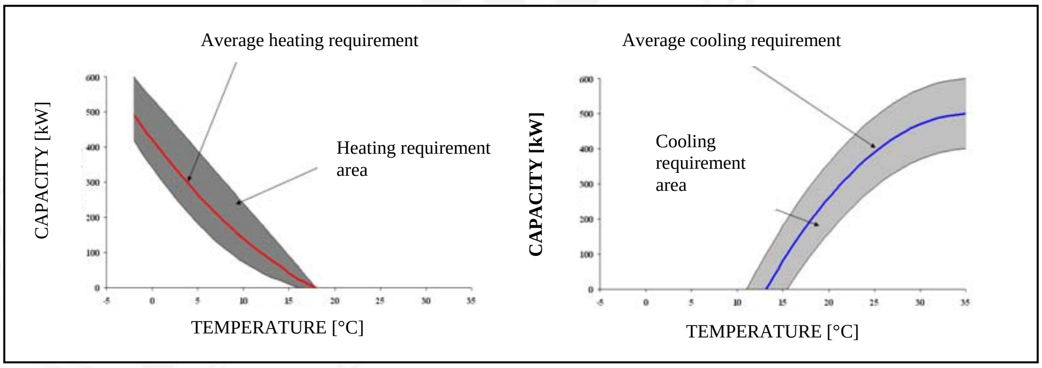


Fig. 2.6 - Third limitation of the Bin Method: there is no univocal instantaneous heating requirement for each single temperature value.

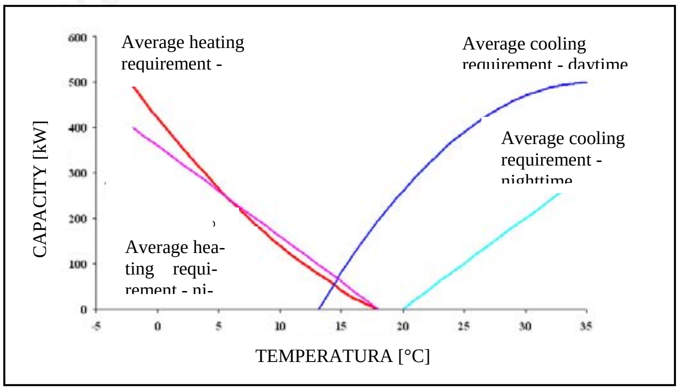


Fig. 2.7 - Overcoming of the third limitation of the Bin Method: average requirement for daytime and nighttime.

Fourth limitation of the Bin Method
It is connected to the third one, for all types of generators, and also to the second, as regards air-to-water heat pumps. The efficiency of a refrigeration system is substantially linked to environmental conditions, to a greater extent for refrigeration systems and air-source heat pumps, and to the load percentage. The Bin Method provides a capacity mean value and a relative humidity mean value for each temperature value. It is not to be taken for granted that the efficiency calculated according to these mean conditions exactly corresponds to the actual average efficiency values that are obtained for those temperature values. The possible mistake may be even greater for air-source heat pumps, due to defrosting that is discussed in Annex I, and for most refrigeration systems, especially if placed in parallel one to the other (this topic is not addressed in the present document, because of its great complexity).
This fourth limitation of the bin method is difficult to be overcome.

Fifth limitation of the Bin Method
This limitation occurs with working profiles that are strongly influenced by the time factor, instead of by temperature values, as is the case with domestic hot water consumption. Since it is impossible to foresee outdoor temperature data at a given time in advance, it is necessary to spread the required load over the 24 hours, as if it were in a steady-state situation. Figure 2.8 shows an example of hourly consumption trends for 24 hours, clearly showing that in the morning hours, the coldest ones, and in the evening energy demand is higher, while it is lower during the day. Most probably, if average values are spread over 24 hours, efficiency values turn out to be higher than the actual ones.
The fifth limitation is still more marked when the working profiles vary in function of the different days in the week or of some periods in the year. Typically this is the case of hotels: the ones in cities are usually fully-booked during the week, but empty on week-ends, while in holiday resorts or arts cities, the opposite is true.

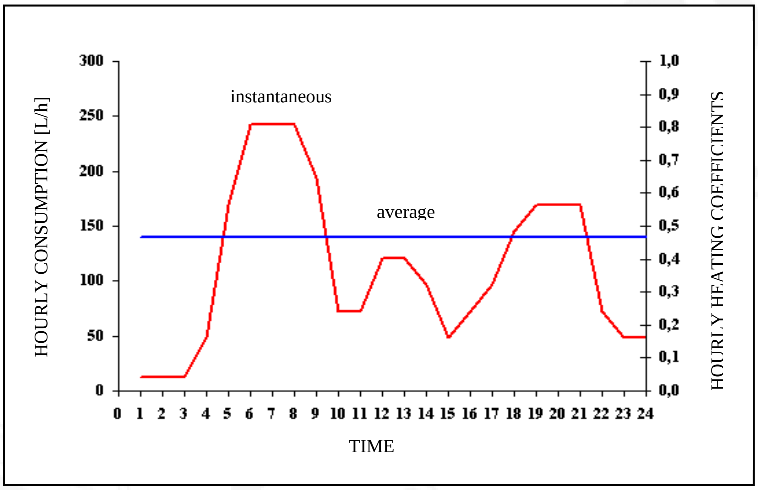


Fig. 2.8 - Fifth limitation of the Bin Method: a correct distribution between different temperature values of the loads that are linked to some times of the day (production of domestic hot water).

Overcoming the fifth limitation may be possible only in the case of large-scale storage, so that production is reduced either in the daytime or in the nighttime. Figure 2.9 shows the production of domestic hot water of a heat pump referred to daytime hours, the hottest ones, so that *COP* values are optimized. In this case, the calculation error is very small.
As can be appreciated, the implementation of the Bin Method results in a series of inaccuracies that synergically act on the overall calculation and reduce reliability. In many cases this is not acceptable, so that it is necessary to construct consumption curves for each of the 365 days of a year.
The second and third limitations of the Bin Method may be overcome using monthly Bins, as adapted by UNI Standard TS 11300-4.

2.5 - AiCARR's proposal for the calculation

AiCARR suggests:

a) with regard to the residential sector in the case of stand-alone systems installed in the individual apartments, a simplified calculation based on SPF_{HP} *or on* $SPF_{HP,C,Syst}$ values may be used. This choice is justified by the fact that in individual apartments, it is not easy to evaluate the actual operation of a system, in particular for the cooling mode. As a matter of fact, the same apartment may be used by a family, by a couple where both partners work, by a single person who works in a different city, by a student whose residence is in another place, and so forth;
b) in all other cases, more complex calculation methods have to be used (the Bin Method as one possible solution), according to what is indicated in previous paragraphs. The greater the complexity, the less homogeneous the operation of a system (hotels are one example, because weekend loads may be different from loads during the week).

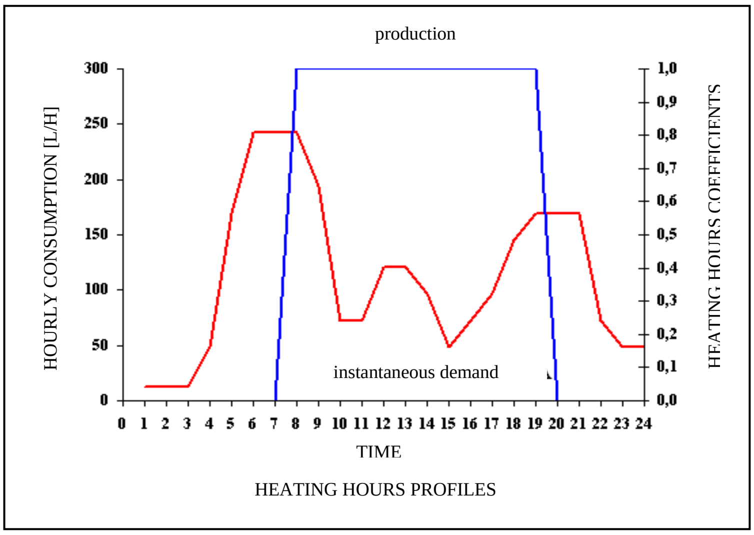


Fig. 2.9 - The fifth limitation of the Bin Method may be overcome only if production, with a system of heat energy storage, is concentrated either in the daytime or nighttime.

ANNEXES

HEAT PUMPS

A.1 - Foreword
Some useful information is provided on heat pumps is this annex.

A.2 - Types of heat pumps
A heat pump is a reverse cycle device and, as such, enables the transfer of thermal energy from a lower temperature source to a higher temperature source, the opposite of the natural heat flow, as shown in Figure A.1. The most common example of a heat pump is the domestic refrigerator: a certain amount of thermal energy is transferred from the food compartment to the kitchen: the food compartment is the cold source of the refrigerator and the kitchen is the hot source. The difference between a refrigerator and a heat pump is not thermodynamic in nature, rather it only depends on the observation point. If the system is observed from the cold source point of view, one can see a refrigerator, while if you observe the system from the hot source point of view, you can see a heat pump.
The operation of a heat pump requires energy to make the system work. This energy can be mechanical, which is the case for compression devices, or else thermal, which is the case for absorption devices.

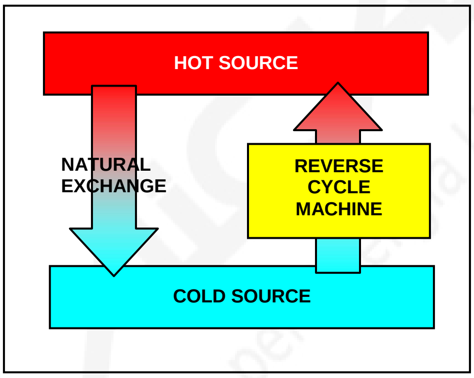


Fig. A.1 - Diagram of a heat pump indicating the directions of the thermal exchange.

Compressor-driven heat pumps are fired with electrical energy or with any fuel, for example natural gas, burnt in an engine that produces mechanical or electrical energy if coupled with an alternator. In this second case, it is also possible to recover the thermal energy from the cooling of the engine and the energy from exhaust fumes. These systems are called Total Energy.

A.3 - Compression heat pumps
Figure A.2 shows a refrigeration cycle in its basic structure, characterized by 4 fundamental elements:
1) the compressor;
2) the condenser, which is the heat exchanger between the refrigerant fluid and the heat source;
3) the organ of lamination, which in refrigeration devices for air conditioning is almost invariably composed of a thermostatic valve;
4) the evaporator, which is the heat exchanger between the refrigerant fluid and the cold source.

In the evaporator, the refrigerant fluid has a lower temperature than the cold source, which is why a natural transfer of heat takes place, from the second to the first. Vice versa, in the condenser the refrigerant fluid is higher in temperature than the heat source to which it releases thermal energy.
The heat transfer, therefore, from the cold source to the hot one takes place through two natural exchanges: the first exchange in the evaporator, from the cold source to the refrigerant, and the second exchange in the condenser, from the refrigerant to the heat source. The compressor and the lamination valve have the function to bring the refrigerant to the required conditions that make the two thermal exchanges possible.
The heat transfer occurs through a change of state of the refrigerant which condense from a vapor into a liquid in the condenser and from a liquid to a vapor in the evaporator. Obviously, these changes of state need a temperature difference, so in the condenser temperature values are higher than in the evaporator, which is colder. For example, in an air condensed refrigerator, that produces water at 7 °C with air outdoor temperature at 35 °C, the

temperature for the change of state inside the condenser (condensation temperature) is 50 °C and the temperature for the change of state inside the evaporator (evaporation temperature) is 2 °C. It is well known that the changes of state can occur at different temperatures: for example, at atmospheric pressure, water boils and evaporates at 100°C; if the pressure is lower, water evaporates at a lower temperature and, vice versa, if the pressure is higher atmospheric pressure, evaporation takes place at a higher temperature.

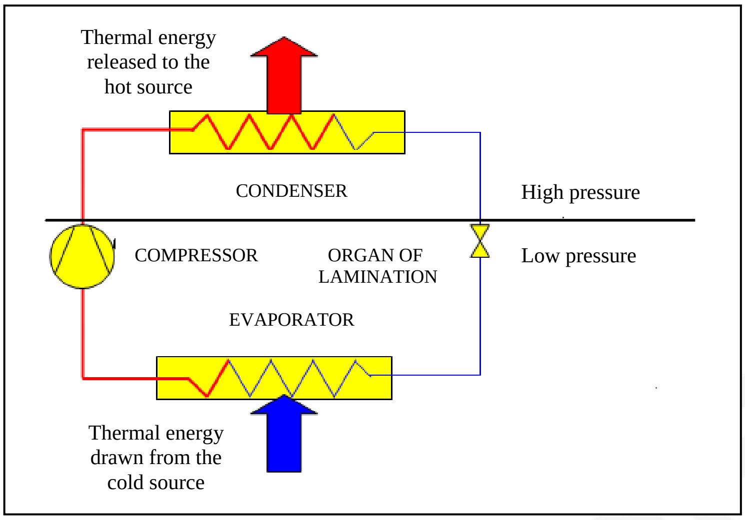


Fig. A.2 - Compression heat pumps.

Refrigerants also are subject to two different pressure levels: higher in the condenser, lower in the evaporator, and the type of refrigerants used have an impact on these values. The compressor has to raise the pressure of the refrigerant which passes from the vapor state to the condensed state, while the organ of lamination acts in the opposite direction and brings the refrigerant from liquid to vapor. In practice, the circuit is divided into two main parts: the first – high pressure - from downstream of the compressor to upstream of the organ of lamination and, the second – low pressure - from downstream of the organ of lamination to upstream of the compressor, as is shown in the diagram of Figure A.1.
In summary, a heat transfer from the cold source, at lower temperature, to the heat source, at higher temperatures, is possible, only if the system is supplied with the input energy that is needed to take the refrigerant from the evaporator pressure level to the condensation pressure level.

A.4 - Total Energy Systems

A Total Energy system is a system that directly produces energy on the spot and on its own to activate the compressor of a refrigeration loop and that, simultaneously, recovers the waste thermal energy which is then used for an air conditioning unit, either directly used, or used once it is transformed to refrigeration energy, through an absorption cycle.
According to this definition, a Total Energy system is made up of a thermal engine, that either generates mechanical energy which is directly supplied to the shaft of the compressor, or electrical energy if coupled to an alternator, by a compression refrigeration cycle, by a device for waste thermal energy recovery or possibly by a refrigeration absorption cycle.
Figure A.3 illustrates the simpler unit, where the compressor is directly coupled to the engine shaft. In this configuration, the heat source is not only supplied with the input thermal energy produced by condensation of the refrigerant in the compression refrigeration cycle, but also with the energy recovered from exhaust gases and from the heat rejected by the engine body, which is given by the product of the energy burned as fuel by the thermal efficiency of the engine. The mechanical energy given off to the compressor is equal to the product of the primary energy burned as fuel by the mechanical efficiency of the engine.
As an alternative to direct coupling, an electrical generator can be used, made up of an engine connected to an alternator that supplies energy to the compressor of a refrigeration cycle. In this case, the Total Energy system is defined as hybrid.
The more complex configuration, typical of trigeneration systems, is shown in Figure A.4: the thermal energy produced by the engine cooling is used to supply energy to an absorption heat pump. In this way the amount of energy that is given off to the heat source goes up, because it adds up to the effect of the absorption heat pump.

A.5 - Absorption heat pumps

An absorption device is made up of four main components, as shown in Figure A.5. In particular, evaporator and condenser have a position and a purpose similar to those of a compression cycle. They too are placed in contact with the two sources, cold and hot, from which they simultaneously draw energy and to which they give off energy, through the change of state of the refrigerant. The compressor is replaced by an organ that is called generator that is in direct contact with the thermal energy source. The organ of lamination, which, as specified above, for vapor-compression devices, typically is a thermostatic valve, is replaced by an organ that is called absorber.
Inside the loop there is a mixture of two components; one has the function of a refrigerant and the other with an absorption function. In the generator, thermal energy is supplied to the mixture, therefore the refrigerant evapo-

rates and separates from the absorbing fluid, goes through the condenser, where it becomes liquid again, only to evaporate once more in the low pressure environment of the evaporator. The absorbing fluid, instead, passes from the generator to the absorber, where it replenishes the mixture with the refrigerant in vapor state, thus keeping the environment at a low pressure level.

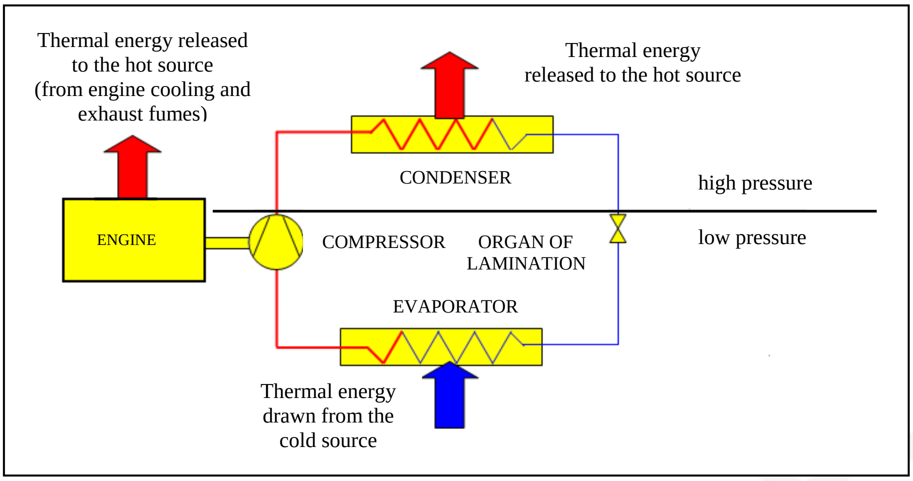


Fig. A.3 - Total Energy System in the simpler configuration.

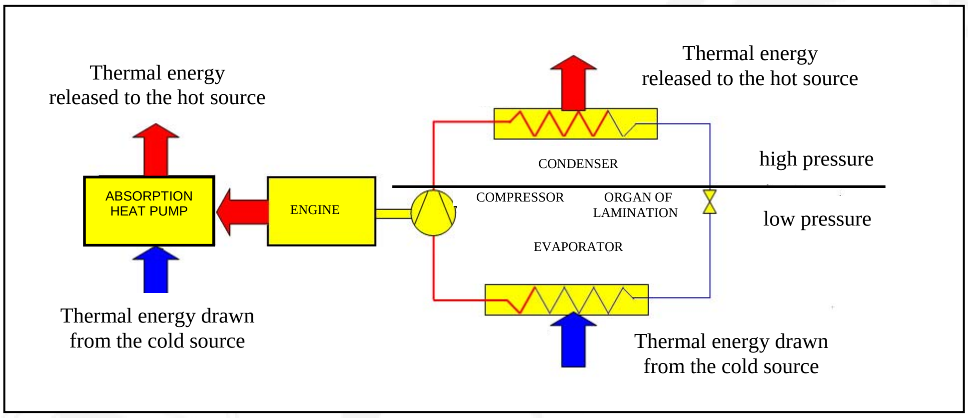


Fig. A.4 - Total Energy System in the more complex configuration (trigeneration system).

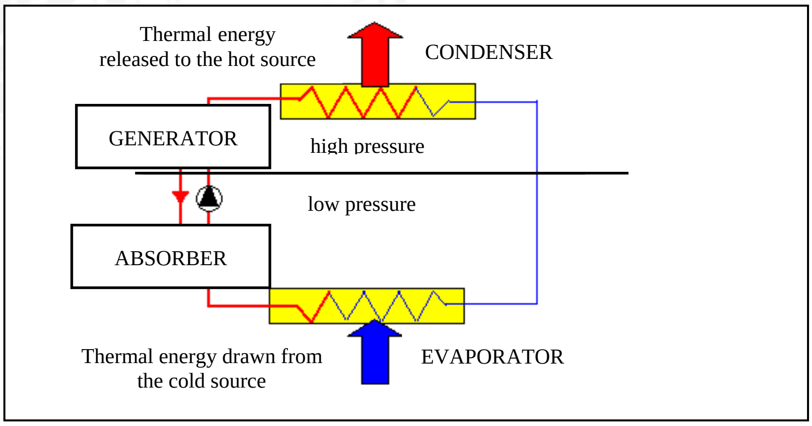


Fig. A.5 - Absorption refrigeration cycle.

The refrigeration loop is divided up in two distinct parts, one at a high pressure, containing generator and condenser, and the other one at a low pressure, containing evaporator and absorber.
There are many types of absorption refrigeration cycles. One classification can be established with the benchmark of the thermal energy source and another classification, on the basis of the mixture contained inside the refrigeration loop.
The thermal energy can be supplied:

- by direct-firing, by directly burning fuel in the generator;
- by vapor;
- by hot water, provided temperature exceeds 75 °C.

With respect to the mixture, there are several solutions that, in theory, can be used. In practice, only two are used:

- water – lithium bromide;
- ammonia – water.

Of the two substances that make up the mixture, obviously one carries out the refrigerant function, while the other has an absorbing function. In the case in question, a lithium bromide/water solution, the water acts as refrigerant and the lithium bromide as absorbent, while in the ammonia/water mixture, ammonia is the refrigerant and water is the absorbent.

There are different limitations for the two options: in the case of a lithium-bromide/water mixture, the temperature value of the cold source has to be above 0 °C while the heat source doesn't have to exceed 38 °C. In the case of an ammonia/water mixture, the temperature value of the cold source can go down to -20 °C, while the heat source can reach 70 °C. Consequently, only water-to-water devices can be built with a lithium bromide/water mixture, while also reverse-cycle air-water plants can be constructed with ammonia/water mixtures.

A lithium-bromide/water device can work with a heat pump thanks to the reversing mechanism offered by this kind of plant. The condensation takes place with water, thus the same pattern adopted for compression refrigeration systems apply: their only limitation is that they can produce water at temperature values that do not exceed 38 °C when the heat pump is used in winter operation.

An ammonia/water device can work on a reverse cycle in cooling mode actually not only in the hydraulic loop, because air can also be used as a thermal source. The advantage is that they are capable of producing water at 70 °C with efficiency values that are sufficiently high even in very low outdoor air temperature ranges (-20 °C).

PERFORMANCE OF HEAT PUMPS

B.1 - Foreword

Whatever the type of heat pump, considering that a heat pump system is always characterized by the energy drawn from the cold source E_F, by the energy that is supplied to the system, E_S, and by the energy that is given off to the heat source, E_C, the energy balance, shown in Figure B.1, is the following:

$$E_C = E_F + E_S \tag{B.1}$$

Starting from (B.1) and accepting the definitions introduced by EUROVENT for coefficients of performance of heat pumps:
- *EER* for summer operation,
- *COP* for winter operation,

the efficiency of the system can be defined by the following equations:

$$EER = \frac{E_F}{E_S} \tag{B.2}$$

$$COP = \frac{E_C}{E_S} = \frac{E_F + E_S}{E_S} = EER + 1 \tag{B.3}$$

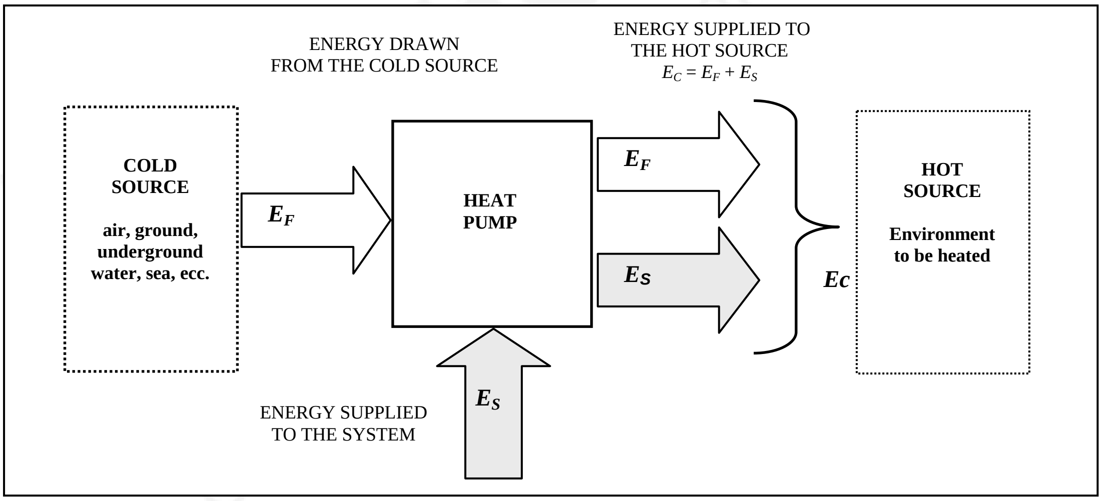


Fig. B.1 - Energy balance of a heat pump system.

B.2 – The performance of compression heat pumps

For compression heat pumps with electrical engine, the energy supplied to the system is always electricity; hence *EER* and *COP* are ratios of the thermal energy produced to electrical input energy, where their values range from 2 to 8 depending upon the parameters that influence them.

B.3 – The performance of absorption heat pumps

For absorption heat pumps, the energy supplied to the system is thermal, so that (B.2) and (B.3) are ratios between thermal energies. This is the reason why, apparently, *EER* and *COP* values of absorption heat pumps look markedly lower than compression heat pumps data, with their range going from 0,7 to around 2. However, this is not true: as repeated many times in this document, primary energy has to be the representative benchmark.

B.4 – The performance of Total Energy systems

Total Energy systems use a compression refrigeration cycle that includes a gas engine. With reference to the configuration of Figure A.3 in Annex A, *EER* and *COP* values are derived from the following ratios:

$$EER = \eta_M \eta_A EER_{RC} \tag{B.4}$$

$$COP = \eta_M \eta_A COP_{RC} + \eta_T \tag{B.5}$$

with:

η_M = mechanical efficiency of the engine;
η_A = electrical efficiency of the alternator (in the case of direct coupling η_A = 1 because the mechanical energy is provided directly);
η_T = thermal efficiency of the engine (share of the thermal energy that is recovered from engine cooling and exhaust gases);
EER_{RC} = *EER* of the refrigeration cycle in the Total Energy system;
COP_{RC} = *COP* of the refrigeration cycle in the Total Energy system.

For the configuration in Figure A.4 of Annex A, one has to allow for the energy that is transferred from the absorption cycle, so that (B4) and (B.5) respectively become:

$$EER = \eta_M EER_{RC} + \eta_T EER_{AD} \tag{B.6}$$

$$COP = \eta_M COP_{RC} + \eta_T COP_{AD} \tag{B.7}$$

where:

EER_{AD} = *EER* of the absorption device;
COP_{AD} = *COP* of the absorption device.

Figure B.2 shows an example of the trends of *EER* and *COP* in a Total Energy system with direct coupling in function of the coefficient of performance in the compression refrigeration cycle.

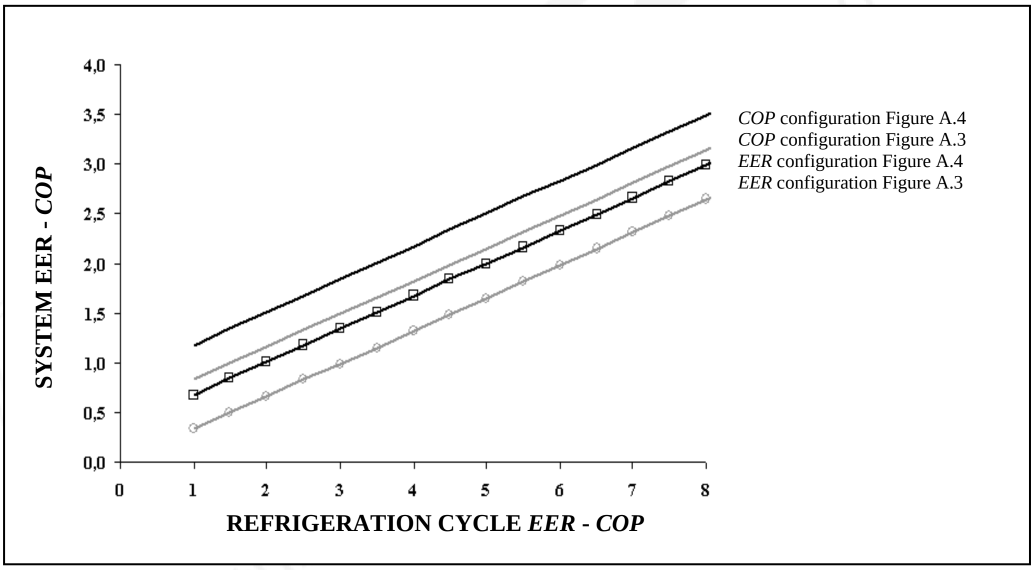


Fig. B.2 - *EER* e *COP* of a Total Energy system with η_M= 0,33, η_T= 0,5, EER_{AS}= 0,7 e COP_{AS}= 1,7.

B.5 – Comparison of performance between the different types of heat pumps: *PER*

Based on what has been discussed, it is evident that the values for coefficients of performance of compression heat pumps are not homogeneous if compared to *COP* values of absorption heat pumps and Total Energy systems, because the form of energy supplied to the system is different, E_S, which in the first case is electrical energy (secondary energy), and in the second, thermal energy, produced by a combustion (primary energy). To overcome this inconsistency, also in the case of compression heat pumps, primary energy has to be assumed as the detailed benchmark, which means that the energy supplied by the system, E_S, has to be transformed to primary energy, E_P, as shown in Figure B.3. Obviously, under identical conditions of primary energy, E_P, the greater the energy provided to the system, the higher the efficiency of the electrical system, η_{SE}, which in turn is equivalent to the product of the output efficiency of an electricity-generating power station by the distribution efficiency of the grid (it is worth noting that in Part 1 this efficiency is defined as distribution and production efficiency of electrical energy and is indicated by the symbol η, as laid down by Italian Decree 28/11). The Primary Energy Ratio, *PER*, can be expressed both in summer operation and in winter operation by the following ratios:

$$PER_{Sum} = \frac{E_F}{E_P} = \frac{\eta E_F}{E_S} \quad \text{(B.8)}$$

$$PER_{Win} = \frac{E_C}{E_P} = \frac{\eta E_C}{E_S} \quad \text{(B.9)}$$

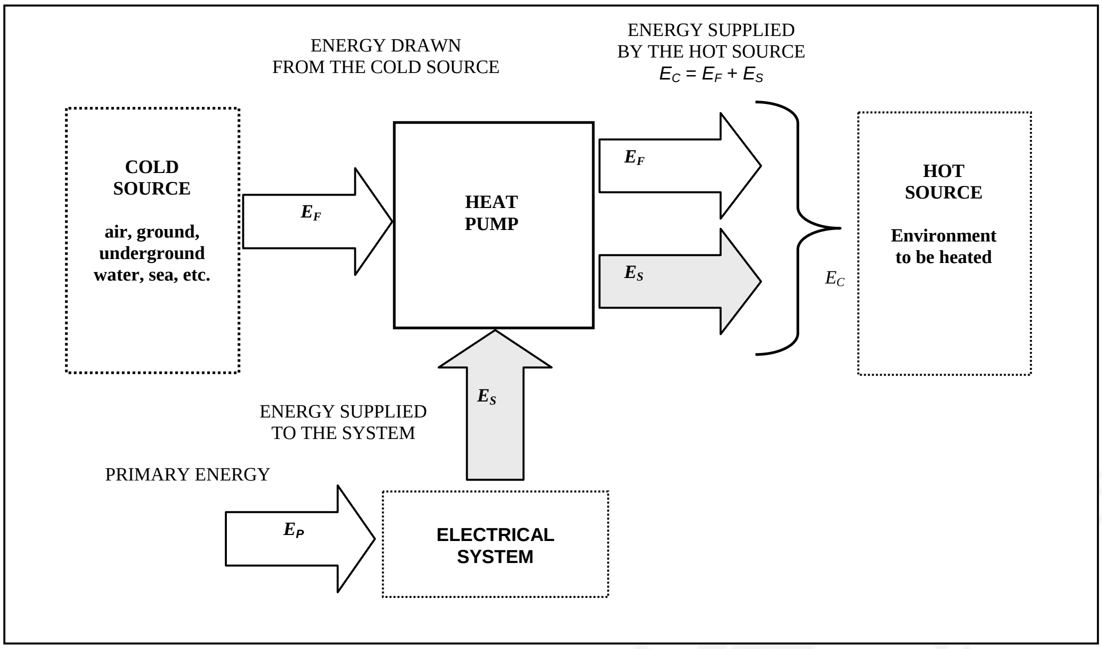


Fig. B3 – Energy balance on a heat pump system with primary energy as a benchmark.

which clearly points to the fact that for absorption devices and for Total Energy systems, the value of the Primary Energy Ratio, *PER*, in both operation conditions, corresponds to the coefficients of performance *EER* and *COP*. Likewise, for a condensing boiler the *PER* corresponds to efficiency, which is calculated based on the higher heating power, or higher calorific value.

FREE-COOLING

C.1 - Foreword

To reduce energy usage in buildings, it is possible to install plants that are capable of exploiting both direct and indirect Free-Cooling systems.
Direct Free-Cooling is obtained through the input of outside air into the environment at lower temperature values than the ambient air (that is with lower enthalpy values).
Indirect Free-Cooling is obtained conveying water to the thermal exchange batteries of an ATU or of a terminal.

C.2 - Direct Free-Cooling

The use of direct Free-Cooling is possible in air source systems, both with variable and with constant air volume, and to a lesser extent also in primary ventilation systems. Generally speaking, the Italian climate is suitable for these systems since, as shown in Figure C.1, in Italy in the months from April to October, relative humidity of outdoor air, which has a strong impact on the performance of direct Free-Cooling, is unlikely to exceed the level of 70% in the daytime, even allowing for the obvious differences between Italian cities.

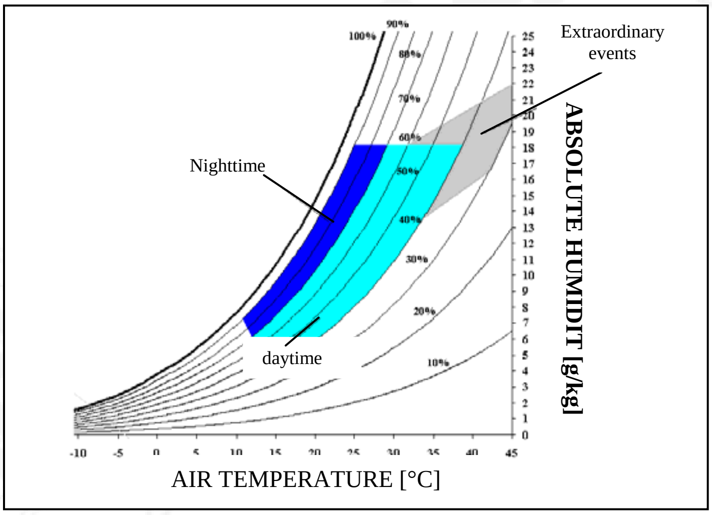


Fig. C.1 - Variation field of temperature and air relative humidity data in Italy between April and October.

The savings that can be achieved with direct Free-Cooling systems are truly considerable, even more so if the input of outside air is suitably pre-cooled with both direct and indirect adiabatic cooling systems.

C.2.1 – Indirect adiabatic cooling

The most efficient solution to improve the performance of exchangers that work only with sensible heat in summer operation and Free-Cooling operation consists in combining them to an indirect adiabatic cooling system (IAC), single stage or double stage, as shown in Figure C.2.
In a single stage system, as shown by the diagram reported in the left part of the Figure, exhaust air starts from condition 1, then enters the sensible heat recovery exchanger, undergoes an adiabatic cooling by humidification, and reaches condition 2. In the exchanger, an exchange takes place between outdoor air, in condition 3, which is cooled until it achieves condition 4, and the exhaust humidified air, in condition 2, that is heated.
In the double stage system, the expulsion line consists of two exchangers equipped with humidifiers. The exhaust air in condition 1 undergoes a first adiabatic cooling by humidification, reaching condition 2 before entering the first sensible heat recovery exchanger, where it is heated and from where it is discharged in condition 3, thanks to an exchange of energy with outside air that, having entered condition 6, is cooled until it reaches condition 7 and is then conveyed to the cooling battery. The exhaust air in condition 3 is cooled again by humidification until it

reaches condition 4, it subsequently enters the sensible heat exchanger where it draws thermal energy from outside air that goes from condition 5 to condition 6.
For both systems the transformations are reported in the psychometric diagrams in the right part of the Figure.
From the conceptual point of view, single stage and double stage systems are similar: it could be argued that the double stage system may enhance efficiency in a sensible heat exchanger (e.g. cross flow static recuperator), the less so the higher the efficiency of each exchanger is. Absurdly, if the exchanger had an efficiency of 1, resorting to a double stage system would be useless. For example, using a double stage system with two exchangers with an efficiency of 0,65 is equivalent to using one single exchanger with an efficiency of 0,8. Instead, using a double stage system with two exchangers with an efficiency of 0,45 is equivalent to using one single exchanger with an efficiency of 0,70 in the single stage. Hence, it is convenient to make use of the double stage only if low efficiency exchangers are used. Moreover, a double stage system allows a greater adiabatic cooling and therefore a greater recovery, but it increases air pressure drops, thus reducing the capacity required for ventilation. This is the reason why it is important to carefully assess its use, since the increase in energy usage for ventilation could reach such a level as to annihilate the advantages that are gained through the greater heat recovery.

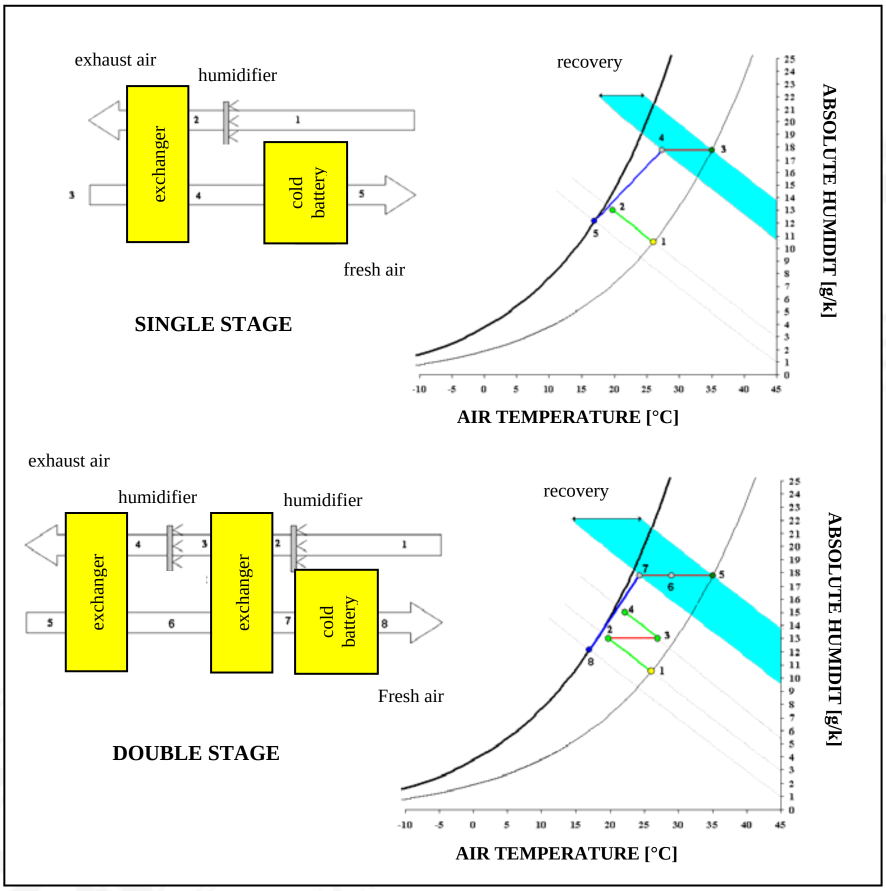


Fig. C.2 – Sensible heat recovery associated with indirect adiabatic cooling.

The system can also be used in the cases when there is no line for exhaust air, for example in the treatment stations of primary ventilation plant. In this case, input air is cooled by using a sort of outside air recirculation, in accordance with the diagram and the transformations that are illustrated in Figure C.3: outdoor air in condition 1 is adiabatically cooled until condition 2 is achieved and subsequently it is conveyed to the exchanger, where it draws sensible heat from the outside air stream that has to be introduced in the plant, cooling the stream until it reaches condition 3, when it is conveyed to the cold treatment battery from where it gets out at condition 4.
Table C.1 reports the maximum level of obtainable energy savings for different temperature and relative humidity values of outdoor air, in the case of direct Free-Cooling with adiabatic pre-cooling of the outside air, considering a recovery device equipped only with sensible heat recovery having net efficiency of 0,5 and a humidifier with efficiency of 0,85. It is evident that the savings are huge, particularly so when relative humidity values of the outdoor air are low, that is in the daytime, which entails that the main savings would be obtained in the tertiary sector.
The Italian climate is much more suitable than is generally assumed to the use of direct Free-Cooling. The values reported in Table C.1, relative to the savings that are obtainable thanks to air temperature and relative humidity values, clearly highlight that direct Free-Cooling is a very good solution also in cities which have an apparently hot but dry climate, which is the case in some cities of southern Italy.
Table C.2 illustrates the behavior of Free-Cooling assisted by adiabatic cooling in the main Italian cities. As can be noted, the influence of humidity and altitude is stronger than the role played by actual weather conditions. The cities where the behavior is most favorable have most commonly low humidity rates, especially in the daytime (Trieste, Reggio Calabria, Bari) or are high above sea level (Enna, Catanzaro, Perugia etc.). Conversely, the two cities that behave worse are - Milano and Palermo - are both characterized by high humidity.

C.3 - Indirect Free-Cooling

As already mentioned, indirect Free-Cooling consists in conveying cold water to the thermal exchange coils of the ATUs or the terminals, thus leading to a free air conditioning system for the building.
Indirect Free-Cooling can originate from two distinct sources:
- geothermal sources (underground water, lakes, sea, ground);

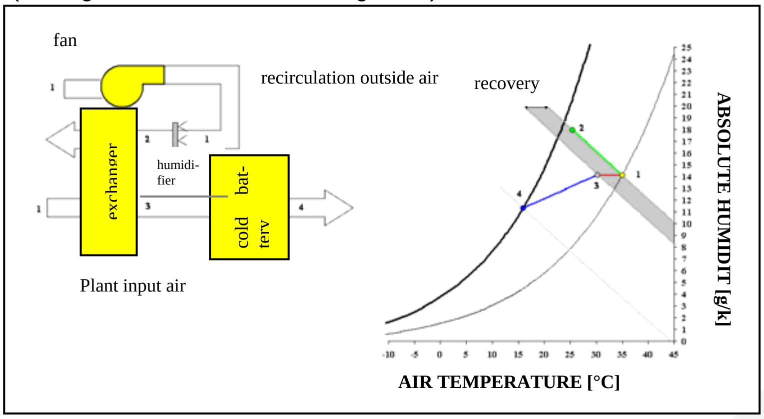


Fig. C.3 – Sensible heat recovery associated with indirect adiabatic cooling and outside air recirculation.

- outdoor air.

A geothermal source has the advantage to have low temperature levels that are in a steady-state situation at all times of the year. Sea water, for example, below 10 meters of depth, is most commonly at constant temperature level ranging from 10 °C to 12 °C. The ground, used as geothermal source thanks to vertical drills, has the yearly average temperature of the desired location, hence ranging between 12 °C and i 16 °C in every part of Italy. Similar temperature levels are also recorded for deep underground water, while surface water (lakes, rivers, lagoons) have more variable temperatures during the year, but in some cases they still are advantageous for indirect Free-Cooling.

Tab. C.1 – Percentage savings that can be obtained thanks to direct Free-Cooling assisted by outdoor air adiabatic pre-cooling (ambient RH = 55%, input air temperature = 16 °C). t_e= outdoor temperature; t_a= indoor temperature ; RH_e= outdoor relative humidity.

		SAVING							
t_e [°C]	t_a [°C]	RH_e 40%	50%	60%	65%	70%	75%	80%	85%
32	26	0%	0%	0%	0%	0%	0%	0%	0%
31	26	10%	0%	0%	0%	0%	0%	0%	0%
30	26	29%	0%	0%	0%	0%	0%	0%	0%
29	26	48%	0%	0%	0%	0%	0%	0%	0%
28	26	69%	10%	0%	0%	0%	0%	0%	0%
27	26	90%	31%	1%	0%	0%	0%	0%	0%
26	26	100%	54%	1%	0%	0%	0%	0%	0%
25	26	100%	77%	24%	0%	0%	0%	0%	0%
24	26	100%	99%	49%	24%	0%	0%	0%	0%
23	25	100%	100%	59%	42%	15%	0%	0%	0%
22	24	100%	100%	50%	49%	38%	8%	0%	0%
21	23	100%	100%	70%	53%	48%	34%	3%	0%
20	22	100%	100%	67%	61%	59%	54%	36%	0%
19	21	100%	100%	81%	71%	66%	61%	56%	35%
18	20	100%	100%	100%	90%	81%	77%	75%	65%
17	20	100%	100%	100%	100%	100%	100%	94%	88%
16	20	100%	100%	100%	100%	100%	100%	100%	100%
15	20	100%	100%	100%	100%	100%	100%	100%	100%

Therefore, geothermal sources can be used only in 3 ways:
- conveying water pre-cooling coils. This system represents a good alternative to a heat recovery system. For example, having water available at a temperature level in the range 15 -17 °C, with a sufficiently low tempera-

ture differential in the range of 3 °C, and with a good by-pass factor, *BF* = 0,05, it is conceivable to achieve an air temperature drop from 35 °C with *RH* = 50% down to 20 °C, reducing enthalpy by 22,4 kJ/kg, as shown in Figure C.4.
Obviously the saving is decreased with the reduction of outdoor air temperature values, but it remains sufficiently high for the most part of the summer season.

Tab. C.2 – Evaluation on the use of Free-Cooling with adiabatic cooling in some Italian cities divided into 3 areas (North, Center, South)

	NORTHERN ITALY	**CENTRAL ITALY**	**SOUTHERN ITALY**
EXCELLENT Free-Cooling can also be used during summertime	Aosta, Sondrio, Trieste, Bolzano, Trento	Perugia, Rieti, L'Aquila, Grosseto	Campobasso, Enna, Avellino, Sassari, Bari, Catanzaro, Reggio Calabria
GOOD Free-Cooling can be used in spring/autumn and, exceptionally in summer	Torino, Alessandria, Como, Genoa, Rimini, Bologna	Macerata, Ascoli	Lecce, Catania, Cagliari
POOR Free-Cooling can be used only in-spring/autumn	Venezia	Ancona, Roma	Napoli, Pescara
BAD Free-Cooling is unlikely to be used except when air temperature levels are very low	Milano		Palermo

It has to be highlighted that a thermal source that is available at 15 °C can be used also in the winter to preheat outdoor air, in a sort of free heating system, or Free-Heating.
The same coil of Figure C.4, if kept at 13 °C in winter (allowing for the presence of a heat exchanger between source and plant), is capable of taking air that is at -5 °C with *RH* = 80% up to a temperature level of 10 °C, with an enthalpy saving of 15 kJ/kg. Hence it is possible to conceive a winter saving similar to the one of an only sensible economizer;

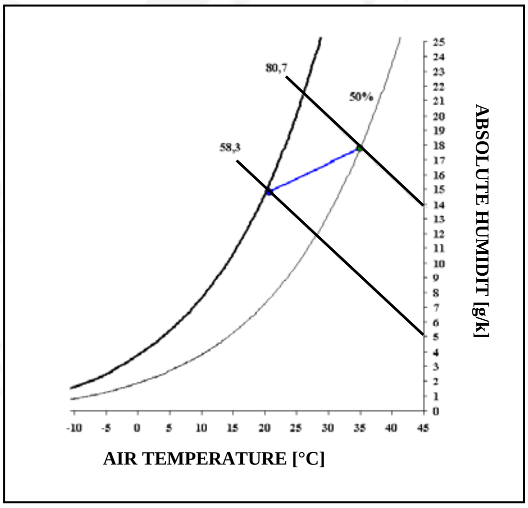


Fig. C.4 – Air pre-cooling by means of Free-Cooling from geothermal source. Case of a coil kept at 17 °C, temperature gap of the water of 3 °C, BF = 0,05. The saving is proportional to an enthalpy gap of 22,4 kJ/kg.

- conveying water to terminals at moderate temperature, such as floor or ceiling radiant panels, or chilled beams. For example, in the case of radiant panels, with this solution it is possible to obtain a only sensible capacity, ranging from 20 to 60 W for surface unit, depending upon the different solutions, be it a radiant floor or ceiling. Considering that this capacity is feasible at all times of the summer, the solution is very interesting.
- conveying water to the fan coils of a primary air plant, only in spring and autumn. For example, for an ambient air temperature value of 22 °C, a fan coil supplied with input water at 15 °C is capable of supplying one fourth of the nominal sensible capacity, or the one that is calculated with temperature values equal to 7 °C for input

water and to 26 °C for air: a fan coil with nominal sensible capacity equal to 1.500 W, is capable of providing 400 W with input water at 15° C and input air at 22° C.

The three options can be used simultaneously. When the generator is a water-water refrigerating system connected to a geothermal source, it is possible to use the scheme drawn in Figure C.5 where water, coming from the exchanger with thermal source, is partially or totally conveyed to the terminals first and then to the condenser of the refrigerating system.

Key:
1: water-water heat pump
2: Free-Cooling terminals
PE: pump evaporator loop
PC: pump condenser loop
PF: pump Free-Cooling loop
PS: pump source
C: collector
SC: exchanger
V_3: overpressure valve
Grey area: radiant panel supply and pre-cooling coils.

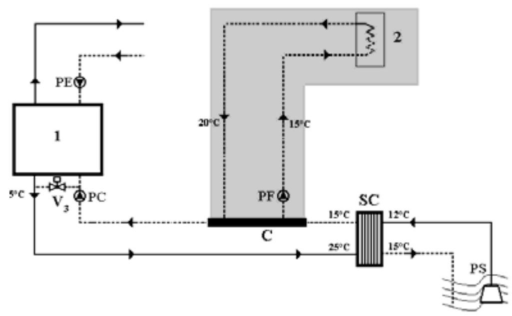
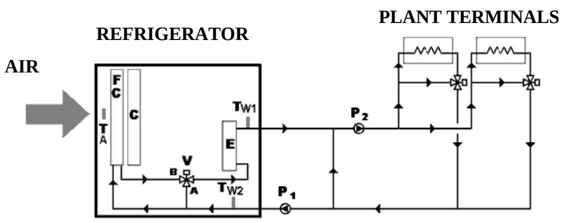


Fig. C.5 – Summer Free–Cooling from geothermal source.

As already mentioned, the thermal source used for indirect Free-Cooling can also be outdoor air, used to pre-cool the water of the plant, provided that the climatic conditions are favorable.

This system is widely used in the industrial sector and in the sector of communication and IT, but in spring and autumn, it can also be used in the tertiary sector, in hotels and hospitals, and be combined with ceiling radiant systems or chilled beam systems. The possible options are water coolers, closed loop evaporative cooling towers, or refrigerating systems with embedded Free-Cooling.

The refrigerating systems with embedded Free-Cooling, shown in Figure C.6, are distinguished from conventional refrigerators for the presence of an finned coil that acts as an air-water exchanger, placed upstream to the condensing battery, through which goes the air moved by the fans before entering the condenser of the refrigerating circuit. The finned coil has the purpose of using the low temperature air to cool the return water from the plant before conveying it to the evaporator of the system. In this way a free cooling is obtained, that also results in a saving of electricity, given that the compressors work less.



Key:

FC	:	Free-Cooling coil	C	:	condensing coil
E	:	evaporator	V	:	valvola a tre vie
T_a	:	air temperature sensor	T_{W1}	:	water temperature sensor
T_{W2}	:	water temperature sensor	P_2	:	system pump
P_1	:	primary loop pump			

NORMAL OPERATION **FREE-COOLING OPERATION**

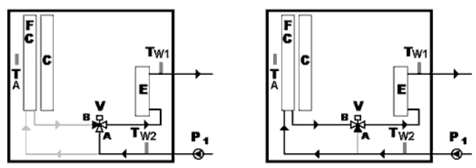


Fig. C.6 – Refrigerating system with embedded Free-Cooling.

OVERLAP OF LOADS

D.1 - Foreword

The increase in thermal insulation thanks to improved building envelopes, the increase in endogenous loads mainly due to systems and to lighting, the greater attention given to the quality of the indoor environment, with the consequent use of an air conditioning plant at all times of the year, imply that designers are increasingly faced with buildings that have simultaneously cooling and heating requirements.

D.2 – Overlap of loads

In a building with simultaneous thermal loads with opposite sign, the air change with outside fresh air and some other areas have to be heated, while other areas of the same building have to be cooled. The diagram in Figure D.1 reports the qualitative trend of heating and cooling loads in function of outdoor air temperature values: it is evident that there is an overlap field of the two different loads.

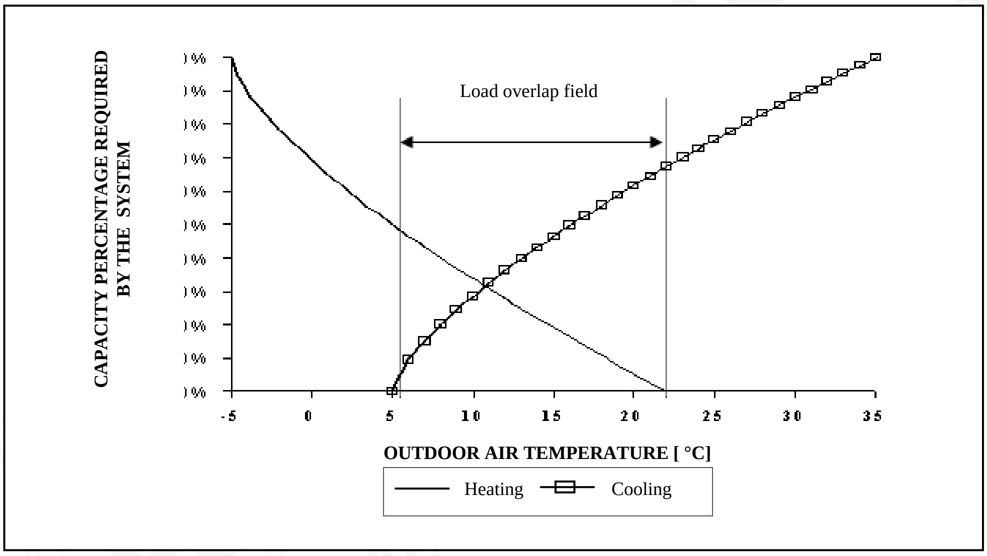


Fig. D.1 – Example of thermal load trends in function of the temperature values of outdoor air.

It is obvious that in actual conditions there is no direct connection between the thermal load and the outdoor air temperature value, because many other variables have an impact on the load, such as solar radiation, occupancy and operation profiles, the outdoor temperature trend, that can have the same value in daytime and nighttime, only in different months. However, for each building it is possible to construct curves similar to the ones in Figure D.1, allowing for average thermal loads.
Moreover, the width of the overlap field greatly varies in function of the type of architecture and the use for which the building is intended. For example, in the shops in a shopping center there is going to be a demand for cooling even when the outdoor temperature value is very low.
It has to be underlined that at the intersection point of the two curves almost invariably each of the single capacities demanded by the plant doesn't exceed 40% - 50% of its maximum value. This has important implications on the yearly overall consumption of the plant, to the extent that the temperatures belonging to the overlap field are those that are more frequent during the year and therefore in the design this overlap has to be accounted for.
Lastly, the load overlap can also take place because of the simultaneous demand for cooling in some spaces and for heating of fresh air and/or production of domestic hot water.

D.3 – Design options

The designer has three different possible options:

- use a conventional system, with double generator, for example two heat pumps, one of which operating in summer cycle and one in winter cycle, or a boiler for the heating energy and a refrigerator for the cooling energy, solution that was frequently adopted until some years ago, but is absolutely reprehensible because of its huge energy usage;
- use a liquid loop system, WLHP (Water Liquid Heat Pumps) with heat transfer to a balancing medium (fluid), again with double generator, but capable of enhancing the energy performance in comparison with conventional systems;
- use a total heat recovery system, both hydronic and with direct expansion. This option has an impact on the total energy consumption of the plant. Figure D.2 shows the total thermal power supplied by generators at the different outdoor air temperatures.

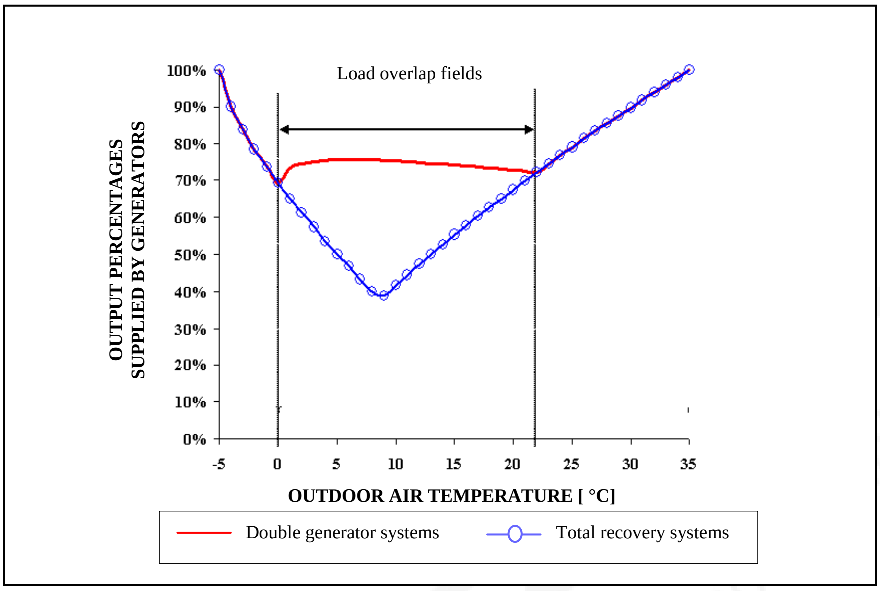


Fig. D.2 – Output supplied by the generators in the case of a system with double generator and of a total recovery system.

The conventional system with double generator is the worse, because it adds up the two capacities required by the plant, the heating one and the cooling one, given that there is no heat recovery.
The heat recovery systems have a better performance, because the total demanded capacity is always the lower of the two, since the other one is recovered during operation of the pumps for free.

HEAT RECOVERY FROM REFRIGERATING SYSTEMS

E.1 - Foreword

As was discussed in Annex A, a reverse cycle transfers heat from the cold source, in contact with the evaporator, to the heat source, in contact with the condenser. Every time that a refrigerating cycle is involved, therefore when a fluid cools down, the heat released by the refrigerant in the condenser can be recovered for free for any purpose, from the heating of the spaces, to the reheating of air or to the production of domestic hot water.

E.2 – Heat recovery

Observing Figure E.1, which depicts a refrigerating cycle on a pressure-enthalpy plane, it shows that heat recovery can be:

- partial: sensible only heat of the refrigerant desuperheating is recovered;
- total: all the condensation heat is recovered.

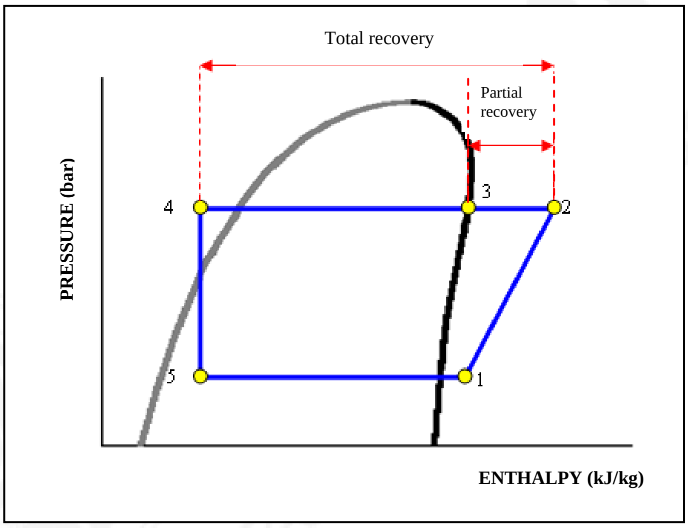


Fig. E.1 – Heat recovery on a pressure-enthalpy plane.

Partial heat recovery

Partial heat recovery takes place through a sensible heat exchanger, called desuperheater, always connected in series to the main condenser. The maximum recovery capacity ranges from 15% to 40% of the refrigerating capacity rendered to the evaporator, in function of the operation parameters and the refrigerant that is used.

Total heat recovery

If the purpose is achieving a greater recovery than the one that is feasible through partial recovery, the only possible solution is the option of a total recovery system. In this case, all the condensation heat is recovered, with a range from 120% to 140% of the refrigerating capacity conveyed to the evaporator.
The total heat recovery can take place through a single compressor cycle or a two compressor cycle with a connection in series, as shown in Figure E.2.
The first case is typical of total recovery refrigeration units, of polyvalent heat pumps with two or four pipes and of direct expansion VRF – VRV systems that don-t produce high temperature domestic hot water. The second case is typical of Water Liquid Heat Pumps and VRF – VRV systems that also produce high temperature domestic hot water.
In the first case, the *COP* to be considered in the calculation is:

$$COP = \frac{\text{delivered heat output}}{\text{electric input to the compressor}} = \frac{P_T}{P_A} \tag{E.1}$$

In the second case, the *COP* to be considered in the calculation is:

$$COP = \frac{\text{delivered heat output}}{\text{sum of power inputs to the compressors}} = \frac{P_T}{P_A} \tag{E.2}$$

The respective *SPF* are calculated by replacing heat output values to energy values

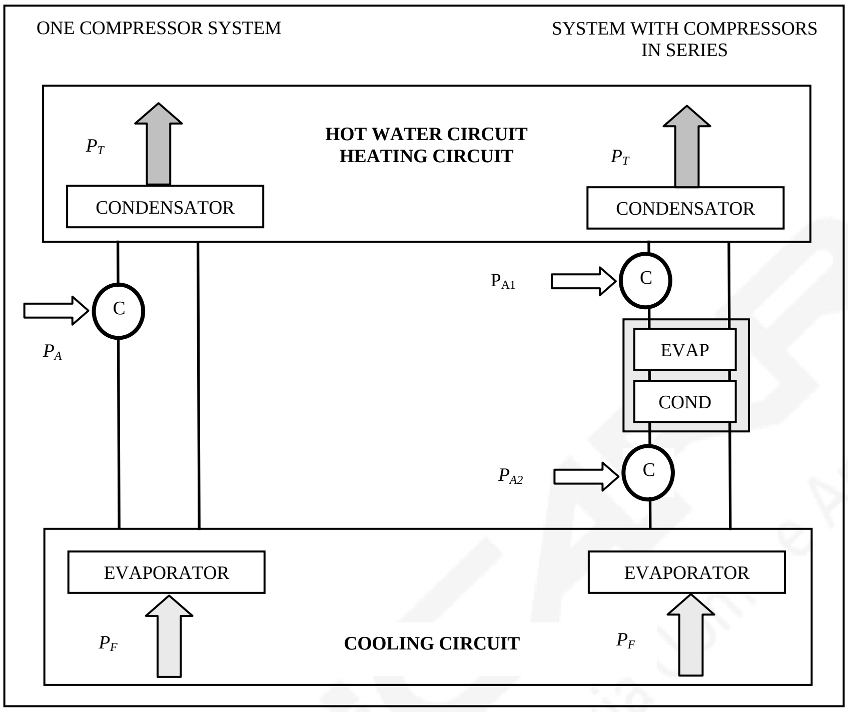


Fig. E.2 – Heat recovery systems.

OPTIMIZATION OF HEAT PUMP PRODUCTION

F.1 - Foreword
Heat pumps can be optimized for the summer season, or for the winter one, for two reasons, one linked to the heat exchanger and the other linked to the compressor.

F.2 – Optimization linked to the operation of exchangers
Heat pumps with reversible refrigeration circuit switch the operation mode by swapping the refrigerant loop in the exchanger when the operation mode changes. Therefore, in an exchanger, in particular in plate exchangers, it is possible to establish that the operation in counterflow takes place either in summer or in winter, as shown in Figure F.1. In the first case, the optimization is relative to the summer operation, because the evaporation occurs at higher temperatures, in the second, to the winter operation, because the condensation takes place at a lower temperature. In any case, the operation in the other season is placed at a disadvantage.

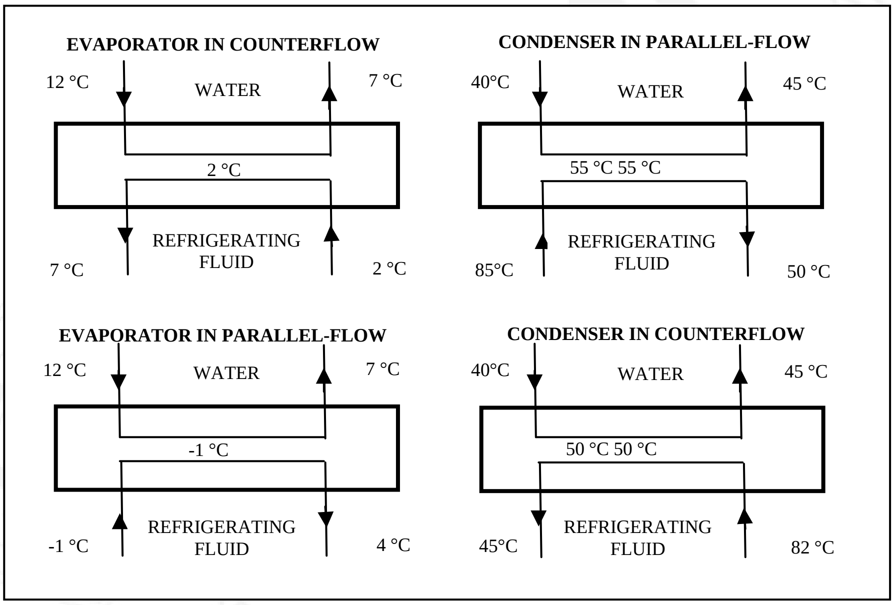


Fig. F.1 – Reversible cycle of refrigerant fluid in heat pump exchangers that reverse the refrigeration cycle.

F.3 – Optimization linked to the operation of compressors
At present, for heat pumps dedicated to HVAC systems, rotary positive-displacement compressors are used, screw and scroll, while piston compressors have been completely abandoned. Screw compressors are designed for higher capacities and work with R134a, while scroll compressors are suitable for medium and small capacities and are combined to R410A or 407C.
Rotary positive-displacement compressors, scroll and screw, conventionally have a design of the compression chamber at the suction point and at the discharge point so that for a certain compression ratio (defined as ratio of the pressure at the beginning of the compression to the value at the end of compression), the level of isentropic efficiency is at the top, as illustrated in Figure F.2. When the compression ratio is different from the most favorable design value, clearly isentropic efficiency is reduced, reason why these compressors are also named “fixed compression ratio compressors”. In general, scroll compressors are optimized for a compression ratio of 3, while screw compressors may have different optimization values.
For the choice of the compressor, it is very important to understand what the working conditions of the heat pump are, or when the demand of energy reaches its top level. For example, using R134a as a refrigerant, the top level

energy consumption is reached in correspondence with a compression ratio of around 3,2 when the operation is for the most part in summer, and around 5 when vice versa the operation is for the most part in winter.

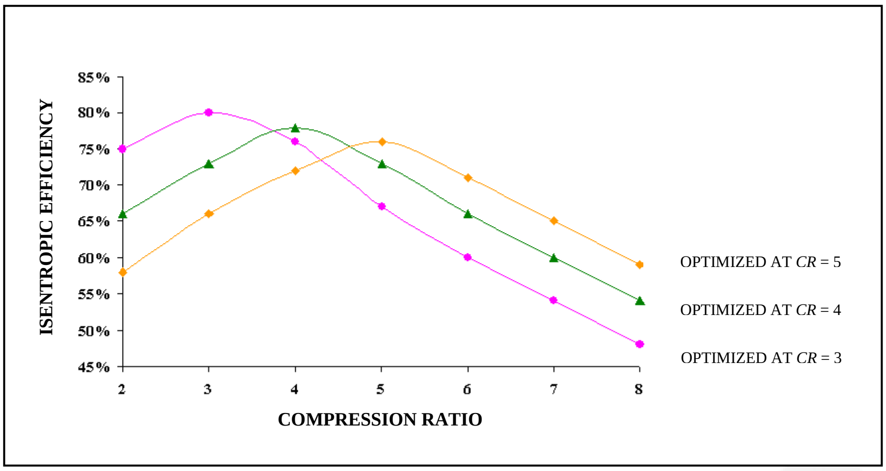


Fig. F.2 – Efficiency of rotary positive-displacement compressors (screw and scroll).

Table F.1 reports some efficiency values in summer and winter operation mode for screw compressors that work with R134a. As can be seen, efficiency has a strong impact on summer performance levels, stronger than on winter performance levels: compressors that are optimized for low compression ratios, compensate with summer gains the losses in winter efficiency. For example, a compressor that is optimized for *CR* = 3, compared with another one that is optimized for *CR* = 5, gains almost 20% of efficiency in summer operation mode, while in winter operation, it loses only 6%. Therefore, it is always worthwhile to optimize for low compression ratios, that are suitable for the summertime. Only heat pumps that exclusively work in winter have to be equipped with compressors that are optimized for high *CR* levels.

Tab. F.1 – EER and COP values in function of CR for screw compressors with R134a

Optimization	Summer operation		Winter operation	
	compressor efficiency	EER	compressor efficiency	*COP*
***CR* = 3**	79%	5,15	67%	4,04
***CR* = 4**	73%	4,76	73%	4,21
***CR* = 5**	66%	4,30	76%	4,28

CONSTRUCTION OF HOURLY FREQUENCY CURVES FOR OUTDOOR RELATIVE HUMIDITY AND TEMPERATURE VALUES

G.1 - Foreword

The construction of the hourly profile for outdoor temperature and relative humidity values is fundamental when a detailed calculation, to enhance energy performances of the selected system is desirable.

G.2 - Method for the construction of profiles

The construction of the hourly temperature values profile of an average day in a typical month requires the knowledge of the monthly average values of maximum temperatures, T_{mMax}, as well as of minimum temperatures, T_{mMIn}. The temperature value at a certain time of the day, T_h, represents a weighted average of the values of those minimum and maximum averages and is derived as follows:

$$T_h = T_{m,Max} - k_h \left(T_{m,max} - T_{m,Min}\right) \qquad \text{(G.1)}$$

where the values of the weight coefficient k_h are reported in Table G.1. It is fundamental to be reminded that in the months when the daylight saving time is in force, the coefficient data have a one-hour gap: the minimum temperature level takes place at 6.00 am, instead of 5.00 am, while the maximum at 4.00 pm instead of 3.00 pm. The hourly trend is sinusoidal, as shown in Figure G.1.

Tab. G.1 - Coefficient k_h values in(G.1) in function of the different times of the day. From ASHRAE Fundamentals 2009, Ch. 28

time	1	2	3	4	5	6	7	8	9	10	11	12
k_h	0,87	0,92	0,96	0,99	1	0,98	0,93	0,84	0,71	0,56	0,39	0,23
time	**13**	**14**	**15**	**16**	**17**	**18**	**19**	**20**	**21**	**22**	**23**	**24**
k_h	0,11	0,03	0	0,03	0,1	0,21	0,34	0,47	0,58	0,68	0,76	0,82

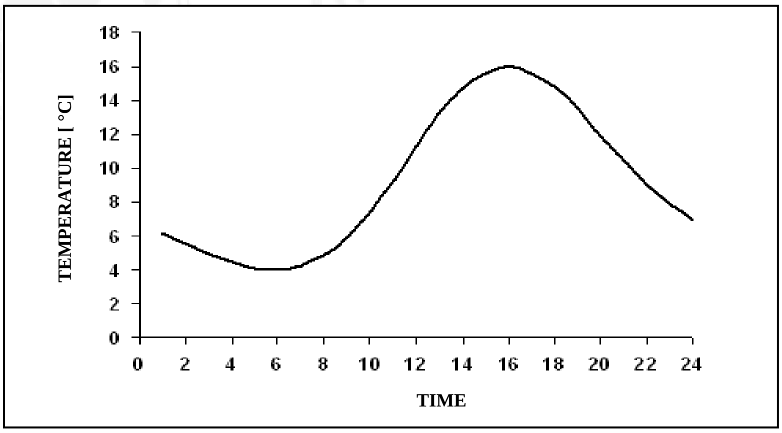


Fig. G.1 – Hourly trend of temperature values.

The construction of a relative humidity values profile is a bit more complex. The relative humidity minimum values are recorded at the same time of the maximum temperature values and vice versa. Having decided that x_{Min} is the absolute humidity value that is recorded at 3.00 pm solar time (4 pm daylight saving time), when the temperature value is at its top and the relative humidity value is at its lowest, and x_{Max} that is recorded at 5 am solar time (6 am daylight saving time) when the temperature value is at its lowest and the relative humidity one is at its top, it is

possible to calculate the hourly absolute humidity values, x_h, in a similar way to what has been done for temperature as follows:

$$x_h = x_{Min} - k_h (x_{Max} - x_{Min}) \tag{G.2}$$

To clarify the procedure, let it be considered one day with T_{Max} = 16 °C, corresponding to RH_{Min} = 60%, e T_{Min} = 4 °C, corresponding to RH_{Max} = 96%. The corresponding absolute humidity values are respectively x_{Min} = 6,66 g/kg and x_{Max} = 4,83 g/kg. Please note that the Min and Max subscripts are referred to relative humidity *RH*, so that the absolute humidity value is higher when it corresponds to of the Min subscript than the Max subscript. At 10 pm, applying (G.2), one obtains x_h = 5,45.
Once the hourly couples of temperature values T_h and absolute humidity x_h are obtained, it is possible to calculate relative humidity trends RH_h values too.
For relative humidity values the trend is sinusoidal too, as shown in Figure G.2.

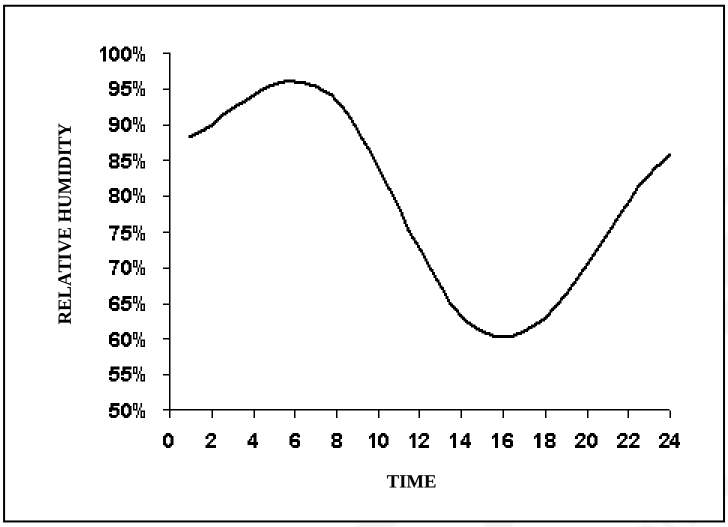


Fig. G.2 – Hourly trend of relative humidity.

In this way, hourly trends of temperature values and relative humidity are constructed, for the average day of each month. Suppose that this takes place on the 15th of each month, that the hottest day in the year is 31st July and the coldest day is 31st January, temperature and relative humidity curves could be constructed for all the 365 days of the year, by simply interpolating between two average days, as is shown in Figure G.3. Therefore for each city it is possible to calculate the hourly frequency of the temperature values to which relative humidity values can be associated.

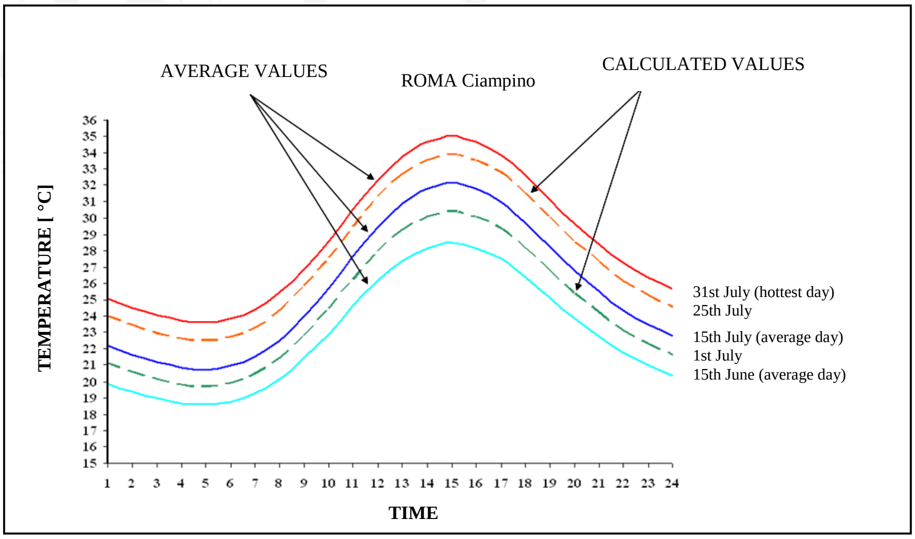


Fig. G.3 – Temperature value hourly trend in intermediate days.

DEFROSTING IN AIR HEAT PUMPS

H.1 - Foreword

During the winter operation of heat pumps frost formation can happen on the surface of evaporators. This phenomenon determines a reduction in the exchanger function that may result in the heat pump being halted.

H.2 – Causes of frost formation

Frost formation on the surface of evaporators occurs when two simultaneous conditions take place:

- reduction in absolute humidity values of the air between inlet and exit of the evaporator, with consequent deposit of the condensate on the surface of the coil;
- surface temperature of the evaporator below 0 °C.

As can be seen in the psychrometric diagram reported in Figure H.1, the reduction in absolute humidity between inlet and exit from the evaporator does not depend very much upon the air temperature, but rather it depends upon its absolute humidity. As a matter of fact, the Figure reports two cases, both characterized by air temperature values equivalent to 4 °C and by surface temperatures of the coil below 0 °C, but by two different *RH* values, 90% and 50%: the variation in absolute humidity values takes place only when *RH* is 90%.

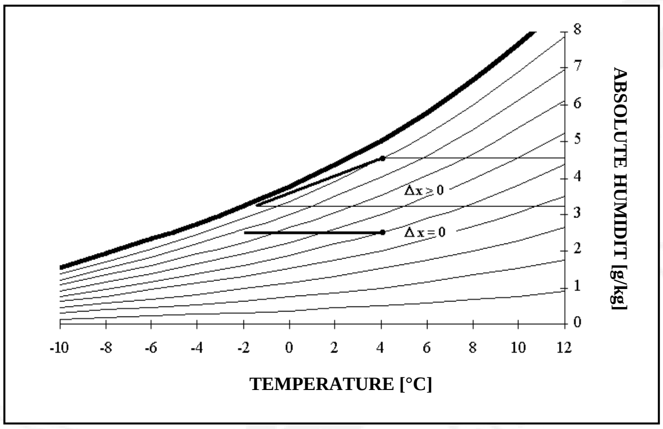


Fig. H.1 – Air trasformations on the evaporator surface for different relative humidity values.

The Figure highlights another very interesting aspect for understanding the problem: since the absolute humidity variation of the transformation, Δx, substantially depends on the slope of the saturation curve and therefore, under identical conditions, it is greater for higher air temperature levels, the amount of ice that is formed on the coils, equal to the product of the mass flow rate times Δx, decreases with the reduction of the air temperature value.
In the case of part load operations, the refrigerant flow rate in the evaporator is reduced, and therefore the thermal exchange, the evaporation temperature rises and in consequence also the surface temperature of the evaporator goes up and the phenomenon of frost formation is reduced. Figure H.2 shows a comparison between the areas inside the device where frost formation on the evaporator develops either in full load or in partial load: frost formation develops only with RH values above 50% and the area is dramatically reduced in the case of a part load operation in the single circuit. The amount of ice is not formed homogenously over the entire surface, rather it is very heavy in the point of initiation of the phenomenon on the saturation curve, and is reduced when moving away from this point and finally disappears along the lower part of the curve: the higher the temperature value of frost initiation, the greater the amount of ice that is formed on the coil. Therefore, the working of evaporation air heat pumps is not really subject to low temperature values, rather it is subject to high relative humidity values associated with low air temperature values, for example around 5 °C, as is the case in in cities in the Po river valley.

H.3 – Consequences of frost formation

The phenomenon of frost formation, if not under control, rapidly results in the halting of the heat pump, due to low pressure at evaporator. As a matter of fact, the ice layer that develops on the surface of the coil impairs both the thermal exchange functions, in that it acts as insulation, and the area where the stream of air flows, which increases pressure drop on the air side: it is as if the exchange surface gets smaller while the phenomenon of the ice formation develops. This double effect produces a drop in evaporation temperature values and, consequently, the result is also a drop in the surface temperature with consequent increase in Δx. In short, the greater and thicker the ice layer, the smaller the exchange surface and the greater the frost formation.

Once initiated, the phenomenon of frost formation exponentially increases in intensity until the coil is completely covered by ice and automatic safety devices halt the heat pump. To avoid this inconvenience, air-source heat pumps are provided with defrosting cycles.

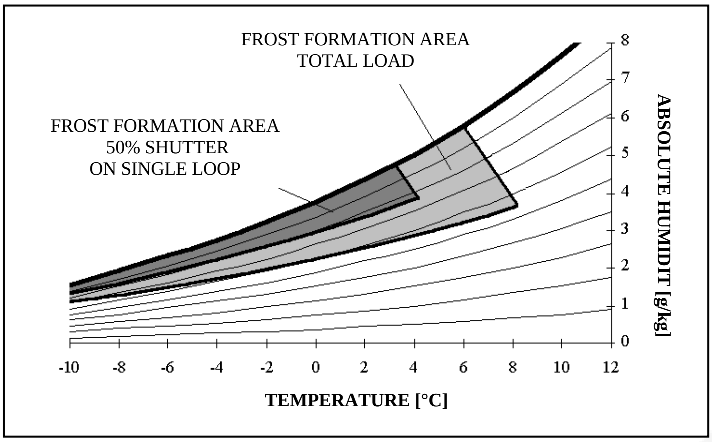


Fig. H.2 – Comparison between the areas where frost is formed on evaporator coils in the case of full load and in the case of part load.

H.4 – Defrosting systems

Generally speaking, it can be said that heat pumps require one defrosting cycle every time that 1 gram of water vapor condensate, for each kilogram of air flowing through the evaporator, is transformed into frost. In practice, if this phenomenon is expressed in terms of specific humidity variations between inlet air and exhaust air in the evaporator coil, the cycle has to be performed for each Δx = 1 g/kg of condensate transformed into ice.

There are typically two systems that are used to perform the defrosting: the reversal of the cycle and the hot gas injection. The reversal of cycle, by far the more widely used system, consists of swapping the cycle, from winter to summer operation: in this way, the exchange coil, used again as a condenser, is crossed by a hot fluid that melts the ice which gets detached from the surface. Also with the hot gas injection system the coil is filled with a hot fluid, but in this case, the refrigerating cycle is not completely reversed.

One has to allow for the effects of defrosting to avoid making serious mistakes both in terms of energy, and in the right sizing of the heat pump.

From the energy point of view, independently of the system used, defrosting cycles are not “free” for the heat pump. Firstly, defrosting means a loss of energy due to the combined effect of the consumption of the compressor during the cycle and the heat amount that is drawn away from the primary heating circuit by the condenser that becomes temporarily an evaporator. This loss may be more or less equivalent to 10% for each cycle. Moreover, defrosting entails a greater wear for the compressors, because during the reversal of the cycle, the two parts of the loop – high pressure and low pressure – are swapped, which creates a thermomechanical stress condition for all organs, problems of oil leakages from the compressor pan and, above all, the risk of a “liquid stroke” due liquid suction for the compressor.

From the sizing point of view, defrosting cycles do not have an impact on the efficiency of the heat pumps, but they do have an impact on the energy output, so that heat pumps have to be oversized to guarantee the necessary energy delivery. For example, if two defrosting cycles an hour are required, considering that, as mentioned, each cycle results in a 10% energy loss, the energy produced is l’80% of the amount of energy that would be produced under the same conditions without defrosting cycles: a heat pump with an output of 100 kW, produces only 80 kWh if it undergoes the process of defrosting for two cycles.

Figure H.3 reports *COP* curves with full load for a high efficiency air source heat pump (EUROVENT Class A) in function of air temperature, for relative humidity values equal to 90% and 70% and below 50% (sensible only exchange) and for temperature values at condenser exit equal to 35 °C e 45 °C. The discontinuity points of the curves with *RH* > 50% represent the situations where surface temperature values of the coil become negative and the condensate is transformed into ice. For the heat pump under discussion, at high efficiency, transition temperatures are 4 °C and 5 °C for a relative humidity of 90%, versus 45 °C and 35 °C, and 6 °C and 7 °C for a relative humidity of 70% at the same exit conditions.

Lastly, the logic that controls defrosting cycles is very important. At present, almost all heat pumps work with dynamic control logics, capable of detecting the actual presence of ice formation.

Table H.1 reports average *COP* values for monthly operation in some of the main Italian cities, both for daytime and for nighttime, that can be obtained with a heat pump with a Class B EUROVENT efficiency with nominal *COP* of 2,9. The values refer to operation at full load that is for heat pumps with one step part load for each refrigerating circuit (On – Off operation), to make clear what the weight of air on *COP* is. Heat pumps equipped with more shutters reach better results. Average *COP* values are above 2,5 in all cities. One interesting case is the city of Bolzano: thanks to the cold but dry climate, in the depths of winter (December, January and February) *COP* values are higher than in other cities in the North of Italy, that have a milder but more humid climate (Torino, Milano, Venezia), precisely because the number of defrosting cycles is limited. The values in the Table apply to water produced at 45 °C and to machines equipped with dynamic logic defrosting systems.

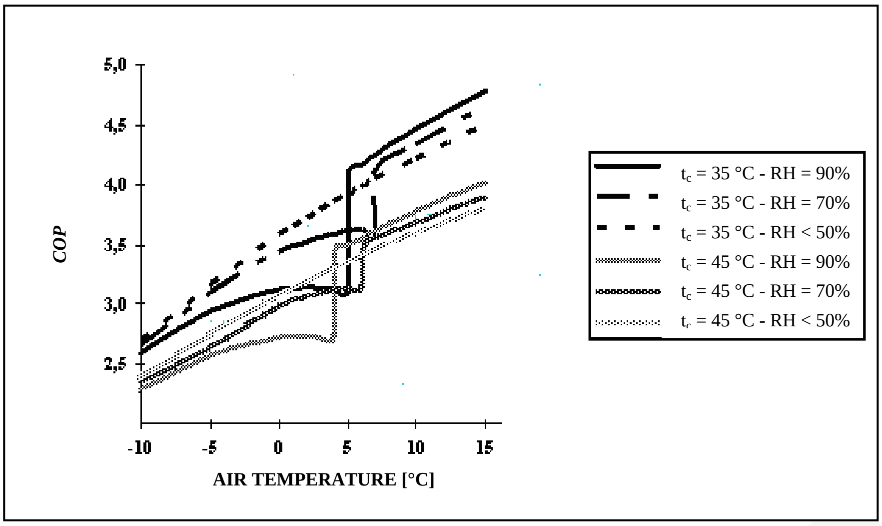


Fig. H.3 - *COP* of an air-water heat pump in function of air temperature values for different relative humidity values, *RH*, and of the temperature at condenser exit, t_c. For *RH* < 50% there is only one sensible exchange.

Tab. H.1 - COP values at full load with class B EUROVENT efficiency heat pumps in some Italian cities

	OCTOBER		**NOVEMBER**		**DECEMBER**		**JANUARY**		**FEBRUARY**		**MARCH**	
	day	***night***	**day**	***night***	**day**	***night***	**day**	***night***	**day**	***night***	**day**	***night***
Torino	3,5	*3,4*	3,0	*2,6*	2,6	*2,5*	2,7	*2,6*	2,6	*2,6*	3,2	*2,9*
Milano	3,6	*3,5*	3,1	*2,9*	2,6	*2,6*	2,6	*2,5*	2,7	*2,6*	3,2	*3,0*
Bolzano	3,4	*2,9*	2,8	*2,6*	2,9	*2,8*	2,8	*2,7*	2,9	*2,8*	3,2	*2,9*
Venezia	3,6	*3,5*	3,3	*3,3*	2,7	*2,6*	2,7	*2,6*	2,7	*2,6*	3,1	*2,9*
Genova	3,7	*3,6*	3,4	*3,4*	3,4	*3,3*	3,2	*3,1*	3,2	*3,1*	3,4	*3,4*
Bologna	3,6	*3,5*	3,1	*2,6*	2,6	*2,5*	2,6	*2,5*	2,5	*2,5*	3,2	*2,9*
Ancona	3,7	*3,6*	3,5	*3,5*	3,1	*2,9*	3,3	*3,1*	3,1	*2,7*	3,4	*3,4*
Roma	3,7	*3,6*	3,5	*3,5*	3,2	*3,2*	3,1	*2,6*	3,2	*2,8*	3,3	*3,1*
Pescara	3,7	*3,6*	3,5	*3,4*	3,1	*2,7*	3,0	*2,6*	3,1	*2,7*	3,2	*2,9*
Napoli	3,7	*3,6*	3,5	*3,5*	3,4	*3,4*	3,2	*2,8*	3,1	*2,7*	3,4	*3,4*
Cagliari	3,7	*3,7*	3,6	*3,6*	3,5	*3,5*	3,2	*3,2*	3,2	*3,2*	3,5	*3,4*