SEASONAL PERFORMANCE CALCULATION AND TRANSIENT SIMULATION OF A NEWLY DEVELOPED 18 KW AIR-SOURCE WATER-AMMONIA GAS HEAT PUMP FOR RESIDENTIAL APPLICATIONS

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Abstract: Within the research project HEAT4U, an 18 kW heating capacity air-source water-ammonia gas heat pump (GAHP) for residential applications has been developed and tested according to standard prEN12309. In this paper, the assessment of the GAHP seasonal gas utilization efficiency is carried out by two different methods, namely TRNSYS simulation and a new simplified method. The first method aims to provide realistic estimates of the seasonal performance by means of dynamic simulations of the GAHP and the building load. The second method aims to reduce the complexity and the amount of information required by dynamic simulations without relevant loss in accuracy. In particular, the simplified method follows the approach defined within standard prEN12309 and extends its applicability to different building loads, building insulation characteristics and weather data not considered within prEN12309. By comparing the two methods for different locations representative of the European climate, it can be concluded that the proposed simplified method does not introduce significant deviations with respect to more detailed dynamic simulations, thus making it a suitable alternative for the development of decision support tools.

Key words: gas absorption heat pump, dynamic simulation, seasonal performance, bin method

1 INTRODUCTION

The realistic evaluation of the seasonal performance of an air-source gas absorption heat pump (GAHP) in space heating applications is a difficult task. The performance of the GAHP, namely capacity (Q_h), fuel (Q_g) and electricity (E_{aux}) consumption, are affected by the source air temperature (T_a) and humidity (ω_a), the water supply temperature (T_s) and flow rate (mw), the current load to full capacity ratio (CR) and the modulation capabilities of the appliance (on-off cycling vs. partial load continuous operation). Moreover, transient effects are likely to occur when the air source is humid and cold. In such conditions, frost formation on the evaporator causes the progressive degradation of the appliance performance, until the periodical cleaning of the evaporator, usually foreseen in air-source heat pump appliances (e.g. by heating the evaporator with hot refrigerant gas), leads to a sudden drop of the heating capacity for a relatively short operational time.

Additional complexity arises when the GAHP copes with a dynamic space-heating load. In principle, building simulation tools allow to determine the idealized dynamic space heating load by setting an indoor temperature schedule (e.g. 20°C during daytime and 18°C during nighttime) and calculating instantaneous heat additions (solar radiation, internal gains) and subtractions (dispersion through envelope and windows, air infiltration) to the building mass (walls, floors, indoor air). However, the load calculated in this way does not always represent the load for an appliance in real applications. The heat delivered by the heat emission system (e.g. radiators) to the indoor space is a function of the mean fluid temperature and the emission characteristic of the indoor heating terminals (radiators, fan-coils, floor heating). In simple hydronic systems, such as the central heating system of a single dwelling, the direct hydraulic connection of the heater to the emission system does not allow a fine control of the heat delivered to the indoor space. The commonly used control strategy is based on room thermostats, in combination with the climatic control of the water supply temperature (i.e. a simple relationship with outdoor temperature). With respect to the idealized load curve, the appliance might deliver a higher or a lower load: the former will cause the indoor space to become warmer than comfortable, the latter colder than desired. However, the end effect is an effective load for the heating appliance different from the idealized load; in the likely situation where heating capacity is higher than the idealized load, the heating appliance is going to operate in on/off mode. Finer controlling devices of the heat emission system include thermostatic valves, variable speed controlled fan coils, temperature controlled floor heating systems. When the generation subsystem and the heat distribution subsystem are hydraulically separated (e.g. by means of a water storage, hydraulic separator or heat exchanger), the temperature levels and flow rates on the generation side can be different from those on the distribution side. Provided that the supply temperature on the generation side is higher than the supply temperature on the distribution side, the heating appliance can deliver heat according to an independent temperature and flow strategy (e.g. constant flow rate and constant supply temperature, variable return temperature). Moreover, a fine control of the heat emission system could be capable to follow the idealized load curve with very high accuracy and to transfer it to the heating appliance. Differences in the load curve followed by the appliance might still arise due to constraints in the heating capacity modulation of the appliance itself, which might induce on/off operation at loads much lower than appliance capacity.

Despite of the variety of real life schemes and control strategies, the methods for the determination of the seasonal performance of heat pumps currently available in European standards (EN14825, prEN12309) are rather simplified. The load is linearly correlated to outdoor temperature with the assumption of a balancing point temperature of 16°C, and the occurrences of the outdoor temperatures are calculated with the bin method for a few reference climates. An indoor heat exchanger is foreseen, so that generation side and distribution side are hydraulically separated. The heat emission system can work at constant or variable supply temperature, and different emission systems and corresponding generation supply temperatures are accounted for, including low temperature (35°C), medium temperature (45°C), high temperature (55°C) and very high temperature (65°C). The standards prEN12309-6 defines the procedure for the seasonal performance calculation of gas driven heat pumps. The building load is set equal to the appliance heating capacity at the design temperature, so that the appliance can always provide the heating load throughout the heating season (monovalent case). For a set of predefined test conditions, the gas utilization efficiency (GUE) and auxiliary energy (electricity) factor (AEF) are determined by experimental measurements. The seasonal performance is calculated according to the bin method under the assumptions of quasi-static operation of the appliance and linear variation of GUE and AEF from one test condition to the next one.

The aforementioned standards provide a method for evaluating seasonal performances in a few reference conditions, which seldom represent the conditions for the system planner. The purpose of this work is to extend the applicability of the bin method suggested by prEN12309-6 to more general conditions, including different climates, different building insulation levels, and different design loads. It is envisaged that the proposed method shall not require additional measurements with respect to those prescribed in the standard. The term of comparison for the validation of the proposed method will be transient simulations carried out in TRNSYS for a newly developed GAHP of 18 kW heating capacity, developed within the project HEAT4U. During the project, the GAHP prototype has been fully characterized in the lab, and the experimental data have been used to calibrate a detailed quasi-static model of the appliance, which accounts for air temperature, water supply temperature, water flow rate and capacity ratio.

2 MATHEMATICAL MODELLING

2.1 TRNSYS simulations

TRNSYS Type56 allows generating the idealized load curve of a given building. Besides the building envelope characteristics, the main inputs to the building model are the outdoor air temperature, the indoor air temperature, the sky temperature, the ground temperature, the internal gains and the solar radiation for different orientations. The main output is the heating load, which has to be covered by the GAHP. The considered internal gains are the ones caused by inhabitants, by technical equipment and lighting. A typical example of heating load is provided in Figure 1. When correlated to outdoor temperature, the idealized load curve will become scattered. This is mainly a consequence of the building thermal capacity, solar gains and variable internal gains.

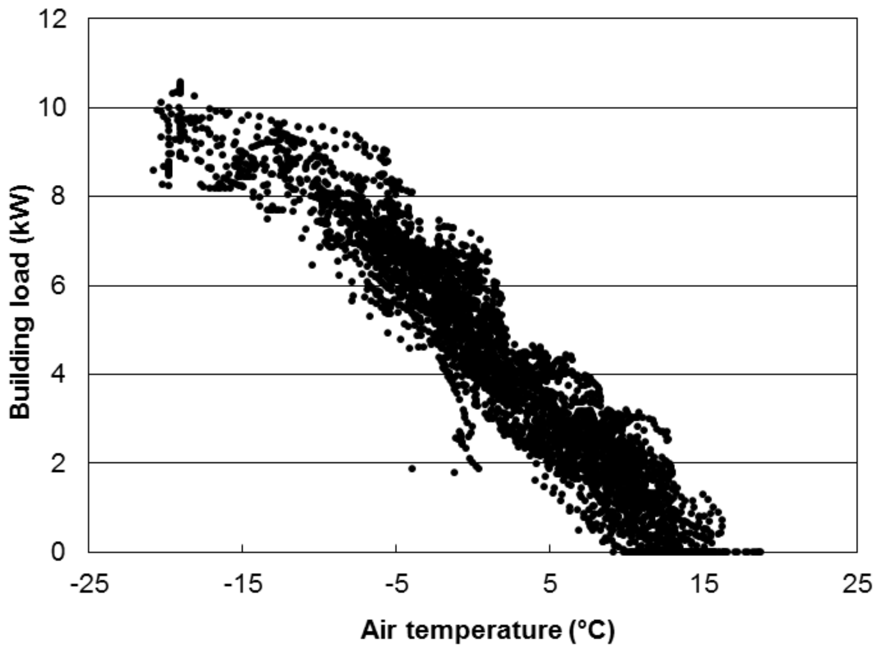

Figure 1: Example of heating load vs. outdoor temperature.

Hydraulic separation between generation and distribution circuits is assumed, so that the water supply temperature in the generation circuit is assigned according to a predefined climatic curve. At any time step, the appliance performance is calculated for the given heating load, supply temperature, flow rate and outdoor temperature, based on a quasi-static GAHP model. It shall be noted that the main difference between TRNSYS simulation and the bin method is the existence of an entire range of heating loads, rather than a single heating load, for each outdoor temperature within the heating season. Concerning the GAHP, its model is based on real performance data measured according to prEN12309-4 under a large range of operating conditions (see Table 1).
The GAHP quasi-static model consists of a set of algebraic equations which allow calculating the GUE_{PL} and the AEF_{PL} at any working conditions.
In particular, the GUE at a given part load condition is calculated from Eq.1

(1)

Table 1: Laboratory test conditions for the GAHP

T_a (°C)	T_s (°C)	T_r (°C)	Q_h (W)
-22	55.0	45.5	11250
-15	49.1	41.2	9154
-15	55.0	44.8	12113
-10	55.0	43.4	13668
-10	55.0	43.8	13221
-7	35.0	28.6	16343
-7	35.2	29.1	16800
-7	44.0	38.2	6807
-7	52.1	42.1	11653
-7	55.0	42.8	14408
-7	55.0	43.2	13889
-7	55.0	50.5	11451
-2	35.0	28.6	16418
-2	55.0	41.9	15369
-2	55.0	50.0	12860
0	54.8	41.9	15318
2	35.0	28.3	17065
2	37.0	33.5	4126
2	42.1	36.0	7165
2	54.9	41.4	15901
2	55.0	42.1	15253
2	55.0	49.6	13697
7	32.0	29.7	2655
7	35.0	28.2	17345
7	36.1	32.1	4698
7	45.0	38.5	16507

T_a (°C)	T_s (°C)	T_r (°C)	Q_h (W)
7	45.0	34.7	17652
7	45.0	29.1	18425
7	45.3	37.8	17339
7	46.1	37.7	9809
7	55.0	41.4	15931
7	55.0	37.0	16482
7	55.0	41.3	16120
7	55.0	41.5	15931
7	55.0	43.4	15950
7	55.0	47.2	15271
7	55.0	49.2	14926
7	57.0	47.1	15300
7	59.1	45.1	15357
7	61.7	42.5	15379
12	28.0	27.0	1187
12	30.0	28.3	2031
12	34.0	30.3	4357
12	35.0	28.2	17286
12	45.0	38.3	17116
12	55.0	49.0	15446
12	55.1	41.3	17519
12	55.1	41.3	17519
15	55.0	41.4	17491
20	55.0	41.4	17691
30	54.9	41.4	17619

The GUE_0 is the GUE at full power, which is estimated by means of Eq.2:

$$\mathrm{GUE_0} = \mathrm{GUE_{th}} \cdot \{\sin[0.0324 \cdot (\mathrm{T_r} - \mathrm{T}_a)]\}^{0.94} \qquad (2)$$

where GUE_{th} is the Carnot efficiency of an appliance working among the same temperatures (Hellmann, 2002), as in Eq. (3).

$$\mathrm{GUE}_{th} = \frac{T_r}{T_{gen}} \frac{T_{gen} - T_a}{T_r - T_a} \qquad (3)$$

In Eq. (3) the temperature in the generator (T_{gen}) is assumed constant and equal to 200°C. This value is close to the actual value in most of the operating conditions. Moreover, through a sensitivity analysis, it has been verified that variations on the generator temperature within the typical temperature range do not affect significantly the model output.
The CR is the ratio between the actual power delivered by the appliance at any part load condition (Q_h) and the maximum capacity (Q_{h0}) at the same working conditions, i.e. at the same outdoor air temperature and water inlet temperature:

$$\mathrm{CR} = \frac{Q_h}{Q_{ho}} \qquad (4)$$

Q_{100} is obtained from the product between the GUE_0 and the gas input at full load, Q_{g0}, which can be expressed as a function of the outdoor air temperature as in Eq. 5:

$$Q_{g0} = -0.032681 \cdot T_a + 11.276 \tag{5}$$

The electrical power consumption varies linearly with the CR. Two different functions are defined according to whether the working condition is in the modulation range or in the on-off zone:

$$E_{aux} = \begin{cases} 360.18 \cdot CR + 58.217 \; for \; CR < 0.35 \\ 431.95 \cdot CR - 9.5032 \; for \; CR \geq 0.35 \end{cases} \tag{6}$$

Moreover, as the fan is switched off when the outdoor air temperature is lower than -15°C, below this point the electrical power consumption is observed to be constant and equal to 220 kW.

2.2 Modified bin method

The calculation of seasonal performance follows from the application of the bin method, as described in prEN15316-4-2. The bin hours b_j are calculated according to the climatic file as the number of hours for which the outdoor temperature falls within the temperature interval T_j± 0.5 K. The seasonal GUE (SGUE) is calculated as the ratio of the overall heating energy to the overall gas consumption:

$$SGUE = \frac{\sum_{i=1}^{N} b_j Q_j}{\sum_{i=1}^{N} b_j (Q_j / GUE_j)} \tag{7}$$

Similarly, the seasonal AEF (SEF) is calculated as the ratio of the overall heating energy to the overall electricity consumption of the appliance:

$$SAEF = \frac{\sum_{i=1}^{N} b_j Q_j}{\sum_{i=1}^{N} b_j (Q_j / AEF_j)} \tag{8}$$

The load corresponding to the bin temperature T_j is defined according to the following linear relationship:

$$Q_j = Q_{des} \frac{T_j - T_{des}}{T_{bp} - T_{des}} \tag{9}$$

Differently from standard prEN12309, here the design heating load can be lower than the capacity of the GAHP at the design temperature (T_{des}). Moreover, the balancing point temperature (T_{bp}) is not fixed. In general, design heating load and balancing point temperature of the building are supposed to be known design values.

The determination of GUE_j and AEF_j is the most critical aspect. According to prEN12309-6, GUE and AEF are determined for certain reference conditions and then linearly interpolated between two adjacent conditions. As an example, the reference conditions for two reference climates (average, warmer) and two application temperatures (55°C, 65°C) are illustrated in Table 2. Note that, for each condition, both GUE(AEF) at part load and GUE(AEF) at full load shall be determined. In the general case of an intermediate climate between two reference climates and an intermediate application temperature between two reference application temperatures, four sets of experimental data are needed for the multi-dimensional linera interpolation of GUE and AEF. For example, a climate with design temperature -5°C and a variable application temperature of 60°C is considered. The four data sets are the combinations of the two climates average (-10°C design temperature) and warmer (2°C design temperature) with the two application temperatures 55°C and 65°C. First, a set of suitable calculation conditions is defined: the outdoor temperatures are selected according the design temperature of the considered climate and the sequence fixed by the standard (-10, -7, 2, 7, 12°C). In the example, the calculation conditions will be as follows: -5, 2, 7, 12°C. Second, the water supply temperatures are defined for each calculation condition, according to the desired climatic curve. For example: 60, 51, 43, 33°C, respectively. Third, Q_h, GUE and AEF at full load at the design temperature -5°C are calculated by two-dimensional interpolation: the three closest conditions are selected and triangular basis functions in the two variables outdoor temperature and water supply temperature are used. Fourth, Q_h, GUE and AEF at full load for the remaining calculation conditions are calculated as the linear interpolation of the two closest conditions selected among the experimental data set. Fifth, GUE part load factor (PLF_g) is determined as the correlation between GUE/GUE_o and CR as provided in the four experimental data sets, and similarly for the AEF part load factor (PLF_e). An example of the calculation of the two part load factors is provided in Figure 2: second order polynomials have been used. Sixth, the heating load is assigned for each calculation condition on the basis of design load, building balancing point temperature and building seasonal heating demand. Within this work, these data are calculated form the hourly building load obtained from the TRNSYS simulations. Conventionally, the design load is assumed to be the maximum hourly load in the heating season, while the balancing point is the outdoor air temperature for which the heating load, calculated form the linear best fitting of the building load with the outdoor air temperature, is zero.

Table 2: Sample test conditions according to prEN12309-6.

Outdoor Temperature (°C)	Water supply temperature (°C)			
	Average climate		Warmer climate	
	High	Very High	High	Very High
-10	55	65	-	-
-7	52	61	-	-
2	42	49	55	65
7	36	41	46	53
12	30	32	34	39

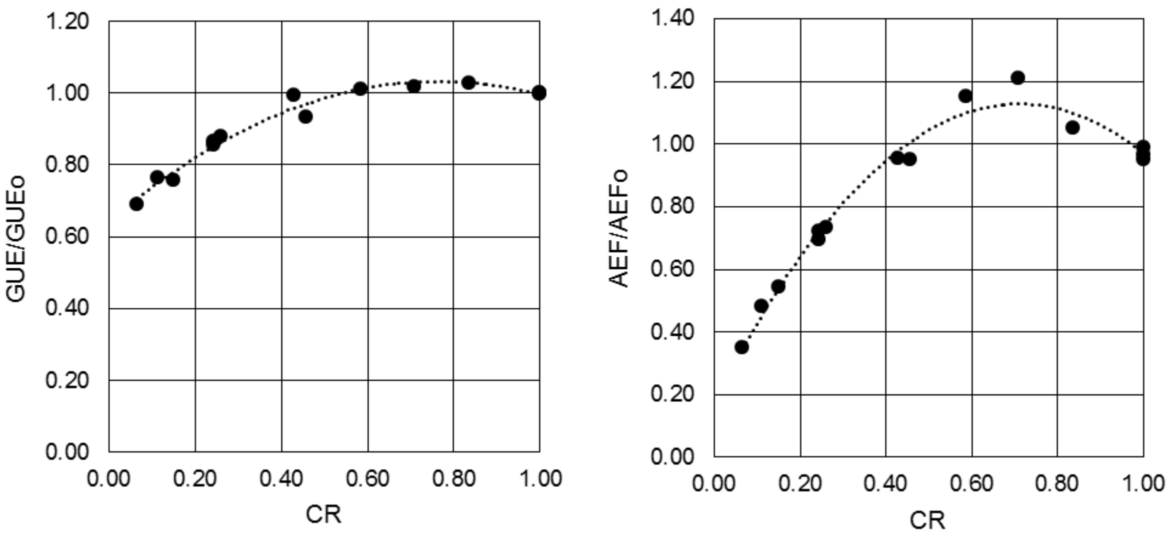

Figure 2: GUE and AEF part load factors represented by second order polynomial correlations.

In the standard, a linear profile of the heating load with outdoor temperature is assigned, so that the design load and balancing point conditions are met. In the modified bin method, a parabolic profile of the load with outdoor temperature is assigned, so that also the condition on the seasonal heating demand can be met. Seventh, the GUE and AEF at part load at the required supply temperature are evaluated for each test condition, as follows:

$$GUE = PLF_g(CR) \cdot GUE_o \qquad (10)$$
$$AEF = PLF_e(CR) \cdot AEF_o$$

Finally, GUE_j and AEF_j are determined by linear interpolation of the two closest GUE and AEF values, respectively.

3 SEASONAL PERFORMANCE CALCULATIONS

A few sample buildings have been considered in this study, with the aim of comparing the GAHP seasonal performance calculated using TRNSYS and the modified bin method (MBM) over a range of very different operating conditions.
For each building, the heating season has been defined according to the building location, in the same way as prEN12309 (see Table 3).

Table 3: Heating season limits according to the building location.

	Helsinki	Athens	Milan	Strasbourg
beginning	01-sep	01-nov	15-oct	01-oct
end	31-may	30-apr	30-apr	30-apr

Internal gains due to inhabitants are set according to the typical occupation profile for apartments. This corresponds to yearly values of 7.0 kWh/m²a of sensible heating. Specific schedule are created for lighting and for technical equipment too, resulting to an average heat emission of 2.6 W/m².
The envelope features of the buildings are illustrated in Table 4, along with the associated climatic file. In particular, the total building surface is reported (including walls, windows, roof and basement), the average U-value, the infiltration rate and the ratio between windows surface and overall building surface.

Table 4: Features of sample buildings and associated climatic file.

Building	Total floor area (m^2)	External surface (m^2)	Average U-value W/(m^2 K)	Infiltration rate h^{-1}	Window surf. ratio (m^2/m^2)	Climate file
Helsinki 1	200	436	0.61	0.5	5.7%	FI-Helsinki-Kaisani-29980.tm2
Helsinki 2	450	728	0.37	0.3	6.9%	
Strasbourg 1	200	436	0.83	0.5	5.7%	FR-Strasbourg-71900.tm2
Strasbourg 2	450	728	0.51	0.3	6.9%	
Athens 1	200	436	1.59	0.5	5.7%	GR-Athinai-167140.tm2
Athens 2	900	1284	0.64	0.3	5.4%	
Milan 1	200	436	1.02	0.5	5.7%	IT-Milano-Linate-160800.tm2
Milan 2	450	728	0.65	0.3	6.9%	

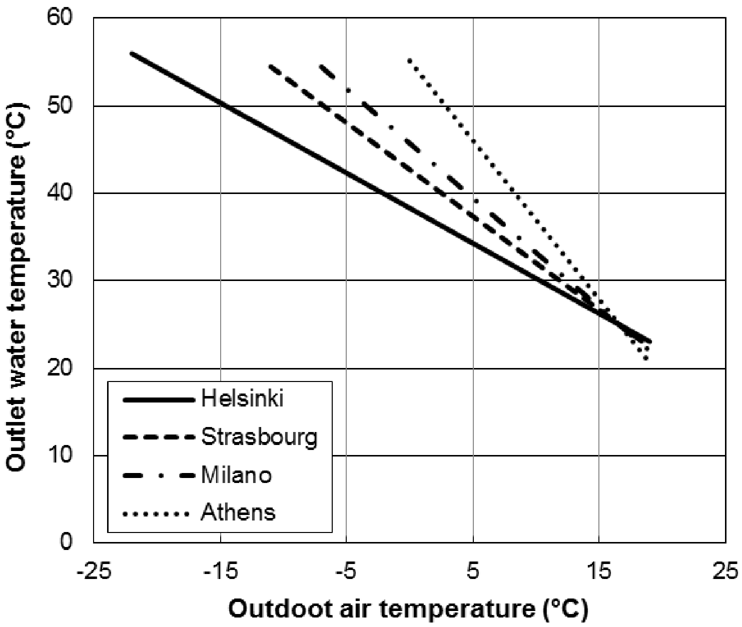

Figure 3: Climatic curves associated to the considered buildings.

Table 5: Seasonal performance calculations according to TRNSYS simulations and modified bin method.

		TRNSYS SIMULATION				MODIFIED BIN METHOD			
	Q_h	E_{aux}	Q_g	SGUE	SAEF	E_{aux}	Q_g	SGUE	SAEF
	MWh	kWh	MWh	-	-	kWh	MWh	-	-
Helsinki 1	31.0	979	24.5	1.27	31.7	985	24.4	1.27	31.5
Helsinki 2	25.3	842	20.6	1.23	30.1	874	20.7	1.22	29.0
Strasbourg 1	23.0	749	17.8	1.30	30.7	745	17.7	1.30	30.9
Strasbourg 2	15.6	598	12.8	1.22	26.1	617	13.0	1.20	25.3
Athens 1	13.1	488	10.0	1.31	26.8	505	10.0	1.31	25.9
Athens 2	9.0	366	7.1	1.26	24.5	399	7.2	1.24	22.5
Milan 1	23.8	730	18.0	1.32	32.6	725	18.0	1.32	32.8
Milan 2	17.4	600	13.8	1.26	29.1	622	14.0	1.24	28.1

A specific climatic curve, corresponding to the high temperature application of standard prEN12309, has been defined for each building, see Figure 3.

Two main seasonal performance figures have been calculated and compared: gas consumption and electricity consumption. In addition, the two seasonal performance indicators SGUE and SAEF were also calculated. As shown in Table 4, the heating demand varies quite a lot among the considered cases, from a minimum of 9.0 MWh in Athens-2 to a maximum of 31 MWh in Helsinki. Despite of these large differences among the heating demands, the (Modified Bin Method) MBM was capable to predict the gas consumption quite well in all cases, with maximum deviation of 2% with respect to TRNSYS simulations (see Table 5). The MBM seems to be less precise for what concerns electricity consumption, with maximum deviations up to 9%. However, it shall be noted that electricity consumption is quite modest in comparison to gas consumption, and a larger percentage deviation in electricity does not imply a big difference in the overall seasonal performance assessment.

Deepening the analysis, the average gas and electricity consumption within each bin are also compared (see Figure 4). It is found that the gas input predicted by MBM is in excellent agreement with the corresponding figure obtained with TRNSYS. Some discrepancies are found for electricity consumption, especially at about -15 °C. These are due to the particular control strategy of the fan. In fact, below -15°C the fan is no longer operated with a large discontinuity in electricity consumption, a fact that cannot be captured by the simple part load factor approach. Nevertheless, the overall trend for the average electricity input by bin is reasonably good.

Table 5: Difference between the results from the two methods

	E_{aux}	Q_g	SGUE	SAEF
Helsinki 1	1%	0%	0%	-1%
Helsinki 2	4%	1%	-1%	-4%
Strasbourg 1	0%	0%	0%	0%
Strasbourg 2	3%	1%	-1%	-3%
Athens 1	3%	0%	0%	-3%
Athens 2	9%	2%	-2%	-8%
Milan 1	-1%	0%	0%	1%
Milan 2	4%	1%	-1%	-3%

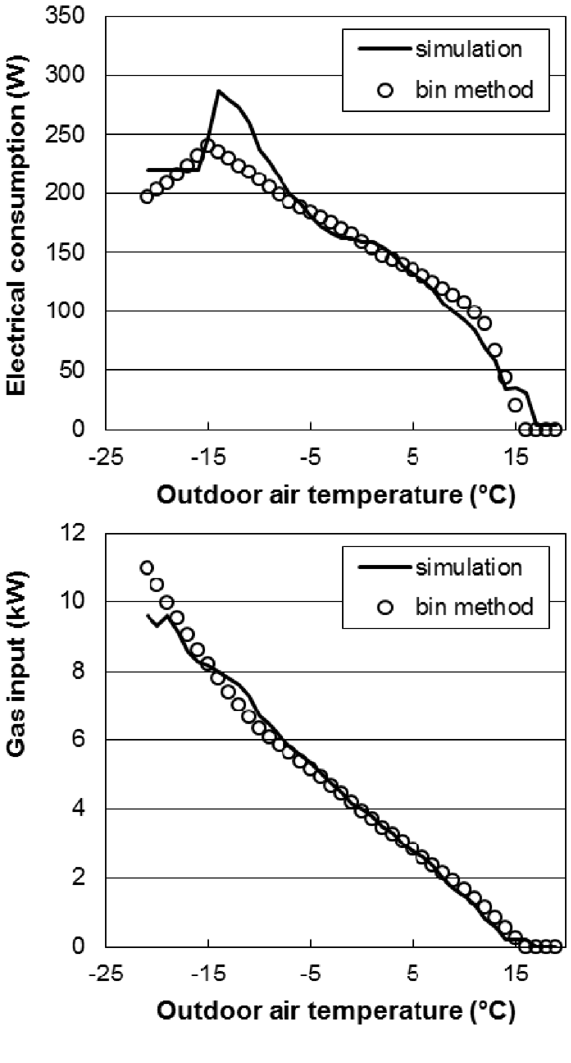

Figure 4: Average gas input and electricity input by bin temperature according to TRNSYS simulations and modified bin method for building Helsinki 1.

4 CONCLUSIONS

The bin method suggested by prEN12309-6 has been modified with the purpose of extending its applicability to more general conditions, including different climates, different building insulation levels and different design loads. The extension of the bin method has been done by maintaining the simple approach and without the need of additional measurements with respect to those originally prescribed in the standard. The proposed method makes use of a parabolic profile, rather than linear, between the building heating demand and the outdoor air temperature. Moreover, the balancing point temperature and the design load are chosen according to the building characteristics, rather than on the basis of the appliance capacity and the climate. Finally, the GAHP performance at any working conditions is derived on the basis of the data collected under a typical test campaign according to prEN12309.

The new method has been tested on a few sample buildings with different insulation levels and located in different climate conditions and its applicability has been verified by comparing its results with the results of a more detailed building and appliance simulation carried out with TRNSYS. A reasonable agreement between TRNSYS simulations and the modified bin method is found, especially for the calculation of the SGUE. This makes the proposed simplified method a suitable alternative for the development of decision support tools.

Future development of this work is the investigation of the effects of an imperfect control of the heat emission systems with respect to the idealized building load.

5 NOMENCLATURE

Q_g	gas input in NCV, W
Q_{g0}	gas input at full load in NCV, W
Q_{des}	design heating load, W
E_{aux}	electricity consumption, W
Q_h	heating load, W
Q_{g0}	heating capacity at full load, W
b_j	number of hours for bin j, h
Q_j	heating load for bin j, W
T_j	temperature for bin j, °C
T_{des}	design temperature, °C
T_{bp}	balancing point temperature, °C
T_{gen}	generator temperature, °C
T_a	ambient or outdoor temperature, °C
T_s	water supply temperature, °C
T_r	water return temperature, °C
m_w	water mass flow rate, kg/s
GUE	gas utilization efficiency = Q_h/Q_g, -
GUE_o	gas utilization efficiency at full load, -
GUE_{th}	gas utilization efficiency of a Carnot cycle working with the same temperatures, -
AEF	auxiliary energy factor = Q_h/E_{aux}, -
AEF_o	auxiliary energy factor at full laod, -
PLF_g	gas utilization efficiency part load factor, -
PLF_e	auxiliary energy factor part load factor, -
NCV	net calorific value, J/kg

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