Investigation of the Air-Heating Concept for Norwegian Passive Houses

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Abstracts

The German definition of the passive house standard is strongly related to the air-heating (AH) concept, while this concept is not explicitly connected with the Norwegian definition (NS 3700 standard). As AH presents an opportunity for space-heating (SH) simplification, the AH potential is here investigated in the Norwegian context. The questions of the required AH temperatures, of the temperature distribution between rooms and the influence of losses from ventilation ducts are investigated using detailed dynamic simulations (here using TRNSYS). This is done using a typical detached house typology, both considering different building construction materials as well as different climate zones (Oslo, Bergen and Karasjok). Simulation results present the potential and limitation of the AH for this common building typology but also enable to derive guidelines for the proper design of AH systems in Nordic conditions. For example, the standard SH design conditions (STD) appear to be the most severe conditions in term of AH temperatures and uneven temperature distribution between rooms.

Keywords: Air heating; Thermal comfort; Simplified distribution; Norwegian context; Simulations

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. In the original philosophy of the passive house [Feist 2005], the minimal requirement for the envelope performance has been strongly 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.

A specific definition of the PH standard has been defined for Norway, the NS 3700 [Standard Norge 2010]. Although not official yet, this standard is often considered as the future legal requirement for new buildings after 2015, while it is also seen as the minimal envelope performance for future

Norwegian net-Zero Emission Buildings (nZEB). Although these objectives are ambitious and at a short term, there is still a lack of experience about AH in Norwegian PH so that this knowledge should be quickly gained. Paradoxically, the NS 3700 is not based on the AH concept although it is explicitly written that it derives the German PH concept to Norway. Therefore, the present contribution investigates the AH potential in a Nordic context, characterized by large differences between climate zones as well as low solar angles. Accordingly, our work takes as a basis assumption that the maximal AH temperature is 55°C, mixing ventilation and that flow rates can only be increased up to ~50% above the nominal rates, based on hygienic considerations, at peak load.

A broad review of the questions regarding the indoor air quality (IAQ) in passive houses can be found in [Thomsen 2012]. Investigations on AH specific to Nordic climates are mostly based on Swedish works [Karlsson 2006; Wall 2006; Isaksson 2011; Molin 2011]. 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 climate zones in Norway, is the maximal AH power actually enough to cover the SH load during all 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 plumes generated by internal heat loads [Krajčík 2012]. 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 shortcuts to the exhaust Air Terminal Device, ATD [Mathisen 1989]. These phenomena are also dependent on the room geometry, AH temperature and ATDs locations. Although, dedicated research is still needed, recent works based on measurements [Feist 2005; Krajčík 2012] did not report any severe issue: these results investigated AH temperatures up to ~45°C.
3. **Distribution losses in 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) [Feist 2005].
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 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?

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 perfect mixing). In this context, investigations are consistently performed using detailed dynamic simulations, here applied to one benchmark Norwegian passive house. A sensitivity analysis is done using a set of ~800 simulations with time steps of 1 to 3 min. Only most representative results are reported in the paper.

Simulation procedure

Building model

Given the above assumptions, investigations are performed using detailed dynamic simulations, here using TRNSYS [Klein et al. 2010]. The multi-zone building model, Type 56, is applied where each zone is consistently represented by one air-node (i.e. perfect mixing). In order to investigate natural convection inside the envelope, the airflow rates between rooms are computed using a ventilation-network model, here TRNFLOW based on the COMIS library. Feist et al. [Feist 2005] indeed demonstrated that the opening of the internal doors is an efficient way to homogenize the temperature within the entire envelope. Doors are modelled using a large opening approximation [Etheridge 1996] 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 [Heiselberg 2006], while a default value of 0.65 is here applied. Doors have an effective section of 1m x 2.1m, with a 1cm opening underneath when closed. Finally, the thermal comfort is evaluated globally using the operative temperature, T_{op} [CEN 2005].

Benchmark passive house

The same detached single-family house geometry is used as a benchmark building for all simulations. It is a typical two-storey's building extracted from a house manufacturer catalogue [Mesterhus 2012], a common typology in Norway. The house has a net heated surface (A_{fl}) of 173.5 m^2. 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.

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 and bedrooms, and is extracted in the *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 [KRD 2010]. As already mentioned, it is here assumed that the airflow rate can only be boosted up to 50% above V_n. By default, constant and space-uniform internal gains with a value of 4.2 W/m^2 comparable to the NS 3700 are applied, while 2.1 W/m^2 is considered in the PHPP tool [Feist 2007] used for the design of German PH.

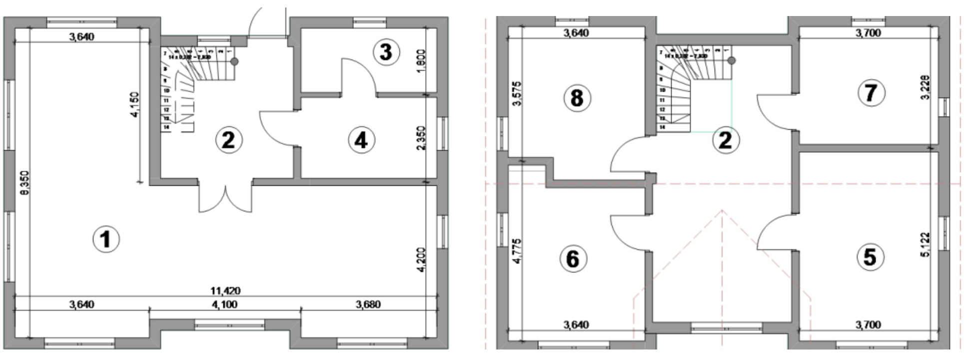


Figure 1 Sketches of the first and second floors: kitchen coupled to the living room (zone1), corridor with an open staircase towards the second floor (zone2), technical room (zone 3), bathrooms (zones 4 and 7) and bedrooms (zones 5, 6 and 8).

Construction modes

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}).

This last criterion is made dependent on the local weather conditions and the building compactness: Q_{max} is adapted as a function of the annual mean outdoor temperature (θ_{ym}) and the heated area, A_{fl}. Three different building locations are considered here, see Table 1: Oslo, Bergen and Karasjok. Although limited, this set of locations enables a coarse estimate of the wide range of weather conditions found in Norway. The SH set-point temperature (T_{set}) is fixed at 21°C by the NS 3700.

Table 1 Weather characteristics for the 3 locations: $I_{tot,rad}$ is the mean total radiation on a horizontal surface, $\theta_{SH,dim}$ is the SH design outdoor temperature.

	Θ_{YM} [°C]	$I_{TOT,RAD}$ [W/M²]	$\Theta_{SH,DIM}$ [°C]	Q_{MAX}* [KWH/M².Y]
Oslo	6.3	110	-20.0	19.2
Bergen	7.5	87	-11.7	19.1
Karasjok	-2.5	79	-48.0	41.6

* *Given in NS 3700, depending on ϑ_{ym} and A_{fl} only*

The building envelope performance has been defined in order to comply with the NS 3700. This has been verified using the building simulation software SIMIEN [ProgramByggerne] equipped with specific modules to check the compliance with Norwegian building standards. As three locations are here considered, three levels of building envelope performance have been defined (see Table 2).

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 level of internal thermal mass have been defined using Norwegian technical literature [Byggforsk], see Table 3. They range from *very-heavy* to *very-light* according to EN 13790 [CEN 2008]. In practice, the different construction modes have different levels of thermal insulation located inside the building envelope (e.g. in partition walls between rooms). This insulation is essentially placed for acoustic reasons. In general, one notices that the higher the thermal mass, the lower the insulation level in internal walls. Finally, it is worth mentioning that wooden constructions are commonly used in Norway.

Table 2 Building envelope performance as a function of the geographic location: 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}), infiltration rate at 50 Pa (n_{50}) as well as net SH needs computed using SIMIEN (Q_{net}) and maximum net SH power (P_{SH}).

	$U_{EXT,WALL}$ [W/M².K]	U_{ROOF} [W/M².K]	U_{SLAB} [W/M².K]	U_{WIN} [W/M².K]	Ψ'' [W/M².K]	H_{EXCH} [%]	N_{50} [1/H]	Q_{NET} [KWH/M².Y]	P_{SH} [W/M²]
Oslo	0.15	0.12	0.11	0.72	0.03	85	0.6	18.9	16.6
Bergen	0.15	0.16	0.11	0.80	0.03	85	0.6	16.0	11.7
Karasjok	0.12	0.09	0.08	0.72	0.03	85	0.6	41.0	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*

Table 3 Building construction modes: overall building inertia using EN 13790, 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

Air heating modelling

The AH temperature (T_{AH}) is adapted to enforce one reference temperature at T_{set}. By default, the air temperature in the living room is here used. Another common strategy is to resort to the mean return temperature of the ventilation air ($T_{vent,r}$), an alternative that is also tested. The power to raise the ventilation inlet temperature (T_{in}) for the AH is controlled using a PI action (requiring to apply a time step of ~1 min for simulations). For the sake of clarity, a constant T_{set} is only considered here. In practice, the extra-power for an intermittent SH will in fact lead to higher T_{AH} (e.g. when using a night setback).

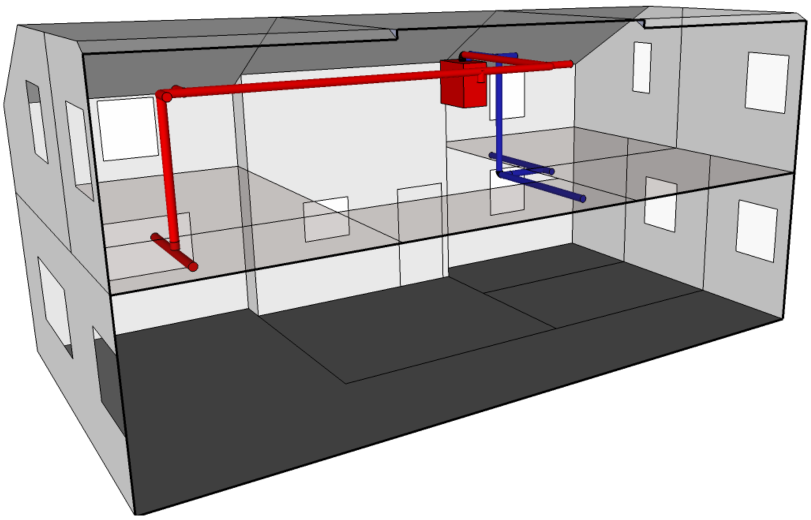


Figure 2 Sketch of the analyzed ventilation network: fresh and hot air in shown in red colour while the return air is in blue.

The ventilation network has been designed by a professional installer using standard products and according to the Norwegian regulation. In this way, realistic ducts lengths and diameters are considered. Ducts are modelled in TRNSYS (using Type 31) and coupled to the building model: ducts losses are injected in the building model as gains while these losses also induce a corresponding reduction of the air temperature in the ducts. The delay for the air to propagate through ducts in also modelled. The overall heat transfer coefficient is computed using convection correlations for circular pipes and the radiative heat transfer is computed analytically assuming that the duct dimensions are small compared to the room. In practice, three scenarios have been considered: without duct insulation, with 5 cm of insulation and without duct losses. Performance is first investigated without thermal losses in order to distinguish their specific effect when they are subsequently introduced.

Performance in STD without ventilation ducts losses

AH performance is here analyzed in Standard Design Conditions (STD): it assumes steady-state conditions with the $\theta_{SH,dim}$ introduced in Table 1 and the building without solar gains.

Maximal AH temperature during STD

The T_{AH} is evaluated in STD and results are reported on the first columns of Table 4. It 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.
- In practice, the AH is only possible in Oslo if the ventilation rate is forced to 50% but with a high T_{AH} of approximately 45°C. The climate of Bergen is milder so that AH is possible using V_n at the condition that the 4.2 W/m² internal gains are applied. T_{AH} are then more acceptable and can be limited to about 40°C if the ventilation is forced at 50%. On the contrary, the Karasjok climate is extremely cold. Even though the performance requirements of NS 3700 are adapted to the local climate, the AH is almost impossible during wintertime.

Table 4 Possibility to implement AH and corresponding T_{AH}: comparison between **STD** and **TMY** conditions considering closed internal doors.

OPERATING CONDITIONS			STD		TMY				
V	GAINS [W/M²]	CLIMATE		T_{AH} [°C]		$T_{AH,MAX}$ [°C]		$T_{AH,95\%}$ [°C]	
						CONSTANT GAINS	VARIABLE GAINS	CONSTANT GAINS	VARIABLE GAINS
V_n	0.0	Oslo	KO	-	Limit	[51.1;55.0]		[46.5;52.3]	
		Bergen	KO	-	OK	[45.6;50.6]		[42.0;45.9]	
		Karasjok	KO	-	KO	-		-	
	4.2	Oslo	Limit	[49.5;55.0]	Limit	[41.2;48.7]	[49.3;55.0]	[36.7;41.4]	[42.5;46.0]
		Bergen	OK	[42.7;47.1]	OK	[35.8;40.0]	[44.0;49.3]	[32.6;35.3]	[37.9;40.3]
		Karasjok	KO	-	KO	-	-	-	-
3/2 V_n	0.0	Oslo	OK	[45.9;50.5]	OK	[40.4;45.6]		[37.6;41.1]	
		Bergen	OK	[41.2;44.6]	OK	[37.0;40.1]		[34.8;37.2]	
		Karasjok	KO	-	OK	[48.0;52.9]		[45.7;49.2]	
	4.2	Oslo	OK	[40.0;44.0]	OK	[34.0;38.7]	[39.0;45.1]	[31.3;34.3]	[34.9;37.1]
		Bergen	OK	[35.3;38.0]	OK	[30.7;33.3]	[35.7;39.0]	[28.6;30.2]	[32.0;33.6]
		Karasjok	Limit	[49.9;55.0]	OK	[45.1;49.8]	[47.4;55.0]	[38.7;44.1]	[42.6;47.0]]

"OK" when AH possible, "KO" when AH impossible and "Limit" when dependent on construction mode.

Table 5 Temperature distribution between rooms during **STD**: maximal and minimal operative temperature, $T_{op,max}$ and $T_{op,min}$, respectively, for all the construction modes*.

INTERNAL DOORS			CLOSED		OPEN		CLOSED	
SH IN BATHROOMS			NO		NO		YES	
V	GAINS [W/M²]	CLIMATE	$T_{OP,MAX}$ [°C]	$T_{OP,MIN}$ [°C]	$T_{OP,MAX}$ [°C]	$T_{OP,MIN}$ [°C]	$T_{OP,MAX}$ [°C]	$T_{OP,MIN}$ [°C]
V_n	4.2	Oslo	[23.7;31.3]	[17.5;19.9]	[21.8;22.7]	[19.7;19.9]	[23.8;31.3]	[19.3;20.5]
		Bergen	[22.9;28.8]	[18.2;20.1]	[21.6;22.3]	[19.9;20.1]	[22.9;28.8]	[20.1;20.2]
3/2 V_n	0.0	Oslo	[23.9;31.3]	[17.4;19.8]	[21.9;22.8]	[19.5;19.7]	[24.0;31.3]	[19.9;20.0]
		Bergen	[23.2;28.9]	[17.8;20.0]	[21.7;22.5]	[19.6;19.9]	[23.3;28.8]	[20.0;20.1]
	4.2	Oslo	[23.2;29.2]	[18.4;20.0]	[21.7;22.5]	[19.7;19.9]	[23.3;29.2]	[19.9;20.0]
		Bergen	[22.5;26.7]	[18.9;20.1]	[21.5;22.1]	[19.9;20.1]	[22.6;26.7]	[20.1;20.2]
		Karasjok	[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 are putted into brackets

Temperature distribution during STD

Using a 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 higher thermal insulation in partition walls present increased temperature differences between zones*.
- The colder the climate, the larger the temperature differences inside the building. This is a direct consequence of higher T_{AH} applied with colder conditions. Accordingly, a same effect is found with internal gains where lower gains lead to higher temperature differences in the building.
- The opening of the internal doors is an efficient way to homogenize heat inside the building. Therefore, 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 the 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 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 building characteristic time scales of PH are 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, it is already the case.

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 mean zone temperature in the building is increased of about ~3°C. It can be easily understood as $T_{vent,r}$ is mainly based on the wet rooms temperatures (e.g. bathrooms) which always present the lowest temperature when using AH. In fact, the temperature difference between rooms is almost unchanged between the two control strategies. As a consequence, this change of reference temperature for the control can essentially be considered as a shift in the mean building temperature.

Performance during TMY without ventilation duct losses

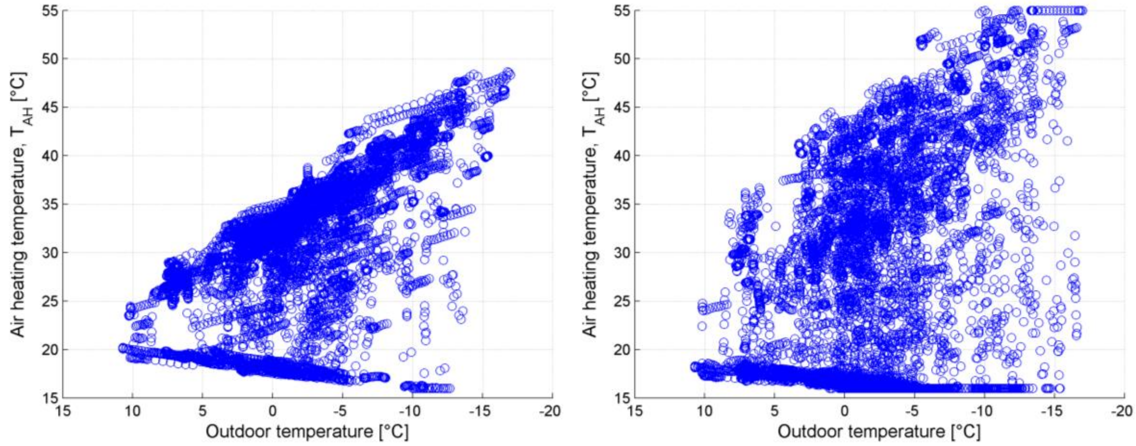
Comparison against STD

An interesting question is to check how standard design conditions (STD) are representative of everyday operating conditions. Furthermore, it should also be confirmed that these STD are representative of the most severe operating conditions. According to [Feist 2005], the maximum heating power should be evaluated for two extreme cases. The first case is a cold day with a clear sky and, thus, solar gains. The second case is a milder day with overcast sky (i.e. with negligible solar gains). This procedure is translated in the PHPP evaluation tool [Feist 2007] used in many countries (e.g. Germany). In our case, STD correspond to the design outdoor temperature $\theta_{SH,dim}$ without solar gains.

In order to investigate this, yearly 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.

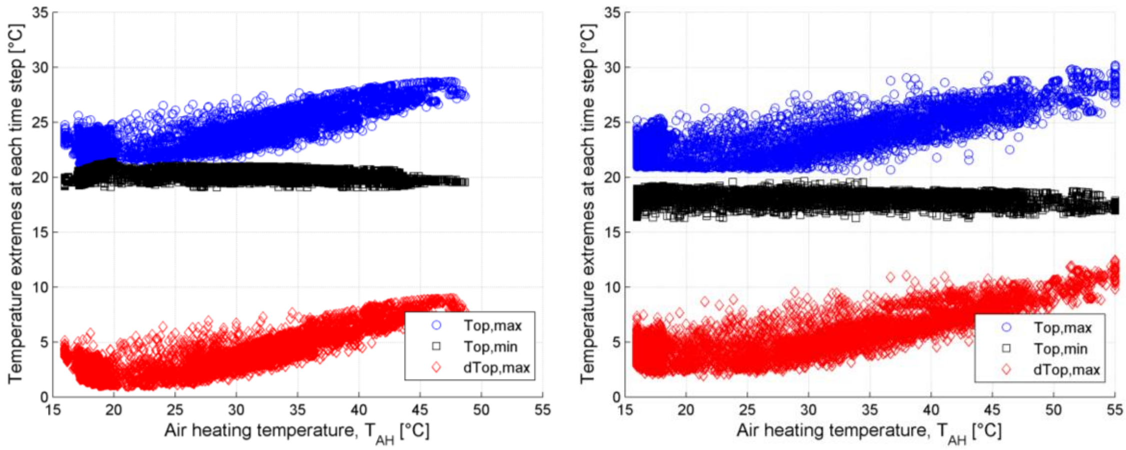
Firstly, the maximal and 95% percentile T_{AH} are reported on Table 4 (termed $T_{AH,max}$ and $T_{AH,95\%,}$ respectively). A distinction is then made between simulations done using **constant** gains and **variable** gains. These variable gains are non-uniform in space (i.e. between rooms) and fluctuating in time (i.e. with 1-h resolution) but also present an average value of 4.2 W/m². They were created artificially but consistently using Norwegian statistics of household use (e.g. average electricity consumption by equipment and their typical cycle length, time-of-use surveys). Using constant gains and a TMY, T_{AH} are lower than the maximal value obtained using STD. Nevertheless, considering variable gains, these temperatures are significantly increased. The variability of gains should then be properly integrated in a AH design procedure.

Secondly, hourly $T_{op,max}$ and $T_{op,min}$ can be analyzed during a TMY along with the maximal temperature difference in the building, $dT_{op,max}$. In Figs. 3 and 4, results are presented for the very-light building located in Oslo, using hygienic ventilation airflow rates (i.e. $V = V_n$) as well as closed internal doors. A similar behaviour was found for all other test cases.



(a) Constant internal gains (b) Variable internal gains

Figure 3 T_{AH} as a function of the outdoor temperature during a **TMY** for the very-light building located in Oslo with closed internal doors and hygienic ventilation flow rates.



(a) Constant internal gains (b) Variable internal gains

Figure 4 $T_{op,min}$, $T_{op,max}$ and $dT_{op,max}$ as a function of T_{AH} during a **TMY** for the very-light building located in Oslo with closed internal doors and hygienic ventilation flow rates.

From Fig. 3, one clearly notices that the maximal T_{AH} is found during the coldest days. The evolution is obviously more regular using constant compared to variable gains. As explained, higher T_{AH} may occur when using variable gains for a given outdoor temperature.

Furthermore, Fig. 4 also shows that the temperature differences in the building are mainly driven by the T_{AH}, the trend is indeed almost linear. Nevertheless, the different definitions of internal gains affect the temperature distribution (see e.g. the level of $T_{op,min}$ in both graphs). Finally, let us mention that rooms that were identified to be the warmer and colder in STD are the same when using a TMY.

As a result, both Figures confirm that a cold day without sun make sense in the AH design. In other words, one should not expect more severe operating conditions. Nevertheless, while the analysis may be restricted to a cold day without sun, the variation of gains should be integrated in the design in order to evaluate the $T_{AH,max}$ and the temperature distribution accurately.

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. Results are first reported for STD conditions in Table 6: 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 [Feist 2005]. In the present ventilation network layout, 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 the 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 are injected in the corridor (i.e. zone 2). As a result, the bathrooms present higher temperatures that are comparable to the living room.

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

Table 6 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*.

INTERNAL DOORS			CLOSED			CLOSED			CLOSED		
DISTRIBUTION LOSSES			NO LOSSES			NON-INSULATED DUCTS			DUCTS WITH 5CM INSULATION		
V	GAINS [W/M²]	CLIMATE	$T_{OP,MAX}$ [°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	[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	[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	[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	[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	[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	[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	[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 are putted into brackets

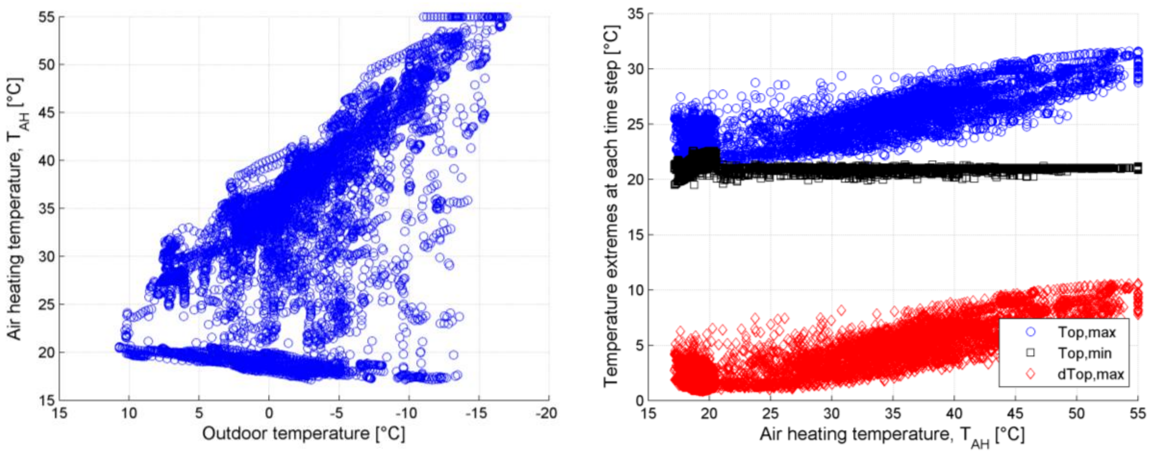


Figure 5 T_{AH}, $T_{op,min}$, $T_{op,max}$ and $dT_{op,max}$ during a **TMY** for the very-light building located in Oslo with non-insulated ducts: with closed internal doors, constant gains and hygienic ventilation flow rates.

The performance in TMY is now introduced. The temperature evolution can here be compared for the case of Oslo, without (Figures 3(a) and 4(a)) and with losses (Figure 5). Again, for a given outdoor temperature, the T_{AH} can be higher with distribution losses. On the contrary, the evolution of temperature extremes 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).

Conclusions

This work investigates air heating (AH) utilized as space heating (SH) of Norwegian passive houses (PH); these houses being characterized by a specific Norwegian standard (i.e. the NS 3700). Although this standard is considered as the future minimal requirements for buildings in the coming years, there is still a lack of experience about AH in Norwegian PH. This paper first presents a list a challenges for the AH, and proposes to investigate questions related to the system thermal dynamics as well as the possibility to apply AH from a conceptual point of view. 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 by an outdoor SH design temperature and no solar gains, and subsequently using a Typical Meteorological Year (TMY). Results, based on a set of ~800 distinct simulations, lead to the following conclusions in terms of physics and implications in AH design:

- In theory, the **maximal value of the AH temperature** (T_{AH}) should be evaluated in STD and without internal gains. This scenario is too conservative and would disqualify the AH approach in too many building projects where the T_{AH} is in fact acceptable during usual operating conditions (here considering a TMY). 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 lower design outdoor temperature.
- The AH based on a centralized heating coil leads to an **uneven temperature distribution** between rooms. Firstly, a cold day without sun leads to the most severe distribution inside the building: in fact, investigations using TMY did not show any other critical configurations. Therefore, the AH design could be limited to these conditions. Secondly, a multi-zone building analysis is required for a proper design. The temperature distribution is strongly related to the building architectonic properties (e.g. level of insulation in internal walls and internal doors opening). Furthermore, the spatial variation of internal gains should be considered if an accurate assessment of temperature distribution between rooms in required.
- **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, unlike other internal gains, thermal losses from ducts may further influence the T_{AH}: a significant temperature drop can occur in the ducts between the air-handling unit and the injection from ATDs, leading to higher T_{AH}. With 5 cm insulation around ducts, thermal losses can be neglected. Nonetheless, with a limited

insulation, losses should be considered for both the evaluation of the temperature spatial distribution and the maximal T_{AH}.

- A **specific building typology was investigated**. If AH covers the entire SH needs (i.e. no peak heating), the AH does not seem to be well adapted, or, at least, not in this standard form. Firstly, except for milder climates (i.e. Bergen), the T_{AH} are quite high **or** significantly higher ventilation flow rates are required (i.e. compared to hygienic flow rates). Secondly, even for the milder climate, 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. A possible solution to homogenize heat inside the building envelope is to open the internal doors. Finally, these results suggest that AH should preferably be combined with a peak-load heating system and/or the number of heating coils should be increased.

Nomenclature

A_{fl} = net heated surface
AH = air heating
ATD = air terminal device
C_d = discharge coefficient for doors
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_{net} = net SH needs
Q_{max} = maximum net SH needs in NS 3700
Ψ'' = normalized cold bridges
SH = space heating
$\vartheta_{SH,dim}$ = SH design outdoor temperature
ϑ_{ym} = annual mean outdoor temperature
STD = standard design conditions
T_{AH} = air-heating temperature (after the heating battery, before the distribution system)
$T_{AH,max}$ = maximal T_{AH} during TMY
$T_{AH,95\%}$ = 95% percentile of T_{AH} during TMY
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
$T_{op,g,min}$ = global minimal T_{op} during TMY
$dT_{op,max}$ = instantaneous maximal difference between T_{op} in building (hourly value)
T_{set} = set-point SH temperature
$T_{vent,r}$ = mean return temperature ventilation air
U = thermal transmittance
V = forced/actual ventilation airflow rate
V_n = nominal ventilation airflow rate

Acknowledgement

The authors acknowledge Mette Maltha from SINTEF Byggforsk for her contribution to define the benchmark passive house. They also want to thank the Norwegian Research Council and several partners for their support, as this work was performed in the framework of the Research Centre for Zero Emission Buildings (ZEB).

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