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AIR HEATING OF PASSIVE HOUSES IN COLD CLIMATES: INVESTIGATION USING DETAILED DYNAMIC SIMULATIONS

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1. ABSTRACT

The passive house (PH) standard was originally defined for Central Europe and has subsequently been applied to many cold climate countries. In these conditions, the relation between this standard and the air heating (AH) is not clear while both concepts are usually associated. Furthermore, the AH provides a way to simplify the space-heating distribution system. The present contribution investigates the feasibility of the AH concept in PH along with its challenges in terms of thermal dynamics: the magnitude of the AH temperature needed, the temperature difference between rooms, the impact of internal gains, the influence of thermal losses from ventilation ducts and the AH control. This is performed using detailed dynamic simulations (TRNSYS) on a typical detached house typology. Practically, four cold climate zones are considered as well as different insulation levels and construction materials. Results show limitations related to a centralized AH as well as provide guidelines for a consistent AH design in cold climates. In addition, a simple analytical method used for the design of German PH is tested and proved accurate enough to estimate the maximal AH temperature during the heating season.

2. INTRODUCTION

The passive house (PH) is a standard that aims at promoting energy efficiency. The main concept of the passive house is based on reduction of the space-heating (SH) needs using a super-insulated envelope. The PH concept and first test cases [1] have been originally developed for temperate climates (i.e. Germany, Austria and Switzerland). Afterwards, the PH concept has been extended to many cold climates such as in Denmark, Sweden, Norway, Estonia and Canada. In many cases, the PH requirements have been adapted to the local context, such as the regulation or the climate. It is for example the case for Norway [2] and Sweden [3].

In the original philosophy of the passive house [4], the minimal requirement for the envelope performance has been strongly and explicitly connected to the air-heating (AH) concept. In practice, the building envelope should be sufficiently insulated so that it is possible to cover the SH by the ventilation air at standard hygienic flow rates. Another underlying assumption is that the maximal inlet AH temperature can be raised up to 50-55°C, representative of the temperature of dust carbonization. A direct consequence of the AH concept is that the passive house should resort to high performance windows (e.g. using triple glazing) in order to prevent excessive cold draft or discomfort induced by an internal cold surface.

As regards cold climates, the relation between the PH requirements and the air heating is often not clear. This is due to the change of climate (i.e. from temperate to cold) as well as differences in PH definition. Therefore, the present contribution investigates the AH potential in cold climates, also characterized by large differences between climate zones and often by low solar angles. Likewise, our work takes as a basic assumption that the maximal AH temperature is 55°C and mixing ventilation. In agreement with the current design practice in mechanical ventilation, flow rates can be increased at peak load up to ~50% above the nominal rates, based on hygienic considerations.

A broad review of the questions regarding the indoor air quality (IAQ) in passive houses can be found in [5]. Investigations on AH specific to cold climates are mostly based on Swedish works [6-10]. In terms of thermal comfort and energy efficiency only, we propose to classify the challenges for the AH of passive houses in the following way:

1. **Design and robustness of the AH concept:** Given the different cold climate zones, is the maximal AH power actually enough to cover the SH load throughout the winter? What are the boundary conditions for the AH design (e.g. outdoor temperature and solar irradiation)?
2. **Air distribution inside a room:** Using AH, the flow is buoyancy driven by the cold draft of windows and by the thermal plumes generated by internal heat loads [11]. In practice, there is still a risk of strong temperature stratification or potentially uncomfortable draft. Furthermore, reduced ventilation effectiveness can be found, e.g. the fresh air can shortcut directly from the inlet to the exhaust Air Terminal Device (ATD) [12]. These phenomena are also dependent on the room geometry, the AH temperature and ATDs location. Although further research is still needed, recent works based on measurements [4, 11] did not report any severe issue. Let us mention that these works investigated AH temperatures below ~45°C.
3. **Thermal losses from ventilation ducts:** Thermal losses from ducts are significant and may affect the thermal comfort in passive houses. Losses should be a part of the design and can also be used to improve the thermal comfort in specific rooms (e.g. bathrooms) [4].
4. **AH thermal dynamics:** Assuming a single heating coil for a centralized AH, there is a non-uniform distribution of temperature between rooms. This distribution is mainly influenced by the building architectonic properties, the climate and the AH control. In addition, it is also worth investigating the influence of a complementary SH emission in bathrooms, the effect of

the internal gains distribution inside the building, the influence of the solar gains (e.g. shading strategy), and the influence of opening the doors inside the building.

5. **Maximal AH temperature:** Is the characteristic temperature before dust carbonization the correct criterion for thermal comfort and IAQ? Are there other health concerns that require AH temperatures well below 50°C?

Even though all these five questions deserve a proper treatment, the present article specifically focuses on points (1), (3) and (4). In parallel, an idealized behaviour for question (2) is assumed (i.e. a fully mixed room). Furthermore, the temperature of duct carbonization is taken as the limit for the AH temperature, leaving question (5) open. In this context, investigations are consistently performed using detailed dynamic simulations, here applied to one benchmark Norwegian passive house. Nevertheless, the envelope performance and the weather conditions are selected such that results are representative for cold climates and not exclusively for Norway. A sensitivity analysis is done using a set of ~1200 simulations with a time step of 1 min. Only most representative results are reported in the paper while preliminary results were communicated during a scientific conference [13].

3. SIMULATION PROCEDURE

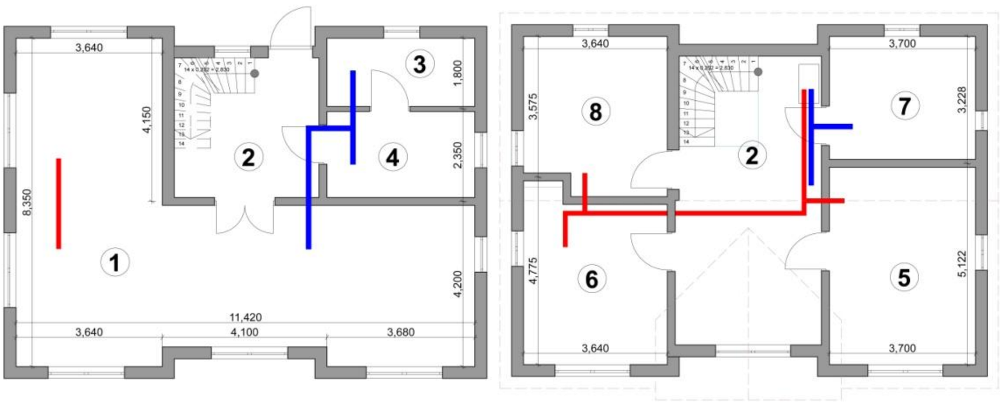
3.1 Building model

Given these aforementioned assumptions, investigations are performed using detailed dynamic simulations, here using TRNSYS 17 [14]. TRNSYS is a transient system simulation software with a modular structure that implements a component-based approach [15]. A dedicated component, termed *Type 56*, computes the heat balance of a building using a multi-zone modelling approach [16]. In each zone, the air temperature is assumed uniform, a simplification that is consistent with the assumptions done in the introduction section (i.e. full mixing in room). The unsteady conduction heat transfer through walls is essentially considered one-dimensional and is computed using the transfer function method (TFM). Finally, a heat balance is performed for the internal and external surfaces of each wall (i.e. opaque medium) or window (i.e. transparent medium). The input data includes the building location, geometry and material characteristics, the meteorological data, the internal gains as well as the infiltration and ventilation airflow rates. Basically, the *Type 56* computes the net SH needs assuming a so-called *perfect or idealized* SH system. Nevertheless, a specific SH system and its imperfections may be investigated by coupling the *Type 56* to other HVAC components through the TRNSYS user interface. It is the approach followed here to model the AH system. Finally, the thermal comfort is evaluated globally using the operative temperature, T_{op}, [17] computed by the *Type 56*.

The ventilation airflow rates are a major input for the building thermal simulation. In practice, they can be simply imposed when internal doors inside the building are close (i.e. unidirectional flow). Nevertheless, Feist et al. [4] demonstrated that the strong bidirectional flow through open internal doors is an efficient way to homogenize the temperature within passive envelopes. In order to investigate this natural convection inside the envelope, airflow rates between rooms are computed using a *ventilation-network model*. Such a model idealizes the building as a network of nodes and airflow links between them. A node represents either a room or the building surrounding. The scope and limitations of the approach for ventilation prediction is well described by Chen [18], Wang and Chen [19] and Srebric [20]. It essentially assumes that the air temperature in each zone is uniform and that the air momentum can be neglected (i.e. quiescent air in zones). In practice, airflow rates are here evaluated using the COMIS software [21] which is properly coupled to the building thermal model (*Type 56*) by the TRNFLOW module [22]. Doors are modelled using a large opening approximation [23] introducing a discharge coefficient, C_d, to tune the model to a specific flow physics. Measurements have shown that C_d for doors typically ranges between 0.4 and 0.8 [24, 25], while a default value of 0.65 is here applied. Doors have a section of 1m x 2.1m, with a 1cm gap underneath when closed.

3.2 Benchmark passive house

A same geometry of detached single-family house is used as a benchmark building for all simulations. It is a typical two-storey's building extracted from a house manufacturer catalogue [26]. This is a common typology for Norway with a net heated surface (A_{fl}) of 173.5 m^2. Statistics also show that detached single-family houses are a common typology in many Nordic countries [27, 28] (e.g. 52% of the total Norwegian building stock). The house and its internal organization are shown in Fig. 1: the building is divided into 8 thermal zones. The living room faces the south. It is assumed that the house is placed on a flat and open terrain without obstacles.



(a) First floor (b) Second floor

Figure 1 Sketches of the first and second floors: kitchen coupled to the living room (zone 1), corridor with an open staircase towards the second floor (zone 2), technical room (zone 3), bathrooms (zones 4 and 7) and bedrooms (zones 5, 6, 8); the layout of the ventilation network is shown in colour (red for the heated fresh air and blue for the exhaust air).

The building has balanced mechanical ventilation equipped with a heat recovery unit. The constant-air-volume (CAV) ventilation operates a cascade-flow: the fresh air is supplied in the living rooms (i.e.

zones 1, 2) and bedrooms (i.e. zones 5, 6, 8) , and is extracted in *wet rooms* (e.g. bathroom). Standard hygienic flow rates (V_n) are imposed, with a mean fresh airflow rate of 1.2 $m^3/m^2.h$ [29] and a minimum ach of 0.5 h^{-1}. This value is representative for many cold climate countries such as Denmark, Sweden and Finland [30]. As mentioned, it is assumed that the airflow rate can only be boosted up to 50% above V_n. By default, time-constant and space-uniform internal gains with a value of 4.2 W/m^2 are applied to the building model. This value corresponds to the mean power of Norwegian PH [2], while 2.1 W/m^2 is considered in the PHPP tool [31] used for the design of German PH.

3.3 Cold climates

Table 1 Weather characteristics for the 4 geographic locations and the corresponding climatic zone following the ASHRAE classification [32].

	θ_{ym} [°C]	**$I_{tot,rad}$ [W/m²]**	**$\theta_{SH,dim}$ [°C]**	**HDD_{18} [°C.day]**	**Climatic zone N° and name**
Oslo	6.3	110	-20.0	4423	6, *Cold*
Bergen	7.5	87	-11.7	3858	5, *Cool*
Tromsø	2.9	72.6	-14.6	5508	7, *Very-Cold*
Karasjok	-2.5	79	-48.0	7538	8, *Subarctic*

Four different building locations are considered here: *Oslo, Bergen, Tromsø* and *Karasjok*. Some important weather characteristics are given in Table 1, as the annual mean outdoor temperature (θ_{ym}), the mean total radiation on a horizontal surface ($I_{tot,rad}$) and the SH design outdoor temperature ($\theta_{SH,dim}$). It is worth mentioning that $\theta_{SH,dim}$ is here defined as the lowest 3-days mean temperature during a 30-years measurement period. These conditions are rather severe and can be considered as a *cold wave*. It is important to remember this point during the discussion of results. For each location, the heating degree-days based on an outdoor temperature of 18°C (HDD_{18}) are also reported. HDD_{18} enable a correspondence with international climatic zones [32]. Although limited, this set of locations covers a wide range of weather conditions found in countries with cold climates: climatic zones indeed continuously range from category 5 to 8.

3.4 Construction modes

A specific PH standard has been defined for Norway, the NS 3700 [2]. Although not official yet, this standard is considered as the legal requirement for new buildings after 2015, while it is also seen as a basis performance for future Norwegian net-Zero Emission Buildings (nZEB) [33, 34]. The NS 3700 defines a minimal performance requirement for each building component (e.g. external walls, windows) as well as a maximum value admitted for the annual net SH needs (Q_{max}). The SH set-point temperature (T_{set}) is fixed at 21°C in each room when evaluating the net SH needs. In the standard, Q_{max} is made dependent on the local weather conditions and the building compactness. In practice, Q_{max} is adapted as a function of θ_{ym} and A_{fl} using Eq. (1) and (2):

$$Q_{corr} = \max\left(\left(250 - A_{fl}\right), 0\right) / 100 , \quad (1)$$

$$Q_{\max} = 15.0 + 5.4 \times Q_{corr} + \left(2.1 + 0.59 \times Q_{corr}\right) \times \max\left(\left(6.3 - \theta_{ym}\right), 0\right). \quad (2)$$

A first set of building envelope performance has been defined in order to comply with the minimal requirements of NS 3700. This has been verified using the building software SIMIEN [35] equipped with specific modules to check the compliance with Norwegian building regulation. The evaluation method of SIMIEN to calculate the building needs complies with the Norwegian standard NS 3031 [36]: it is essentially based on the simplified dynamic model of the EN 13790 [37] and is validated using the EN 15265 [38]. As four locations are considered, four levels of building envelope performance are defined. This set will be further termed with the acronym NS and is reported in Table 2.a: the net SH needs (Q_{net}) and the maximum net SH power (P_{SH}) are computed using SIMIEN with a constant T_{set} and the default 4.2 W/m² constant gains.

Table 2.a Building envelope performance for the NS case: U-value of external walls ($U_{ext,wall}$), the roof (U_{roof}), the slab (U_{slab}) and the windows (U_{win}), normalized thermal bridges (ψ”), efficiency of the heat recovery (η_{exch}) and infiltration rate at 50 Pa (n_{50}).

NS case	$U_{ext,wall}$ [W/m².K]	U_{roof} [W/m².K]	U_{slab} [W/m².K]	U_{win} [W/m².K]	ψ” [W/m².K]	η_{exch} [%]	n_{50} [1/h]	Q_{net} [kWh/m².y]	Q_{max} [kWh/m².y]	P_{SH} [W/m²]
Oslo	0.15	0.12	0.11	0.72	0.03	85	0.6	18.9	19.2	16.6
Bergen	0.15	0.13	0.11	0.80	0.03	85	0.6	16.0	19.1	12.9
Tromsø	0.14	0.11	0.11	0.72	0.03	85	0.6	26.7	27.8	13.4
Karasjok	0.12	0.09	0.08	0.72	0.03	85	0.6	41.0	41.6	26.3
NS 3700*	0.15	0.13	0.15	0.80	0.03	80	0.6	-		-

**Minimal requirement by building component imposed by the Norwegian PH standard, NS 3700.*

By definition, NS cases are based on a local definition of the passive house. In practice, they lead to moderate envelope performance for milder climate. To make conclusions more general, a second set has been considered with improved envelope performance. This high-performance case, termed HP, is reported in Table 2.b and is taken equivalent to the Karasjok case in Table 2.a. Investigating both sets of building performance leads to a wide range of P_{SH} and weather conditions so that results may be considered as representative for many countries with cold climates.

Table 2.b Building envelope performance for the HP case, as a function of the geographic location.

HP case	$U_{ext,wall}$ [W/m².K]	U_{roof} [W/m².K]	U_{slab} [W/m².K]	U_{win} [W/m².K]	ψ” [W/m².K]	η_{exch} [%]	n_{50} [1/h]	Q_{net} [kWh/m².y]	Q_{max} [kWh/m².y]	P_{SH} [W/m²]
Oslo	0.12	0.09	0.08	0.72	0.03	85	0.6	14.6	19.2	14.5
Bergen	0.12	0.09	0.08	0.72	0.03	85	0.6	10.8	19.1	10.6
Tromsø	0.12	0.09	0.08	0.72	0.03	85	0.6	22.4	27.8	12.1

Furthermore, different construction modes may lead to the same envelope performance (e.g. using masonry or wood). Five possible construction modes that correspond to five different levels of internal thermal mass have been defined using Norwegian technical literature [39], see Table 3. They range from *very-heavy* to *very-light* according to EN 13790 [37]. In practice, the different construction modes have different levels of thermal insulation located inside the building envelope (i.e. in partition walls and ceilings between rooms). Essentially, thermal insulation is often placed in light partition walls in order to improve their acoustic performance. On the contrary, heavy partition walls in concrete do not require acoustic insulation. Therefore, one generally notices that the lower the thermal mass, the lower the thermal transmittance of internal walls. Finally, it is worth mentioning that wooden constructions are commonly used in Nordic countries.

Table 3 Building construction modes: overall building inertia using EN 13790 [37], U-value of floor/ceiling (U_{floor}), partition walls (U_{part}) and bearing walls ($U_{bearing}$).

Construction Type	**Inertia Type**	**Inertia [MJ/K]**	**U_{floor} [W/m².K]**	**U_{part} [W/m².K]**	**$U_{bearing}$ [W/m².K]**
Masonry heavy	*Very-heavy*	86	1.6	3.2	2.8
Mixed wood-masonry	*Heavy*	41	1.6	0.33	2.8
Wooden heavy	*Medium*	35	0.23	0.33	2.8
Masonry light	*Light*	26	0.21	0.33	1.1
Wooden light	*Very-Light*	14	0.21	0.33	0.25

3.5 Air heating modelling

The AH temperature (T_{AH}) is controlled to enforce one reference temperature (T_{ref}) at T_{set}, here taken at 21°C. By default, the air temperature in the main living room is here used (i.e. zone 1). Another common strategy is to use the mean return temperature of the ventilation air ($T_{vent,r}$), an alternative that is also tested. The fresh air temperature after the heat recovery unit and before the heating coil is termed T_{in}. The power to raise T_{in} for the AH is controlled using a Proportional-Integral (PI) action (requiring to apply a time step of ~1 min in simulations). Only a constant T_{set} is considered here. In practice, the extra-power for an intermittent SH will lead to higher T_{AH} (e.g. during the restart phase after a night setback). Consequently, if T_{AH} are too high with a constant heating, the AH will systematically fail with an intermittent SH strategy (as T_{AH} will be even higher).

The ventilation network has been designed by a professional installer using standard products. In this way, realistic ducts lengths and diameters are considered, see Figures 1 and 2. Ducts are modelled using the *Type 31* component [14] and have been coupled to the building component (*Type 56*) using the TRNSYS user interface: ducts thermal losses are injected into the building model as gains while these losses also induce a corresponding temperature drop of the air in the ducts. In the *Type 31*, the air temperature distribution inside the duct is assumed one-dimensional (along the streamwise direction).

The duct length is divided into segments with a uniform temperature. An energy balance is then performed on each segment during their travel through the duct at a finite speed. Therefore, the delay for the temperature to propagate along the duct is also modelled. However, the thermal mass of the duct shell itself is neglected by the *Type 31*. The method here developed to compute the overall heat transfer coefficient from a duct to a room is reported in Annex.

In practice, three scenarios have been considered: without duct insulation, with 5 cm of mineral wool and without duct thermal losses. Performance is first investigated without thermal losses in order to distinguish their specific effect when they are subsequently introduced.

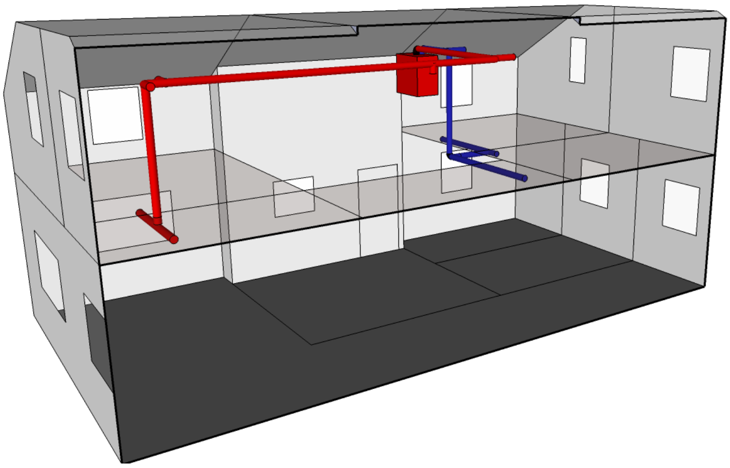


Figure 2 Sketch of the analyzed ventilation network: the heating fresh air is shown in red colour while the return air is in blue.

3.6 Mono-zone analytical approach

The PHPP [31] is a design tool used for German PH, even though it is also employed in many other countries (e.g. Belgium). It assumes a mono-zone model and is based on analytical formula. For instance, the net SH needs (Q_{SH}) are evaluated using a quasi-steady-state approach with a monthly resolution (see e.g. [37]). As explained before, internal gains of 2.1 W/m² are then applied. Regarding the maximal heating power (P_{SH}), the PHPP evaluates it for two critical weather conditions (both in pure steady state). In temperate climates, given the high contribution of solar gains for the SH of passive houses, the P_{SH} is either found in a day with clear sky (i.e. with solar gains) or during a milder day with overcast sky (i.e. with negligible solar gains) [4]. To account for potential variations of the building use, internal gains are reduced to 1.6 W/m² when computing P_{SH} (e.g. during a period of reduced occupancy or activity level) [31]. Knowing P_{SH} and the $\theta_{SH,dim}$ (here specific to the PHPP method), it is then possible to estimate the resulting maximal T_{AH} ($T_{AH,max}$) found during operation:

$$P_{SH} = \alpha V_n C_p \left(T_{AH} - T_{in}\right), \qquad (3)$$

$$T_{in} = \eta_{exch} \left(T_{set} - \theta_{SH,\dim}\right) + \theta_{SH,\dim}, \qquad (4)$$

$$\alpha = V/V_n \ . \qquad (5)$$

where V is the forced airflow rate while V_n is the nominal value. This method is simple and enables to quickly evaluate the $T_{AH,max}$ for a given project. Therefore, the consistency and accuracy of this design approach will be investigated for cold climates. It will be further termed *mono-zone analytical* approach.

4. PERFORMANCE IN STANDARD DESIGN CONDITIONS

The AH performance is here analyzed in Standard Design conditions (STD). They are termed *standard* because these conditions are usually employed to dimension the power of SH systems. They are defined as steady-state conditions with the $\theta_{SH,dim}$ introduced in Table 1 and the building without solar gains. The internal gains are here deliberately left as a free parameter that can be adapted.

4.1 AH temperature during STD

Using the heating power (P_{SH}) in STD from the passive house assessment, the resulting T_{AH} can be evaluated using the *mono-zone analytical* approach. The T_{AH} can also be evaluated numerically using the TRNSYS *multi-zone* building model. Comparison between both methods is reported on Table 4 and leads to the following conclusions:

- For the multi-zone simulations, T_{AH} is ranging from a few degrees depending on the construction mode considered. This is due to different temperature distributions inside the building giving rise to different thermal losses through the envelope.
- The T_{AH} using the mono-zone design approach is close to the multi-zone results. In fact, if the mono-zone approach concludes that the AH is impossible, the same conclusion is found using the multi-zone approach even though some critical cases exist when the T_{AH} is close to 55°C.
- The colder the climate, here $\theta_{SH,dim}$, the higher the P_{SH} and the higher the resulting T_{AH}. Even for the NS cases where the performance requirements of the NS 3700 are adapted to the local climate (i.e. the colder the climate, the better the insulation), P_{SH} increases significantly with decreasing $\theta_{SH,dim}$ so that T_{AH} is still strongly dependent on the local climate. With hygienic flow rates of 0.5 h^{-1}, test cases with a resulting P_{SH} lower than 16 W/m² lead to a T_{AH} lower than 55°C while P_{SH} lower than 10 W/m² gives T_{AH} lower than 40°C. With a forcing of 50% above V_n, a P_{SH} lower than 24 W/m² leads to a T_{AH} lower than 55°C and a P_{SH} lower than 15 W/m² to a T_{AH} lower than 40°C. These numbers can be easily extrapolated to any specific country by adjusting results with the corresponding ventilation airflow rates. For instance, German PH have typically ach of 0.3-0.4 h^{-1} [4]. Then, the P_{SH} should be lower than ~10 W/m² to prevent T_{AH} higher than 55°C. A criterion that is extensively used in the German definition of the PH standard.
- In practice, the AH is possible using hygienic flow rates for Bergen, Oslo and Tromsø if the 4.2 W/m² gains are applied. On the contrary, no requirement on internal gains appears if a 50%

forcing of the ventilation flow rates is applied. For the extreme case of Karasjok, only the application of a 50% ventilation forcing in combination with 4.2 W/m² internal gains makes the AH possible even though the resulting T_{AH} is critical

*Table 4 Possibility to implement AH and corresponding T_{AH}: comparison between the **analytical mono-zone**, the **multi-zone** methods in **STD** as well as the **multi-zone** model during a **TMY** (using closed internal doors).*

Building model Operating conditions				**Mono-zone STD**			**Multi-zone STD**		**Multi-zone TMY**		
V	**Gains [W/M²]**	**Location**	**Case**		**P_{SH} [W/M²]**	**T_{AH} [°C]**		**T_{AH} [°C]**		**$T_{AH,MAX}$ [°C]**	**$T_{AH,95\%}$ [°C]**
V_n	0.0	Oslo	NS	⊗	20.8	-	⊗	-	⊙	[51.1;55.0]	[46.5;52.3]
			HP	⊗	18.7	-	⊙	[52.6;55.0]	⊕	[46.2;53.3]	[41.6;46.3]
		Bergen	NS	⊗	17.1	-	⊗	-	⊕	[45.6;50.6]	[42.0;45.9]
			HP	⊕	14.6	52.0	⊕	[46.0;50.9]	⊕	[40.8;45.1]	[37.9;41.2]
		Tromsø	NS	⊗	17.6	-	⊙	[51.8;55.0]	⊙	[49.0;55.0]	[46.0;51.7]
			HP	⊗	16.3	-	⊕	[48.3;53.8]	⊕	[45.6;51.2]	[43.0;47.8]
		Karasjok	NS	⊗	30.5	-	⊗	-	⊗	-	-
	4.2	Oslo	NS	⊗	16.6	-	⊙	[49.5;55.0]	⊕	[41.2;48.7]	[36.7;41.4]
			HP	⊕	14.5	50.0	⊕	[43.6;48.2]	⊕	[36.3;42.3]	[32.1;35.8]
		Bergen	NS	⊕	12.9	44.9	⊕	[42.7;47.1]	⊕	[35.8;40.0]	[32.6;35.3]
			HP	⊕	10.6	41.7	⊕	[36.9;40.8]	⊕	[31.5;34.6]	[28.5;30.7]
		Tromsø	NS	⊕	13.4	48.5	⊕	[42.7;47.5]	⊕	[39.3;44.5]	[36.6;40.5]
			HP	⊕	12.1	45.2	⊕	[39.2;43.7]	⊕	[36.2;40.7]	[33.7;37.2]
		Karasjok	NS	⊗	26.3	-	⊗	-	⊗	-	-
3/2 V_n	0.0	Oslo	NS	⊕	22.0	50.8	⊕	[45.9;50.5]	⊕	[40.4;45.6]	[37.6;41.1]
		Tromsø	NS	⊕	18.6	46.0	⊕	[41.2;44.9]			
		Karasjok	NS	⊗	32.4	-	⊗	-	⊕	[48.0;52.9]	[45.7;49.2]
	4.2	Oslo	NS	⊕	17.8	43.9	⊕	[40.0;44.0]	⊕	[34.0;38.7]	[31.3;34.3]
		Karasjok	NS	⊗	28.2	-	⊙	[49.9;55.0]	⊕	[45.1;49.8]	[38.7;44.1]

Symbol ⊕ when AH possible, ⊗ when AH impossible and ⊙ when limit case dependent on construction mode.

4.2 Temperature distribution during STD

Using the multi-zone building model, it is also possible to investigate the temperature distribution within the passive house during STD. Typical results, reported on Table 5, can be summarized in the following way:

- With closed internal doors, large temperature differences take place in the building. Zones with the highest temperature ($T_{op,max}$) are bedrooms while the lowest temperatures ($T_{op,min}$) are found in the bathrooms (without SH). These differences are again dependent on the construction mode: constructions with a high thermal insulation in partition walls have higher temperature differences between zones.

- The higher the T_{AH}, the larger the temperature differences inside the building. It can be checked by comparing different envelope performance for a same climate (i.e. HP compared to NS sets), different climates with a same envelope performance (i.e. HP set), or a same test case by reducing the internal gains from 4.2 to 0.0 W/m².
- The opening of the internal doors is an efficient way to homogenize temperature inside the building. Therefore, the difference in temperature distribution between construction modes is reduced (because the thermal conduction through walls is less dominant). Nevertheless, highest temperatures are still found in bedrooms (typically ~22°C).
- Applying a SH at the same T_{set} in bathrooms significantly reduces $T_{op,min}$ and, thus, improves the thermal comfort. Nevertheless, it does not affect significantly $T_{op,max}$ found in the building.
- From a practical point of view, the centralized AH generates high temperatures in bedrooms while it is known that users may require lower temperatures during the night. A SH night setback does not solve the problem as the characteristic time scale of PH is large (i.e. the temperature does not have the time to decrease significantly during one night). Furthermore, the temperature may even be prohibitive if internal doors are closed and a light building structure is considered. Even for the milder climate of Bergen, this is already the case.

*Table 5 Temperature distribution during **STD** using the **multi-zone** building model: maximal and minimal operative temperature for all the construction modes[*] ($T_{op,max}$ and $T_{op,min}$, respectively).*

Internal doors				**Closed**		**Open**		**Closed**	
SH in bathrooms				**NO**		**NO**		**YES**	
V	**Gains [W/M²]**	**Location**	**Case**	**$T_{op,max}$ [°C]**	**$T_{op,min}$ [°C]**	**$T_{op,max}$ [°C]**	**$T_{op,min}$ [°C]**	**$T_{op,max}$ [°C]**	**$T_{op,min}$ [°C]**
V_n	0.0	Bergen	HP	[22.7;27.7]	[19.2;20.3]	[21.6;22.2]	[19.5;20.4]	[22.7;27.7]	[20.3;20.4]
	4.2	Oslo	NS	[23.7;31.3]	[17.5;19.9]	[21.8;22.7]	[19.7;19.9]	[23.8;31.3]	[19.3;20.5]
			HP	[23.4;30.7]	[18.6;20.2]	[21.8;22.6]	[20.0;20.1]	[23.4;30.7]	[20.1;20.2]
		Bergen	NS	[22.9;28.8]	[18.2;20.1]	[21.6;22.3]	[19.9;20.1]	[22.9;28.8]	[20.1;20.2]
			HP	[22.7;27.7]	[19.3;20.3]	[21.6;22.2]	[20.2;20.3]	[22.7;27.7]	[20.3;20.4]
3/2 V_n	0.0	Oslo	NS	[23.9;31.3]	[17.1;19.8]	[21.9;22.8]	[19.5;19.7]	[24.0;31.3]	[19.9;20.0]
			HP	[23.7;30.3]	[18.2;20.0]	[21.9;22.8]	[19.8;20.2]	[23.7;30.3]	[20.0;20.1]
		Bergen	NS	[23.2;30.3]	[17.1;20.0]	[21.7;22.5]	[19.6;19.9]	[23.3;28.8]	[20.0;20.1]
			HP	[23.0;28.1]	[18.7;20.2]	[21.7;22.4]	[19.9;20.2]	[23.1;28.2]	[20.2;20.3]
	4.2	Oslo	NS	[23.2;29.2]	[16.8;19.4]	[21.7;22.5]	[19.7;19.9]	[23.3;29.2]	[19.9;20.0]
			HP	[23.0;28.2]	[19.2;20.2]	[21.7;22.4]	[20.0;20.1]	[23.0;28.2]	[20.1;20.2]
		Bergen	NS	[22.5;26.7]	[18.9;19.6]	[21.5;22.1]	[19.9;20.1]	[22.6;26.7]	[20.1;20.2]
			HP	[22.4;26.0]	[19.7;20.3]	[21.5;22.0]	[20.2;20.3]	[22.4;26.0]	[20.3;20.4]
		Karasjok	NS	[25.2;34.1]	[17.7;19.6]	[22.3;23.4]	[19.4;19.5]	[25.2;34.1]	[18.4;19.6]

[*]*As all the construction modes are considered, the range of values spanned by them is reported between brackets.*

By default, the living room was taken as the reference temperature for the AH control. Another common strategy is to use the mean return temperature of the ventilation air ($T_{vent,r}$). For a same T_{set}, this strategy essentially generates higher building temperatures: the average temperature of the building increases of about ~3°C. It can be easily understood as $T_{vent,r}$ is mainly based on the wet rooms temperature (e.g. bathrooms) which always presents the lowest temperature when using AH. In fact, the temperature difference between rooms is almost unchanged between the two control strategies. Therefore, this change of reference temperature for the control can essentially be considered as a shift in the average building temperature.

4.3 Influence of the internal gains definition

As only one temperature is used to control the centralized heating coil, one may suspect that the distribution of the internal gains may generate configurations between zones that are less favourable in terms of thermal comfort. Furthermore, internal gains are not constant so that periods with lower gains may occur.

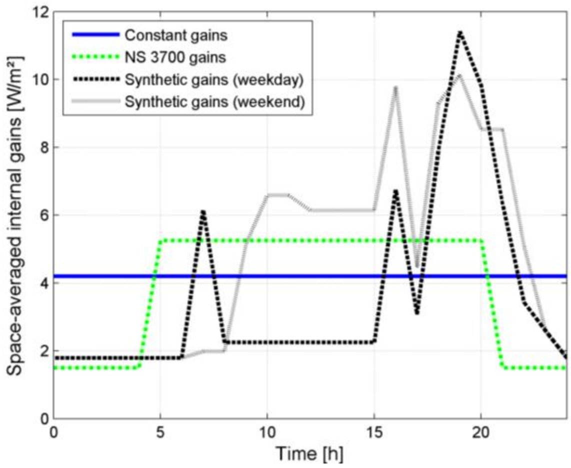


Figure 3 Daily profiles of the three different internal gains: for the sake of the comparison, synthetic gains have been space-averaged while the constant and the NS 3700 gains are by definition space-uniform.

At this stage, only time-constant and space-uniform internal gains were used in the building model. Two alternative profiles for internal gains are now introduced:

- Firstly, the internal gains of the **NS 3700** are strictly applied. These are also space-uniform but using different levels during the day and night periods: the night is characterized by lower internal gains from lighting and equipments, see Fig. 3.
- Secondly, time- and space-varying internal gains are created (with 1-hour resolution). These **synthetic** gains have been generated in a way that (a) they have a mean value of 4.2 W/m² as the NS 3700, (b) the yearly energy consumption of electric appliances is compatible with measurements [40, 41] (also using typical operating cycle lengths), (c) as well as daily profiles are

consistent with a Norwegian time-of-use survey [42]. The location of the different electric appliances has been allocated in a meaningful way according to the building spatial organization, see Fig. 1. The time-averaged value of the synthetic gains in the living room is 6.1 W/m², ~4 W/m² in bedrooms and < 1.0 W/m² in bathrooms. A distinction is done between weekdays and weekends, see Fig. 3. Hourly profiles are defined on a daily basis and are unchanged throughout the year.

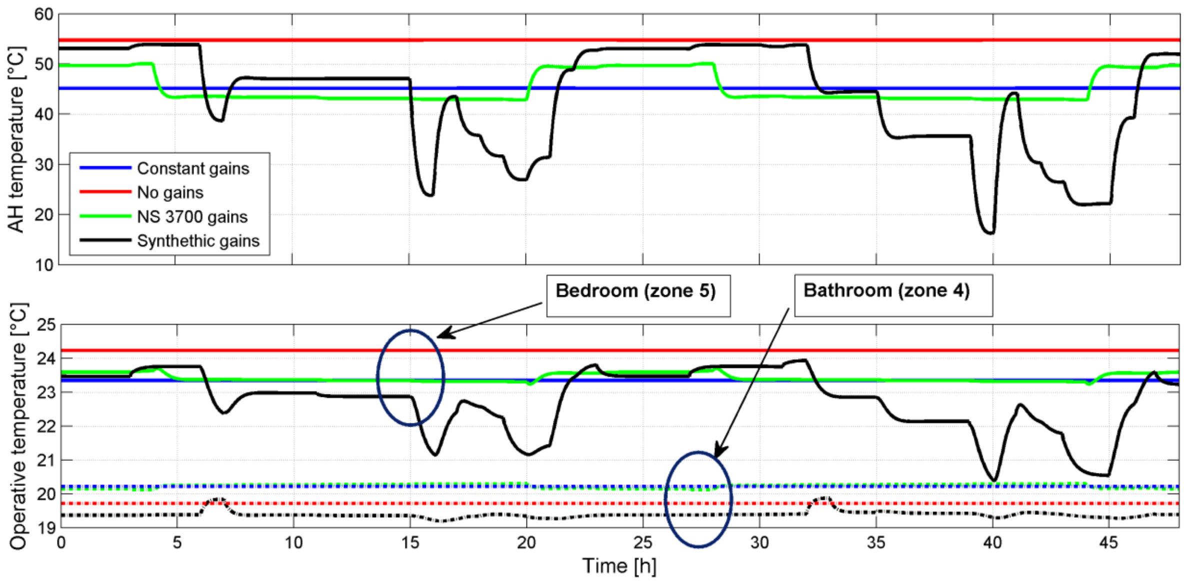


*Figure 4 T_{AH} and resulting T_{op} for the different internal gains during **STD** for the very-heavy HP building located in Oslo (with closed internal doors and hygienic ventilation flow rates): solid line for the bedroom (zone 5) and dashed line for the bathroom (zone 4) in the picture below.*

The influence of the internal gains on T_{AH} is well illustrated with Fig. 4. Three interrelated effects can be observed:

1. In period of lower internal gains, the T_{AH} increases accordingly. Comparing the constant and the NS 3700 gains, one clearly notices that the T_{AH} increases during the night when internal gains are lower. Synthetic gains are very small in the living room during the night so that T_{AH} converges to value without internal gains (i.e. 0 W/m²). As the T_{AH} increases, the T_{op} rises accordingly in other rooms with an inlet ATD as bedrooms (e.g. zone 5). Inversely, the T_{AH} decreases in period of higher internal gains.
2. Nevertheless, synthetic gains vary quickly in time and are variable in space. With synthetic gains, strong internal gains occur between 4 PM and 8 PM in the living room. T_{AH} is significantly reduced (e.g. 16°C) and this induces large variations in rooms equipped with inlet ATD. It clearly shows that there is a risk if the zone taken as reference temperature T_{ref} for the AH control is prone to large fluctuating gains. The impact is not significant for wet rooms with an outlet ATD as bathrooms: the temperature remains almost constant. They are indeed located downstream in the cascade-flow and thus not directly affected by the T_{AH}.

Nevertheless, it should not be concluded prematurely that bathrooms or the return temperature ($T_{vent,r}$) are better candidates to place the sensor for the T_{ref}. As mentioned, these rooms are not directly affected by the T_{AH} so that a small change in these reference temperatures may require a large modification of the T_{AH} upstream. In practice, no improvement as regards the sensitivity to internal gains fluctuations has been monitored using the bathroom temperature or $T_{vent,r}$ as the reference temperature. In fact, interactions between physical phenomena are very complex so that it is difficult to predict the best room temperature to use as a reference. In present case, the corridor (zone 2) with its low internal gains and its strong coupling with the bedrooms and the living room appeared to be the best T_{ref} to limit fluctuations. Let us mention that fluctuations reported for synthetic gains are strongly linked to the PI control strategy. Other control approaches may reduce this effect, for example, by letting the living room temperature fluctuate to some extent around the set-point temperature T_{set}. Another solution is to establish the reference building temperature as a combination of several temperature sensors. Preliminary results have shown that the resulting T_{AH} would be less affected by the variations of internal gains. Nevertheless, as in point (1), an overall change of the internal gains in the buildings will always influence the T_{AH} magnitude.

3. Another effect is dependent on the internal gains variations. In fact, the temperature in bathrooms is not minimal when internal gains are null. In practice, the minimal value is obtained with the non-uniform synthetic gains. Compared to other internal gains, synthetic gains are large in the living room and relatively low in the bathrooms. The spatial distribution of internal gains between rooms is thus important if a proper evaluation of the thermal comfort has to be done.

5. PERFORMANCE DURING TYPICAL METEOROLOGICAL YEAR

5.1 Comparison against STD

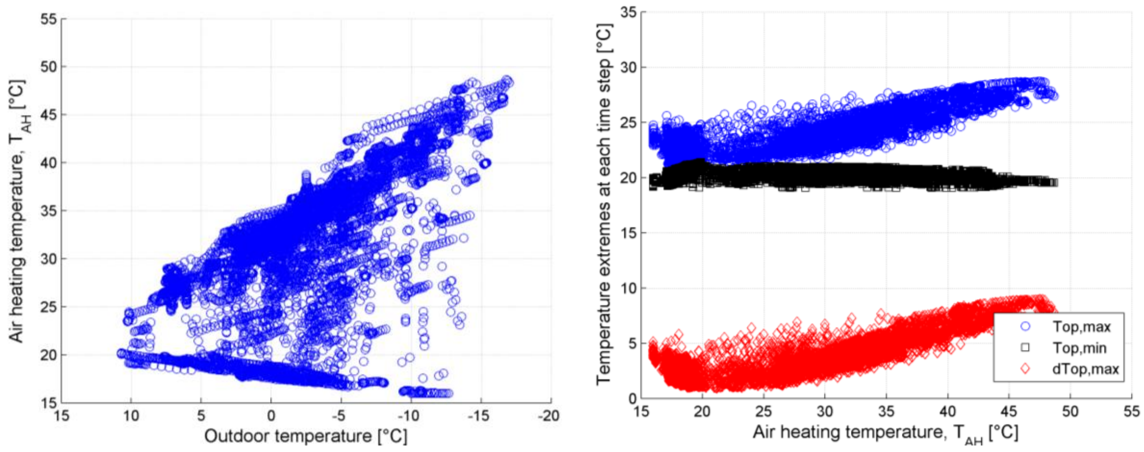
An interesting question is to check how standard design conditions (STD) are representative of everyday operating conditions. Furthermore, it should also be confirmed that STD are representative for the most severe operating conditions. In the PHPP, design conditions are a mild day with an overcast sky and a cold day with clear sky. In our case, STD only correspond to the design outdoor temperature, $\theta_{SH,dim}$ in Table 1, without solar gains.

In order to investigate this, all-year simulations are performed using a Typical Meteorological Year (TMY), here generated using Meteonorm. In the following considerations, no solar shading strategy is applied during the heating period while internal gains are strictly constant and uniform.

Firstly, the maximal and 95% percentile T_{AH} are reported on Table 4 (termed $T_{AH,max}$ and $T_{AH,95\%,}$ respectively). These temperatures are significantly lower than values obtained using STD conditions.

It somehow shows that STD conditions are severe when evaluating the maximal T_{AH}. It is not surprising given the definition of $\theta_{SH,dim}$ as a cold wave. In practice, some cases that were critical in STD (i.e. T_{AH} close to 55°C) present acceptable temperatures in TMY.

Secondly, rooms that were identified as the warmest and coldest in STD are the same when using a TMY.



(a) (b)

*Figure 5 Hourly T_{AH}, $T_{op,min}$, $T_{op,max}$ and $dT_{op,max}$ during a **TMY** with **constant gains** for the very-light NS building located in Oslo (with closed internal doors and hygienic ventilation flow rates).*

Thridly, hourly T_{AH}, $T_{op,max}$ and $T_{op,min}$ can be analyzed during a TMY along with the maximal temperature difference between rooms, $dT_{op,max}$. In Fig. 5, results are presented for the very-light NS building located in Oslo, using hygienic ventilation airflow rates as well as closed internal doors. A similar behaviour was found for all other test cases. From Fig. 5(a), one clearly notices that $T_{AH,max}$ is found during the coldest day. The scattering of points in the curve is essentially induced by the variation of solar gains (as internal gains are here constant). A deeper analysis of results demonstrates that highest T_{AH} are found during hours without sun. Furthermore, Fig. 5(b) also shows that the temperature differences in the building are mainly driven by the T_{AH}, the trend is indeed almost linear.

For cold climates, it confirms that selecting a cold day without sun as AH design conditions makes sense. In other words, one should not expect more severe operating conditions. The STD conditions belong to this category. Given the definition of $\theta_{SH,dim}$ as the coldest event in 30 years, STD can be considered as the most conservative for a AH design. Nevertheless, this security margin induces significantly higher $T_{AH,max}$ so that it could disqualify the AH in situations where T_{AH} would be acceptable during usual operation.

5.2 Influence of the internal gains definition

The internal gains resolution is now investigated for the TMY. Compared to the discussion using STD, the effect of the internal gains is now combined with solar gains and a changing outdoor temperature. Sample results are reported on Tables 6 to 8 for the different gains, using the Oslo climate and the NS case. The overall maximal and minimal temperatures found in the building during the heating season is here considered as the most severe event ($T_{op,g,max}$ and $T_{op,g,min}$, respectively).

The three effects reported for STD in Subsection 4.2 can be found for the TMY. For instance, the $T_{AH,max}$ is increased for the NS 3700 gains and the synthetic gains compared to the constant gains. This is due to a lower magnitude of gains during the nighttime. The effect of the internal gains variation on T_{AH} can also be illustrated by comparing Fig. 5(a) and 6(a): for a same outdoor temperature, the simulation with synthetic gains can have higher T_{AH}. If the $T_{AH,max}$ increases, the maximal temperature in the bedrooms will increase accordingly, see $T_{op,g,max}$ in Tables 6 to 8. On the contrary, the minimal temperature ($T_{op,g,min}$) is strongly affected by the synthetic gains. As already mentioned, a different temperature distribution in the building occurs due to different levels of internal gains applied to the different rooms (i.e. here lower in bathrooms). Comparing Fig. 5(b) and 6(b), one notices a shift of the $T_{op,min}$ curve due to reduced gains in the bathrooms, while the impact on the $T_{op,max}$ curve is rather limited.

*Table 6 Thermal comfort during a **TMY** for the Oslo climate with **constant gains, no shading** (with the NS case, closed doors and hygienic ventilation flow rates).*

Constr. Mode	**$T_{op,g,max}$ [°C]**	**$T_{op,g,min}$ [°C]**	**$T_{AH,max}$ [°C]**
Very-heavy	22.4	20.0	41.2
Heavy	23.6	18.3	42.6
Medium	25.5	19.0	42.8
Light	26.1	18.9	45.7
Very-light	27.6	19.0	48.6

*Table 7 Thermal comfort during a **TMY** for the Oslo climate with **NS 3700 gains, no shading** (with the NS case, closed doors and hygienic ventilation flow rates).*

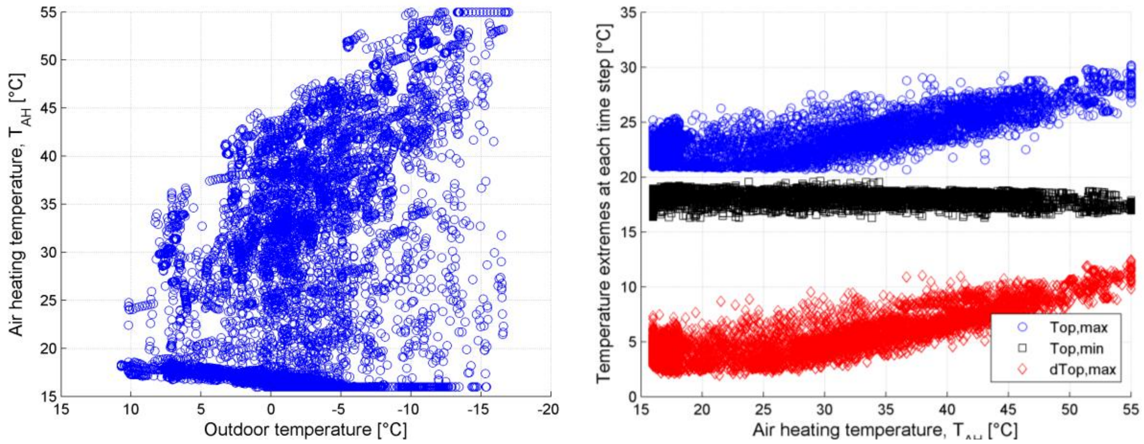
Constr. Mode	**$T_{op,g,max}$ [°C]**	**$T_{op,g,min}$ [°C]**	**$T_{AH,max}$ [°C]**
Very-heavy	22.6	19.9	45.8
Heavy	24.1	18.0	47.6
Medium	26.1	18.9	48.2
Light	26.8	18.8	51.6
Very-light	28.3	19.0	55.0

*Table 8 Thermal comfort during a **TMY** for the Oslo climate with **synthetic gains, no shading** (with the NS case, with closed doors and hygienic ventilation flow rates).*

Constr. Mode	**$T_{op,g,max}$ [°C]**	**$T_{op,g,min}$ [°C]**	**$T_{AH,max}$ [°C]**
Very-heavy	22.8	18.8	49.3

Heavy	25.6	16.5	51.2
Medium	27.1	16.9	52.4
Light	27.6	16.3	54.6
Very-light	28.6	15.9	55.0

In conclusion, coldest days without sun still give the more severe temperature configuration in the building. $T_{op,max}$ is bounded by simulations without internal gains (i.e. 0 W/m²). On the contrary, the $T_{op,min}$ depends on the internal gain distribution between rooms.



(a) (b)

Figure 6 Hourly T_{AH}, $T_{op,min}$, $T_{op,max}$ and $dT_{op,max}$ during a ***TMY*** *with* ***synthetic gains*** *for the very-light NS building located in Oslo (with closed internal doors and hygienic ventilation flow rates).*

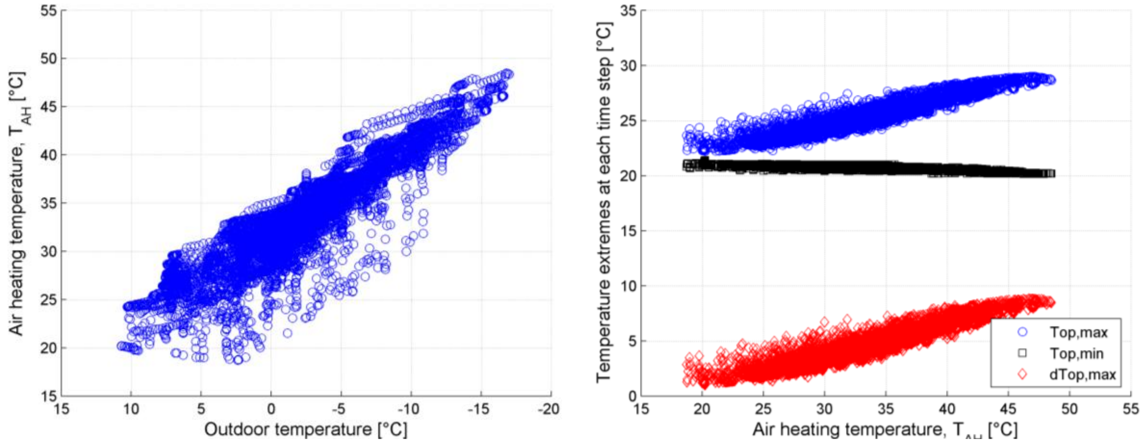
5.3 Influence of the shading strategy

Results up to now were obtained without solar shading. Since low solar altitudes often characterize cold climates, users commonly use shading to prevent glare. The effect of the shading on the AH performance is then analyzed. An external blind is applied at each window and is closed when the solar radiation on the corresponding façade is higher than 140 W/m². When closed, the shading factor is equal to 0.9. Results are reported in Table 9 and should be compared to Table 6. This shading strategy is quite effective: for the Oslo NS case, it increases the net SH needs from [16.2;17.5] to [19.8;21.6] kWh/m².an.

The shading does not reduce the overheating. It translates the fact that, in wintertime, solar gains do not significantly contribute to the most severe events generating the largest temperature differences. This behaviour is well illustrated when comparing the shading influence between Fig. 5 and 7. The highest T_{AH} are found when the sun is absent, a situation already existing without solar shading. The effect of the shading is only translated by a reduced scattering of the points. These investigations suggest that an extended analysis of the shading influence is not required for the AH design. Nevertheless, a larger set of shading configurations should also be tested in order to rigorously confirm this last conclusion.

*Table 9 Thermal comfort during a **TMY** for the Oslo climate with **constant gains and shading** (with the NS case, closed doors and hygienic ventilation flow rates).*

Constr. Mode	**$T_{op,g,max}$ [°C]**	**$T_{op,g,min}$ [°C]**	**$T_{AH,max}$ [°C]**
Very-heavy	22.6	20.0	41.6
Heavy	23.6	18.3	42.5
Medium	25.9	19.0	42.5
Light	26.6	18.9	45.3
Very-light	27.9	20.0	48.5



(a) (b)

*Figure 7 Hourly T_{AH}, $T_{op,min}$, $T_{op,max}$ and $dT_{op,max}$ during a **TMY** with **constant gains and solar shading** for the very-light building located in Oslo (with closed internal doors and hygienic ventilation flow rates).*

6. INFLUENCE OF VENTILATION DUCTS LOSSES

The influence of the ventilation duct losses is now investigated. Two levels of duct insulation are considered: *without* and *with* 5 cm insulation (using mineral wool). Results are first reported for STD conditions in Table 10. The configuration with closed internal doors is only reported as it gives rise to the largest temperature differences between rooms.

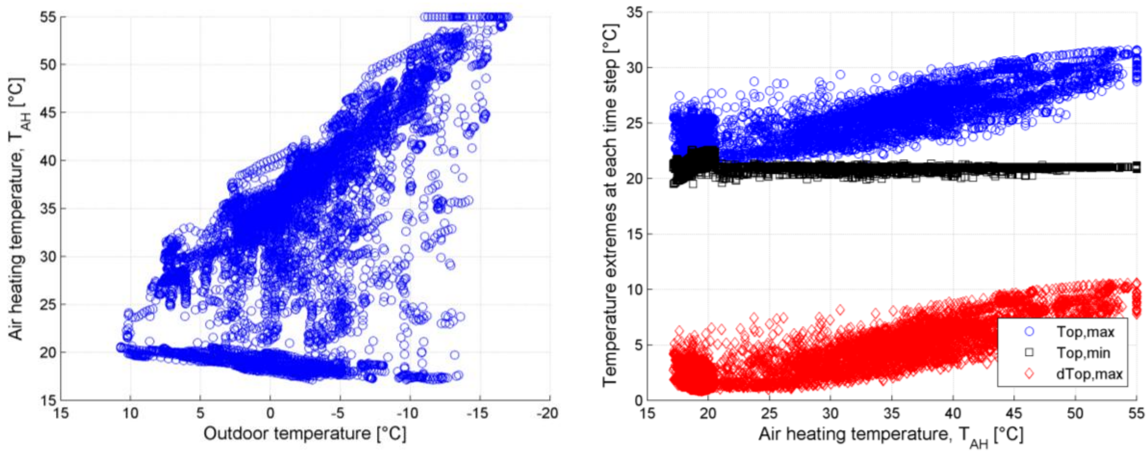
With non-insulated ducts, losses are significant [4]. In the present layout of the ventilation network, there is a long distance between the heating coil (i.e. in the air-handling unit) and ATDs in the living room. In practice, with an inlet temperature of 50°C, a temperature drop of ~15°C can take place before the fresh air is injected in the living room. Even though this drop is large, the T_{AH} is only increased by 2°-5°C by thermal losses (compared to the case without losses). It means that losses contribute significantly in the heating of the building. Particularly, losses contribute to the living-room heating by an increase of the temperature in neighbouring rooms. In the present test case, a large amount of ventilation duct losses is injected in the corridor (i.e. zone 2). Furthermore, this increases the bathrooms temperature that is now comparable to the living room.

With 5 cm insulation, the influence of duct losses is reduced to a large extent. The performance is indeed very close to the reference case without losses.

*Table 10 Influence of duct losses on thermal comfort during **STD** with closed internal doors: T_{AH}, $T_{op,max}$ and $T_{op,min}$ for all the construction modes*[*].

Distribution losses				**No losses**			**Non-insulated ducts**			**Ducts with 5cm insulation**		
V	**Gains [W/m²]**	**Location**	**Case**	**$T_{op,max}$ [°C]**	**$T_{op,min}$ [°C]**	**T_{AH} [°C]**	**$T_{op,max}$ [°C]**	**$T_{op,min}$ [°C]**	**T_{AH} [°C]**	**$T_{op,max}$ [°C]**	**$T_{op,min}$ [°C]**	**T_{AH} [°C]**
V_n	4.2	Oslo	NS	[23.7;31.3]	[17.5;19.9]	[49.5;55.0]	[24.6;30.3]	[16.7;20.2]	[52.2;55.0]	[23.9;31.1]	[17.9;20.1]	[50.5;55.0]
		Bergen	NS	[22.9;28.8]	[18.2;20.1]	[42.7;47.1]	[23.6;31.2]	[19.4;20.4]	[44.5;52.4]	[23.1;29.5]	[18.5;20.3]	[43.1;48.5]
3/2 V_n	0.0	Oslo	NS	[23.9;31.3]	[17.4;19.8]	[45.9;50.5]	[24.7;32.6]	[18.7;20.1]	[47.6;53.4]	[24.1;31.9]	[17.7;19.9]	[46.3;51.6]
		Bergen	NS	[23.2;28.9]	[17.8;20.0]	[41.2;44.6]	[23.9;30.8]	[19.0;20.3]	[42.7;48.3]	[23.3;29.3]	[18.1;20.1]	[41.6;45.5]
	4.2	Oslo	NS	[23.2;29.2]	[18.4;20.0]	[40.0;44.0]	[23.2;31.1]	[19.4;20.1]	[41.3;47.5]	[23.4;29.7]	[18.6;20.0]	[40.4;44.8]
		Bergen	NS	[22.5;26.7]	[18.9;20.1]	[35.3;38.0]	[23.0;28.2]	[19.6;20.3]	[36.3;40.7]	[22.6;27.1]	[19.0;20.2]	[35.5;38.7]
		Karasjok	NS	[25.2;34.1]	[17.7;19.6]	[49.9;55.0]	[26.1;33.3]	[16.0;19.8]	[51.6;55.0]	[25.4;33.9]	[18.0;19.6]	[50.3;55.0]

[*]As all the construction modes are considered, the range of values spanned by them is reported between brackets.



(a) (b)

*Figure 8 T_{AH}, $T_{op,min}$, $T_{op,max}$ and $dT_{op,max}$ during a **TMY** with **constant gains and non-insulated ducts** for the very-light building located in Oslo with (with closed internal doors and hygienic ventilation flow rates).*

The performance in TMY is now investigated. The temperature evolution can here be compared for the case of Oslo, without (Fig. 5) and with losses (Fig. 8). Again, for a given outdoor temperature, the T_{AH} can be higher with distribution losses. On the contrary, the evolution of $T_{op,max}$ as a function of T_{AH} is not strongly affected by losses. Furthermore, lowest outdoor temperatures are still representative of the most severe temperature distributions in the building. Finally, as in STD conditions, 5 cm insulation around ventilation ducts significantly reduces the influence of distribution losses (but results are not reported here).

7. CONCLUSIONS AND DISCUSSION

This work investigates the air heating (AH) utilized as space heating (SH) for passive houses (PH) in cold climates. This paper first presents a list of challenges for the AH. Then, questions related to the system thermal dynamics and to the possibility to apply AH from a conceptual point of view are investigated. This is done using detailed dynamic simulations (here TRNSYS) on a typical detached house typology. The AH is here based on one centralized heating coil. Its performance is first analyzed for standard design conditions (STD), defined as an outdoor SH design temperature ($\theta_{SH,dim}$) and no solar gains, and subsequently using a Typical Meteorological Year (TMY). Results, based on a set of ~1200 simulations, lead to the following conclusions in terms of physics and AH design. Even though investigations are initially based on the Norwegian context, test cases have been selected enough general to be representative for other countries with a cold climate:

- For the wide range of conditions investigated, results showed that the **maximal value of the AH temperature** ($T_{AH,max}$) is found during the coldest days without sun.
- The **maximal value of the AH temperature** ($T_{AH,max}$) can be evaluated analytically using the maximal SH power computed during the envelope design (P_{SH}) and assuming a mono-zone building. Although very simple, it is an easy and convenient way to screen the AH potential for a specific project (i.e. building type and location). Using an air change rate of 0.6 h^{-1} representative of many countries with a cold climate, simulations have shown that a P_{SH} lower than ~16 W/m² leads to a T_{AH} lower 55°C while a P_{SH} lower than ~10 W/m² leads to a T_{AH} lower than 40°C. A same approach is used in the PHPP for the German PH standard. The air change rate is then 0.3-0.4 h^{-1} so that the P_{SH} should then be lower than 10 W/m² in order to limit the T_{AH} below 55°C.
- In theory, the **maximal value of the AH temperature** ($T_{AH,max}$) should be evaluated in STD without internal gains. The standard $\theta_{SH,dim}$ is here taken as the lowest 3-days mean temperature in 30 years and thus corresponds to a cold wave. This scenario appears too conservative and would disqualify the AH approach in many building projects while the T_{AH} is in fact acceptable during usual operating conditions. Furthermore, results have shown a strong impact of the internal gain magnitude on T_{AH}. During usual building operation, they vary in time as they are strongly related to the user behaviour. It is thus wise to keep the internal gains at a low level when evaluating the $T_{AH,max}$ (e.g. no gains at all). On the contrary, the outdoor design temperature could be taken higher than the standard $\theta_{SH,dim}$. For a given project, this selection is a trade-off between security and the $T_{AH,max}$ that could be accepted. Furthermore, the presence of a complementary peak-heating system can justify the selection of a lower design outdoor temperature. Finally, for PH with a time constant longer than 3 days [37], the damping effect of the thermal mass would justify the use of a less severe definition of the $\theta_{SH,dim}$. It is consistently done in the Swedish PH definition [3] where the standard design temperature can be based up to a 12-days average temperature.

- The AH based on a centralized heating coil leads to an **uneven temperature distribution** inside the building. A multi-zone building analysis is then required for a proper design. This temperature distribution is strongly related to the building architectonic properties (i.e. level of insulation in internal walls).
- Furthermore, the **internal gains and their variations** should be considered to have an accurate assessment. They have a strong impact as they modify significantly the thermal behaviour of the building. They influence the temperature distribution in the building but also impact the control of the AH. In design phase, the reference building temperature (T_{ref}) for the AH control should be selected considering this effect. Finally, present investigations have not shown a dominant effect of solar gains.
- As higher T_{AH} lead to larger temperature differences inside the building, coldest days without sun also lead to the most severe **temperature distribution**. Therefore, the AH design could be limited to these conditions. In fact, investigations using TMY did not show any other critical configurations, even considering thermal losses from ventilation ducts.
- **Thermal losses from ventilation ducts** can be large and should be integrated in the AH design. By definition, these losses are internal gains so that their spatial variation influences the temperature distribution between rooms. Nevertheless, thermal losses from ducts may further influence the T_{AH}. Firstly, a large temperature drop can occur in the ducts between the air-handling unit and the injection in rooms from ATDs. Secondly, the thermal losses can significantly contribute to the SH by rising the temperature in thermal zones including long duct sections. When combined, these two opposite effects can lead to higher T_{AH} compared to the case without losses. Consequently, with a limited insulation, losses should be considered for both the evaluation of the temperature distribution and the $T_{AH,max}$. For this last application, the mono-zone analytical formula is not able to capture the physical phenomena anymore. With 5 cm insulation around ducts, the influence of thermal losses can be neglected. Finally, given the magnitude of losses, ducts should be well insulated if they have to cross an unheated building zone (e.g. a basement).
- A **specific building typology** have been investigated along with a standard centralized AH strategy. If the AH covers all the SH needs, it does not seem to be well adapted. In practice, the T_{AH} are acceptable for the climate of the Bergen, Oslo and Tromsø but are prohibitive for the case of Karasjok. Nevertheless, even for the milder climates, high temperatures are found in bedrooms, temperatures that are most probably not acceptable for users. Furthermore, the AH control does not give any flexibility for the user to reduce the temperature locally in bedrooms (even though opening the internal doors can be a part of the solution). These results suggest that AH should preferably be combined with a peak-load heating system and/or that the number of heating coils should be increased. In these cases, the SH simplification induced by the AH strategy would be reduced so that its opportunity can be questioned.

Investigations were performed using dynamic simulations assuming a perfect mixing in each zone. Given the number of test cases, this was a necessary simplification in order to have an insight into the whole-year thermal comfort at an acceptable computational cost. Nevertheless, this assumption should be further investigated even though some existing works confirm that this hypothesis is reasonable (see e.g. [11]). Furthermore, field measurements could also complement results and crosscheck conclusions. The present research can serve as a basis in order to define the critical test cases that deserve an extended analysis, as detailed measurements.

8. NOMENCLATURE

A_{fl} = net heated surface

Ach = air change rate

AH = air heating

ATD = air terminal device

C_d = discharge coefficient for doors

HDD_{18} = heating degree-days based on an 18°C outdoor temperature

η_{exch} = efficiency of the heat recovery unit

n_{50} = infiltration rate at 50 Pa

$I_{tot,rad}$ = mean total radiation on horizontal

PI = proportional-integral control

P_{SH} = maximal net SH power with constant T_{set}

Q_{SH} = net SH energy needs

Q_{max} = maximum net SH needs in NS 3700

Ψ'' = normalized cold bridges

SH = space heating

$\theta_{SH,dim}$ = SH design outdoor temperature

θ_{ym} = annual mean outdoor temperature

STD = standard SH design conditions

T_{AH} = air-heating temperature (after the heating coil)

$T_{AH,max}$ = maximal T_{AH} during TMY

$T_{AH,95\%}$ = 95% percentile of T_{AH} during TMY

T_{in} = fresh air temperature (before the heating coil)

TMY = typical meteorological year

T_{op} = operative temperature

$T_{op,max}$ = instantaneous maximal T_{op} among thermal zones (hourly value)

$T_{op,min}$ = instantaneous minimal T_{op} among thermal zones (hourly value)

$T_{op,g,max}$ = global maximal T_{op} during TMY (yearly value)

$T_{op,g,min}$ = global minimal T_{op} during TMY (yearly value)

$dT_{op,max}$ = instantaneous maximal difference between T_{op} in building (hourly value)

T_{ref} = reference building temperature enforced at T_{set} by the AH control

T_{set} = set-point SH temperature

$T_{vent,r}$ = mean return temperature of ventilation air

U = thermal transmittance

V = forced ventilation airflow rate

V_n = nominal ventilation airflow rate

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10. APPENDIX

Thermal losses from ventilation ducts are computed using the *Type 31* component of TRNSYS [14]. This one-dimensional model requires defining the overall transfer coefficient of the pipe. The model assumes it constant and uniform so that it was evaluated using a typical mean temperature inside the duct (i.e. 35°C). Its evaluation is performed analytically by combining the heat transfer coefficient inside the duct, outside the duct and through the duct envelope:

- The radiative heat transfer is neglected inside the duct. The convection coefficient for a fully turbulent developed flow in a circular pipe is evaluated using the Dittus and Boelter correlation [43].
- For the external flow, the convection coefficient is estimated using the Churchill and Chu correlation [44] for a free convection around a long horizontal cylinder. The radiation to the room is evaluated analytically assuming that the dimensions of the pipe are small compared to the room geometry. The radiative heat flux is then only dependent on the area of the duct external surface, its emissivity as well as the duct and room temperatures.
- The thermal mass of the duct envelope is neglected. The radial conduction in the cylindrical shell is evaluated by putting in series the thermal resistance of each layer [45].

Using this method, the magnitude of thermal losses are equivalent to those reported in Feist et al. [4] and to field measurements done in a Norwegian passive house [46].

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