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Variable-Speed Air-to-Water Heat Pumps for Residential Buildings: Evaluation of the Performance in Northern Italian Climate

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Abstract

In this paper we evaluated the seasonal performance of air-to-water heat pumps, which is strongly dependent on operating temperatures (of source and sink) and on the load. A new TRNSYS type (subroutine) modeling variable refrigerant flow heat pumps has been developed and it has been used to simulate a heating system coupled with a building, in Northern Italian climate. We analyzed the issue of the degradation of the heat pump performance during the partial load operations. Additionally, the seasonal performance obtained by means of dynamic simulation was compared with the predictions of the bin method, currently required by the Italian standard UNI/TS 11300-4 (similar to the detailed method described in standard EN 15316-4-2). Results highlight that the appropriate sizing of these units is important in order to reach a high efficiency of the building-plant system, and that use of the design heat demand based on the standard EN 12831 can lead to oversizing heat pumps.

Keywords - heat pump, SCOP, VRF, heating, dynamic simulation

1. Introduction

The mandatory provisions about the use of renewable energy introduced by the Italian Decree D. lgs. 28/2011 and by the EU Directive 2009/28/CE greatly increase the requirement to use renewable energy sources to cover the energy consumption of a building. The Directive also acknowledges air to water vapor-compression heat pumps (HPs) as using renewable energy sources. Therefore, this technology is considered an option to reach the European target for a sustainable energy supply: according with the IEAs blue map scenario they will account for 63% of CO_2 total savings (heating and cooling equipment) in 2050 [3].

In this paper we analyze Air Source Heat Pumps (ASHPs) in heating applications. This HVAC solution is made possible by the adoption of low temperature heating terminals (e.g. radiant panels) and by the better insulation of the envelope in new (and refurbished) buildings.

The performance of these devices is quite different from that of a (condensing) boiler that has been the heat generation system of choice up to a couple of years ago: in fact, the *COP* is strongly dependent on the outside temperature and on the load and it is necessary to take into account the specific applications of the unit. The actual seasonal coefficient of performance (*SCOP*) is usually different from the one declared in manufacturer's data and also from the *SCOP* evaluated with the calculation procedure described in the Italian Standard UNI TS 11300-4 ("bin method") combined with standard EN 14825 [1]. Moreover, variable speed units' performance is affected by the *COP* variation at part load conditions, that is rarely declared by manufacturers.

2. Methods

2.1 Modeling a Variable-Speed Air-to-Water Heat Pump

With a dynamic simulation program, such as TRNSYS, it is possible to assess the impact of these aspects in order to perform a better evaluation of the seasonal performance of ASHPs. The currently available TRNSYS subroutines (Types) are capable to model only air-source heat-pumps with a constant speed compressor. Hence, a new Type (Type VRF Variable Refrigerant Flow) has been coded to model variable refrigerant flow heat pumps, modifying an existing "Type" (Type 941 by J. Thornton, 2005 [4]). Very few data about part load operation of inverter controller HP are available in the literature [5], [6] even if some manufacturer data begin to appear in the technical documentation, often showing a slight increase of the COP in the initial stages of part load operation, peaking at CR values around 0,6-0,55. Three operating modes were identified as a function of the capacity ratio (*CR*, defined as the ratio of the required thermal capacity over the nominal capacity of the unit at the operating temperatures):

1. HP working at nominal speed;
2. HP working part load with the *COP* as declared in the manufacturer's data;
3. HP working in on/off mode because *CR* is below the modulating capacity limit.

According to the Italian technical specification UNI TS 11300-4 (which is based on the standard EN 14825) *COP* at part load conditions for variable capacity can be evaluated multiplying the COP value by a correction factor f_{corr} that depends on the capacity ratio (*CR*), as follows:

$$f_{corr} = \frac{CR}{C_c\, CR + (1 - C_c)} \quad if\ CR < CR_{lim} \tag{1}$$

$$f_{corr} = 1 \quad if\ CR \geq CR_{lim}$$

In case of missing manufacturer's data, the default degradation coefficient C_c is assumed equal to 0.9 and CR_{lim}, i.e. the minimum capacity ratio modulation limit, is assumed equal to 0.5.

In this paper, we considered three alternatives:

a) no *COP* degradation at part load (i.e. constant $f_{corr} = 1$);

b) a pump capable to modulate up to CR 0.3 with a slight COP increase (up to f_{corr} = 1,2 around CR 0.55 as can be the case with the last generation of variable speed heat pumps commercially available);

c) *COP* degradation at part load as proposed by the standard UNI/TS 11300-4.

In case b) and c), when the minimum modulating capacity of the unit has been reached, the unit is assumed to work in on/off mode and, consequently, the *COP* is assumed to vary according to (1). Curves b) AND c) have been modelled within Type VRF. They are shown in figure 1.

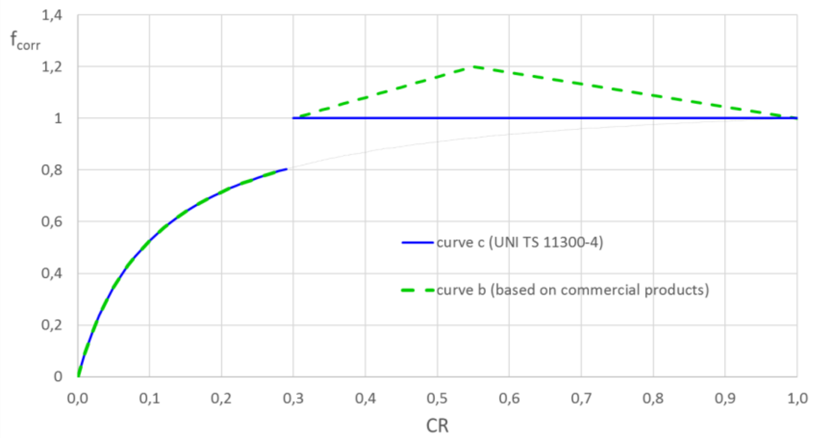

Fig. 1 COP degradation functions at part load. The CR value where the COP begins to degrade (here, 0.3) and the CR value where the maximum of f_{corr} occurs (0.55), as well as the maximum f_{corr} (1.2), can be modified as parameters of the Type, depending on the specific performance of the unit.

2.2 Bin Method and Dynamic Simulations

In this paragraph we analyze how the seasonal performance varies whether the COP variation with CR is taken into account and the extent to which it is affected by the choice of the correction function. We compared two different calculation procedures: the “bin method”, implemented according to the Italian standard UNI TS 11300-4 (that is similar to the detailed method described in the standard EN 15316-4-2), and a dynamic method, using the software TRNSYS with the mentioned type variable-speed HPs (Type VRF).

To this purpose we investigated a two-floor wooden building with a very simple geometry (S/V=0.90 m^{-1}) and a well-insulated envelope. The total area of heated thermal zone is 80 m^2 and the heated volume is 275 m^3. Climatic conditions are typical of the mountain area in Northern Italy (outdoor conditions were obtained from the test reference year of Trento, Northern Italy). With the reference design temperature of -12°C, the design load, calculated according to EN 12831 is *4.1 kW*. Nonetheless, the value resulting from the dynamic simulation (or through the Building Energy Signature trend) is quite lower (2.7 kW), as shown in figure 2.

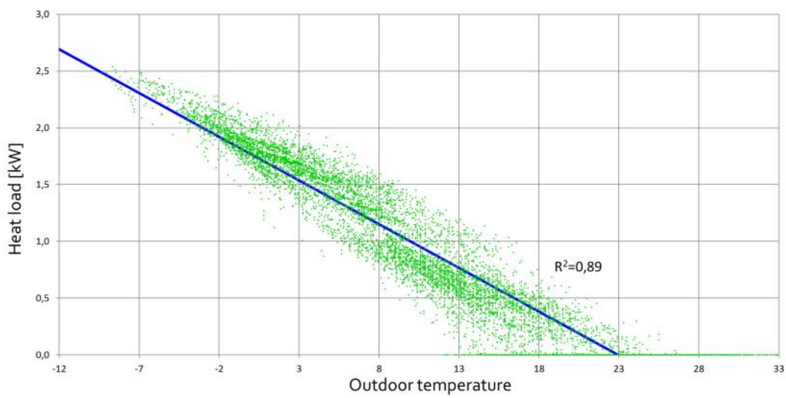

Fig. 2 Heat power required by the building as a function of the external air temperature. The blue line is the linear regression (i.e. Building Energy Signature).

A first run of simulations were carried out with a simplified configuration, in order to highlight the discrepancies and to appreciate the differences between the bin method and the dynamic simulation. The model did not include any thermal capacity (e.g. storage tank and/or envelope mass and the building load has been assumed to vary according to the Energy Signature curve of the building (shown in blue in figure 2, consequently the test reference year (TRY) of the location has been used as input.

According to the "bin method" procedure described in the UNI TS 11300-4, starting from the monthly average temperatures (obtained from the same TRY data in order to be able to compare the results) and the standard deviation, a temperature distribution for every month was obtained and then split in bins having a constant temperature. The standard requires to use bin intervals of 1 K and a monthly normal distribution, whose standard deviation can be estimated with formula G.2 in Appendix G of UNI TS 11300-4. This procedure, suggested by Italian standards, is typically used by designers.

Hence, the heating demand has been distributed among the bins and the overall performance has been calculated with the (2):

$$SCOP = \frac{\sum_{j=1}^{N} h_j (P_h(T_j) - elbu(T_j))}{\sum_{j=1}^{N} h_j \frac{P_h(T_j) - elbu(T_j)}{COP_{PL}(T_j)}} \tag{2}$$

where h_j, $P_h(T_j)$, $elbu(T_j)$ and $COP_{PL}(T_j)$ are, respectively, the number of bin hours occurring at the corresponding temperature T_j, the heating demand of the building, the required capacity of an electric backup heater and the *COP* values of the unit, at part or full load. The (2) is provided in the EN 14825 ($SCOP_{net}$): it does not consider the power of a supplementary electric backup heater, in order to determine only the HP performance.

3. Results

3.1 Influence of Degradation Function on Seasonal Performance

As mentioned before, the described procedures to evaluate *SCOP* have been applied to the three different cases (a, b and c) explained in paragraph 2.1. In Fig. 3 we compared *SCOP* in these three conditions, using both the defined approaches and calculated for two different HP nominal capacities (corresponding to 80% and 60% of the design load).

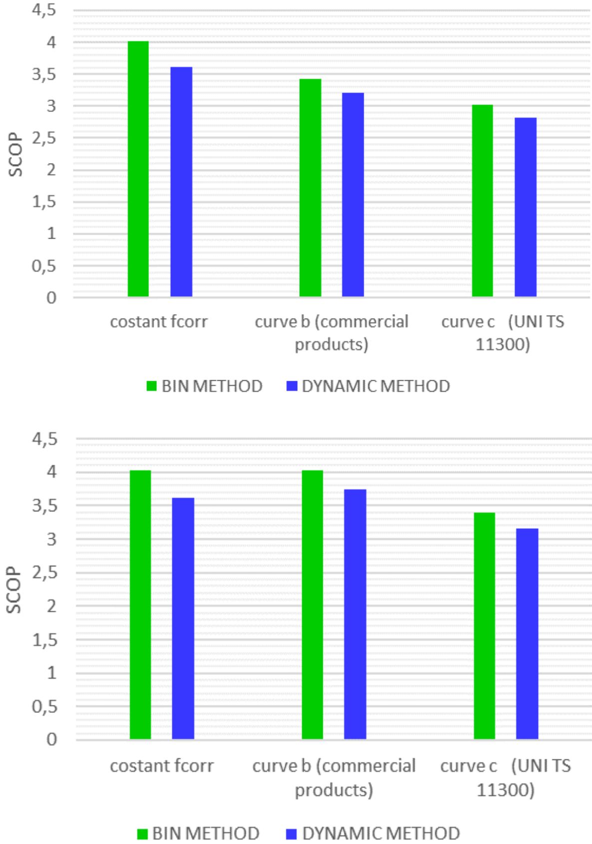

Fig. 3 Seasonal performance of two different HP sizes (i. e. thermal power): 80% of design load (above) and 60% of the design load (below). Calculation with bin method and dynamic method.

The graph shows that, considering a degradation function, the resulting *SCOP* is lower. The value, as expected, depends on the chosen function: this curve should be given by manufacturers, but if not available, the Italian standard allow to use equation (1). This curve was derived based on the performance of HP available on the market some years ago but is probably a bit conservative considering the last generation of heat pumps available on the market. Obviously, if COP degradation is not considered, the performance does not depend on the capacity ratio: as shown by the graph the *SCOP* is the same for the two different sizes considered, whether calculating it by means of the dynamic simulation or with bin method but the bin method seems to be a little more optimistic.

3.2 Influence of HP Size on Seasonal Performance

In the previous paragraph, we mentioned the effects of the size of HP on their performance. Introducing other elements into the model, it is possible to carry out more realistic dynamic simulations. In the next evaluations the model has been integrated with a small water storage tank (0.5 m^3), a climatic control on the supply temperature of the HP, and a building heating load evaluated considering the thermal capacity of the structures (but no internal or solar heat gains).

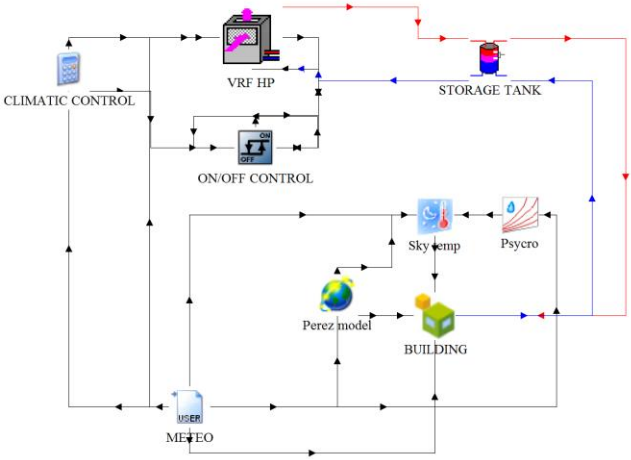

Fig. 4 Building and heating system model (TRNSYS)

Since the size (rated thermal capacity) seems to be quite important in order to achieve higher seasonal coefficients of performance, we run some simulations where the HP capacity is the only variable parameter. The COP part load correction factor was evaluated according to the aforementioned curve b) (last generation VRF HP). The calculated *COP* values, as a function of dry bulb outdoor temperature, are shown in figure 4, for different HP capacities (corresponding to 80%, 60% and 40% of the design load for the considered building). In these calculations, the *COP* (and the *SCOP*) does not take into account the backup heating demand and, consequently, these data should be considered together with the energy needed from the backup heating system (Q_{BU}) in order to supply the total heating demand (Q_{tot}).

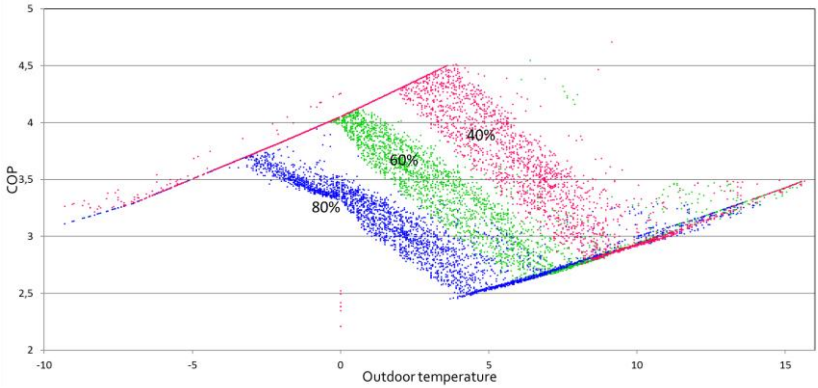

Fig. 5 *COP* for different heat pump size (percentage of rated power over design heat load) depending on outdoor temperature.

Despite the improved part load performance of the last generation of HP, larger capacity units (compared to the design load) still suffer a *COP* degradation for a greater range of temperatures (thus, for a longer period of time during the operating season). The capacity of these devices cannot be selected based on the absolute maximum heating load evaluated according to EN 12831, since, in that case, the HP will be grossly oversized and will work most of the time with a suboptimal capacity ratio and, consequently, with a low performance.

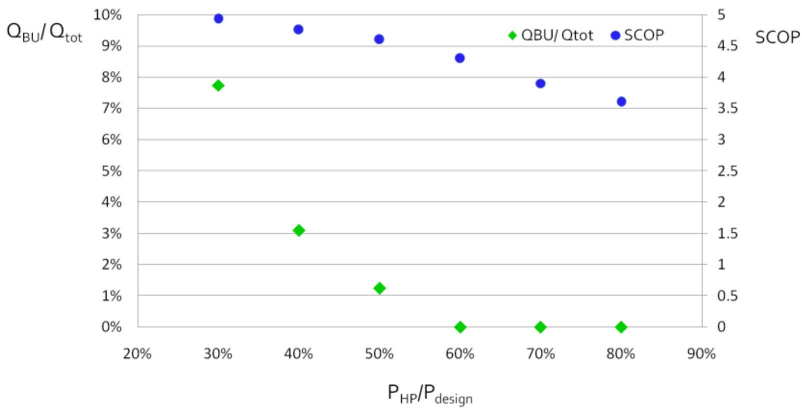

Fig. 6 SCOP and backup heat percentage for different HP sizes.

The *SCOP* values for different HP capacities are shown in figure 6, combined with the corresponding ratio between the heating energy supplied by the backup generator(Q_{BU}) vs the total heating demand (Q_{tot}).

The rated thermal capacities of different HP sizes considered have been shown as a percentage of the design load calculated with EN 12831 (4.1 kW). It is clear that, since the design data suggested by EN 12831 considers a peak heating load substantially larger than the one found in the typical test reference year (TRY), the resulting *CR* is very low during most of the operating season. If the HP capacity is evaluated as a percentage of the maximum thermal load resulting from the TRY used for the simulation (2.7kW), the ratios are quite higher. For example, a capacity P_{HP} equal to 60% of the design load P_{design} (according to EN 12831) rises to 91% (based on TRY data). With such capacity, the heating demand is still completely covered. With a capacity corresponding to 50% of P_{design} (i.e. 76% based on TRY data), a small contribution (less than 2%) is required from the backup heating system.

However, we want to remark that calculations in this paper have some limits. First, we have considered a test reference year for a single location whose minimum temperature is higher the -10°C, so the design temperature (-12°C) has never been reached in these simulations. Second, at the current development stage of the described variable-speed HP model, the code does not fully model power losses due to defrosting cycles, that would furtherly decrease the overall performance.

4. Conclusions

In the first part of the paper we compared the results obtained by means of the bin method suggested by UNI/TS 11300-4 and the dynamic simulation of a heating system based on an air-source heat pump, in terms of seasonal coefficients of performance (*SCOP*). As already deduced by other authors [2] the bin method is able to give numerical results in good agreement with dynamic hourly simulation, as long as the dynamic model does not consider the benefits arising from inertial elements, such as thermal water storage systems and a massive envelope and from a climatic control of the system. When such effects are considered the overall performance improves: this condition can only be simulated with a dynamical model.

The second issue that should be highlighted is that the appropriate sizing of the HP is still a key issue in order to achieve the most efficient operating conditions not only for the on/off units (as underlined by other authors [2]), but also for inverter driven VRF HPs. The best approach seems to be to undersize the heat pump capacity, carefully optimizing through dynamic simulation the best combination of an HP and an auxiliary heating source (that presumably will operate for a limited time).

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