STEADY STATE CALCULATION OF HEAT PUMP LONG TERM PERFORMANCES BASED ON UTILIZATION FACTOR METHOD

P. Romagnoni, L. Schibuola,
Istituto di Fisica Tecnica - Università di Padova - Via Venezia, 1 - 35131 Padova (Italy)

Abstract
A hourly simulation in dynamic conditions of building-plant system is able to give a rigorous evaluation of seasonal performances of a heat pump using outside air as cold source. However in professional applications we have often the necessity of speed estimation of the energetic and economic opportunities of its introduction whose advantages are not always real and sure. For this reason various simplified methods have been proposed. A new procedure to estimate seasonal performances of an air source heat pump is proposed here, based on utilization factor method. This method, elaborated by CEN TC 89, permits to evaluate climatization energy requirements of buildings by steady-state calculations referring to monthly average weather data.
The results obtained by this procedure are compared with those obtained by comprehensive dynamic simulations for different climate conditions.

1. Introduction

The real convenience of the installation of a heat pump in a building climatization plant must be verified in each application case. This analysis is particularly critical when the cold source of the heat pump is the outside air because of its variability during the heating season.
The use of comprehensive computer programs able to simulate the building-plant system behaviour in dynamic conditions permits to obtain a correct evaluation of long term performances of the machine. But these calculations result binding and extremely laborious especially for professional applications. Therefore various simplified methodologies have been proposed to obtain quickly a reliable estimation. Among all, the Bin Method [1] is perhaps the most famous. The fundamental limit of this method is to neglect solar radiation contribution through glass and on opaque surfaces. The consciousness of this gap has caused the modification of the original Bin Method [2]. The essence of the modification consists in estimating this solar contribution by a linear function of outdoor air temperature. But, as the comparison with comprehensive computer programs has point out, the modification of the method doesn't involve always an improvement of the reliability of the results because of the arbitrariness of solar effect linear correlation [3].
On the other hand the most recent proposals of European Standard regarding the calculation of energy requirement for building heating have led to the elaboration of a simplified method named utilization factor method [4], [5]. In fact the contribution of solar radiation and internal heat gains in this case are taking into account by a suitable utilization factor. In this paper a procedure for a simplified estimation of the seasonal performances of an air source heat pump is presented which is a logical consequence of the CEN proposal.

The results obtained by this way are here compared with those by dynamic simulation with ASHRAE Transfer Function Method for some typical application cases and in different geographical localities.

2. The utilization factor method

This calculation method is based on a steady-state energy balance of each single zone of the building referred to monthly values where the presence of the utilization factor permits to consider also the dynamic effects of internal heat gains and solar contributions. In this way the heating monthly requirement Q_h of the zone is calculated by the following equation:

$$Q_h = Q_L - \eta_u (Q_I + Q_s) \qquad (1)$$

where:
Q_L = monthly energy loss due to ventilation and conduction through the building envelope
Q_s = monthly solar gains
Q_I = monthly internal heat gains
η_u = utilization factor

In presence of intermittent or reduced heating the previous algorithm is modified in this way:

$$Q_h = k \,[\, F_{il}\, Q_L - \eta_u\, F_{ig}\, (Q_I + Q_s) \,] \qquad (2)$$

where:
k = coefficient for different heating mode (k=1 intermittent heating)
F_{il}, F_{ig} = correction factors evaluated on the basis of heating schedule and zone thermal inertia (F_{il}, F_{ig} = 1 for continuous heating).

In any case utilization factor η_u is calculated in the following way:

$$\eta_u = \frac{1 - \gamma^{\tau}}{1 - \gamma^{\tau+1}} \qquad \text{if} \quad \gamma \neq 1 \qquad (3)$$

$$\eta_u = \frac{\tau}{\tau+1} \qquad \text{if} \quad \gamma = 1 \qquad (4)$$

$$\gamma = \frac{Q_I + Q_s}{Q_L} \qquad (5)$$

where:

$\tau = \tau_0 + t_c/t_0$
$\tau_0 = 1$
$t_0 = 16$
t_c = time constant of the building in hours: $t_c = \dfrac{C}{H_k\, 3600}$

C= thermal capacity of the zone

H_k = global loss coefficient: $H_k = \dfrac{QL}{86400\, N\, \Delta\vartheta}$

N = number of days in the month (normally assumed 30 for all months)
$\Delta\vartheta$ = monthly average temperature difference between inside and outside ambient

A more detailed version of this formulas is presented in Italian National Standard (UNI-CTI 10344). It consists in the separation of the solar contribution Q_{se}, absorbed by the external surfaces of the envelope and transmitted to the interior by conduction, from the solar contribution through windows Q_{si}. In this way relations (1), (2), (5) are changed as follows:

$$Q_h = (Q_L - Q_{se}) - \eta_u (Q_I + Q_{si}) \qquad (6)$$

$$Q_h = k\,[\,F_{il}\,(Q_L - Q_{se}) - \eta_u\, F_{ig}\,(Q_I + Q_{si})\,] \qquad (7)$$

$$\gamma = \frac{Q_I + Q_{si}}{Q_L - Q_{se}} \qquad (8)$$

3. Design data for the application cases.
The reliability of the method has been verified in some application cases rather frequent and so particularly significant. In detail we have considered three situations:
a) an office unit with intermittent heating from 7 AM to 7 PM (12 h)
b) a residential unit with continuous heating
c) a residential unit with intermittent heating from 7 AM to 9 PM (14 h).
The building, sketched in fig. 1, is subdivided in 8 rooms and is exposed, for three sides, to the outside, while the fourth side and the floors are bounded by rooms climatized in the same way of the investigated unit. Typical design data have been assumed in terms of building structures, utilization and management. For the office windows are 50% of external walls, instead for the residential case, windows are reduced to about 12% of the floor area of the relative room. The office is characterized by a forced ventilation equal to 2 vol/h and by the presence of internal heat gains for persons (max. 1 person/10 m^2), lighting (max. 20 W/m^2, fluorescent) and computers (max. 12 W/m^2) during plant working period. For the residential unit, internal heat gains have been neglected and we have an average natural infiltration estimated equal to 0.5 vol/h.

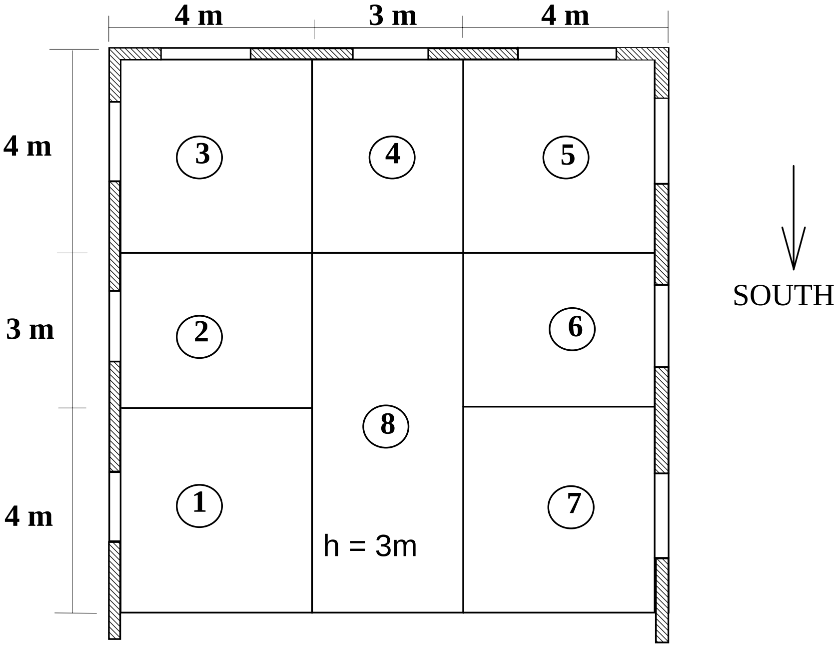

Fig. 1 Sketch of the residential unit

Heating season analysed is 6 months long, from October to March, in this period the plant ensures an internal design temperature of 20°C. First the heat requirement calculations have been carried on by dynamic simulation using ASHRAE Transfer Function Method. In order to extend the analysis to the climatic conditions of 15 Italian cities distributed along the national territory, we have chosen to utilize monthly mean days available in technical literature [6]. From the same meteorological data, for each month, a mean day can be evaluated, characterized by temperatures and solar radiation for various orientations equal to arithmetic means of the same values in the same hour of the day during the whole month. In this way for each month we considered, for simplicity, 30 days equal to monthly mean day. On the basis of few days, it is possible to obtain easily a dynamic analysis referred to the whole heating season with an error tolerance normally acceptable for professional purposes.

We have considered three types of heat pumps which uses outside air as cold source: air/water with on/off control, air/air on/off, air/air with the control based on two different speeds to drive the compressors. In order to evaluate heat pump performances we have used working curves from a builder of small capacity heat pumps, absolutely standard, with reciprocating compressors and refrigerant R22.

For example, in fig. 2 the trends of COP and real capacity of an air/water heat pump are reported as functions of outside air and for a supply water temperature of 45°C. In fig. 3 one can see the part load coefficient PLF to correct the previous COP in presence of part load with on-off control of the machine. This penalization is due to the intermittent working of the compressors. But actually a limit to the maximum number of on-off cycles frequency is always present by the protection systems of electric motor or caused by the effect of the inertia of the machine and of the plant. Normally we can therefore estimate the maximum possible penalization due to part load with the corrective coefficient indicated for a 50% of the full capacity. For our heat pumps, then the value of this coefficient is 0.85 with on-off control and 0.925 with two speeds [0.925=1-0.5x(1-0.85)].

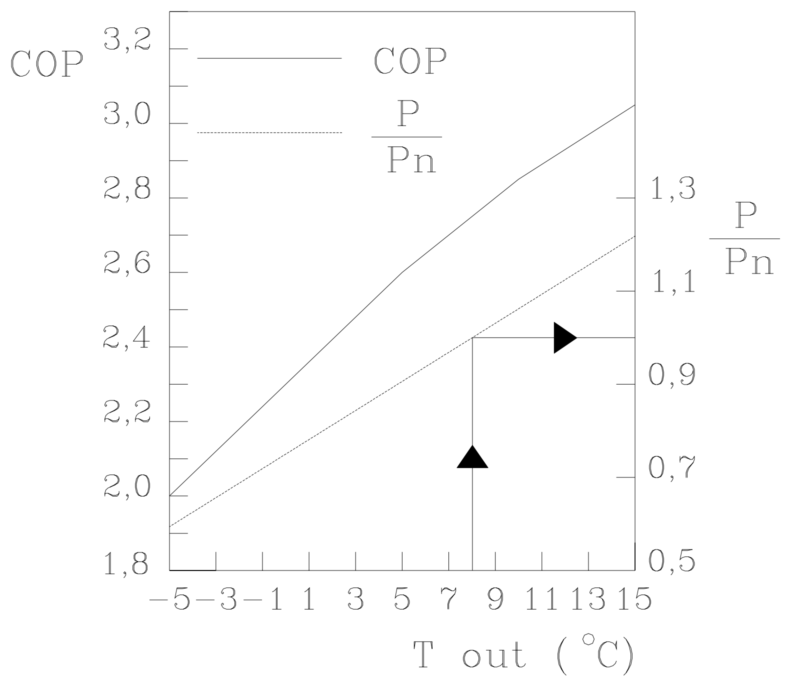

Fig. 2 Coefficient of performance (COP) and ratio between real and nominal capacity at 8°C (P/Pn) as a function of outside air temperature (Tout). This data are referred to an air/water heat pump with a supply temperature of 45°C.

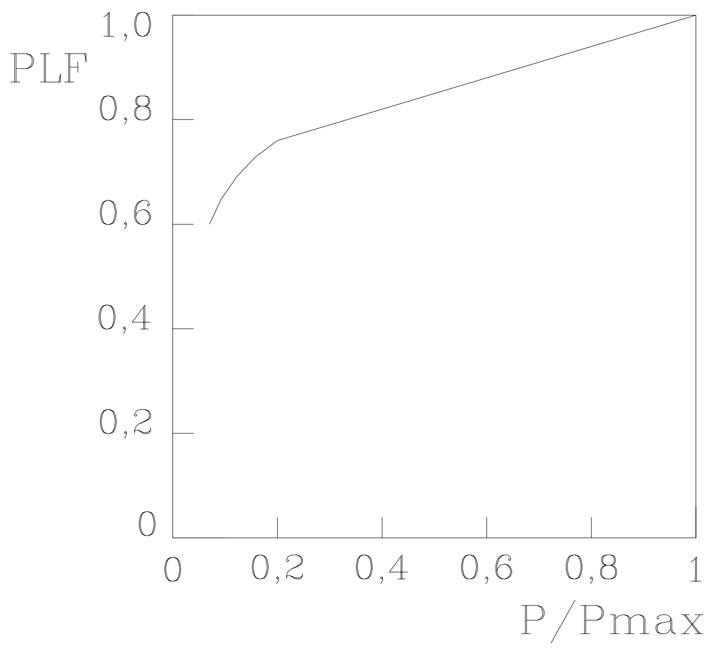

Fig. 3 Part load factor (PLF) as a function of the ratio of part load to full load capacities (P/Pmax) of the heat pump.

We have also taken into account the effect of the defrosting of evaporator coil, by cycle inversion, which is required when the outside temperature is less than 5°C. The subsequent simulations have however shown a modest influence of defrosting on monthly performances.

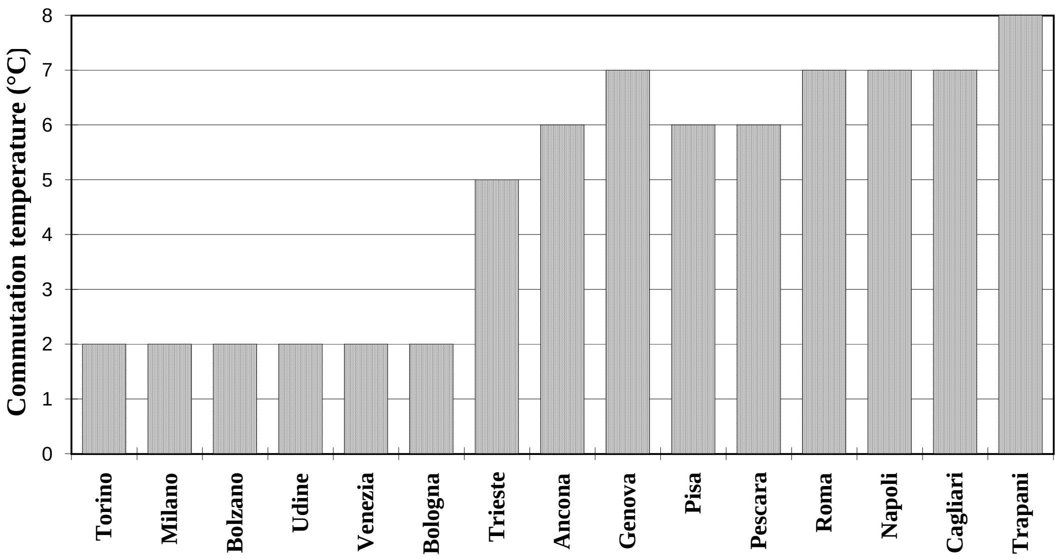

Fig.4 Commutation temperatures between boiler and heat pump working for the 15 Italian localities here considered.

In order to ensure acceptable COP, the air source heat pump normally works only when outside air temperature is greater then a fixed value named commutation temperature. Under this value a boilers is used, above heat pump is able to provide the whole heating load. In fig. 4 commutation temperatures between heat pump and boiler working assumed in this study are reported for the various Italian cities.
The choice of this temperature value involves comparisons between electric and fuel energy costs but also considerations about the money available to install a heat pump of a suitable capacity. In conclusion the final design decision about the value of the commutation temperature can be very different also for the same locality on the basis of criteria also subjective. In this study they have been therefore chosen in a way absolutely arbitrary and, if anything, in some cities we have privileged the need to avoid the easiest case of a heating completely provided by the heat pump. This reason explains the high commutation temperatures considered in the localities with milder climate.

4. The calculation of the covered fraction

As the heat pump works only in some period, it is therefore necessary to estimate the fraction of total heating requirement really covered by the heat pump.
For this aim we introduce the term covered fraction F which can be defined in a mathematical way as follows:

$$F = \frac{Q_{hp}}{Q_{tot}} \qquad (9)$$

where Q_{hp} is the heat really provided by the heat pump and Q_{tot} is the total heating requirement in the same period. Depending on this period, when the energy fluxes are calculated, we can have an instantaneous value of the fraction F or a mean value over a time longer, for example a monthly mean value or a seasonal mean value, if referred to the entire heating season.

A correct design of the heating plant, where the air source heat pump works in alternative to the boiler when climatic conditions are favourable, must foresee a nominal capacity of the machine able to satisfy the whole building requirement from the commutation temperature. The calculation of the fraction is reduced than to the calculation of the ratio of partial heating requirement during heat pump working, i.e. when outside temperature is not less than that temperature limit, to total heating requirement.

In order to obtain a simplified method to estimate this fraction in accordance with utilization factor method we started with the comparison of the results by CEN proposal with those of dynamic simulations. In the case of the office building, with remarkable internal heat gains, the European procedure is resulted indeed deficient and needy of an integration when solar contribution is strongly increasing [7]. For this reason we have considered only 6 cities in the northern part of Italy for the office case. Instead for the two residential cases, we have considered all the 15 cities.

After the verification of the substantial coincidence of the calculated building requirements, we have tried to find an algorithm able to estimate the fraction covered by the heat pump. The simplest procedure can be to express this fraction as the ratio of the heating period when outdoor air temperature is above the commutation temperature to the total heating hours. Naturally the hourly requirement is not constant but instead it is decreasing in the period more favourable of the day. In this way the fraction F is overvalued. In the end the algorithm more satisfactory for the coincidence with the results of dynamic simulations and for its easily application seems to be:

$$F = \frac{k\,[\,F_{il}\,(Q_L - Q_{se})_{hp} - \eta_u\,F_{ig}\,(Q_I + Q_{si})_{hp}\,]}{k\,[\,F_{il}\,(Q_L - Q_{se})_{hea} - \eta_u\,F_{ig}\,(Q_I + Q_{si})_{hea}\,]} \qquad (10)$$

Here Qse, QI, Qsi are evaluated in the monthly mean day, starting from the hourly value averaged over 24 hours and then multiply for the real working hours of the heat pump (hp) in the numerator and for the total heating hours (hea) in the denominator. In the same manner QL is calculated only in the working hours of the heat pump (hp) or in the total heating period (hea). With regard to the original method, this calculation requires now only the knowledge of the hourly trend of outside air temperature in the monthly mean day. You can note that the difference between the expression at the denominator and that one proposed by utilization factor method to estimate total monthly requirements (7) is caused by the necessity to have F equal to unit when the heat pump can work always during heating period (here for example in Trapani).

For example the evaluation of F for the month of January in Venice for residential building with intermittent heating is reported here in detail. By the application of the utilization factor method described in paragraph 2 and using the same symbols, in this case we have:

Q_{se} = 22.5 10^6 J/month
Q_{si} = 13.7 10^8 J/month
Q_I = 0
H_k = 145.3 W/K
k = 1 (intermittent heating)
F_{il} = 0.853
F_{ig} = 0,919
γ = 0.2091 and τ = 7.87 h, by equation 3 we calculate η_u = 0.9999.

In this way the heating monthly requirement Q_h is 43.2 10^8 J/month.

From the hourly trend of outside temperature in January, monthly mean day in Venice, we have that for 11 hours this temperature is greater of the chosen commutation temperature (2°C). In this period we can calculate a mean temperature of 4.4 °C, in the 14 hours of heating period the mean outside temperature is 3.7°C. The energy losses Q_L can be then calculated in these two daily intervals:

$$(Q_L)_{hp} = 3600\ 11\ N\ H_k\ (20-4.4) = 26.9\ 10^8 \text{ J/month}$$

$$(Q_L)_{hea} = 3600\ 14\ N\ H_k\ (20-3.7) = 35.8\ 10^8 \text{ J/month}$$

By application of equation 10 then we have:

$$F = \frac{0.853 \cdot (26.9 \cdot 10^8 - 22.5 \cdot 10^6 \cdot \frac{11}{24}) - 0.919 \cdot 0.9999 \cdot (0 + 13.7 \cdot 10^8) \cdot \frac{11}{24}}{0.853 \cdot (35.8 \cdot 10^8 - 22.5 \cdot 10^6 \cdot \frac{14}{24}) - 0.919 \cdot 0.9999 \cdot (0 + 13.7 \cdot 10^8) \cdot \frac{14}{24}} = 0.74$$

By multiplying F for the monthly heating requirement Q_h, we calculate the heat provided by the heat pump in January. The seasonal fraction F can be then evaluate as the ratio of the sum of the heat pump contributions estimated for all months to the total seasonal heating requirement.
In fig. 5 the seasonal fractions F obtained in this way are compared with those from dynamic simulations for the three different application cases and for the various geographical localities.
In fig. 6 the relative percentage deviations are reported. They are referred to the results of dynamic simulation. You can notice that these differences are normally less than 10% (except Bologna, office case). With the residential unit then they are usually less than 4% (except Bolzano). On the other hand the same forecast of the heating requirements on the basis of utilization factor method results more precise with reduced heat gains. The deviations are however normally acceptable for professional purposes.

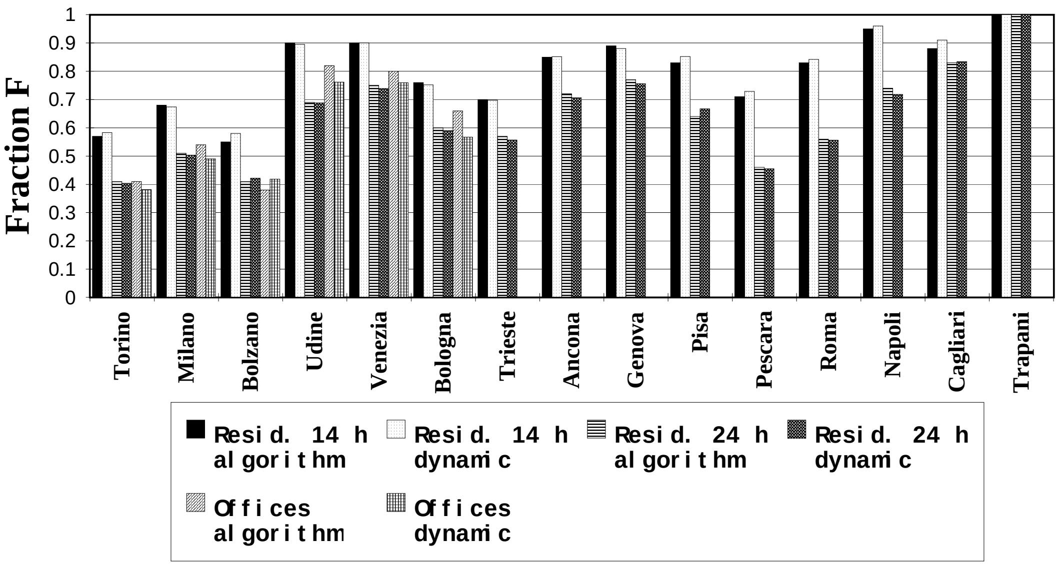

Fig. 5 Comparison between seasonal requirement fraction F covered by the heat pump calculated with the proposed algorithm or with dynamic simulation in the cases studied and for the various cities.

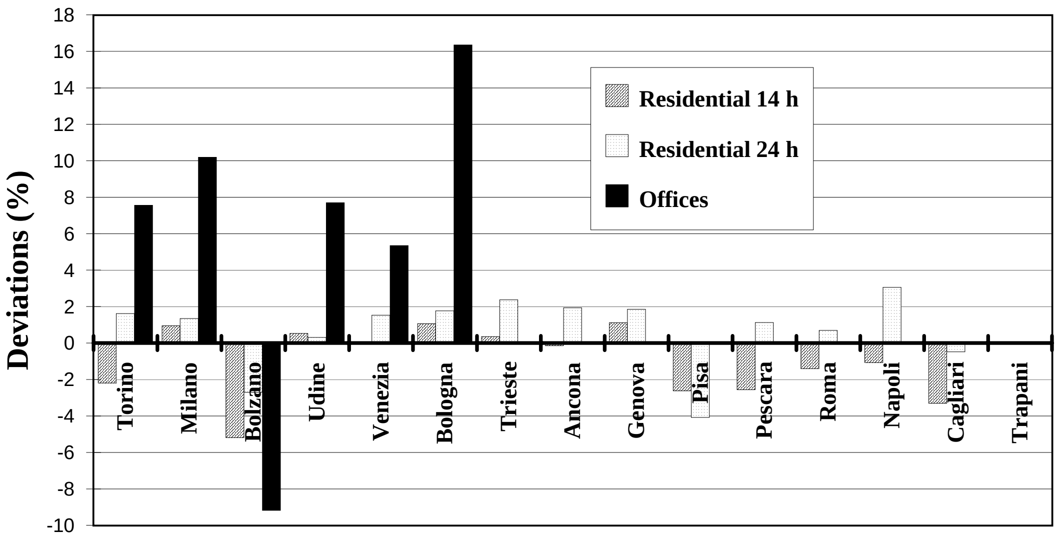

Fig. 6 Percentage deviations of seasonal requirement fractions F covered by the heat pump calculated by the proposed algorithm in the various cases studied. These deviations are referred to the results obtained by dynamic simulations.

5. The calculation of seasonal average COP

Under the same other conditions, the real behaviour of the air source heat pump results variable during the working hours on the basis of the outdoor air temperature and also of the load level of the machine. In order to calculate correctly a mean coefficient of performance (COP) is then necessary to evaluate the influence of these two parameters during heating period. In accordance with the CEN method, the calculation of COP is developed on a monthly basis. In this case we can utilize the mean value of outside temperature averaged only the real working period of the heat pump. This temperature can be estimated in the monthly mean day by considering only the hours when the temperature is above the commutation temperature during the daily heating period. Referring to the CEN method, the only further data necessary are again the hourly trends of outside temperature in the monthly mean days of the heating season.

As regard to the part load influence, because the installed capacity is able to satisfy the heating need at the commutation temperature, it means that the machine works for the most time at part load. In effect the simulations have shown that the mean corrective coefficient of the COP can be well approximated with that in correspondence with the maximum intermittence of the compressors (in our cases 0.85 with on-off control and 0.925 with the two speeds). At monthly level if the COP and fraction F are known, it is then possible to evaluate electric consumption of the heat pump. By adding these monthly values correspondent seasonal values can be obtained. The seasonal COP is then the ratio of the total heat provided by heat pump during winter to the electric absorption of the machine in the same period. In fig. 7 the mean coefficients of performance obtained with the proposed procedure are reported together with the correspondent values provided by dynamic simulations of residential intermittent heating and for the three heat pumps studied. The precision of this calculation (mode c) has been compared with a simpler method which uses only the monthly outside temperature averaged the whole day long and neglect the influence of part load working (mode a). In the fig. 8, 9, 10 percentage deviations referred to dynamic results of the COPs obtained by these two modes are reported for the three application cases. Mode c usually provides results more precise and acceptable than mode a. The differences between real mean outside temperature during heat pump working and monthly mean outside temperature really affected the COP. The same is for the part load mean factor which cannot be neglected.

In pursuit of possible simplifications, we have considered also an intermediate possibility (mode b): i.e. the calculation of COP on the basis of the whole month average outside temperature as in mode a, but taking into account also partial load effects as in mode c. In fig. 11, 12, 13 the percentage deviations of COP referred to dynamic simulations are compared again, but this time considering mode b and c. The procedure b provides better results than mode a but clearly worse than mode c. Its use therefore is not justified because the need of hourly trends of outside temperature in the monthly mean day is anyway just present for the calculation of the fraction F.

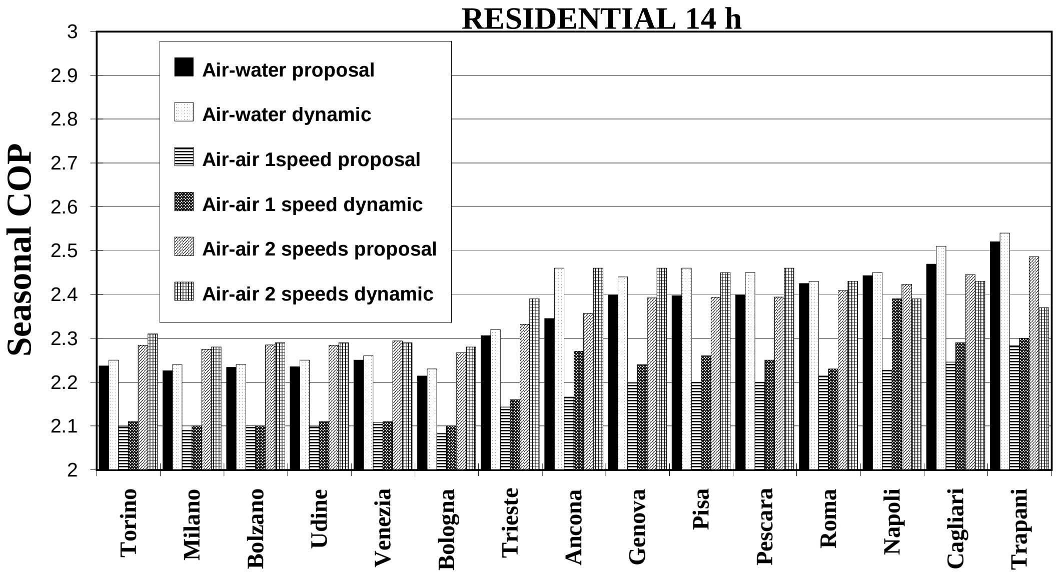

Fig. 7 Seasonal COP evaluated by the proposed method and with the dynamic simulation in the residential case with intermittent heating for the three types of heat pump and in the various cities.

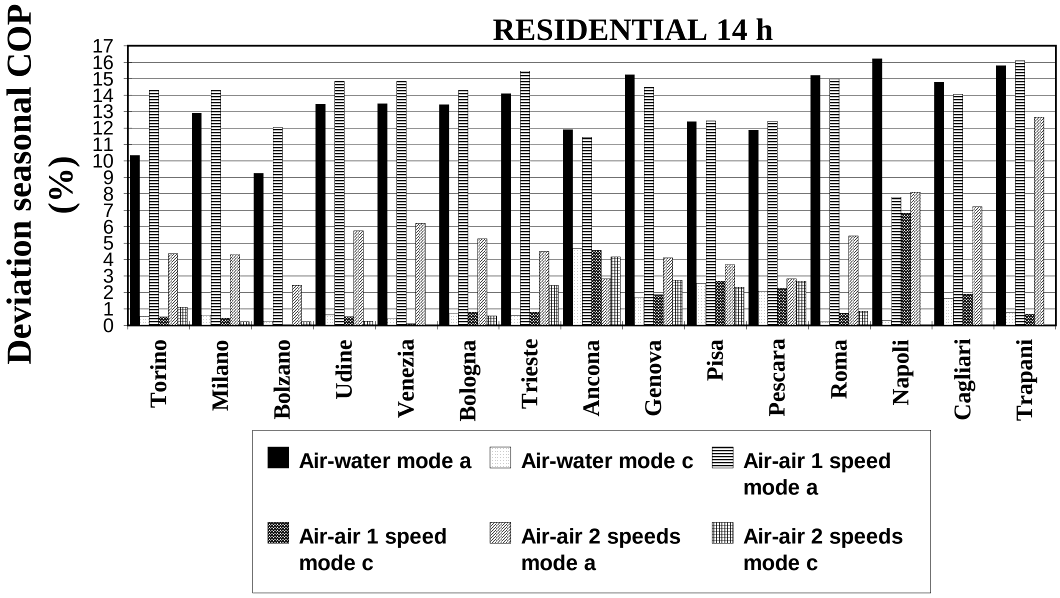

Fig. 8 Percentage deviations of seasonal COP calculated by mode a or c for the three types of heat pump in the residential case with intermittent heating. The deviations are referred to the results of dynamic simulations.

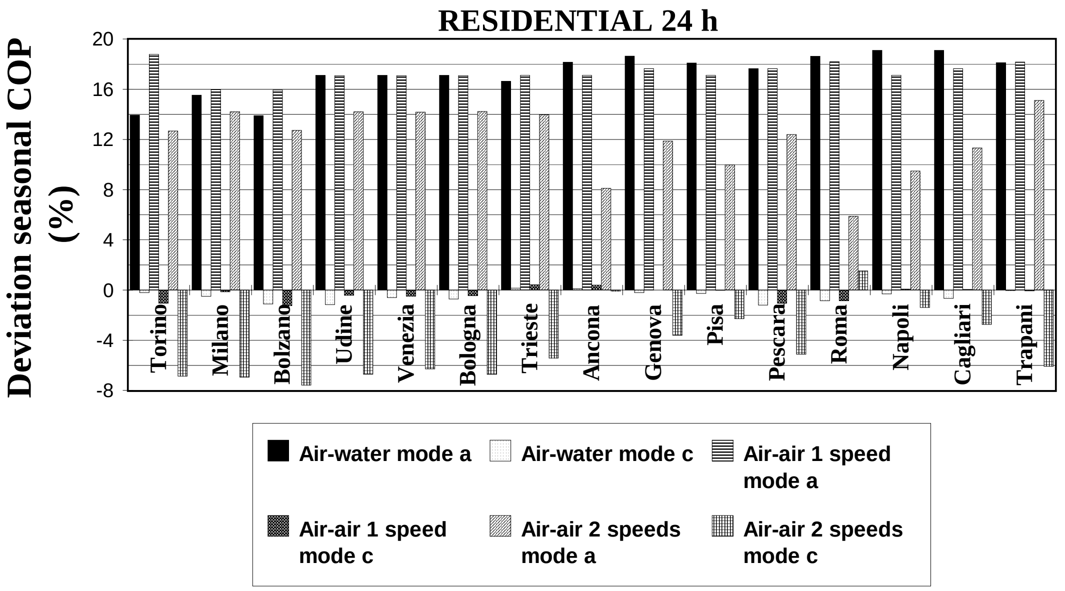

Fig. 9 Percentage deviations of seasonal COP calculated by mode a or c for the three types of heat pump in the residential case with continuous heating. The deviations are referred to the results of dynamic simulations.

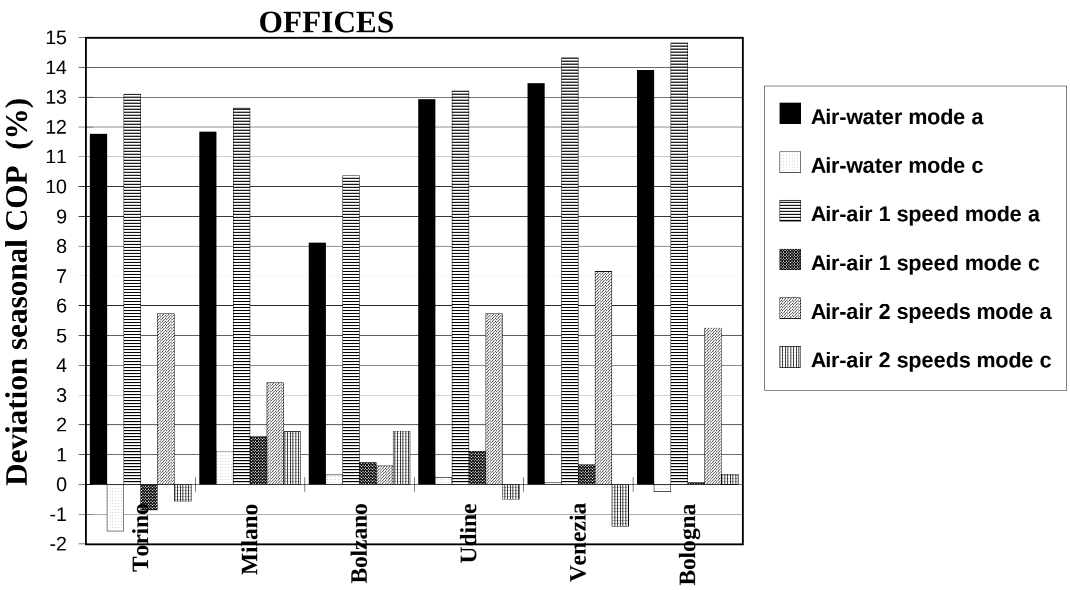

Fig. 10 Percentage deviations of seasonal COP calculated by mode a or c for the three types of heat pump in the office case. The deviations are referred to the results of dynamic simulations.

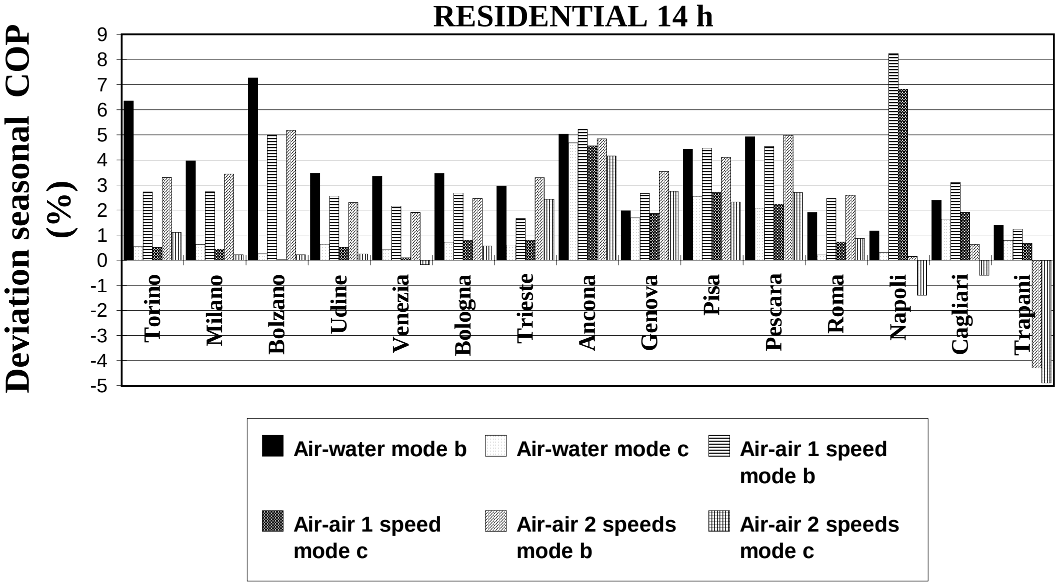

Fig. 11 Percentage deviations of seasonal COP calculated by mode b or c for the three types of heat pump in the residential case with intermittent heating. The deviations are referred to the results of dynamic simulations.

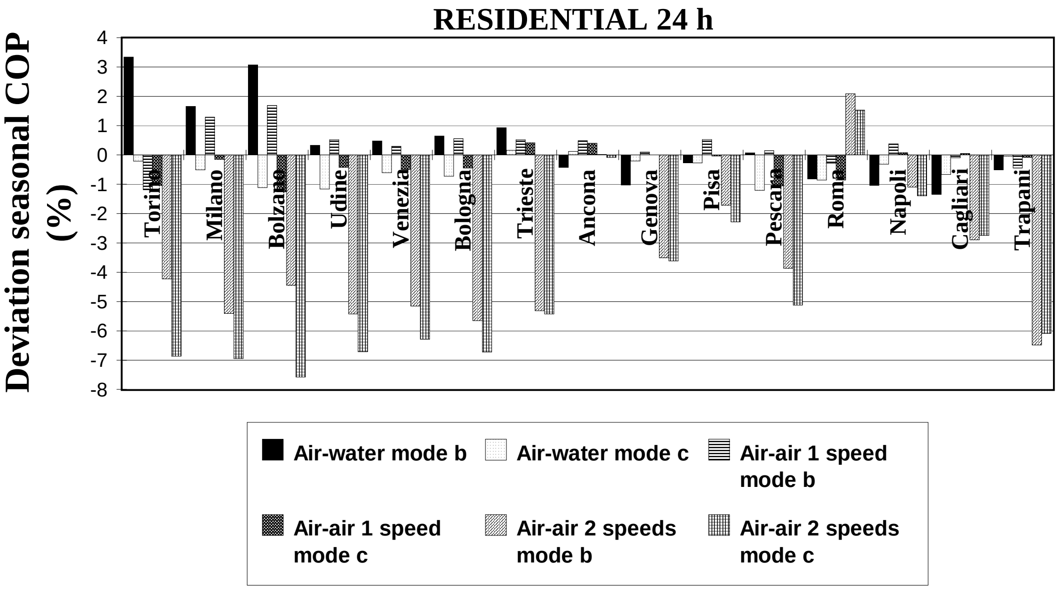

Fig. 12 Percentage deviations of seasonal COP calculated by mode b or c for the three types of heat pump in the residential case with continuous heating. The deviations are referred to the results of dynamic simulations.

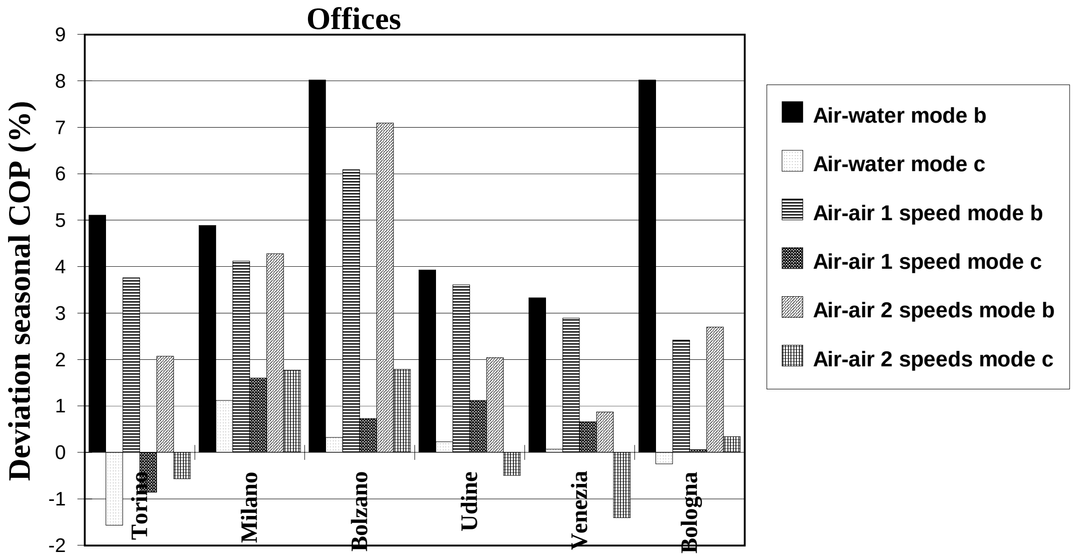

Fig. 13 Percentage deviations of seasonal COP calculated by mode b or c for the three types of heat pump in the office case. The deviations are referred to the results of dynamic simulations.

6. Conclusions

A new proposal has been presented for the calculation of seasonal COP and of the fraction of the heating requirement of a building covered by a heat pump driven by electric motor which uses outside air as cold source. The procedure develops as a consistent extension of utilization factor method proposed by CEN TC89 for the evaluation of building heating needs. As regard to this one, the proposed procedure requires, in addition, only the availability of the hourly trends of the outside temperature in the monthly mean days.

References

[1] ASHRAE Fundamentals Handbook, chapter 26, ASHRAE, New York, 1989.

[2] ASHRAE, Simplified Energy Analysis Using The Modified Bin Method, ASHRAE report, Atlanta 1983.

[3] L. Schibuola, Comprehensive Computers Programs and Simplified Procedure for Building-Plant System Behaviour Evaluations, CIB Workshop Proceedings, Udine 1995.

[4] CEN/TC89/WG 4, prEN 832, Thermal Performance of Buildings Calculation of Energy Use for Heating - Residential Buildings, final draft, Brussels 1995.

[5] CEN/TC89/WG 4, Thermal Performance of Buildings - Calculation of Energy Use for Heating, Brussels 1995.

[6] C.N.R.-PFE, Dati Climatici per la Progettazione edile ed impiantistica, Appendice 1 alla Guida per il Controllo Energetico della Progettazione, Roma febbraio 1982.

[7] L. Agnoletto, P. Romagnoni, L. Schibuola, The Evaluation of the Utilization Factor for Heat Gains: a Critical Analysis of the CEN Proposal, 1996 International Symposium of CIB W67 Proceedings, Wien 1996.