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Air source heat pump for domestic hot water supply: Performance comparison between individual and building scale installations

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ABSTRACT

Air source heat pump (ASHP) for domestic hot water (DHW) preparation under cold climatic conditions is studied by TRNSYS simulation. Emphases are focused on comparing different strategies of installation using energy efficiency performance criteria. Studied strategies include individual (apartment-scale) and collective (building-scale, centralized) installations. Different daily hot water consumption profiles are considered in the annual simulation program. Simulation results show higher annual system efficiency from individual installations than centralized system. In the latter case, pump consumption during hot water recirculation makes the overall system less efficient. Due to lower over-sizing factor, the centralized installation is more favourable to the ASHP performance than individual ones. In both the two cases, GHG emissions can be significantly reduced by 7 thanks to ASHP. This paper provides decision-making information regarding energy efficiency incentive policies of heat pumps for a more sustainable energy supply.

1. Introduction

Domestic Hot Water (DHW) is a behaviour-driven consumption [1] and addressing the energy consumption linked to its supply requires consideration both to the energy conversion system and the draw-off profile. Considering the whole urban water cycle, domestic hot water (DHW) preparation is by far the highest energy consumer, representing approximately 85% of total energy needs [2,3]. In the residential sector, contrary from energy demand in space heating that decreases distinctly thanks to strict energy regulations in building envelope insulation and energy system management, the energy need in DHW remains the same. As a result, on the one side, the energy consumption in DHW preparation should be reduced through using higher efficiency system or introducing other renewable sources; on the other side, basic DHW draw-off patterns should be considered that could have strong temporal variation [4] and thus influence to the system performance.

Air-Source Heat Pump (ASHP) water heater is considered as an efficiency way of preparing hot water in the building sector. Under European [5] and WHO guidelines [6], DHW is heated to 60–65 °C to combat bacterial hazards, particularly *Legionella spp.* Given that the water inlet is between 10° and 15 °C [7], the temperature must be raised by 45–55 °C on average throughout the year, not accounting for seasonal variations. DHW tank should be permanently maintained at 60 °C, especially during the standby mode (no flow). At the end-user side, flowing DHW is generally consumed at 45 °C. Since the COP (Coefficient of Performance) of ASHP depends highly on the temperature rise as well as the ambient climatic conditions, a rigorous dynamic study is critical to provide annual performance indicators.

Moreover, scaling-up individual DHW supply to centralized systems (building or district) may be a better solution while requires more complex regulations. On the one hand, high diversity factor in a centralized installation helps reduce the over-sizing factor (both to ASHP and to cumulus storage tank) compared with an individual scale application. On the other hand, the recirculation of DHW in a centralized system should be maintained for a better DHW availability while is not necessary in an individual installation. A comparative study between individual and collective solutions based on energy efficiency criteria has strategic societal interest [8].

This study aims at comparing the energy efficiency performance of the two strategies by all-year operating dynamic simulation

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approach. Main determinants such as local meteorological data, DHW temperature, Urban Heat Island (UHI) effect [9] as well as tank sizing criteria are considered. Our results provide decision-making support for public decision makers, HP fabricants, property holders and energy service companies.

The rest of the paper is organized as follows: first, the comparative dynamic simulation approach is described with special attention to the system configuration and component model. Then, annual simulation is conducted for 8 families in Paris both for individual and collective solutions. Annual Energy Factor of ASHP itself or the system as a whole, together with Greenhouse Gas (GHG) emission savings are used as evaluation criteria. We also investigate the sensibility of our results regarding the air temperature in high density urban centers due to UHI effect and possible lower DHW preparation temperatures.

2. Methodology

In this study, we construct an annual performance simulation program to compare individual and collective ASHP installations following the methodology shown in Fig. 1. We model the ASHP-DHW system by annual dynamic analysis through TRNSYS [10], considering key influencing factors such as tank sizing, draw-off profiles and their diversity factor, meteorological data as well as control strategy regarding auxiliary heater. The annual simulation program comprises of four main parts: energy production by heat pump, DHW storage, DHW consumption and system control. Simplified empirical ASHP model is employed with performance data retrieved from certificated product specifications provided by a fabricant. Meanwhile, the water storage tank is described by a detailed stratification model. For the comparison of the two installation scenarios, we use parameters such as energy losses, annual energy factor and annual HP production efficiency.

What else, meteorological data of the Paris region (cold climatic condition) are used as a reference for the determination of instantaneous COP of ASHP. Four draw-off profiles are considered in individual scale model. While at the building level, we summed up those draw-off profiles. Control strategies are based on cumulus heat preparation, i.e. DHW preparation out of draw-off hours. Electrical auxiliary heater is considered in case of DHW shortage during the day. For the total system performance evaluation, we calculate annual efficiency based on a year cycle production-consumption dynamics.

2.1. ASHP model

An ASHP water heater is mainly composed of a compressor, an evaporator, a condenser, and an expansion valve. The hot side of heat pump, i.e., the condenser, is connected to a DHW storage tank through an immersed coil heat exchanger.

A simplified empirical heat pump model is used in the simulation. Theoretically the COP (Coefficient of Performance) of an ASHP depends on different functional points of the reversed Carnot cycle and the instantaneous COP value is highly dynamic. Since an annual performance simulation should cover all 8760 h of a year, the real dynamic simulation considering the thermodynamic cycle would be too time-consuming. Thus in this study, the heat pump system is simplified by an empirical COP dataset which depends merely on cold side temperature (air, evaporator side), as well as the hot side temperature (water, condenser side).

The heat pump performance data used in this study is retrieved from the commercialized model from fabricant's catalogue [11]. Its empirical performance is integrated in TRNSYS 17 [10] through the Type 941. The performance data are given by standard field test, with the air-side inlet temperature ranging from −10 °C to 35 °C. For the condenser side, corresponding inlet temperatures are between 10 °C and 60 °C. The performance data shown in Fig. 2 are linear regression COP variation with respect to load (water tank) and source (ambient air) temperatures. Under nominal operating conditions (7 °C for air source and 35 °C for the condenser side), the value of COP is 4.3. The risk of freezing in low temperatures and corresponding defrosting operation are not considered in current study.

By using this model we not only avoid detailed dynamic calculations from a physical modelling of the heat pump, but also make the most of real performance data of a commercialized product. In this way, annual simulations with a short time step are realized.

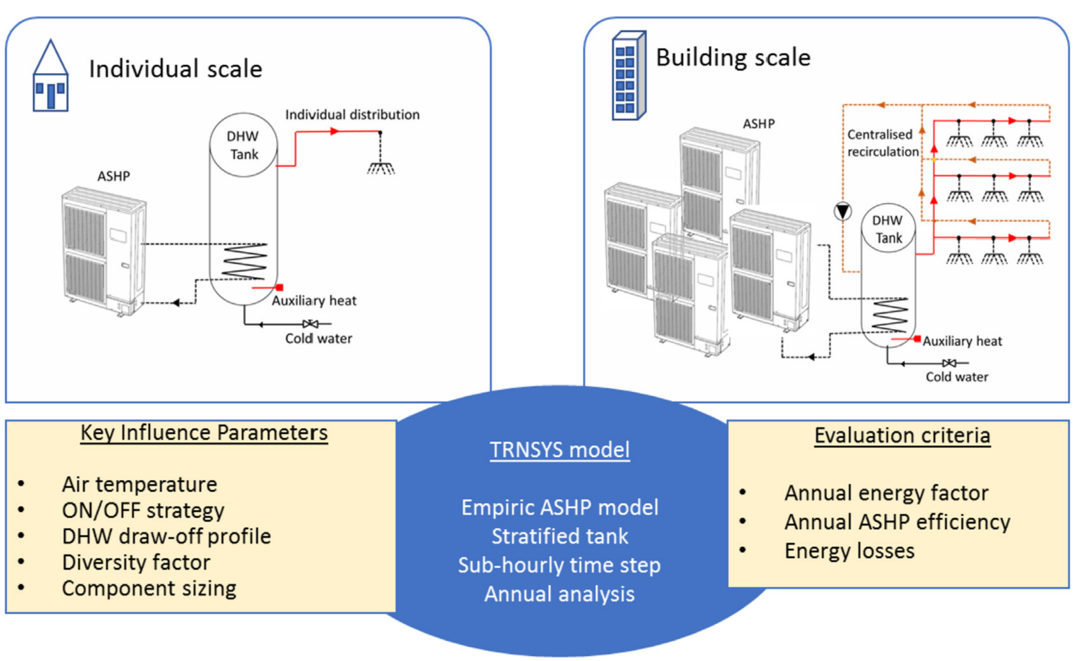

Fig. 1. Comparative simulation approach – what is the best heat pump integration scale for a residential need?.

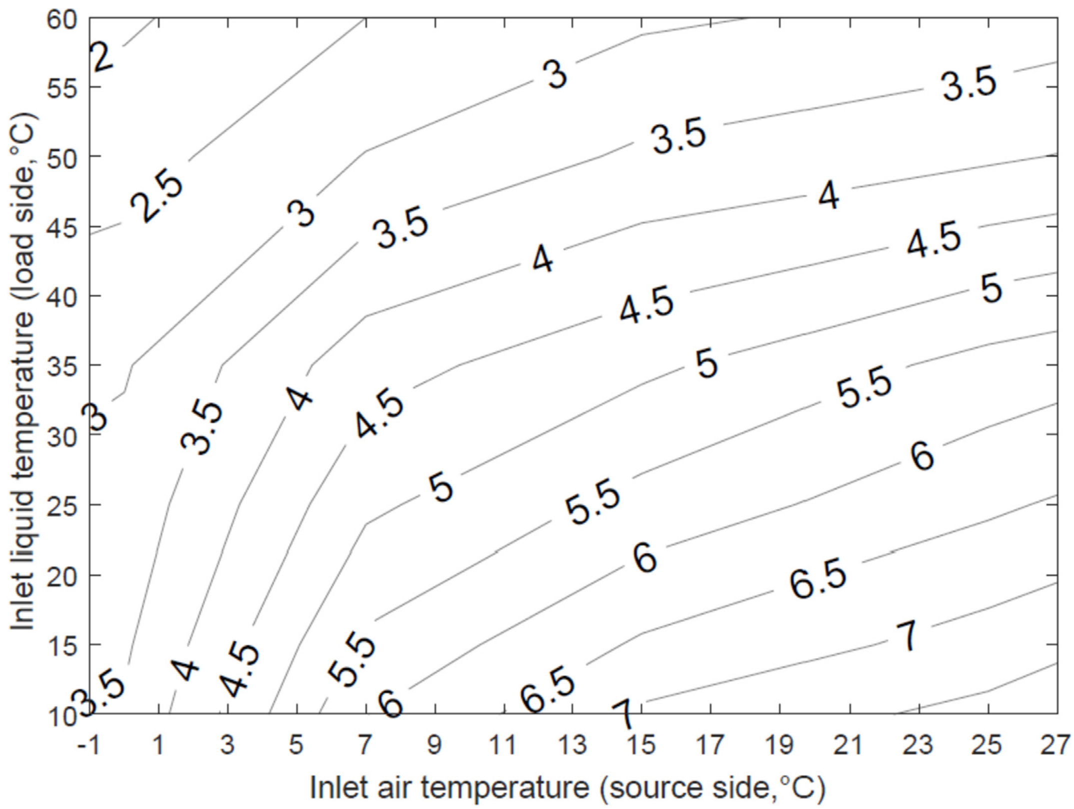

Fig. 2. COP variations as a function of hot and cold side temperatures given by the HP fabricant.

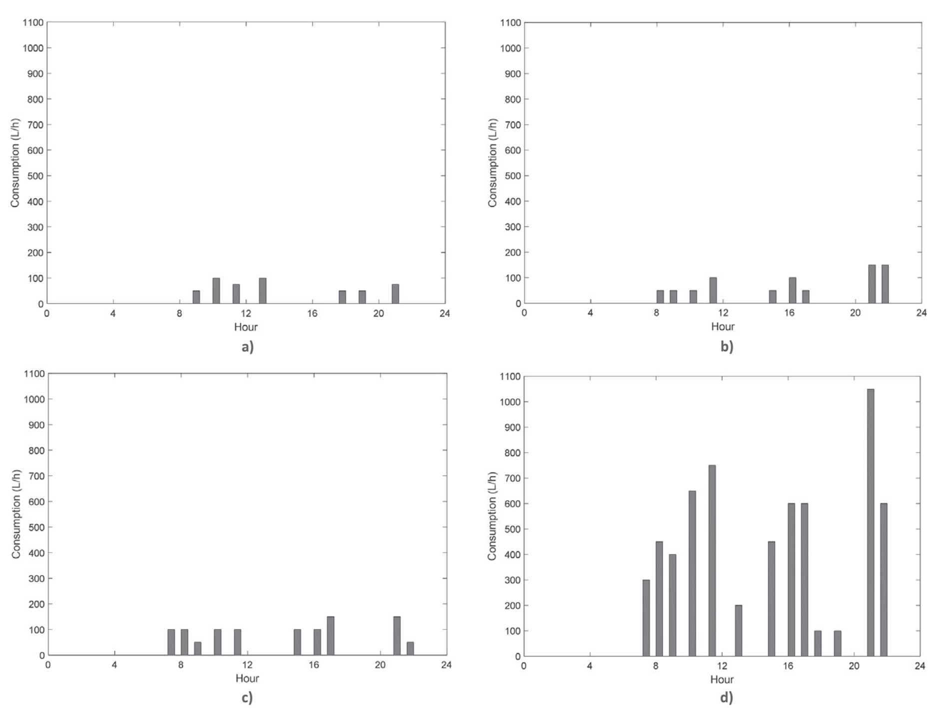

Fig. 3. DHW consumption profiles in a daily basis for three different families and a building consumption a): family A (100 L/d, consumption predominantly morning and evening). b): family B (150 L/d, consumption predominantly morning, midday and evening). c): family C (200 L/d, consumption predominantly morning, midday and evening). d): building 2A3B3C (1250 L/d).

2.2. Draw-off profiles

Shown in Fig. 3(a–d) are hourly residential DHW draw-off profiles for three individual families (a-c) and the whole building (d). These profiles are obtained from several previous studies done by Krauss et al. [12], Spur et al. [4] and as well as the European industrial standard EN 16147:2017 [13]. The daily DHW consumptions range from 100 L/d to 200 L/d for individual families, with different peak draw-off hours in each pattern. The consumption in building (d) is the sum of hourly consumption of a number of 8 families including two A, three B and three C. The total daily DHW consumptions range from 100 L/d to 200 L/d for individual families and that for the building is 1250 L/d.

2.3. Storage tank

A stratified thermal storage tank model is necessary in our study especially regarding the availability of DHW. The tank is operated in cumulus mode and is supposed to cover all-day DHW demand even in the coldest day. The model considers principal physical phenomena as described in Fig. 4: i) heat flux due to "piston" liquid flow through the equivalent inlet/outlet flowrates, ii) heat gain from coil heat exchanger and electrical resistance, iii) heat losses to the ambient air through the wall, iv) heat diffusion between hot and cold layers through water and v) heat diffusion between hot and cold layers through the wall. We do not consider the hydrodynamic flow field inside the tank but only suppose a perfect piston flow along the height. This is realistic since some available stratification enhancement devices such as inlet diffusers, thermal diode valves can prevent fluid from mixing [14].

To precisely represent the tank thermal stratification, we choose to use the Type 534 with 30 layers (nodes). The stratified temperature distribution is essential to guarantee the DHW availability during the phase of standby (without consumption or heating) and draw-off (consumption). During the heating phase, however, the tank becomes fully mixed because of the vertically arranged coil heat exchanger.

2.4. DHW delivery

For individual installation, a length of 14 m between pressurized hot water tank and the water tap is accounted. No circulating pump is necessary. For building-scale installation, a recirculation of hot water is generally needed to guarantee a continuous hot water supply for each apartment. This has to be taken into account by an electrical circulation pump. In the present case, calculations are based on a circulating length of 116 m, representing a three floor building. Between the recirculation pipe and individual tap/shower head, we consider a piping distance of 8 m. The schematic configurations of the two delivery types are shown in Fig. 5.

Heat losses during DHW distribution are taken into account both two cases. The loss can still be divided into distribution loss $\dot{Q}_d$ and dead-leg loss $\dot{Q}_{dl}$.

The distribution heat loss is a function of the distance of delivery and expressed by Eq. (1):

$$\dot{Q}_d = K_d \times L_\text{d} \times (T_{DHW} - T_\text{air}) \tag{1}$$

where T_{DHW} is the DHW temperature and $T_{air} = 20°C$ is that of indoor air; L_d is the delivery length, m, and K_d is the coefficient of losses, $W\ m^{-1}{}^\circ C^{-1}$.

According to the EU Standard EN 12828A+ [15], proper installations should have a K_d value according to pipe outer diameter d_o:

$$K_d = 2.6 \times d_\text{o} + 0.2 \tag{2}$$

The dead leg loss is associated with the cooling down of

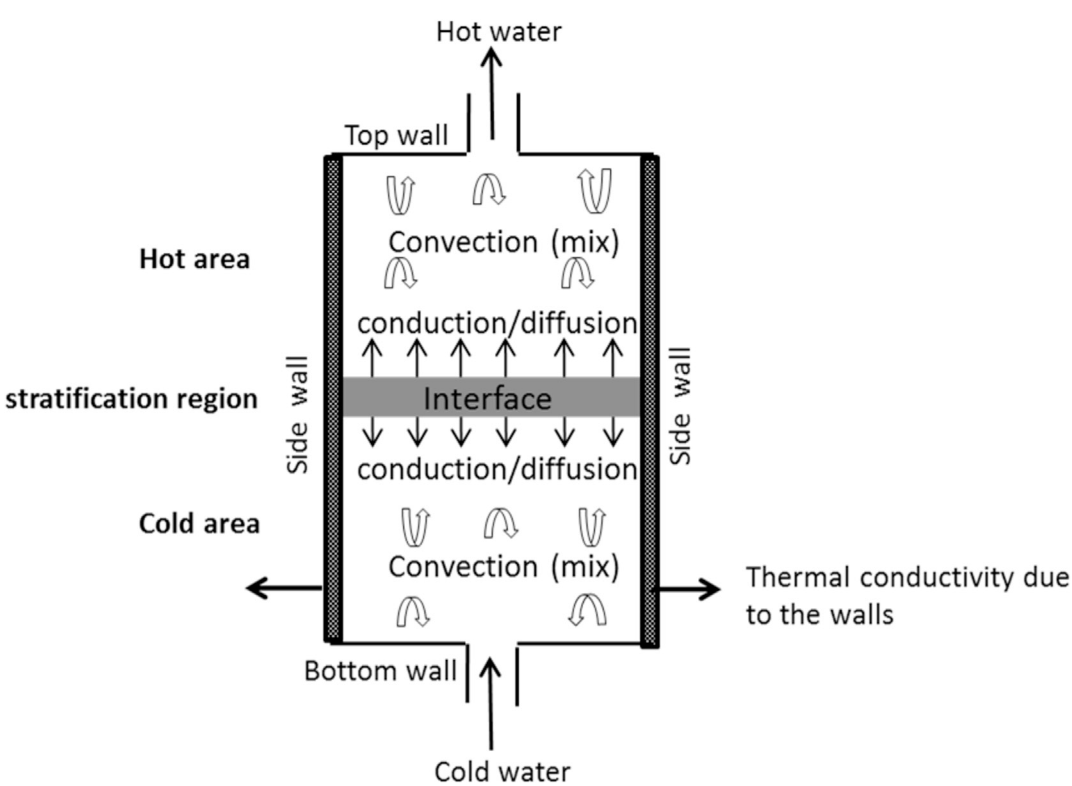

Fig. 4. Stratified hot water storage tank model.

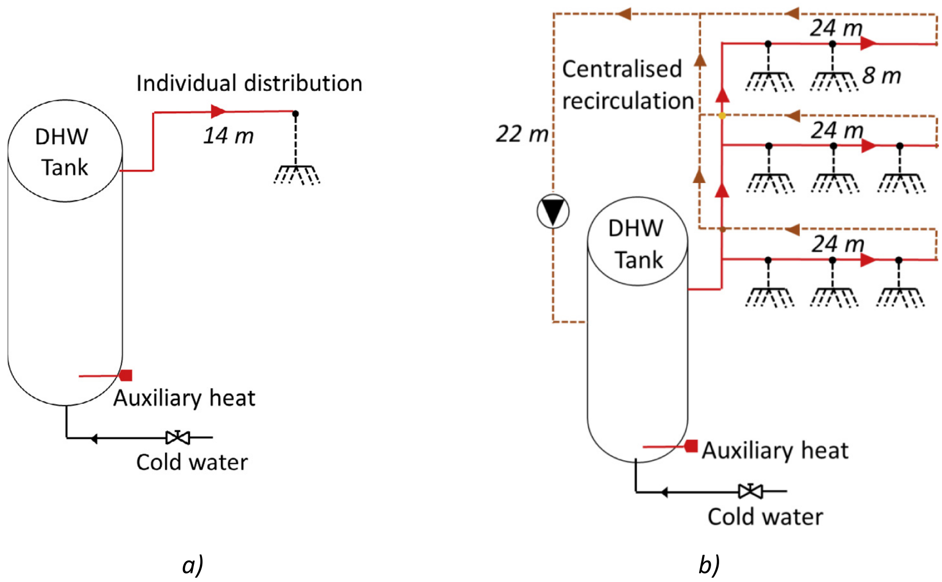

Fig. 5. DHW distribution and recirculation for cases of a) individual and b) centralized installations.

immobile DHW contained in the connecting pipes after each draw-off. These losses are proportional to the length of dead-leg L_{dl} and the number of draw-offs $N_{drawoff}$, as shown by Eq. (3):

$$\dot{Q}_{dl} = \left(\frac{\pi \times C_p \times d_j^2 \times L_{dl}}{4}\right) \times \rho \times N_{drawoff} \times (T_{DHW} - T_{air}) \qquad (3)$$

where d_i is the inner diameter of distribution pipes, m, and ρ and C_p are respectively the density and the massive heat capacity of water.

2.5. Meteorological data

The meteorological data of Paris, France, are used in the simulation process. This region has an average annual temperature of 16 °C, with statistical highest and lowest temperature in a normal year being respectively 35 °C and −7 °C. Shown in Fig. 6 is a typical annual ambient temperature profile of Paris, given by METEONORM [16], via the Type TM-Y in TRNSYS. Besides the historical air temperature, we also consider the UHI effect in high density urban center by adding +1 °C or +2 °C all through the year on the above-mentioned ambient temperature.

2.6. System sizing and control

The tank sizing is based on accumulation storage. With the aforementioned draw-off profiles, we simulate DHW draw-off during the coldest day of the year by assuming a fully charged tank at 6:00 a.m. Then, we look for the least tank volumes according to two criteria. The first concerns individual installations and the highest temperature of the storage tank should be equal or higher than 50 °C at the end of day (22:00). With this limit, the storage tank can cover the needs of DHW at 45 °C by taking into account distribution loss. The second criterion represents the rule of protection against bacteria in distribution loop in case of collective installation. According to the French guidelines about dimensioning sanitary water facilities AICVF [17], the DHW temperature must be maintained at above 50 °C at any point of draw-off.

The results of tank discharge simulation in TRNSYS (Type 534)

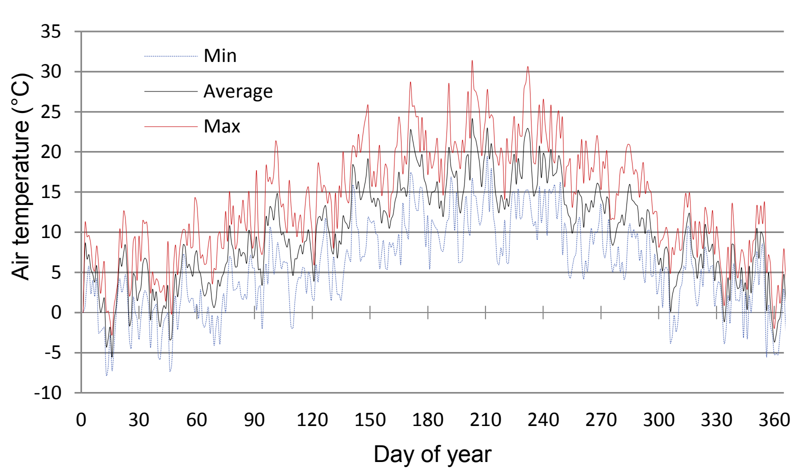

Fig. 6. Meteorological data used in the simulation (Paris, France).

show that the same consumption is satisfied with an over-sizing factor (between the storage tank and daily consumption) of 1.5–2 for individual cases while the ratio is only 1.36 for the collective one. Based on the chosen tank volume, ASHP electric power is selected through product catalogues. The ASHP sizing ensures DHW preparation under cumulus mode (between 12:00 p.m. and 6:00 a.m.) only by ASHP even during the coldest day of the year. Details of dimensioning can be found in Table 1.

Furthermore, identical control strategy for the ASHP system is used for individual and collective installations. Heat pump runs only between 12:00 p.m. and 6:00 a.m. when the average temperature in storage tank is below 45 °C. The ASHP stops when tank temperature reaches 60 °C.

The simulation is processed on an annual basis (8760 h), and with a time-step of 0.01 h between iterations. At the beginning of the year, the initial temperature of the tank is set to 60 °C for every scenario.

3. Results and discussions

3.1. Energy balance and GHG emission

We distinguish system- and production-level annual energy efficiencies based on energy balance analysis. For the system efficiency, we consider the thermal energy delivered to the end-users (water tap) above the overall electricity use, including ASHP, auxiliaries, and circulating pumps if there is one. This efficiency can also be named annual *Energy Factor* (*EF*) through some definitions. The production efficiency, however, represents the annual thermal energy produced by HP (condenser output) above the overall electricity consumption of the ASHP itself (excluding distribution part). The difference is due to that energy losses during DHW storage and delivery (distribution and recirculation) are produced by HP but not consumed by end-users. In the case of a collective installation, these losses are inevitable regardless of how DHW is produced (gas boiler or ASHP). The two efficiencies are useful respectively for householders who pay the energy bills and energy system contractors who need to make a choice between ASHP and gas boiler solutions.

Eq. (4) describes the annual ASHP efficiency EF_{HP}, as the ratio of the thermal energy produced by the heat pump and the electrical energy required to operate the heat pump.

$$EF_{HP} = \frac{\int_0^t \left(\rho C_p \dot{V}_{HTF}\left(T_{o,\,cond} - T_{i,\,cond}\right)\right)\cdot dt}{\int_0^t \left(P_{comp} + P_{aux,HP}\right)\cdot dt} \quad (4)$$

The annual system efficiency EF_{syst} is the ratio of the energy consumed by the user and the electrical energy consumed by the whole system (heat pump, auxiliaries and recirculation pump) and it is given by Eq. (5):

$$EF_{syst} = \frac{\int_0^t \left(\rho C_p \dot{V}_{conso}\left(T_{cons} - T_w\right)\right)\cdot dt}{\int_0^t \left(P_{comp} + P_{aux,tot}\right)\cdot dt} \quad (5)$$

HP and system *Energy Factors* are obtained with the same calculation methods for individual and centralized installations.

To be able to compare all individual systems with the collective one, we calculate a weighted average $EF_{av,ind}$ by taking into account the individual consumption volumes through Eq. (6). It consists of attributing a weighted-value for individual efficiencies. This value is the ratio between the volume consumed and the total volume which is multiplied by the inverse of the efficiency.

$$EF_{av,ind} = \frac{1}{\sum_{i=1}^{8} \frac{1}{EF_i} \times \frac{V_i}{\sum V_i}} \quad (6)$$

Besides, to distinguish primary and electric energy, we compare the above efficiencies with the Primary Energy Factor (PEF) of 2.58, generally applied in France.

Simulation results reveal a higher system *EF* in individual installation than that of collective one. Annual energy flow chart in Fig. 7 shows lower electricity consumption from ASHP and auxiliaries in the collective case (5765 kWh + 1545 kWh) than all individual ones (6540 kWh + 1840 kWh). However, the collective system requires a supplementary 4750 kWh electricity to power the recirculating loop. Thermal energy produced by ASHP comprises of DHW consumptions (16 770 kWh for both cases) and diverse heat losses. Considering the only useful energy, DHW consumption, above the overall electricity consumption, the collective one gives lower system Energy Factor ($EF_{syst} = 1.39$ compared to $EF_{syst} = 2$). It worth noting that the recirculation loop is inevitable in collective installations even with other traditional sources (natural gas or biomass).

Focusing on the ASHP, however, the collective installation gives better *EF*. At the individual level, EF_{HP} equals 2.83; while for the collective case, we obtain an EF_{HP} of 3.02. The higher ASHP performance is mainly due to the smaller tank over-sizing factor used in the collective installation and this will be discussed later with dynamic COP analysis.

Furthermore, a building centralized DHW system has less tank loss than all individual ones together. This is thanks to the “scale effect” of thermal energy storage [14], since larger storage volumes are less exposed to heat loss at the boundaries (see Fig. 7).

Concerning the GHG emission savings calculation, we consider 90 g CO_2 per kWh electricity consumption. For comparison we take the gas boiler as an example, whose CO_2 emission is 220 g/kWh. The gas boiler is supposed to provide all useful and loss energy: DHW draw-off, distribution and storage losses, etc. In the collective case, the recirculation electric consumption is still accounted even for gas boiler. The results presented in Table 2 show that emission saving of CO_2 for collective and individual installations are

Table 1
Sizing details of water storage tank and ASHP.

Strategy	Families	DHW demand (L/d)	Tank volume (L)	Over-sizing factor (–)	Electric power of HP (W)	COP 7 °C/35 °C (–)
Individual	A	100	200	2.00	0.756	4.3
	B	150	250	1.67	0.756	
	C	200	300	1.50	0.834	
Collective	2A3B3C	1250	1700	1.36	3.813	

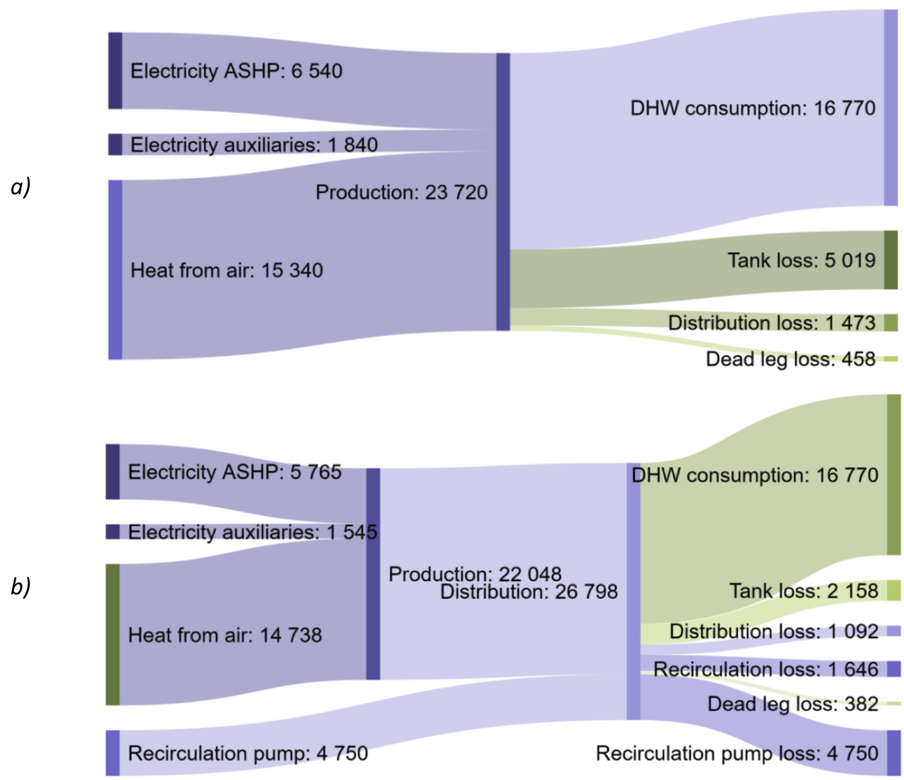

Fig. 7. Performance comparison between individual (a) and centralized (b) installation.

Table 2
Electricity consumption and CO_2 emission saving comparison.

	Individual				Collective 2A3B3C
	A	B	C	Total 2A3B3C	
Compressor consumption (kWh)	573	835	963	6540	5765
HP Auxiliaries (kWh)	167	235	267	1840	1545
Recirculation pump (kWh)	0	0	0	0	4750
Total electricity consumption (kWh)	740	1070	1230	8380	12060
Total GHG emission saving (kg.CO_2)	388	541	688	4464	4193

considerable. For the individual case, a saving of 4464 kg CO_2 per year can be achieved with ASHP. More specifically, the gas boiler would have resulted in a 5218 kg CO_2 emission per year, while that of ASHP has only 754 kg CO_2 per year, representing a carbon emission divided by 7.

The collective case also has consequent CO_2 emission reduction but slightly less effective than the individual one, due to the recirculation system that is nevertheless driven by electricity regardless of gas boiler or ASHP.

3.2. Production-consumption cycle analysis

DHW preparation in an accumulative principle should be studied in separated phases: production, storage and delivery. Fig. 8 shows typical cycles of the whole system during two winter days. Individual case A and the collective case are compared. At the beginning (6:00 a.m. of the day-1), both two tanks are filled with 60 °C hot water. After the first draw-off begins around 7:00 a.m., the tank temperature drops gradually following each draw-off. Until midnight, the average tank temperature will be respectively 20 °C for the collective case and 35 °C for the individual family A. The DHW preparation by ASHP will then be at different regimes: from 35 °C to 60 °C for the individual system and from 20 °C to 60 °C for the collective one. From the COP, we can observe that the ASHP COP ranges between 4 and 1.8 for the individual family. While the collective ASHP shows better performance between 4.7 and 2.3. This result confirms the tank size as the key influence factor for the performance of ASHP.

3.3. Sensibility analysis

Table 3 shows the comparison of system and heat pump Energy Factors in case of lower-temperature DHW supply (59 °C or 58 °C instead of 60 °C) or higher ambient temperature (1 °C or 2 °C above the historical data). The latter is mainly to consider the UHI effect in dense urban areas where higher air temperatures are generally witnessed.

Reducing the DHW preparation temperature by 1 °C has an average positive effect of 0.09 over the Energy Factors. In the case of producing DHW at 58 °C instead of 60 °C, the annual EF_{sys} might increase from 1.39 to 1.57 for the collective case. Similar effects seem to confirm to the individual case as well as to the ASHP

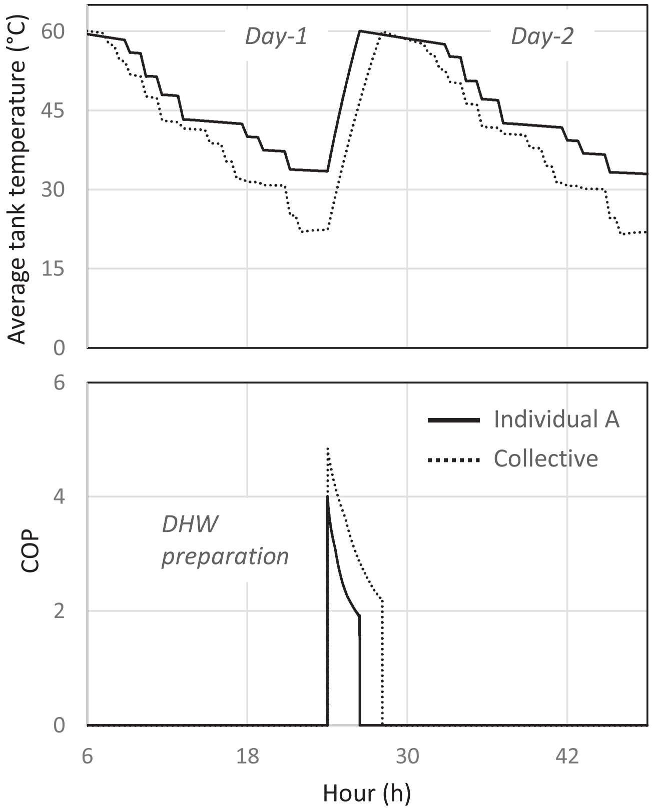

Fig. 8. DHW draw-off and preparation cycle for two winter days.

Table 3
Sensibility study of Energy Factors in case of lower-temperature DHW supply or higher ambient temperature.

Scenarios	Energy Factor	Individual				Collective 2A3B3C
		A	B	C	Total 2A3B3C	
Scenario DHW (Normal air temperature)						
DHW 60 °C	EF_{HP}	2.65	2.71	2.95	2.80	3.02
	EF_{sys}	1.81	1.88	2.18	2.00	1.39
DHW 59 °C	EF_{HP}	2.74	2.81	3.04	2.89	3.12
	EF_{sys}	1.91	1.97	2.25	2.09	1.47
DHW 58 °C	EF_{HP}	2.82	2.89	3.12	2.98	3.21
	EF_{sys}	1.99	2.05	2.35	2.18	1.57
Scenario air (UHI, DHW 60 °C)						
T_{air}	EF_{HP}	2.65	2.71	2.95	2.80	3.02
	EF_{sys}	1.81	1.88	2.18	2.00	1.39
T_{air}+1 °C	EF_{HP}	2.70	2.75	3.02	2.85	3.06
	EF_{sys}	1.86	1.92	2.24	2.05	1.44
T_{air}+2 °C	EF_{HP}	2.75	2.81	3.06	2.90	3.12
	EF_{sys}	1.92	1.98	2.27	2.10	1.50

preparation Energy Factor EF_{HP}. Potentially, significant energy efficiency improvement can be obtained if the permanent regulation on 60 °C for *Legionella* concerns can be revised. For instance, recent studies [18,19] seem to confirm that proper heat shock can be useful to fight against the bacteria development.

Another sensibility factor, the air temperature increase due to UHI or global warming effect, is also favourable to the energy efficiencies. Compared to historical climatic condition, an increase of 2 °C in the air bulb temperature results in an improvement of 0.1 for the system Energy Factor (2.1 instead of 2.0 for the case of individual installation). While we do not expect the arrival of a 2 °C global warming, we can reconsider the climatic conditions in the case of dense urban area with UHI effect during ASHP incitement.

4. Conclusions

We perform rigorous dynamic simulation of ASHP system assessment considering key influencing factors with a comparative approach. By comparing individual installation with centralized one, the study provides strategic decision-making information regarding energy efficiency incentive actions that are ongoing in Europe.

Our results show that encouraging ASHP should be applied firstly in favour of small-size machines at the individual level. They should give higher global performance than building scale installation thanks to the system simplicity and minimal heat loss. At the building scale, replacing gas boilers by HP is still promising since it gives a production COP as high as 3. DHW recirculation results in higher electricity consumption as well as heat loss but is inevitable in collective installations.

However, considering purely primary energy, neither of the two ASHP installations can give an annual energy efficiency higher than 2.58 (French PEF between electrical and primary energy). This means ASHP are not competitive from primary energy saving point of view than fossil sources. Advantage in terms of GHG emission reduction, however, is more evidently in favour of heat pumps since they contribute to 6/7 GHG emission saving.

Sensibility study reveals higher Energy Factors in dense urban areas considering the UHI effect. Lowering DHW water supply by 1 or 2° has positive effects to the energy factor too. Further studies in this topic will allow the development of linear or higher-order regression models considering the two variables, similar to the study of Oussama et al. [20].

Our future work will be focused on the integration of water-source HP in waste heat recovery. Mainly at the district scale, waste heat recovery from different sources can improve the annual efficiency with enhanced energy efficiencies thanks higher source-side temperature. Dynamic simulation will help quantify these waste heat sources more precisely than simple estimations reported earlier [21].

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