Quasi-steady state and dynamic simulation approaches for the calculation of building energy needs: Part 1 thermal losses

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Abstract

In order to assess the coherence between the dynamic simulation and the EN ISO 13790:2008 quasi-steady state method, the authors, differently from previous works in literature, analysed the discrepancy sources directly for thermal losses and gains instead of considering only the final result in term of energy needs. In this first part, the deviations between the thermal losses are evaluated by means of an extensive use of simulation. More than 2000 configurations obtained by the factorial combination of different values for the building shape, envelope insulation and composition, window type and size, ventilation rate and climatic conditions, have been considered to identify the most important discrepancy sources and to improve the correspondence between simulation and quasi-steady state methods.

1. Introduction

According to the Energy Performance of Building (EPB) Directive 2010/31/EU and the former 2002/91/EC (European Parliament, 2010; European Parliament, 2002), in order to implement the energy labelling of buildings, the energy performance can be evaluated either with analytical approaches or with enhanced simulation tools. As observed by Tronchin and Fabbri (2008), the coherence of the methods is of crucial importance in order to obtain a perception of the reliability of this instrument by the market and to ensure the EPB Directive effectiveness. Moreover, the European Standard EN ISO 13790:2008 (CEN, 2008) suggests to use the dynamic simulation in improving and tuning the proposed quasi-steady state method, by refining the estimation of the utilization factor (i.e., the dynamic parameters that reduce the thermal gains for heating need calculation and the thermal losses for cooling). The utilization factor is considered as a function of the ratio between the thermal losses and the thermal gains. Those are calculated through conventional expressions. Extending the method of studying the dynamic factor by van Dijk and Arkesteijn (1987), the Standard proposes the recourse to dynamic simulation also in determining the thermal losses and gains. Many authors have already made some efforts in calibrating the EN ISO 13790:2008 approach, such as Jokisalo and Kurnitski (2007), Corrado and Fabrizio (2007), Orosa and Oliveira (2010) and Oliveira Panão *et al.* (2011). They proposed some changes on the correlations in order to adapt the method to the climatic conditions, especially for the cooling season, and the building stock characteristics in their respective countries but the general problem appears to be still unsolved, as large discrepancies have been found.

The comparison between the dynamic simulation results and the analytical approaches (detailed or simplified) can be helpful in looking for disagreement sources, which can lead to mismatches and errors in the quasi-steady state method refinement. Referring to the thermal losses, among the causes of error or discrepancies, the ones depending on the definition of the boundary conditions and on the calculation of the thermal losses appear to play a crucial role (Judkoff *et al.*, 2008).

For the transmission losses, which represent the first component of thermal losses, the quasi-steady state model linearizes and considers the internal long wave radiation exchanges in parallel with the convection exchange with the air node, assuming an equivalent operative temperature setpoint. This is a weighted average of the air temperature of the

conditioned zone and the mean radiant temperature of the envelope delimiting the zone itself. In contrast, many of the simulation codes perform a detailed analysis of the internal long wave radiation exchange and refer to an air heat balance approach. This method considers an air temperature setpoint, as it is generally more reliable when simulating real operative conditions.
As regards the ventilation losses, the actual driving temperature difference is given by the one between the internal and external air temperatures. Using an operative setpoint for ventilation losses evaluation, as indicated by the quasi-steady state approach of the technical Standard, leads to incorrect results, in particular with large ventilation rates. Previous studies (Pietrzyk, 2010) have stressed the importance of distinguishing the total losses into those by transmission and the ones by ventilation in defining statistical models. The evaluation of the link between the transmission heat losses and ventilation losses has also been investigated in relation to the reduction of the building energy need (Zhou *et al.*, 2008). Other authors (Soleimani-Mohseni *et al.*, 2006) have studied the ventilation flow rate in order to derive some interactions with the operative temperature.
In a previous work, Gasparella and Pernigotto (2012), the authors compared the thermal losses calculated in accordance with the quasi-steady state method to the ones simulated with TRNSYS 16.1, considering the effect of using an air temperature or an operative temperature setpoint, calculated both with balanced and unbalanced weights. In the present work, in addition to the sources of discrepancies between the air and the operative temperature already analysed, the presence of envelope insulation and the ventilation rate, different amounts of adiabatic surfaces have been considered. A statistically derived correction factor has been determined in order to improve the estimation of an operative equivalent temperature starting from the air temperature setpoint. This allowed to improve the agreement with the results of simulation by TRNSYS and air temperature setpoint.

2. Methods

2.1 EN ISO 13790:2008 model

In accordance with the Standard EN ISO 13790:2008, the thermal losses Q_{ht} through the envelope and by ventilation can be calculated with the Eq. (1).

$$Q_{ht} = Q_{tr} + Q_{ve} \tag{1}$$

The thermal transmission losses through the envelope directly exposed to the outdoors are:

$$Q_{tr} = H_{tr} \cdot \left(\theta_{i,set} - \theta_e\right) \cdot t \tag{2}$$

Considering only dispersions of the heated zone towards the outside environment and neglecting the thermal bridges, the overall transmission heat transfer coefficient is:

$$H_{tr} = H_D = \sum_{k=1}^{n} A_k U_k \tag{3}$$

Due to the adopted simplifications, as the surface long wave radiation exchanges are linearized and superimposed to the convective ones, and in coherence with the definition of H_D and of the thermal transmittance U, the setpoint should be considered an operative temperature. It can be assumed as a weighted average of the air and mean radiant temperatures, considering equal weights if complying with the EN ISO 13790:2008:

$$\theta_{op} = 0.5 \cdot \theta_{air} + 0.5 \cdot \theta_{mr} \tag{4}$$

The ventilation thermal losses are defined as:

$$Q_{ve} = H_{ve} \cdot \left(\theta_{i,set} - \theta_e\right) \cdot t \tag{5}$$

where:

$$H_{ve} = \rho_a c_a \left(\sum_{k=1}^{n} b_{ve,k} \dot{V}_k \right) \tag{6}$$

In the considered cases the temperature adjustment factor $b_{ve,k}$ is 1 because the supply air temperature is equal to the external air temperature.
The use of an operative temperature setpoint also for the calculation of ventilation losses is not strictly correct. For that reason it is expected that some discrepancies arise between simulation and simplified calculation results even if the same operative temperature setpoint is used, and that

those differences increase for increasing ventilation rates. The second issue is that generally the operative temperature is not known when using the method, in particular when air temperature setpoints are considered.

2.2 TRNSYS air heat balance

TRNSYS, as many of the most widespread simulation codes, solves the heat balance calculation with respect to the air node, considering separately the convective and the radiative exchanges. The air node balance is expressed as function of the convective thermal exchanges:

$$\phi_{c,i} + \phi_{ve} + \phi_{gc} + \phi_{sys} = C_{air} \frac{d\theta_{air}}{dt} \quad (7)$$

The surface convective exchange is determined solving the surface heat balance, per unit of surface, for the internal side:

$$\dot{q}_{c,i} + \dot{q}_{sol,i} + \dot{q}_{gswr,i} + \dot{q}_{IR,i} + \dot{q}_{glwr,i} + \dot{q}_{tr,i} = 0 \quad (8)$$

In particular, the internal long wave radiation is evaluated by Seem's equivalent star network approach (Seem, 1987). The conduction heat through the envelope is usually calculated by the simulation codes with a numerical approach, such as the transfer function method (TFM): in TRNSYS the method implemented is the Direct Root-Finding (DRF). The external boundary conditions are defined by the balance equation:

$$\dot{q}_{c,o} + \dot{q}_{sol,o} + \dot{q}_{IR,o} + \dot{q}_{tr,o} = 0 \quad (9)$$

The external long wave radiation is calculated considering the exchanges with the external surrounding elements (ground, other buildings, sky vault).

2.3 Thermal losses calculation procedure with the dynamic simulation approach

In order to evaluate the thermal losses by means of dynamic simulation, the EN ISO 13790:2008 prescribes to calculate the energy needs setting to zero the internal gains, the solar gains and the infrared extraflow to the sky vault. The simulation heating and cooling setpoints have to be the same (null regulation band). The thermal losses can be calculated from the heating energy need and the cooling energy need:

$$Q_{ht} = Q_{H,nd} - Q_{C,nd} \quad (10)$$

In order to compare the simulated losses to the quasi-steady state results, boundary conditions and calculation parameters for the simulation have to be coherent with the ones assumed in the quasi-steady state approach. Regarding the external conditions, the hourly weather data have been calculated by means of the subroutine Type 54 starting from the monthly average values reported by the Italian technical Standard UNI 10349:1994 (UNI, 1994). Regarding the internal conditions, because in the TRNSYS subroutine Type 56 only an air temperature setpoint is allowed, an iterative approach was adopted in the simulation in the cases with an operative temperature setpoint:

- imposing the weighting factors to the internal air temperature and the mean radiative temperature calculated at each timestep, the resulting operative temperature was calculated;
- the air temperature setpoint in Type 56 was then corrected given the target operative temperature setpoint, repeating the calculations again, till convergence.

Since the aim of this work is to deepen the analysis of the elements of disagreement and to make the two approaches coherent, only balanced weights have been considered, as defined in Eq. (4).

According to the EN ISO 6946:2007 (CEN, 2007) for quasi-steady state methods, the global surface heat transfer coefficients are distinguished in convective and radiative coefficients. Due to the detailed long wave radiation models adopted by TRNSYS, only the convective coefficients could be set to the values prescribed by the Standard also in the simulation: 20 W m^{-2} K^{-1} for the external side, and 5.0, 0.7 or 2.5 W m^{-2} K^{-1} respectively for upward, downward and horizontal flow on the internal side.

As far as the radiation exchanges are concerned, both internal (ε=1) and external emissivity values (ε=0.9) are non-modifiable in TRNSYS. In principle, attempting to improve the coherence between

detailed simulation and quasi-steady state calculation, the internal long wave radiation heat transfer coefficient used in the quasi-steady state approach could be calculated according to:

$$h_r = 4 \cdot \sigma \cdot \varepsilon \cdot T_{mr}^3 \quad (11)$$

The same unitary internal emissivity used in TRNSYS can be assumed, but the surfaces temperature is not known in advance and can only be approximated with the temperature setpoint, as suggested by the Standard itself. Thus, different surface radiative heat transfer coefficients have been considered coherently with the chosen setpoints.

2.4 Reference building model and set of configurations

The difference between the air and the operative temperature, which impacts on the correspondence between the transmission losses calculated with quasi-steady state approach or detailed simulation when using an air temperature setpoint, is largely affected by the insulation level of the envelope. Moreover, as is also pointed out by the Standard EN ISO 13790:2008 itself, it is expected that also large ventilation rates lead to relevant discrepancies with the quasi-steady state methods, and not only when using air temperature setpoint for the simulation. Therefore, in this first part of the analysis we focused on different ventilation rates and insulation levels of the envelope, as well as on the kind of temperature setpoint. The simulations were performed considering air temperature and operative temperature setpoints and then compared with the ones calculated with quasi-steady state method. Concerning the values of setpoint, a typical heating season setpoint temperature for residential applications (20 °C) and the second one with a typical cooling setpoint temperature (26 °C) have been assumed, in accordance with the prescriptions by the EN ISO 13790:2008.

A single base building module has been considered and a selected group of parameters has been varied within a predefined set of values, obtaining a variety of configurations. With the perspective of the second part of this work on thermal gains, we also paid attention to some component properties, such as the thermal capacity of the walls or the SHGC of the glazings, not related to the thermal losses but affecting the selection the components themselves.

The considered module is single-storey with 100 m² of floor area and a horizontal roof. The opaque envelope is composed by a two-layer structure, whose thermo-physical characteristics are reported in Table 1. An insulation layer, with a thickness depending on the simulation plan, is positioned on the external side. Three possible materials have been considered for the internal layer (timber, clay-block or concrete) with a thickness chosen to have a thermal resistance around 0.8 m² K W^{-1}, as 0.2 m of clay-block. The window frame is a timber frame with a low performance (U_f = 3.2 W m^{-2} K^{-1}) if coupled with the single glass and high performance (U_f = 1.2 W m^{-2} K^{-1}) in the other cases. The frame area covers about the 20 % of the whole window area.

	Timber	**Clay-Block**	**Concrete**	**Insulation**
λ	0.13	0.25	0.37	0.04
c	1880	840	840	1470
ρ	399	893	1190	40
s	0.1	0.2	0.3	0/0.05/0.10
R	0.77	0.80	0.81	0/1.25/2.5

Table 1 – Properties of the opaque components

The following geometrical and thermo-physical characteristics have been determined in accordance with the factorial plan:

1. the amount of envelope surface exposed to the external conditions;
2. the ventilation rate;
3. the level of insulation added to the internal layer;
4. the base material of the opaque envelope (taken into account in this part of the analysis because of the small differences of the three layers thermal resistances);
5. the percentage ratio of glazings A_{gl} to floor area A_{fl};
6. the kind of glazings;
7. the climatic conditions.

For each of the above factors, a certain number of

alternatives (levels) were considered as reported in Table 2. The presence of thermal bridges has been neglected in this study, as they can be considered to play a neutral role comparing the simulation and the quasi-steady state approaches. As the aim was to evaluate the losses by thermal transmission through opaque and transparent elements directly exposed to the outdoor air (i.e. with external air convection boundary conditions), when the floor is not adiabatic, it has been assumed as directly in contact with the external air without any solar contribution, as if it was on a well-ventilated cavity.

The first factor allows to consider different ratios between the dispersing surface and the effect of different percentages of adiabatic surface in the total envelope. The second one analyses the ventilation rates, taking into account also its absence (e.g., thermal losses only by transmission). The variation of the thickness of the insulation layer from 0 to 10 cm (factor 3) and the kind of glazings (factor 6), allow to evaluate configurations ranging from non-insulated buildings to well insulated ones. The factors 1, 5 and 6 allow to analyse the internal infrared exchanges between the glazings and the adiabatic surface and their effects on the mean radiant temperature of the thermal zone. Two climates have been considered to calculate the thermal losses for different profiles of external temperatures

Considering 3 shape ratios, 4 ventilation rates, 3 possible insulation thicknesses, 3 base materials, 2 different ratios between the window surface and the floor and 5 types of glazings, 1080 different configurations have been evaluated for each month of each climate. 25920 monthly values have been elaborated for each of the 4 setpoint conditions (air temperatures 20 °C and 26 °C, operative temperatures 20 °C and 26 °C).

3. Results and discussion

The different setpoint temperature strategies have been considered, assuming the EN ISO 13790:2008 results as a benchmark. Firstly the different setpoint strategies have been compared for a null ventilation rate, in order to investigate the deviation induced by choosing air temperature setpoint or operative setpoint in dynamic simulation. Then the effects of different the ventilation rates have been analysed.

1	a	one wall, floor and ceiling adiabatic; S/V=0.30 m^{-1}
	b	one wall and floor adiabatic; S/V=0.63 m^{-1}
	c	one wall adiabatic; S/V=0.97 m^{-1}
2	a	0 ACH (no ventilation rate)
	b	0.3 ACH (as for residential dwellings in accordance with the Italian technical Specification UNI/TS 11300-1:2008)
	c	0.6 ACH
	d	0.9 ACH
3	a	0 cm – U_{env} = 1.03 W m^{-2} K^{-1}
	b	5 cm – U_{env} = 0.45 W m^{-2} K^{-1}
	c	10 cm – U_{env} = 0.29 W m^{-2} K^{-1}
4	a	Timber
	b	Clay-block
	c	Concrete
5	a	11.7%
	b	23.4%
6	a	(S) single glass U_{gl} = 5.68 W m^{-2} K^{-1}, $SHGC$ = 0.855
	b	(DH) double glazing high solar transmittance U_{gl} = 1.140 W m^{-2} K^{-1}, $SHGC$ = 0.608
	c	(DL) double glazing low solar transmittance U_{gl} = 1.099 W m^{-2} K^{-1}, $SHGC$ = 0.352
	d	(TH) triple glazing high solar transmittance U_{gl} = 0.613 W m^{-2} K^{-1}, $SHGC$ = 0.575
	e	(TL) triple glazing low solar transmittance U_{gl} = 0.602 W m^{-2} K^{-1}, $SHGC$ = 0.343
7	a	Messina – HDD_{20}: 707 K d
	b	Milan – HDD_{20}: 2 404 K d

Table 2 – Factors and levels in the simulation plan

3.1 Transmission heat losses

In Figure 1, the transmission thermal losses simulated with the air temperature setpoints and with the operative temperature setpoints have been plotted against the quasi-steady state results for the case of 20 °C. The cases with a 26 °C setpoint present similar trends and behaviours.

For both setpoint strategies, the results have been first distinguished by S/V ratio. The cases without insulation have been highlighted in a darker colour and regression lines have been added to distinguish trends and deviations. The linear regressions always show a very high index of determination R-squared in all the considered conditions, given that the results have been distinguished by shape ratio and presence of insulation. All other factors, such as window size, kind of glazings, climate and also the thickness of insulation for the insulated cases, seem rather ineffective in spreading the results away from the trendlines. Apart from the coefficient of determination, the equations of the trendlines reported in the charts enable to quantify the deviation of the results of the simulation from the ones of the quasi-steady state method: the more the slope coefficient is different from 1, the more the deviation of the simulation results will be.

The largest deviations are shown when using an air temperature setpoint for the simulations, whose results underestimate the absolute value of thermal losses. As one could expect, the differences are particularly large for the non-insulated cases with a high S/V ratio, with an undervaluation around 22%, but also in the insulated cases, the use of air temperature setpoint leads to absolute simulated losses larger than 10% with respect of the ones of the quasi-steady state method. Reducing the S/V ratio (i.e., increasing the percentage of adiabatic surface), the mean radiant temperature becomes higher, together with the operative temperature, and so the deviations between the simulations and the EN ISO 13790:2008 method get lower. With a S/V equal to 0.3, the underestimation is around 7% for the uninsulated cases and less than 3% for the insulated ones. The operative temperature setpoint enlarges the difference between internal and external air temperature, giving higher absolute transmission and ventilation losses. For the high S/V, the underestimation is less than 7% for the uninsulated cases and around 1% for the insulated ones. For more compact structures (e.g., S/V=0.3), instead, there is a slight overestimation in the simulated results, around 2% for the uninsulated cases and more than 6% for the insulated ones.

3.2 Effects of the ventilation rate

The histograms in Figure 2 represent the percent deviation of the linear trendline slopes from the unitary value for a 20 °C setpoint.

When considering an air temperature setpoint, increasing the ventilation rate reduces the difference with the estimation of the thermal losses by the simulation approach, whatever the amount of adiabatic surfaces. This is due to the fact that larger absolute ventilation losses tend to compensate more the difference between transmission losses.

For S/V larger than 0.3, the air temperature setpoint still remains critical for all the considered ventilation rates, in particular in the non-insulated cases whose deviations are always larger than 10%.

When using an operative temperature setpoint, the trend is generally the same but in that case the effect does not compensate for the already positive deviation of transmission losses. The ventilation losses are underestimated in absolute value by the quasi-steady state approach: the more the ventilation rate increases, the lower the absolute ventilation losses in the quasi-steady state method than in the simulations. The only exception to this behaviour is for the insulated cases with aspect ratio equal to 0.3 and 20 °C setpoint and the insulated ones with the same aspect ratio but a 26 °C setpoint and ventilation rate larger than 0.6 ACH, where increasing the ventilation rate the deviations slightly decrease. Since the mean radiant temperature is independent from the ventilation rate and it is larger than the operative temperature setpoint in many of these last cases, the air temperature setpoint used by TRNSYS in the air-heat balance is lower than the setpoint of 20 or 26 °C (indeed, the thermal losses estimated by TRNSYS are lower than the ones calculated with the EN ISO 13790:2008 method). That also affects the ventilation thermal losses which are overestimated by the quasi-steady state approach and make the global percentage deviation decrease in absolute terms. With the operative temperature setpoint, the deviations are generally within a range of 5%, except for the uninsulated cases

without ventilation and S/V=0.97 and for the insulated ones with S/V=0.3.

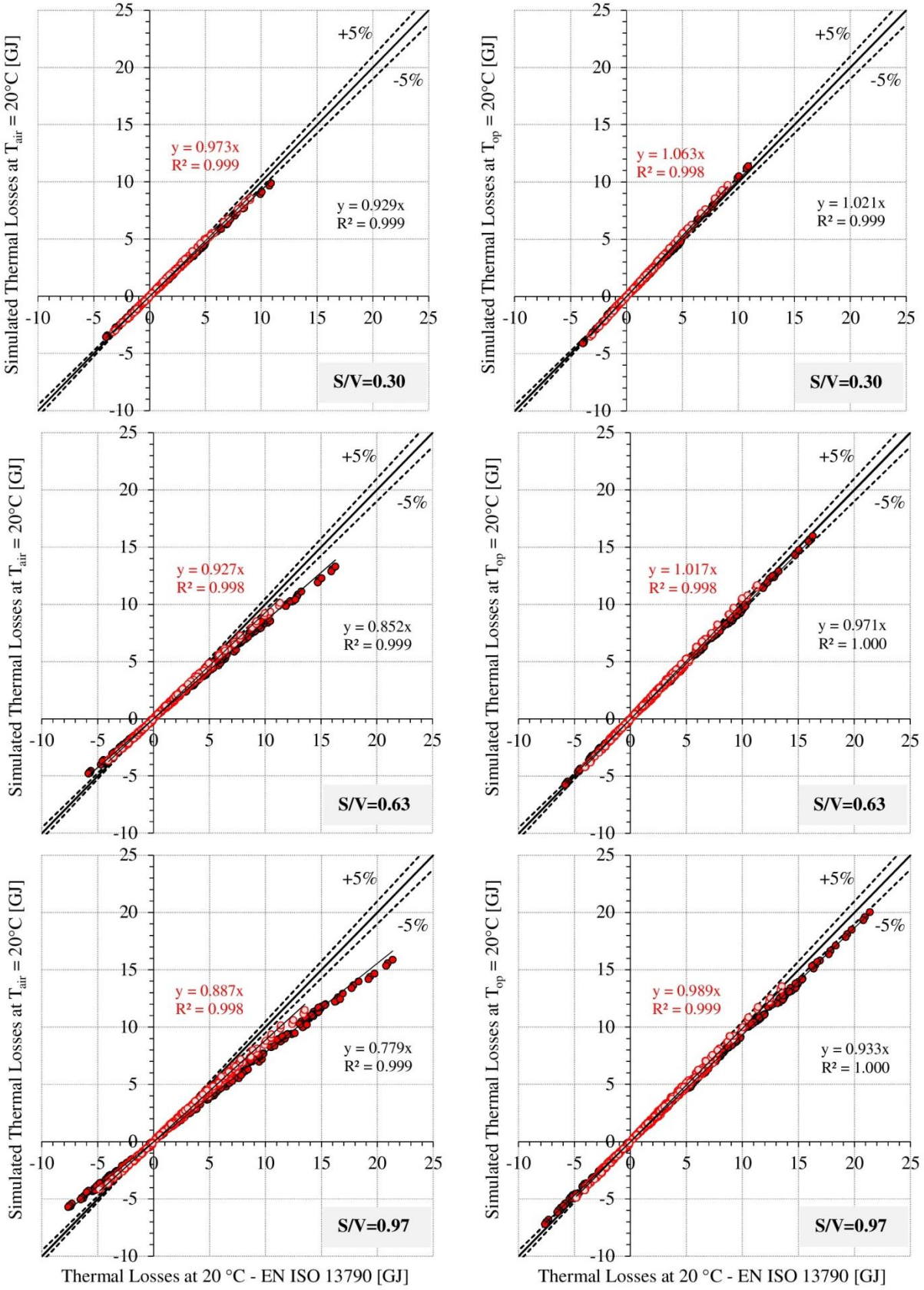

Fig. 1 – Simulated thermal losses with air temperature setpoint (on the left) and operative setpoint (on the right), without ventilation a for the different S/V. Insulated cases in lighter colours

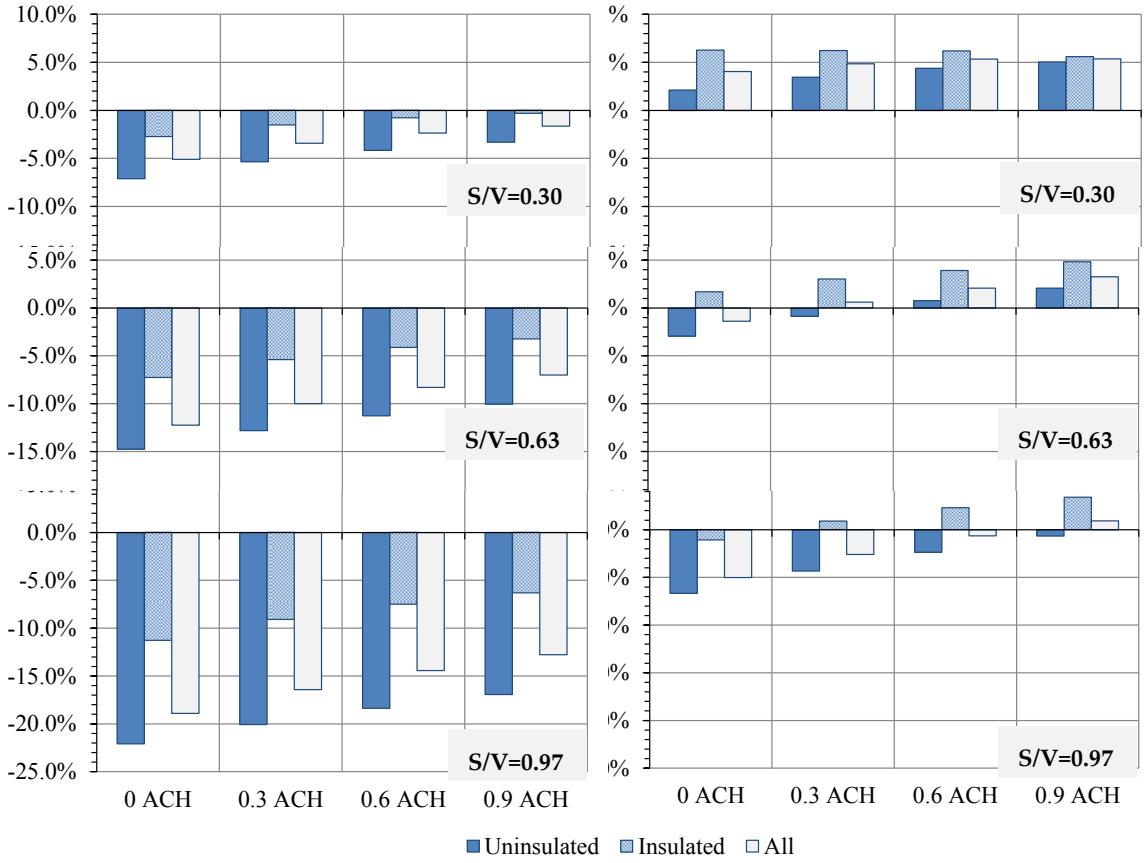

Fig. 2 – Percent deviation of the simulated thermal losses with air temperature setpoint (on the left) and operative setpoint (on the right) respect to the losses calculated with the quasi-steady state approach for different airchange rates and S/V ratios

k_0	k_1	k_2	k_3	k_4	k_5	k_6	k_7	k_8	k_9	k_{10}
1.031	-1.456E-3	2.091E-3	2.451E-3	-0.01465	-0.2093	0.01022	-0.1044	4.518E-3	0.05882	-2.084E-3

Table 3 – Regression coefficients

3.3 Setpoint correction factor for the calculation of thermal losses by transmission

As observed in the previous paragraphs, the main source of discrepancy is the kind of temperature considered as setpoint. The operative temperature setpoint is not realistic in most of the applications and so a correction factor has been calculated for all cases and a regression analysis has been performed in order to find a general equation for determining the correction starting from the envelope characteristics. The developed model is reported in Eq. (12) and it is characterized by an adjusted determination index R2adj equal to 0.85. The 11 coefficients have been reported in Table 3 and Figure 3 represents an example of its application. The sample used for the regression consists mainly in positive thermal losses (e.g., when the average external temperature is lower than the air temperature setpoint of 20 or 26 °C: negative values are, actually, thermal gains). In consequence of that, Eq. (12) should be used only for setpoints close to the range 20 – 26 °C and when the average external temperature is lower than the setpoint.

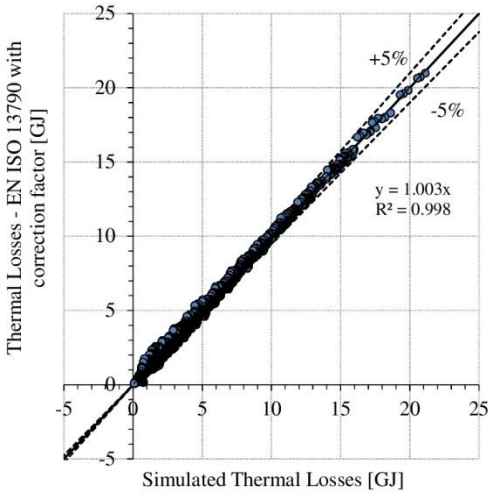

Fig. 3 – Comparison between the corrected EN ISO 13790 method and the air temperature setpoint simulations

The correspondence between the new thermal losses and the simulated ones is good (+0.3%), with larger errors for low values of the thermal losses.

$$\begin{aligned} f &= k_0 + \theta_e \cdot \left(k_1 + k_2 \cdot x_{env} + k_3 \cdot x_{win}\right) + \\ &+ U_{env}\left(k_4 + k_5 \cdot x_{env} + k_6 \cdot x_{env} \cdot \theta_e\right) + \\ &+ U_{win}\left(k_7 \cdot x_{win} + k_8 \cdot x_{win} \cdot \theta_e\right) + \\ &+ k_9 \cdot x_{ad} + k_{10} \cdot \theta_{i,set} \end{aligned} \quad (12)$$

4. Conclusion

In the present work two thermal losses estimation methods have been analysed: the EN ISO 13790:2008 quasi-steady state approach and dynamic simulations, both with air and operative temperature setpoints. In addition 301rdert kind of setpoint, the most important sources of discrepancies, the building aspect ratio, the envelope insulation and the ventilation rate, have been analysed. A correction 301rdert estimation of the operative temperature for the quasi-steady state method has been proposed in 301rdert o correctly evaluate the thermal losses by transmission in presence of air temperature setpoint.

5. Nomenclature

Symbols

A	area (m^2)
c	specific heat capacity (J kg^{-1} K^{-1})
f	correction factor (-)
h	heat transfer surf. Coeff. (W m^{-2} K^{-1})
H	overall heat transfer coeff. (W K^{-1})
HDD	heating degree-days (K d)
Q	energy (GJ)
$\dot{q}$	thermal flux (W m^{-2})
R	thermal resistance (W m^{-2} K^{-1})
s	thickness (m)
S	dispersing surface (m^2)
$SHGC$	solar heat gain coefficient (-)
t	time (s)
U	thermal transmittance (W m^{-2} K^{-1})
V	conditioned volume (m^3)
$\dot{V}$	air change flow (m^3 s^{-1})
x	surface fraction (-)
ε	surface emissivity (-)
ϕ	thermal flow (W)
λ	thermal conductivity (W m^{-1} K^{-1})
ρ	density (kg m^{-3})
σ	Stefan-Boltzmann constant ($5.67 \cdot 10^{-8}$ W m^{-2} K^{-4})
θ/T	temperature (°C) (K if absolute)

Subscripts/Superscripts

a/air	internal air
ad	adiabatic
c	convective
C	cooling
D	towards external air
e/o	external/external side
env	opaque envelope
f	frame
fl	floor
gc	convective gains
gl	glazing
H	heating
ht	heat transfer
i	internal/internal side
i,set	setpoint
IR	infrared radiation
glwr	longwave radiative gains
gswr	shortwave radiative gains

op	operative
mr	mean radiant
nd	energy need
r	radiative
sol	solar
sys	system
tr	transmission
Ve	ventilation
win	window

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