JSACE 1/37

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Received 2024/09/17

Accepted after revision 2024/12/11

Optimisation of Heating System Powered by Air Heat Pump and Gas Condensing Boiler Hybrid Unit

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https://doi.org/10.5755/j01.sace.37.1.38843

Abstract

Gas condensing boiler and air to water heat pump hybrid unit is an optimal way to introduce renewable energy resources in existing buildings. Two energy sources (gas & electricity) give hybrid unit higher flexibility in comparison to typical air to water heat pump. In hybrid solution air heat pump can be used in locations with low temperature heating seasons. Hybrid unit can output higher heat carrier temperatures, because of this, it can be used in combination with older radiator heating systems.

There are many parameters that can influence the performance of hybrid heating unit. This paper investigates heat terminal type, heat carrier temperature, and outdoor switchover temperature setting (outdoor temperature at which hybrid unit switches from electricity to fossil fuel) influence on air to water heat pump and gas condensing boiler hybrid heating unit performance parameters (total efficiency - η_{hybrid} and primary energy factor - PEF_{hhp}). Hybrid heating units performance is evaluated by using a computer model created in program IDA ice 4.8. The created computer model represents a real building, located in Latvia, that uses the previously mentioned hybrid heating unit. The model is verified by comparing its results with energy meter data from the real building, for time period from 01.03.2022 to 28.02.2023.The verified model is used to simulate how hybrid heating units performance is influenced by changes in heating terminal type, heat carrier temperature and outdoor switchover temperature setting.

According to simulation data, at constant heat carrier temperature, heat terminal type has no influence on hybrid heating unit's performance parameters. It has been found that increased heating system volume can reduce hybrid heating unit's run time. In this case replacing panel radiators with floor heating, there is a 33% reduction in unit's annual running time. In simulated scenarios, heat carrier temperature reduction by 15°C, increases η_{hybrid} by 8.7% and decreases PEF_{hhp} by 17.5 % (at temperature graph 40/35°C). Switch over temperature increase from -7 to 3°C decreases η_{hybrid} by 47% and increases PEF_{hhp} 7 %. Switch over temperature increase also reduces η_{hybrid} and PEF_{hhp} change magnitude, when changing heat carrier temperature graphs. When changing the temperature graph from 40/35 to 55/50 °C the changes are as follows: at switchover temperature setting of -7°C, η_{hybrid} drops by 14,33 %, but PEF_{hhp} increases by 23,42%; at switchover temperature setting of -2°C (actual setting), η_{hybrid} drops by 8,7 %, but PEF_{hhp} increases by 17,51%; at switchover temperature setting of +3°C, η_{hybrid} drops by 1,74 %, but PEF_{hhp} increases by 6,45%.

Keywords: air-to-water heat pump; gas condensing boiler; space heating; hybrid heating unit; heat carrier temperature graph.

Introduction

Building heat supply sector consumes a substantial portion of primary energy resources. 79 % of Europe's total energy resources are consumed for building heating and hot water supply (RESHeat, 2022.). The use of alternative energy sources is one of the best ways to reduce CO2 emissions, as well as to increase energy efficiency.

An air to water heat pump is a popular and easily accessible alternative energy source for building heating, however, its energy efficiency degrades at lower outdoor temperatures. The usage of a hybrid heating unit that combines air to water heat pump with a fossil fuel boiler can be beneficial in colder climates and can also allow us to take advantage of fossil fuel and electricity price fluctuations.

Additionally, the usage of hybrid heating unit increases heating systems energy resilience. Energy resilience defines the ability for building energy services to operate during major disruptions and is becoming more important in the current global situation (Building energy codes program, 2024.). Because hybrid units use two energy resources to produce heat (in this case natural gas and electricity), it is possible to switch between these in case of power grid or gas supply problems. To achieve maximum energy resilience the hybrid heating unit should be used together with a heat accumulation tank, and the building should have adequate thermal envelope, to reduce peak heat energy demand.

One of many hybrid heating unit types is an air to water heat pump and gas condensing boiler hybrid heating unit. Compared to a typical air to water heat pump that is coupled with electrical heater, this hybrid unit can reduce primary energy consumption during cold weather and during domestic hot water production (Beccali et al., 2022).

Hybrid heating unit can be a good solution for heating system modernization. In one study (Asaee et al., 2017) it was found that 71 % of Canadian private housing fund could be equipped with hybrid heating unit, without extra modernization. As a result, primary energy resource consumption could be reduced by 36 %.

There are many parameters that influence the performance of hybrid heating units. Both external conditions and hybrid unit's internal parameters are important. Outdoor air temperature and relative humidity influences air to water heat pumps COP value, which has a direct impact on hybrid unit's total efficiency. Work (Park et al., 2014) analyzed how various parameters influence performance of a hybrid heating unit consisting of air to water heat pump with nominal heating capacity of 7 kW (at 7°C) and condensing gas boiler with nominal capacity 23 kW. It was found that a decrease in ambient temperature from +10°C to 0°C, reduces heat pump COP value from 4,7 to 3.8 (21,2%), which in return decreases the total efficiency of hybrid unit from 1,12 to 1,03 (8,4 %). At certain temperature range increased relative humidity can cause frost formation on heat pump's outdoor unit (evaporator) which causes further performance drop due to reduction of heat transfer and defrost cycles. Research (Di Perna et al., 2015) found that if the air humidity is 70 % and higher a sharp drop in heat pump COP value (~18,2%) can happen at outdoor air temperature of ~5°C. If the relative humidity of air is 80 %, then the COP value drop is a staggering 40 %.

The temperature of the heating system is also important. A lower heat carrier temperature will increase both the efficiency of gas condensing boiler and of air to water heat pump. In papers (Park et al., 2014; Hu et al., 2019) it was observed that hybrid unit will achieve higher performance with low heating system temperatures (40°C and lower). It is preferable to couple hybrid heating unit with low temperature heat terminals, for instance, floor heating.

Internal parameters of hybrid heating unit, like switch over temperature and heat pump and gas boiler nominal power ratio, also have a considerable influence on total efficiency. Research (Dongellini et al., 2021) studied how air to water heat pump and gas condensing boiler hybrid heating unit's annual performance changes at different outdoor temperature switchover settings.

It was found that the highest annual performance can be achieved if switch over temperature is 0 to 3°C.

Influence of air to water heat pump and condensing gas boiler nominal power ratio was studied in work (Klein et al., 2014). It was found that if air to water heat pumps nominal power ratio in hybrid unit is only 0.6. At this ratio considerable primary energy resource savings (26 % compared to only condensing gas boiler) can be achieved. However, to maximize total hybrid unit's efficiency heat pumps power ratio must be as high as possible. In work (Bagarella et al., 2016)was found that hybrid unit can achieve higher seasonal performance if heat pump element has higher nominal power. With higher nominal heating power heat pump can satisfy building heating demand at lower outdoor air temperatures, so hybrid unit can run in heat pump mode longer.

Methods

To investigate how an air to water and gas condensing boiler hybrid unit is influenced by heat terminal type, heat carrier temperature and switchover temperature setting, a simulation model for a single-family house heating system using hybrid unit as a heat source has been created. The model has been created using IDA ice 4.8 software. IDA ice has many features that allow detailed energy simulation of a building and its HVAC systems. It is possible to create a detailed description of buildings envelope, set dynamic zone temperature setpoints, define various elements of heating system and create a detailed model of buildings heating center, that can be controlled user created algorithms. Various other authors have used IDA ice in similar papers. Authors in work (Maivel & Kurnitski, 2015) used IDA ice to simulate heating system return temperature effect on heat pump performance. In similar work (Clauß et al., 2019), authors used IDA ice to determine how various control strategies affect operation and performance of air to water heat pump with electric heater.

The simulation model that has been created in this work represents a real single-family house using the previously mentioned hybrid heating unit. The building thermal envelope and heating system element parameters have been obtained from the building design project. Heating center elements have been modeled after manufacturers data.

The model has been verified by comparing simulated gas and electricity consumption data with the data from smart energy meters in the real building. Energy meters are connected to "Metbox" system that allows to save the measured data and gives remote access through internet. The following data can be accessed:

- Electricity meter: electricity consumption (kW), hourly electricity consumption (kWh), total electricity consumption (MWh).
- Gas meter: gas consumption (m^3), hourly gas consumption (m^3/h); total gas consumption (m^3).
- 2x heat meters (separate for space heating and domestic hot water): current flow/power (m3/h; kW), supply/return heat carrier temperature (°C), total consumed heat energy (MWh).

The comparison of data has been made for period from 01.03.2022 to 28.02.2023. For buildings heat load simulation, climate data for this period have been used in the simulation. More detailed information on climate data can be found in the following pages.

After the verification, various parameters of the model have been changed and repeated simulations have been run (for the same period), to study changes in the simulated operation of hybrid heating unit.

The IDA ice model, that has been created in this work, consists of a building climate model, heat center model and control algorithm. The general schematic of simulation can be seen in Figure 1.

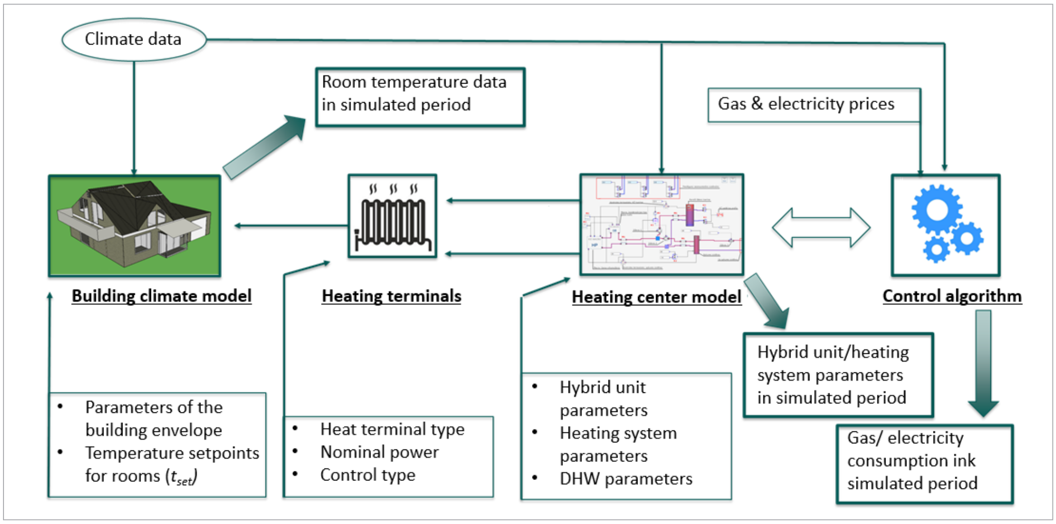


Fig. 1

General schematic of simulation

The building climate model is used to describe the building under study and generate heating demand based on outdoor temperature and indoor temperature setpoint fluctuations. The model consists of the envelope thermophysical parameters, temperature setpoint parameters and defined heat terminals. The building has a total floor area of 179,8 m^2. It consists of two floors – first floor with living room/ kitchen, work room, WC and heating equipment room, second floor – two bedrooms and bathroom.

The Thermal transmittance *U* of building envelope is as follows:

- Outer wall: 0,17 W/(m^{2}K);
- Floor (facing ground): 0,23 W/(m^{2}K);
- Roof: 0,32 W/(m^{2}K);
- Windows: 1,46 W/(m^{2}K).

Thermal bridges (ISO 13789) and infiltration are also considered when calculating heat demand.

The maximum heat demand of the building for simulation period, is 5,2 kW (at minimum outdoor temperature of -13.8°C).

The simulation uses climate data from Center of Environmental Geology and Meteorology of Latvia and ASHRAE, period 01.03.22 to 28.02.23. Climate data is defined in ASHRAE IWEC2 format. Data consists of the following values for each hour of the simulation period:

- Outdoor air, dry bulb temperature, °C;
- Outdoor air relative humidity, %;
- Wind speed, m/s;
- Direct solar radiation, W/m^2;
- Diffuse solar radiation on horizontal surface, W/m^2.

Outdoor air temperature and relative humidity are sourced form Center of Environmental Geology and Meteorology of Latvia, but the rest of data from ASHRAE data available in IDA ice.

The building has a variable temperature setpoint: from 8.00 to 17.00 heating setpoint is 19°C, but for the remaining time 21°C.

The next part of the model is the heating center. The heat center consists of the hybrid heating units indoor block (consists of condensing gas boiler, heat pump's condenser, and circulation pump for heating system connection), 300 l domestic hot water tank, control valves and smart electricity, gas and heat meters.

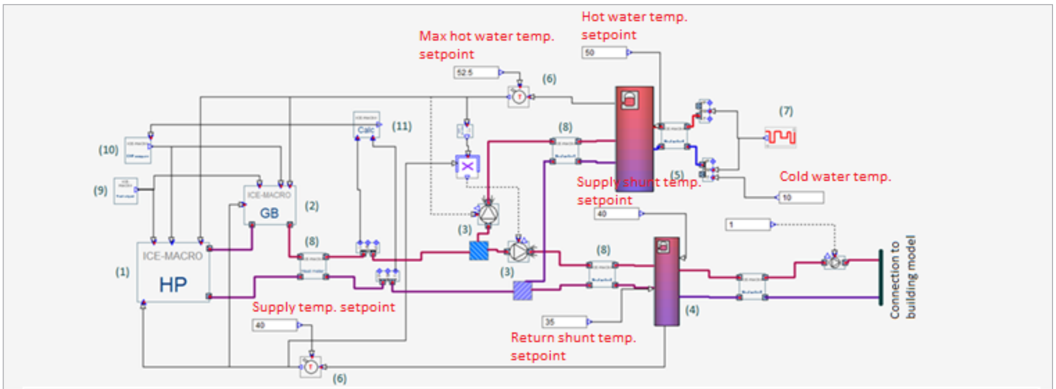
Hybrid heating unit consists of condensing gas boiler with nominal heating capacity of 8,2 kW and efficiency coefficient of 98,7% (at temperature graph 37/30°C, 30 % load).

Air to water heat pump has 9,81 kW nominal capacity and COP of 3,58 (at 7°C outdoor air temperature).

Figure 2 shows the heating center in IDA ice 4.8.

Fig. 2

Heating center model in IDA ice



(1) Air to water heat pump macro	(4) Tank object: heating system volume	(7) Hot water consumption data macro	(10) COP comparison algorithm macro
(2) Condensing gas boiler macro	(5) Tank object: hot water	(8) Flow/heat energy logging macro	(11) Ideal COP calculation macro
(3) Pump	(6) Temperature control macro	(9) Outdoor temperature data macro	

The schematic seen in **Figure 3** is used to describe building heat center in IDA ice environment. It consists of objects available in IDA ice and user created macros. All IDA ice objects have parameters that can be customized by the user, to create more accurate simulation. Macros are a collection of Ida ice objects and are used only to organize the schematic and reduce clutter. For instance, heat pump macro (1) contains IDA ice air heat pump object, and some additional control elements. Thick lines signify hydraulic connections between objects. Thin black lines signify data connection. For instance, heat pump macro (1) is connected to elements (6), (9) and (10), that generate control signal (1 or 0) based on conditions in heating system.

In the model, hybrid heating unit is described with macros HP – air to water heat pump (1) and GB – condensing gas boiler (2). Both heat sources are described using elements available in program IDA ice. Elements "Air to water heat pump" and "Simple boiler" are used, respectively.

Air to water heat pump objects is customized using available data from manufacturer. Some data that was not mentioned in manufacturer's technical documentation has been assumed based on similar nominal power air to water heat pump data available in IDA ice. Only one working point (Heating power; COP value; inlet/ outlet temperatures at one outdoor temperature) can be used to define heat pump in IDA ice 4.8. In this case the working point at outdoor air temperature 2°C and heat carrier temperature graph 40/35°C is used. At these temperatures air to water heat pump has nominal capacity of 8.35 kW and COP value is 8.35. When important parameters like outdoor air temperature, heating system temperature and heating load change during simulation, a new COP value is calculated by the program using a mathematical model. A working point of 2°C temperature is chosen because this is closer to heating seasons average temperature. The chosen working point can influence COP simulation accuracy. Custom heat pump calibration parameters were also used to increase simulation accuracy. It was necessary to use different calibration parameters, due to default values yielding too high COP values during simulation. Parameters were sourced from work (Niemelä et al., n.d.), that investigated IDA ice heat pump model calibration.

Most of the parameters for "simple gas boiler" were sourced from manufacturers data. Fuel type for the boiler is natural gas and nominal heating capacity – 8.2 kW. Two heat carrier temperature graphs are set for boiler object – one for space heating mode (40/35°C) and one for domestic hot water preparation (70/50°C). Efficiency is set to be 0.997 and was measured using flue gas analyzer “DELTA smart”.

Domestic hot water tank (5) is modeled using ideal stratified tank model with internal heat exchanger for heating system. The internal volume of the tank is 300 l. DHW load is defined using schedule element (7), by inputting flow in l/s for a chosen time period. Cold water temperature is assumed to be 10°C.

Even though the real heating system does not have an accumulation tank, another ideal stratified tank model is used to define heating system (4). The volume of this tank is 100 l, which is equal to the total volume of the real heating system. This is necessary to simulate heat inertia of heating system. Tank (4) is connected to building climate model, where heating demand is simulated. Flow in heating system branches is simulated by using pump objects (3). The real heating center has a single pump that is located inside hybrid heaters indoor unit, and flow diverting between space heating and DHW heating is achieved with a three-way valve. For simulation, two pump elements have been used to make modeling simpler. With these elements it is possible to define different heat carrier flow for each circuit. Pump operation is controlled by simulation control logic explained ahead. Both pumps never work simultaneously.

Using the logic elements available in IDA ice a control algorithm for hybrid heating unit has been created. Based on space heating / domestic hot water demand and outdoor air temperature, the optimal heat source is chosen. The created algorithm can be seen in Figure 3.

Hybrid heating unit is controlled by the following parameters:

- Heating demand for domestic hot water preparation;
- Heating demand for space heating;
- Heat pump control parameters:
  - Outdoor air temperature;
  - Air to water heat pump boundary COP value (COP_{BE}).

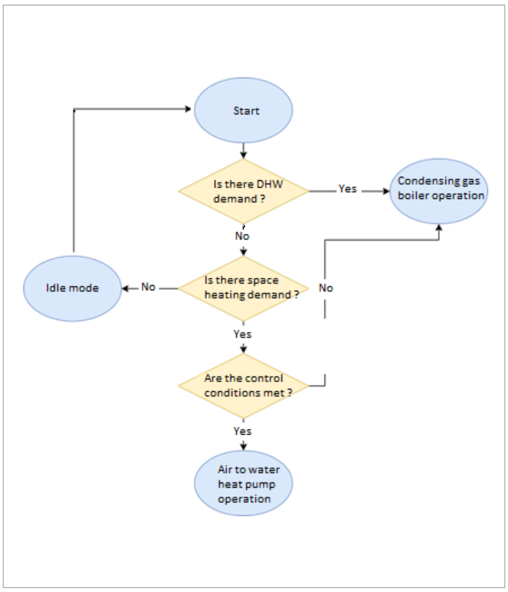


Fig. 3

Hybrid unit control algorithm in IDA ice

Depending on the previously mentioned parameters the optimal heating source is chosen. Only one heating source is active at a time in simulation. In a real hybrid heating unit, a true bivalent operation is possible (when both heat sources are on at a given moment). Unfortunately, due to time constraints, it was not possible to model this operation mode in simulation. When the real hybrid unit operates in this mode supply heat carrier is heated with heat pump up to temperature where $COP \leq COP_{BE}$. Afterwards the actual heat carrier temperature setpoint is reached using condensing gas boiler. It’s not really stated how much energy can be saved using bivalent operation mode. Perhaps it is valuable to explore the benefits of this operation in future.

Hot water demand is controlled by simulation element (6). A water temperature setting of 55°C is maintained in DHW tank. If the hot water temperature drops to 50°C, hybrid heating unit goes into DHW preparation mode. Condensing gas boiler is used to heat the DHW tank through an inbuilt heat exchanger. During DHW heating, the heat carrier supply temperature is 70°C.

Space heating system demand is checked by monitoring temperature in tank (4). The heat carrier supply temperature maintained in this tank is 40°C. If the temperature in tank drops to 35°C, a signal for space heating demand is sent to hybrid heating unit. If there is no DHW demand, then hybrid heating unit will divert heat carrier with 40°C temperature to space heating system.

Heat for space heating will be produced with either air to water heat pump or condensing gas boiler, based on hybrid heating unit control parameters. Heat pump is used if outdoor air temperature is not lower than user setpoint (in current simulation -2°C) and if COP of heat pump is not lower than COP_{BE}. COP_{BE} is calculated based on energy resource prices and gas condensing boiler efficiency value (**equation 1**).

$$COP_{BE} = \frac{Electricity\ price\ (\frac{Eur}{kWh}) \cdot \eta_{boiler}}{Gas\ price\ (\frac{Eur}{kWh})} \quad (1)$$

where: – COP_{BE} - break even COP value for heat pump; η_{boiler} – efficiency of gas condensing boiler.

The current simulation period has a constant electricity and gas price. Because boiler efficiency is also assumed to be constant, COP_{BE} is also constant. With gas price of **98 Eur/MWh** and electricity price of **150 Eur/MWh** the resulting value of COP_{BE} is **1.5**.

Table 1 lists all the scenarios that have been created in the scope of this work.

Table 1

Simulation scenarios

№.	Description (simulation №)	Heat carrier temperature graph	Heat terminal type	Min. outdoor air temperature setting for HP operation
1.	Base scenario (1)	40/35°C	Panel radiators (for low heat carrier temperature)	-2°C
2.	Heat terminal influence on hybrid heating unit (2-3)	40/35°C	Floor heating	-2°C
			Fan coil	
3.	Heat carrier temperature influence on hybrid heating unit (4-6)	40/35°C	Floor heating	-2°C
		45/40°C	Panel radiators	
		50/45°C		
		55/50°C		
4.	Minimal outdoor temperature setpoint, heat carrier temperature influence on hybrid heating unit (7-14)	40/35°C	Floor heating	3°C -7°C
		45/40°C	Panel radiators	
		50/45°C		
		55/50°C		

Parameters (heat carrier temperature, heat terminal type and minimal temperature for heat pump operation) in base scenario (1) are modeled after the real situation. This scenario is used to validate the simulation. The rest of the scenarios are created with parameters that differ from the real situation to study hybrid heating units operation under different conditions.

Using IDA ice model, it is possible to acquire many heating system, hybrid unit and building parameters, for a set period of time:

- Energy resource consumption (gas/electricity for heat pump);
- Generated heat energy for space heating and DHW production;
- Heat center operation data (heat carrier flows/ temperatures/ device operation times, etc.);
- Room temperature data;
- Etc.

To describe hybrid heating unit's efficiency the total efficient coefficient and primary energy factor have been used.

Because hybrid heating unit consists of two heating sources, total heating efficiency must be calculated using **equation 2.** (Poredoš et al., 2017)

$$\eta_{hybrid} = \frac{Q_{GS} + Q_{GKK}}{Q_{gas} + W_{GS}} \tag{2}$$

where: Q_{GS} – heat energy produced with air to water heat pump, kWh; Q_{GKK} – heat energy produced with gas condensing boiler, kWh; Q_{gas} - consumed gas, kWh; W_{GS} – electricity consumed by air to water heat pump, kWh.

Another parameter that can be used to describe hybrid unit's performance is the primary energy factor – *PEF*. Primary energy factor describes how efficiently a heating equipment transfers energy resources in heat energy. *PEF* is the ratio between consumed primary energy and produced heat. The higher is *PEF* value, the more inefficient is a heat source is. If *PEF* value is >1, then part of primary energy is lost when producing heat energy.

To calculate *PEF* of hybrid heating energy, **equation 3** can be used. (Poredoš et al., 2017).

$$PEF_{hhp} = \frac{PEF_{gas}}{\eta_{boiler}} \cdot R_{gas} + \frac{PEF_{el}}{SPF} \cdot R_{el} \tag{3}$$

where: PEF_{gas} – primary energy factor of natural gas; η_{boiler} efficiency of condensing gas boiler; PEF_{el} – primary energy factor of electricity; SPF - seasonal performance factor or air to water heat pump; R_{el} – heat energy produced by heat pump - ratio; R_{gas} heat energy produced by condensing gas boiler - ratio.

The primary energy factor for natural gas has been assumed to be 1,1, but for electricity 2,5. These values have been sourced from local standards.

(Building energy efficiency calculation methods and building energy certification rules, 2021.)

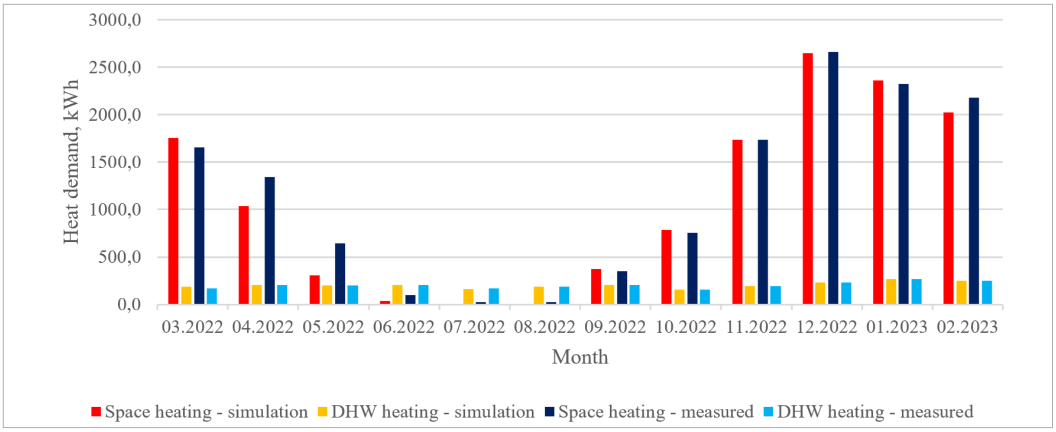
Results

Previously listed scenarios have been run using the created simulation. Each scenario is run for period from March 2022 to February 2023. To verify simulation, the results of 1st scenario have been compared to actual energy meter data for this period.

In **Figure 4**, a comparison between the real and simulated building's heat energy consumption can be seen.

Fig. 4

Heat consumption in measured data and 1st scenario



From **Figure 4**, it can be seen that the simulated space heat demand is very close to the measured. In most months the difference between simulated and actual space heating is no larger than 100 kWh. Months with lowest space heating demand simulation accuracy are:

- March: 96 kWh (5%);
- February: 155 kWh (7%);
- April: 306 kWh (26%);
- May: 334 kWh (70%).

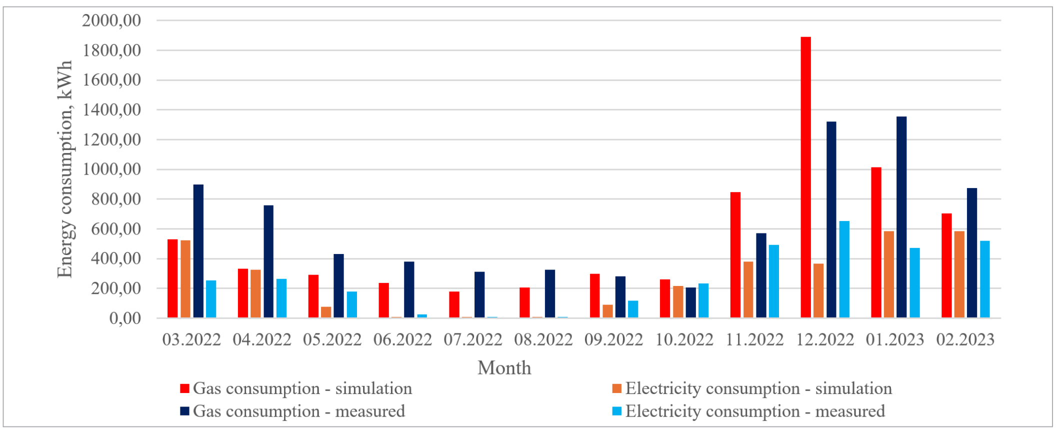
Most likely these inaccuracies can be blamed on imprecisions in used climate data. Outdoor temperatures used in simulations were sourced from public data, from the closest climate monitoring station (~10 km). It is possible that the actual temperatures in the experiment site were different.

DHW heat consumption is not simulated but inserted in simulation through a time graph. There are minimal differences in DHW heat consumption between simulated and measured data due to simulated heat losses.

Simulated and real energy resource consumption has been compared in the graph seen in **Figure 5**.

Fig. 5

Energy resource consumption in measured data and 1st scenario



From the graph, it can be seen that the accuracy of energy resource consumption simulation varies with each month. Most accurately simulated months are September, October and February, in these months difference between simulated and actual energy resource consumption is in range from 6 - 26 %. November, December and January have a higher difference between simulated and actual data 29 - 56 %. Spring and summer months have the highest error (53 - 80 %). Summarized values for whole period are close (13%).

The total efficiency coefficients calculated from simulation and actual data have been compared in a graph, which can be seen in **Figure 6**.

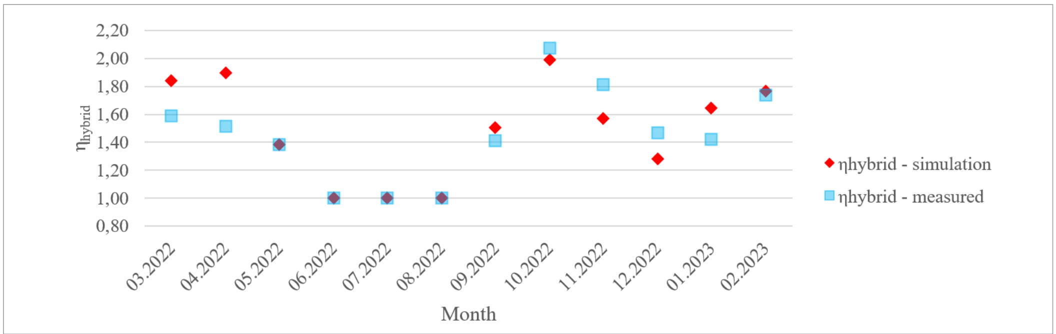


Fig. 6

Hybrid unit efficiency in measured data and 1st scenario

In warmer months simulation outputs a higher total efficiency than the real unit had in that period. In winter period the simulation mostly generates worse total efficiency than in real data. The difference between simulated and measured total efficiency is from 1,3 to 22,4 %. A correlation between coefficients calculated from measured and simulated data can be seen. Similar correlation can be seen when comparing the rest of performance parameters.

The simulated COP values from each hour of simulation have been compared to the COP values from the heat pump manufacturer's data sheet. The comparison can be seen in **Figure 7**.

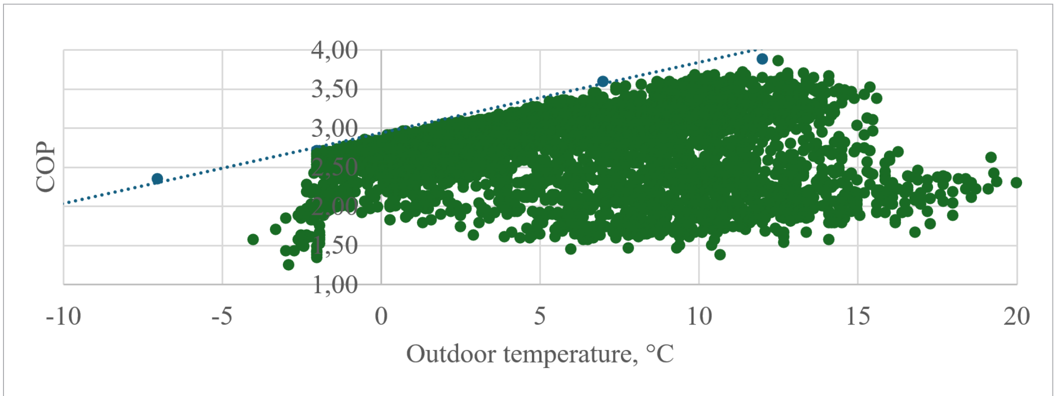


Fig. 7

Hybrid unit total efficiency -simulation vs measured data

It can be seen that simulated COP value never exceeds maximum defined COP value from manufacturers data (blue line). It can also be seen that a lot of values are much lower than maximum manufacturer's COP – this is due to other varying heat pump operation parameters – heat carrier temperature fluctuations and heat pump operation in part – load condition.

It is hard to pinpoint the actual reason for simulation inaccuracies in certain months. It is possible that used climate data deviates from actual data. It is possible that the algorithm used in simulation could be simplified compared to the one used in real hybrid heating unit.

It must also be mentioned that air to water heat pumps COP simulation in IDA ice is idealized and will never be able to completely predict real heat pump operation. Despite the errors, there is correlation between simulated and measured data.

Current simulation was used to compare how hybrid unit's performance parameters would change in combination with different heating terminals (but with same heat carrier temperature graph as in base scenario 40/35°C). Base scenario (panel radiators) was compared with floor heating and heating fan coils. Between the 3 scenarios, the following differences in hybrid heating unit can be listed.

In simulated period floor heating increases total space heating demand by 265,7 kWh (2%) in comparison to panel radiator system. This is due to increased heat inertia of the floor heating system. Floor heating system heat output can't be adjusted as quickly as in the case of panel radiator system.

At constant heat carrier temperature heating terminal type has almost no influence (~1% change or less) on hybrid heating unit's parameters (ratio of heat energy produced by gas, η_{hybrid} . PEF_{hhp}). The only exception is the ratio of heat energy produced by gas in floor heating scenario, where a decrease of 4,65% can be seen in comparison to panel radiator system. This is because the total heat demand that can be satisfied with heat pump increases (Space heating demand increases, DHW heating demand stays the same).

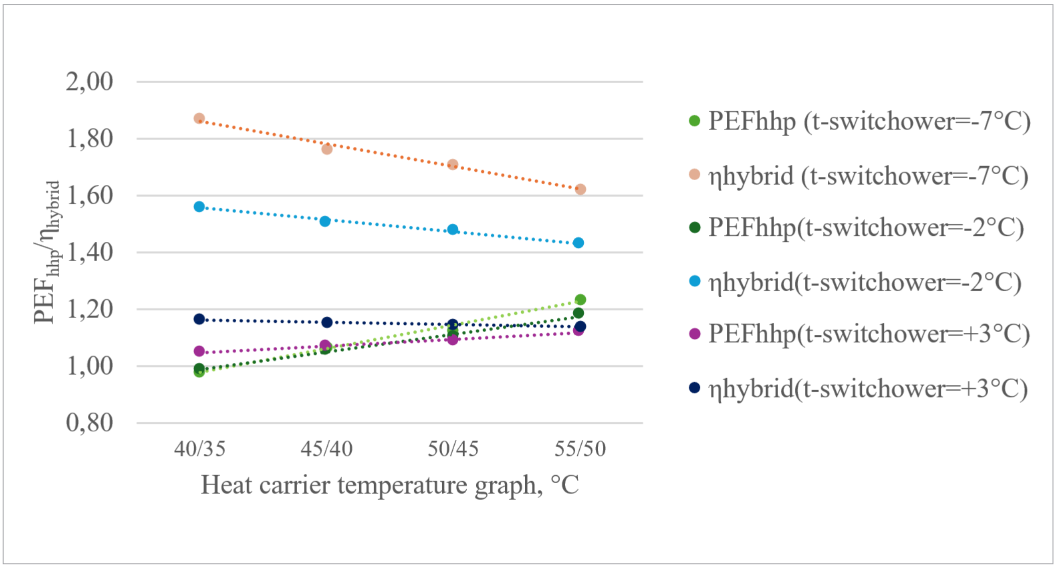
Floor heating increases heat system volume and thermal inertia. Because of this hybrid unit's total operation time in simulation period decreases by 3445 h (33%).

Heating fan coil usage in this building is not preferable. There is no change in hybrid unit's performance or operation time. Total electricity bill, however, is increased by 376 kWh (+12 %) due to fan operation.

In further scenarios hybrid heating unit's performance was simulated with different heat carrier temperatures. Simulations were carried out with heat carrier temperatures of 40/35; 45/40; 50/45 and 55/50°C. The resulting yearly performance parameters of these simulation scenarios can be seen in Figure 8.

Fig. 8

Hybrid heating unit performance at different heat carrier temperatures and switchover temperature settings



As expected, hybrid heating unit's performance parameters worsen with increased heat carrier temperature graph (total efficiency decreases and primary energy factor increases). It can also be observed that the magnitude of performance parameters change is influenced by switchover temperature setting. When changing the temperature graph from 40/35 to 55/50 °C the changes are as follows:

- At switchover temperature setting of -7°C, η_{hybrid} drops by 14,33 %, but PEF_{hhp} increases by 23,42%;
- At switchover temperature setting of -2°C (actual setting), η_{hybrid} drops by 8,7 %, but PEF_{hhp} increases by 17,51%;
- At switchover temperature setting of +3°C, η_{hybrid} drops by 1,74 %, but PEF_{hhp} increases by 6,45%.

However, increased switchover temperature also decreases performance parameters. At the actual temperature graph (40/35°C) η_{hybrid} drops from 1,87 to 1,16 (47 %) and PEF_{hhp} increases from 0,98 to 1,05 (7%) if outdoor switchover temperature setting is changed from -7 to +3 °C.

Hybrid heating unit's performance fluctuations are caused mainly by change in air to water heat pump operation. If the switchover temperature is higher, during the heating period, the ratio of heat produced by gas condensing boiler will increase, and heat pump module will be used less. Because condensing gas boilers performance is not influenced as much as heat pump by the heat carrier temperature, the magnitude of performance parameter change is smaller.

With current simulation parameters the ratio of energy produced by gas boiler changes by meaningful amount only, if the switch-over temperature is adjusted, and is not really influenced by heat carrier temperature.

The ratio of heat energy produced by gas boiler is as follows:

- For switchover temperature -7: 0.22 to 0.26;
- For switchover temperature -2: 0.41 to 0.44;
- For switchover temperature +3: 0.78 to 0.79.

Previously mentioned values are dependent on simulation parameters. In the current simulation the heat sources are switched based on outdoor temperature setting and calculated minimal air to water heat pump COP_{BE}.

In the used climate data, the minimal outdoor air temperature is only -13,8°C, and with this period energy prices (gas price - 98 Eur/MWh; electricity price - 150 Eur/MWh), the COP_{BE} value is **1.5**. During the simulation this threshold is never reached. At the highest temperature graph of 55/50°C and switchover temperature of -7°C, the minimal SPF value is still 1.86 for December.

If the heating period outdoor air temperatures were lower, the energy prices different (electricity cheaper in comparison to gas), or the switchover temperature even lower than heat carrier temperature change could influence energy source ratio more. There is also a possible scenario with dynamic energy resource prices – this scenario would yield even more different results.

When comparing different switchover temperatures after total energy resource cost (for heat carrier temperature 40/35°C), the following results can be observed (table 2).

Table 2

Money spent on energy resources at different switch over temp.

Parameter	Switch - over temperature, °C		
	-7	-2 (actual)	3
Electricity consumption (kWh)	3972.90	3160.00	1115.00
Gas consumption (kWh)	4344.00	6789.90	12218.00
Electricity price (€/kWh)	0.15	0.15	0.15
Gas price (€/kWh)	0.098	0.098	0.098
€ total	**1021.65 (-11%)**	**1139.41**	**1364.61 (+18%)**

With current energy prices, the amount of money spent on energy resources is decreased if the ratio of heat energy produced by condensing gas boiler is smaller. +3°C temperature would not be favorable for the current system. Theoretically -7°C switchover temperature is a little more profitable than the actual -2°C. Realistically, however, -2 °C temperature is more favorable – air to water heat pumps operation is less dependable at lower air temperatures. Several factors could disturb heat pump operation - for instance, evaporator freezing.

Discussion and Conclusions

From the results of current simulation, it can be concluded that the heat terminal type at constant temperature graph has no influence on hybrid heating unit's performance parameters. Positive results can be archived by increasing heating system volume and thermal inertia. In the current simulation the usage of floor heating decreased the total run time of hybrid heating unit by 33 %. Such a result could also be reached by using an accumulation tank.

Hybrid heating unit will operate at increased performance parameters if heat carrier temperature and outdoor switchover temperature setting is lower. Lower heat carrier temperature increases air to water heat pump's COP, which in return increases total hybrid heating unit's efficiency and reduces primary energy factor. At actual switchover temperature (-2°C), change in temperature graph from 55/50 °C to 40/35°C increases η_{hybrid} from 1.43 to 1.56 (8.7%) and decreases PEF_{hhp} from 1.18 to 0.99 (17.5 %).

A higher switchover temperature setting will decrease the total hybrid heating unit's efficiency and increase primary energy factor. At the actual temperature graph of 40/35°C, changing switchover setting from -7 to 3 will reduce nhybrid from 1.87 to 1.16 (47%), but will increase PEFhhp from 0.98 to 1.05 (6.9 %).

Current simulation has been conducted at gas price - 98 Eur/MWh and electricity price - 150 Eur/MWh. With these energy prices COP_{BE} is very low (1.5) and is never reached during simulation – it is not profitable to switch from heat pump to gas before the hard outdoor switchover temperature setting is reached. If electricity price per MWh were to be more expensive relative to gas (compared to used prices) then COP_{BE} would be higher and could trigger heat source switch before the hard outdoor temperature setting is reached. While this algorithm guarantees profitable hybrid unit operation, it would actually worsen previously mentioned performance parameters and make unit less environmentally friendly (increase PEF_{hhp}), because the share of energy produced by heat pump annually would be reduced. Additional simulations must be created to test how electricity and gas price ratios influence hybrid units performance.

It could also be valuable to simulate hybrid heating unit's operation under dynamic energy resource prices. It is possible that in this scenario, current performance parameters and the static switchover temperature settings would have lesser impact, and the COP_{BE} comparison would be the most important control parameter.

In the end several limitations of the simulation, that could have reduced the accuracy of results, must be mentioned. The climate data that was used in the simulation, was not measured on site, but instead was sourced from the closest weather measurement station. The station is not a great distance away from the experiment site (~10 km), this could have created some inaccuracies in heat demand simulation (May and April). Also, wind speed, direct solar radiation, and diffuse solar radiation on horizontal surface was assumed based on data available in IDA ice database, that is less accurate for simulated period.

Another noteworthy point is that simulated hybrid units operation algorithm is not entirely the same as the one for the real unit. As mentioned before in text, due to technical difficulties and time constraints, it was not possible to incorporate bivalent operation in the simulated control algorithm (when both heat pump and gas boiler work simultaneously). Without simulations it is difficult to predict how much this would influence the results, because specific outdoor temperature conditions and hybrid unit control parameter setting values must be present for this operation mode to be triggered.

Lastly, the heat pump model (IDA ice air to water heat pump object) simulates COP based on theoretical formula, that doesn't take into account outdoor unit defrost cycles, caused by humid/cold weather. Heat pump object was calibrated based on heat pumps manufacturers technical data sheets. To increase heat pumps simulation accuracy it would be necessary to acquire heat pump performance data experimentally (for instance as in work Di Perna et al., 2015).

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