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ABSTRACT

One of the aims of the European Directive 2010/31/EU (formerly 2002/91/EC) is to reduce the energy consumption of buildings introducing an energy labeling protocol which is expected to capture the attention and reorient the market, sustaining the diffusion of more efficient solutions. In order to evaluate the building energy performance, either analytical approaches or enhanced simulation tools are allowed. The coherence of the methods is important in order to avoid misleading results which can affect the evaluation by the market and eventually compromise the Directive effectiveness.
The European Standard EN ISO 13790:2008 suggests to use the dynamic simulation in improving and tuning the quasi-steady state method proposed, and in particular to refine the correlation used to calculate the utilization factors (i.e., the dynamic parameters which reduce the thermal gains for heating need calculation and the thermal losses for cooling). Many efforts in calibrating the EN ISO 13790:2008 led to some changes on the correlations proposed in order to adapt the method to the climatic conditions, especially for the cooling season, and the building stock's characteristics in different countries, but large discrepancies have been found.
Differently from the previous works, the authors analyze the discrepancy sources focusing firstly on thermal losses, instead of considering directly the final result in term of energy needs.
In this paper, the deviations between the thermal losses are evaluated, by means of an extensive use of simulation, analyzing a set of a 960 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. Six different setpoint conditions were considered for the simulations.
The analysis allowed the authors to identify the relevance of the deviations and to suggest ways to improve the correspondence between simulation and quasi-steady state methods in tuning processes.

1. INTRODUCTION

The Energy Performance of Building (EPB) Directive 2010/31/EU and the former 2002/91/EC (European Parliament, 2010; European Parliament, 2002) attempt to improve the energy performance of buildings through the introduction of the energy labelling which is expected to catch the attention and reorient the choice of the potential buyer or renter, sustaining the market diffusion of more efficient solutions. According to the EPB Directive the building energy performance can be evaluated either with analytical approaches or with enhanced simulation tools. As observed by Tronchin and Fabbri (2008), the coherence between methods is of crucial importance to avoid misleading results which can affect the evaluation by the market and its perception of the reliability of the energy label, eventually compromising the EPB Directive effectiveness.

Moreover, the European Standard EN ISO 13790:2008 (European Committee for Standardization, 2008) suggests to use the dynamic simulation in improving and tuning the quasi-steady state method proposed, and in particular to refine the correlation used to calculate the utilization factors (i.e., the dynamic parameters which reduce the thermal gains for heating need calculation and the thermal losses for cooling). The utilization factors depend on the ratio between the thermal losses and the thermal gains, which are conventionally calculated through analytical expressions. Differently from the method of studying the dynamic factor by Van Dijk and Arkesteijn (1987), the EN ISO 13790:2008 encourages the use of the simulation software in all steps, 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). Some changes on the correlations have been proposed in order to adapt the method to the climatic conditions, especially for the cooling season, and the building stock's characteristics in their respective countries. Anyhow the general problem appears to be still unsolved as large discrepancies have been found. Referring to the thermal losses, a comparison between the simulated ones and those analytically calculated can be helpful in looking for disagreements elements which can lead to mismatches and errors in the quasi-steady state method's refinement.
Finally the comparison between the dynamic simulation results and the analytical approaches (detailed or simplified) is also the first relevant step in the validation procedures of the simulation codes (Judkoff *et al.*, 1982).
Between the causes of error or discrepancies depending on the definition of the boundary conditions and on the calculation of the thermal losses (Judkoff *et al.*, 2008), the choice of the internal reference conditions appears to play a crucial role. As refers to the transmission losses, which represent the first component of thermal losses, the quasi steady state model linearizes and considers in parallel with the convection exchange with the air node the internal long wave radiation exchanges and so it assumes 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 allows us to use an air temperature setpoint, as it is generally more realistic when simulating real operative conditions. Some authors (Corrado and Fabrizio, 2007) have pointed out this issue, noticing that the difference between the operative temperature and the external one should be considered as the real driving force in heat transfer in dynamic simulation, instead of the difference based on the air temperature.
As regards the second component of thermal losses, that are the ventilation losses, the actual driving force is given by difference 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, would lead to incorrect results in particular in presence of 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 the present work the effect of the use of air setpoint temperature against operative setpoint temperature for total thermal losses has been investigated comparing dynamic simulation to quasi steady state methods. TRNSYS 16.1 was used for detailed simulation while the quasi steady state procedure was directly implemented without the support of any commercial calculation software.
Two of the most important sources of discrepancies between the air and the operative temperature have been analysed: the insulation level of the envelope, which reduces the difference between the air and the operative temperature, and the ventilation rate. The weights considered for averaging air and mean radiant temperature of the envelope when calculating the operative temperature have been analysed by comparing two different hypotheses: balanced weights, i.e. 0.5 and 0.5 respectively for the air and the mean radiant temperature, as indicated by the EN ISO 13790:2008, and unbalanced weights, i.e. 0.3 and 0.7 respectively for the air and the mean radiant temperature.

2. METHOD

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 easily calculated with the Eq. (1).

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

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

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

The overall transmission heat transfer coefficient is:

$$H_{tr}=H_D+H_g+H_U+H_A \tag{3}$$

In the evaluated cases, only dispersions of the heated zone towards the outside environment are considered. Thus, the only term different from zero is H_D. Neglecting the thermal bridges, as it was assumed to do, H_D can be expressed as:

$$H_D=\sum_{k=1}^{n} A_k U_k \tag{4}$$

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:

$$\theta_{op}= 0.5\cdot\theta_{air}+ 0.5\cdot\theta_{mr} \tag{5}$$

Values of 20 °C for the heating period and 26 °C for the cooling period are generally assumed.
The ventilation thermal losses are similarly defined as:

$$Q_{ve}=H_{ve}\cdot\left(\theta_{i,set}-\theta_e\right)\cdot t \tag{6}$$

where:

$$H_{ve}=\rho_a\, c_a\left(\sum_{k=1}^{n} b_{ve,k}\, \dot{V}_k\right) \tag{7}$$

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.
It is pretty clear that 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.

2.2 TRNSYS air heat balance

Differently from the method proposed by EN ISO 13790:2008, 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:

$$\phi_{c,i}+\phi_{ve}+\phi_{g\text{-}c}+\phi_{sys}= C_{air}\frac{d\theta_{air}}{d\tau} \tag{8}$$

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

$$\dot{q}_{c,i}+\left(\dot{q}_{sol,i}+\dot{q}_{g\text{-}swr,i}\right)+\left(\dot{q}_{IR,i}+\dot{q}_{g\text{-}lwr,i}\right)+\dot{q}_{tr,i}=0 \tag{9}$$

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 particular, in TRNSYS (Solar Energy Laboratory, 2005) the method implemented is the Direct Root-Finding (DRF). The external boundary conditions are defined by the balance equation:

$$\dot{q}_{c,o}+\dot{q}_{sol,o}+\dot{q}_{IR,o}+\dot{q}_{tr,o}=0 \tag{10}$$

In particular 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:

$$Q_{ht}=Q_{H,nd}-Q_{C,nd} \tag{11}$$

In order to compare the simulated losses to the quasi steady state results, some boundary conditions or calculation parameters for the simulation have to be chosen in coherence with the ones assumed in the quasi steady state approach.
As regards 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 (Ente Nazionale Italiano di Unificazione UNI, 1994).

As regards the internal conditions, due to the fact that only an air temperature setpoint is allowed in the TRNSYS subroutine Type 56, an iterative approach was adopted in the simulation when an operative temperature setpoint had to be considered:
- imposing the right weighting factors to the internal air temperature and the mean radiative temperature calculated ad 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.

According to the EN ISO 6946:2007 (European Committee for Standardization, 2007) for quasi steady state methods, the global surface heat transfer coefficients are distinguished in surface convective coefficient and surface radiative coefficient. 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 concerns the radiation exchanges, 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:

$$h_r = 4 \cdot \sigma \cdot \varepsilon \cdot T_{mr}^3 \qquad (12)$$

The same unitary internal emissivity used in TRNSYS can be assumed, but the surfaces temperature is not known in advance and can only be approximately assumed as the same of the temperature set point as suggested by the standard itself.

2.4 Plan of calculations and reference building model

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 thermal losses were calculated by simulation considering both standard setpoints on the air-node and operative temperature setpoints and assuming different ventilation rates, and the simulation results were compared with the ones obtained with quasi steady state method. The operative temperature was calculated either averaging the air and mean radiant temperature with equal weights as in Equation (5) or considering different weights of 0.3 and 0.7 respectively for the air and the mean radiant temperatures.
As concerns the considered setpoints 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. As regards the ventilation, three different ventilation rate in addition to the standard rate of 0.3 ach/h indicated for residential dwellings were considered: 0 ach/h (i.e., without ventilation), 0.6 ach/h and 0.9 ach/h.
In order to compare the simulation and quasi steady state approaches over a wide range of cases, 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.
The considered module is single-storey with 100 m^2 of floor area and a horizontal roof. Previous studies confirmed that the presence of internal partition that could modify the mean surface temperature does not affect the results significantly (Gasparella and Pernigotto, 2011), so a single zone building was considered.
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), also the floor of the base module has been assumed as directly in contact with the external air without any solar contribution.
A two-layers composition has been considered for the opaque envelope with an internal massive layer equivalent to a 0.2 m thick clay block layer and an external insulation layer with properties similar to the polystyrene (Table 1).

Table 1: Thermophysical characteristics of the opaque envelope

		Internal layer (clay block)	External layer (insulation)
Thermal conductivity	W m^{-1} K^{-1}	0.25	0.04
Density	kg m^{-3}	850	40
Specific heat	J kg^{-1} K^{-1}	840	1 470

Table 2: Variables considered in the simulation plan

1. Shape		Side ratio	S/V (net dim)
	a. Square	1:1	1.066
	b. Rectangular #1	3:4	1.071
	c. Rectangular #2	1:2	1.091
	d. Rectangular #3	1:4	1.166
2. Percentage ratio A_{gl}/A_f	a.	0.00%	
	b.	11.66%	
	c.	23.34%	
3. Opaque envelope insulation thickness	a.	0 cm	$U = 1.03$ W m^{-2} K^{-1}
	b.	5 cm	$U = 0.45$ W m^{-2} K^{-1}
	c.	10 cm	$U = 0.29$ W m^{-2} K^{-1}
	d.	15 cm	$U = 0.21$ W m^{-2} K^{-1}
4. Glazings	a. Double (4/15/4)	$U_{gl} = 1.10$ W m^{-2} K^{-1}	
	b. Triple (4/8/4/8/4)	$U_{gl} = 0.61$ W m^{-2} K^{-1}	
5. Location		Town	HDD_{20}
	a.	Messina	707
	b.	Rome	1 415
	c.	Milan	2 404

The presence of thermal bridges has been neglected in this study, as it can be considered to play a neutral role comparing the simulation and the quasi steady state approaches.

The different configurations to be analysed have been obtained by changing in the ranges summarised in Table 2 the following parameters :

1. shape ratio of the building floor
2. percentage ratio of glazings A_{gl} to floor area A_f
3. composition of the envelope (presence and thickness of insulation)
4. kind of glazings (the frame considered was a wooden one with thermal transmittance of 1 W m^{-2} K^{-1})
5. climatic conditions

Each different shape of the building floor, with fixed the height of the module and the floor area, corresponds to a different ratio between the dispersing surface of the envelope and the volume.

The variation of the thickness of the insulation layer from 0 to 15 cm, allowed us to evaluate configurations ranging from non-insulated buildings to well insulated ones.

The three climates have been considered to calculate the thermal losses for different profiles of external temperatures.

Considering the 4 shape ratios, 3 different ratios of window and floor surface, 4 possible insulation thicknesses and 2 types of glazings, 80 different configurations have been evaluated for each month of each climate. A total of 2880 monthly values have been elaborated for each of the 6 set point conditions (air temperatures 20 °C and 26 °C, operative temperatures 20 °C and 26 °C with equal weighting for air and mean radiant temperatures and operative temperatures 20 °C and 26 °C with weighting 0.3 for air and 0.7 for mean radiant temperatures). Moreover the calculations have been repeated for the 4 different ventilation rates.

3. RESULTS

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

3.1 Effects of the temperature setpoint

In Figure 1, the thermal losses simulated with the air temperature setpoints and with the two operative temperature setpoints have been plotted against the quasi steady state results, both for the case of 20 °C and 26 °C. The cases without insulation, have been distinguished in darker colour from the insulated ones and regression lines have been added to distinguish trends and deviations.

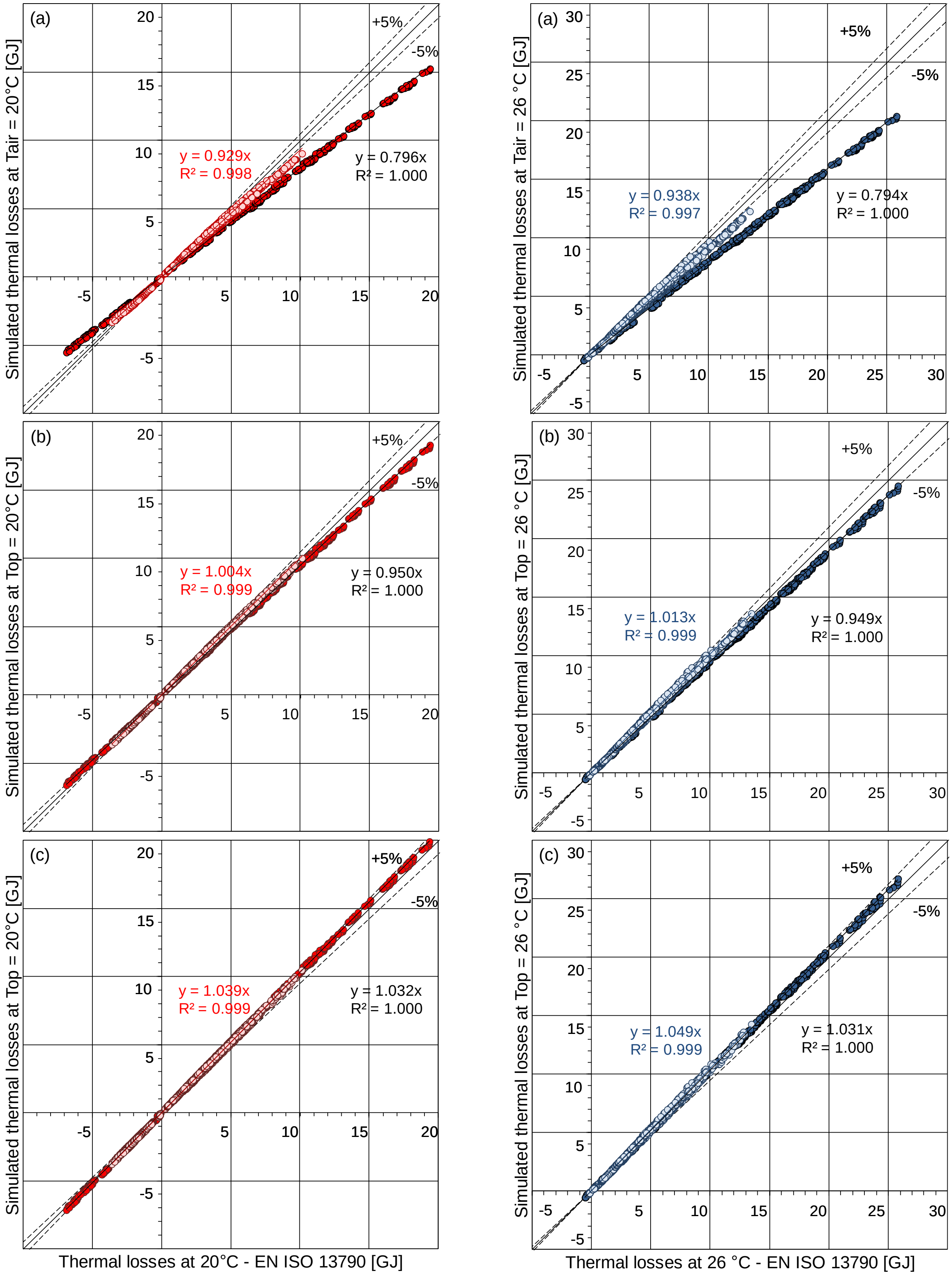

Figure 1: Simulated thermal losses with (a) air temperature setpoint; (b) operative setpoint with equal weights; (c) operative setpoint with weights 0.3 for air and 0.7 for mean radiant temperatures vs thermal losses calculated with the quasi steady state approach. Ventilation rate 0.3 ach h^{-1}. Insulated cases in lighter colours.

The linear regressions always show a very high coefficient of determination R^2 in all the considered conditions, given that the non-insulated cases are distinguished from the insulated ones.
All other factors, such as shape ratio, windows 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 will be the deviation of the simulation results.
The largest deviations are shown when using an air temperature setpoint for the simulations. Both with 20 °C and 26 °C the simulation results underestimate the absolute value of thermal losses. As one could expect, the differences are particularly large for the non-insulated cases, with an undervaluation around 21%, but also in the insulated cases, the use of air temperature setpoint leads to absolute simulated losses around 6-7% lower than the ones of the quasi-steady state method.

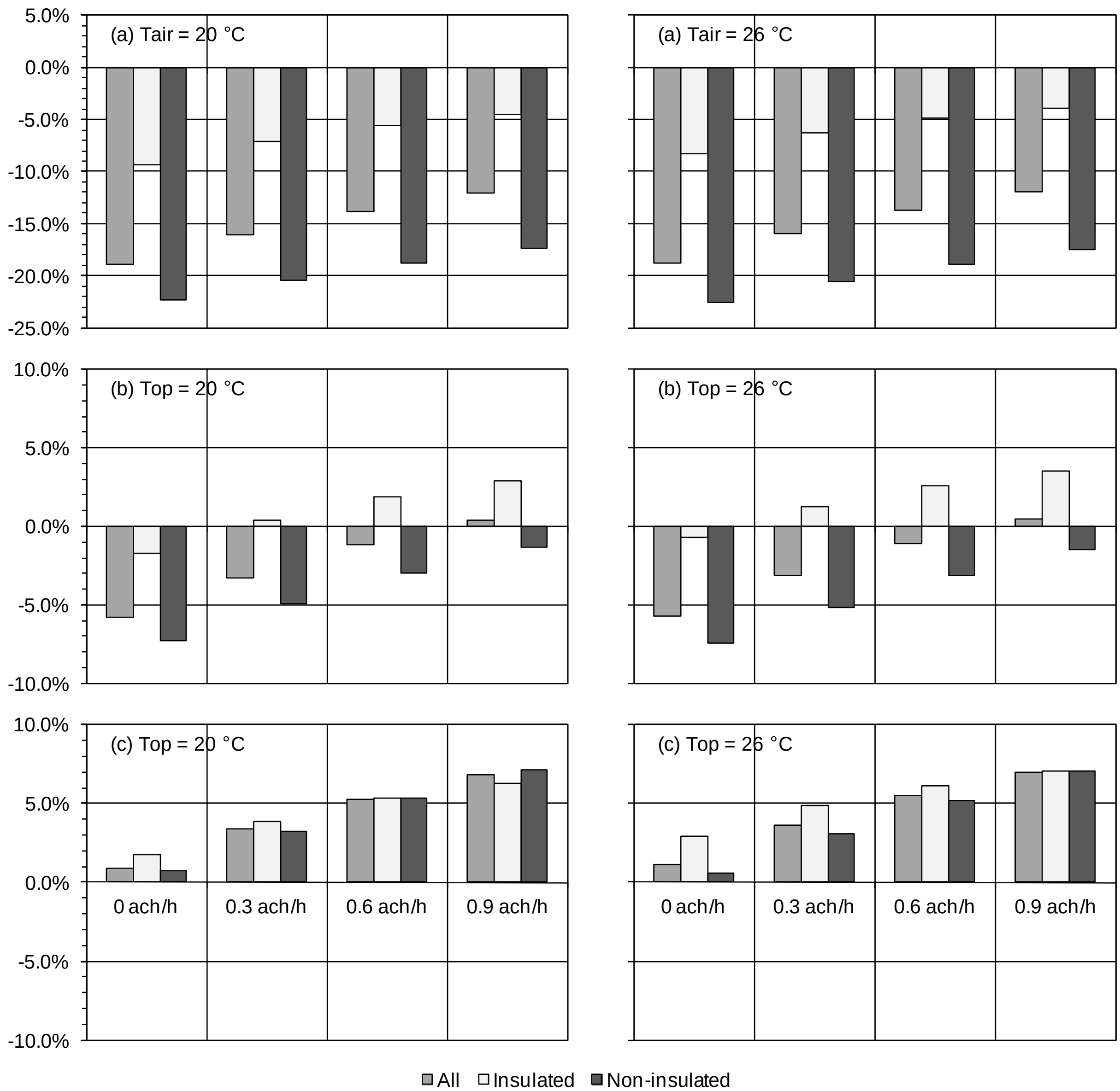

Figure 2: Percent deviation of the simulated thermal losses with (a) air temperature setpoint; (b) operative setpoint with equal weights; (c) operative setpoint with weights 0.3 for air and 0.7 for mean radiant temperatures respect to the losses calculated with the quasi steady state approach for different airchange ratios.

When using the operative temperature setpoint, this enlarges the difference between internal and external air temperature, giving higher absolute transmission and ventilation losses. Especially in the insulated cases, the use of an operative temperature as equally weighted average of the air and mean radiant temperatures gives the best correspondence between simulation and quasi steady state results, at least for the given ventilation rate. Also in the non-insulated cases the underestimation by the simulation losses reduces to around 5%.
This behavior is emphasized when the weighting factor of the air temperature is limited to 0.3. Simulated thermal losses enlarge in absolute value, giving a slight overestimation (between 3% and 5%) of the quasi-steady state losses, but in this case the non-insulated cases tends to become almost undistinguishable from the insulated ones. The slight overestimate could be seen as the effect of the larger undervaluation of the ventilation losses by the quasi-steady state method when using a larger weight for the mean radiant temperature.

3.3 Effects of the ventilation rate

In order to investigate the relation between the setpoint strategy and the ventilation rate the calculation were repeated for different ventilation rates. The histograms in Figure 2 represent the percent deviation of the linear trendline slopes from the unitary value.
Firstly, it is possible to confirm that increasing the ventilation rate reduces the underestimation or produces slight overestimation of the modulus of the thermal losses by the simulation approach. When considering air temperature setpoint this is due to the fact that larger and larger absolute ventilation losses tend to compensate more and more the difference between transmission losses. In contrast, when using operative temperature setpoint, this is mainly due to the fact that the ventilation losses are underestimated in absolute value by the quasi steady approach. Larger ventilation rate increases less the absolute ventilation losses in the quasi-steady state method than in the simulations.
Secondly, the air temperature setpoint still keeps critical for all the considered ventilation rates, in particular in the non-insulated cases for which the deviations are always under -15% in the case of air temperature setpoint, and considering all the cases never rise over -10%.
Finally, the choice of the weighting factors for the air and mean radiant temperatures when calculating the operative temperature is crucial for the correspondence of simulated and quasi-steady-state results. With equal weights as suggested by the EN ISO 13790:2008, the deviations between the thermal losses is in the range +/- 5% for all the cases with ventilation. However there are differences between insulated and non-insulated cases. In contrast, when using 0.3 as weighting factor for the air temperature and 0.7 for the mean radiant temperature, the transmission losses are well aligned as is shown in Figure 2 (c) for null ventilation rate, independently from the insulation level. In that case the total thermal losses tend to be overestimated by the simulation when introducing a ventilation rate.

4. CONCLUSIONS

In the present work the effect of the use of air setpoint temperature against operative setpoint temperature for total thermal losses has been investigated comparing dynamic simulation to quasi steady state methods. TRNSYS 16.1 was used for detailed simulation while the quasi steady state procedure was directly implemented without the support of any commercial calculation software.
Two of the most important sources of discrepancies between the air and the operative temperature have been analysed: the insulation level of the envelope, which reduces the difference between the air and the operative temperature, and the ventilation rate.
Considering the 4 shape ratios, 3 different ratios between the window surface and the floor, 4 possible insulation thicknesses and 2 types of glazings, 80 different configurations have been evaluated for each month of each climate. A total of 2880 monthly values have been elaborated for each of the 6 setpoint conditions (air temperatures 20 °C and 26 °C, operative temperatures 20 °C and 26 °C with equal weighting for air and mean radiant temperatures and operative temperatures 20 °C and 26 °C with weighting 0.3 for air and 0.7 for mean radiant temperatures). The calculation have been repeated for the 4 different ventilation rates.
The comparison allowed to come to some conclusions. Firstly, the quasi-steady-state method should not be applied to cases with air temperature setpoint, in particular for poor insulated configurations. Ventilation losses are correctly estimated but transmission losses are overvalued, leading to overestimated thermal losses.
Secondly, the simulation methods give similar transmission losses when using an operative temperature setpoint. When using equal weights for the air and the mean radiant temperature as suggested by the standard EN ISO 13790:2008 the thermal losses are sensitive to the level of insulation and to the ventilation rate but are generally comprised between -5% and +5%.

Finally, the right weighting for air temperature to have a good correspondence between the transmission losses with simulation and operative setpoint is probably somehow lower than the suggested 0.5, and in case of unitary emissivity of the internal surfaces is rather near to 0.3. Although that choice gives good agreement independently from the level of insulation, the underestimation of the absolute ventilation losses with the quasi-steady state approach gives place to simulated absolute thermal losses generally overestimated.

NOMENCLATURE

A	area	(m^2)
b	temperature adjustment factor	(-)
c	specific heat	(J kg^{-1} K^{-1})
C	heat capacity	(J K^{-1})
H	heat transfer coefficient	(W K^{-1})
h	surface heat transfer coefficient	(W m^{-2} K^{-1})
HDD_{20}	heating degree days referred to 20 °C	(K d)
Q	thermal energy (losses, gains or needs)	(J)
$\dot{q}$	specific thermal flux	(W m^{-2})
R^2	determination coefficient	(-)
S	dispersing envelope surface	(m^2)
T	temperature	(K)
t	time	(s)
U	thermal transmittance	(W m^{-2} K^{-1})
V	conditioned volume	(m^3)
$\dot{V}$	volumetric airflow	(m^3 s^{-1})

Greek

ε	surface emissivity	
θ	temperature	(°C)
ρ	density	(kg m^{-3})
σ	Stefan-Boltzmann constant	(5.67·10^{-8} W m^{-2} K^{-4})
τ	time	(s)
ϕ	thermal power	(W)

Subscripts

A	other conditioned zone
air	internal air
c	by convection from the surfaces
C	cooling
D	direct transfer to the external environment
e	external
f	floor
g	ground
$g\text{-}c$	internal gains, convective part
gl	referred to the glazings
$g\text{-}lwr$	internal gains, longwave radiation part
$g\text{-}swr$	internal gains, shortwave radiation part
H	heating
ht	heat transfer
i	inside surface
IR	surface longwave radiation
k	component k of the envelope or element k of airflow (infiltration, natural or mechanical ventilation)
mr	mean radiant
nd	energy need

o	outside surface
op	operative
r	radiative
set	setpoint
sol	solar
sys	heating or cooling system
tr	transmitted by conduction
ve	ventilation
U	unconditioned zone

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