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Summer performances of reversible air-to-water heat pumps with heat recovery for domestic hot water production

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Abstract

A numerical model for the seasonal performance evaluation of electric air-to-water reversible heat pumps during summer is presented. The model employs the bin-method, as indicated by the standards EN 14825 and UNI/TS 11300-4, but also considers domestic hot water (DHW) production through condensation heat recovery. The model evaluates the heat pump Seasonal Energy Efficiency Ratio (*SEER*) as function of the heat pump typology (multi-compressor, inverter-driven). The energy saving potential of DHW production integrated with the heat pump cooling function with respect to traditional separate cooling and DHW devices is analyzed as function of the building demand and of the heat pump typology.

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Keywords: reversible heat pumps; seasonal performances; bin-method; *SEER*; heat recovery

1. Introduction

In warm climates the diffusion of heat recovery chillers able to couple air-conditioning and Domestic Hot Water (DHW) production has been strongly increasing in this last period and many studies have been addressed to the evaluation of the seasonal performance indexes of these systems (e.g. [1,2]). Following the indications of the European standard EN 14825 [3] and of the Italian standard UNI/TS 11300-4 [4], the authors have recently presented a numerical model, based on the bin-method, to calculate the seasonal efficiency of air-to-water heat pumps in heating mode ([5,6]). In this paper this numerical model is extended to reversible air-to-water heat pumps operating during summer by taking into account the production of domestic hot water (DHW) obtainable by using

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condensation heat recovery. The model is applied to two different typologies of reversible heat pumps (multi-compressor and inverter-driven) by considering the presence of a back-up system for DHW production. Once set the building demand for cooling and DHW, the model is able to evaluate: the Seasonal Energy Efficiency Ratio (*SEER*), the cooling energy delivered by the reversible heat pump, the thermal energy recovered at the condenser for DHW production, the energy delivered by the back-up system for DHW (if any), the heat pump and back-up primary energy consumption. The numerical results obtained by applying the proposed model to a test case are shown as a function of the building demand for cooling and DHW. In order to show the energy saving potential of these systems, their performances are compared with those obtainable by using a traditional gas boiler for DHW production combined with a traditional electric chiller.

2. Mathematical model

2.1. Bin distribution and building energy need

The seasonal energy performance of an air-source heat pump is strongly influenced by the temperature variation of the air source, which can change the heat transferred at the evaporator during the winter season and at the condenser during summer. The European standard EN 14825 and the Italian standard UNI/TS 11300-4 suggest to take into account the effect of the climate features of a specific site on the seasonal performance of an air-source heat pump by using the bin-method. The green histogram in Fig. 1 represents the bin profile obtained for Palermo (Italy, 38° 06′ N, 13° 21′ E), considering a cooling season from May 15th to September 15th, by applying the method proposed by the UNI/TS 11300-4. It can be noticed from the chart that the outdoor temperature (T_{ext}) in Palermo during summer runs from a minimum temperature ($T_{ext,min}$) of 9°C to a maximum one ($T_{ext,max}$) of 35°C, with a mode of the distribution equal to 23°C.

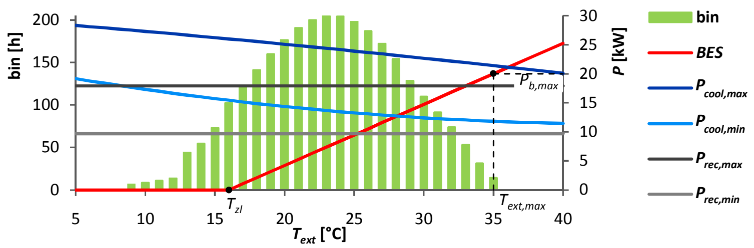

Fig. 1. Bin distribution (Palermo), Building Energy Signature and heat pump power in cooling and heat recovery mode.

For the characterization of the building cooling loads the summer Building Energy Signature (*BES*) has been used, which gives the cooling power required by the building as a function of a modified outdoor temperature, which takes into account also the building loads due to solar radiation. Typically, a straight *BES* line (see the red line in Fig. 1) can be univocally identified by the value of the modified outdoor temperature in correspondence of which the building load becomes zero (T_{zl}) and by the value of the cooling power ($P_{b,max}$) required by the building in correspondence of the summer design outdoor temperature ($T_{ext,max}$). The cooling energy required by the building ($E_{b,cool}$) in correspondence of each *i*-th bin can consequently be obtained through Eq. (1):

$$E_{b,cool}(i) = P_{b,\max}\left[\frac{T_{zl} - T_{ext}(i)}{T_{zl} - T_{ext,\max}}\right] t_{bin}(i) \tag{1}$$

where $t_{bin}(i)$ is the *i*-th bin duration. Once set the daily energy need for DHW, the energy required in each bin ($E_{b,DHW}$) is obtained distributing the daily need according to the *i*-th bin duration.

2.2. Heat pump characterization

In this work two different kinds of heat recovery chillers will be considered: Multi-Compressor (MCHPs) and Inverter-Driven Heat Pumps (IDHPs). The main difference of these heat pumps with respect to the classic on-off chiller is that ON-OFF HPs must activate on-off cycles if the building demand is lower than the heat pump capacity, with consequent efficiency losses; on the contrary, MCHPs and IDHPs are able to reduce the number of compressors activated, or the inverter frequency, in order to follow the building load, thus delaying the on-off cycles activation. In these cases, the heat pump can be characterized by a family of curves equal to the number of compressors (MCHPs), or which represent the different heat pump performances when the inverter frequency is varied between its maximum and minimum value (IDHPs). Blue lines in Fig. 1 are examples of MCHPs and IDHPs characteristic curves at the maximum capacity (all the compressors activated, or maximum inverter frequency) and minimum capacity (only one compressor activated, or minimum inverter frequency). These curves can be obtained by interpolation of the manufacturer technical data and starting from these curves it is possible to know the value of the cooling power delivered by the chiller (P_{cool}) and the corresponding value of the Energy Efficiency Ratio (EER_{cool}) as a function of the *i*-th bin. It is important to observe that the heat recovery chiller operates like an air-to-water heat pump only when DHW production is absent. On the contrary, during the heat recovery mode, the heat pump does not release the condensation heat to the outdoor air, but to a storage tank for DHW production, working as a water-to-water heat pump. As a consequence, in this mode the heat pump cooling power (P_{rec}) and *EER* (EER_{rec}) depend only on the temperatures of the cold water ($T_{w,cool}$) and hot water ($T_{w,DHW}$) and they are not influenced by the bin considered (see the grey lines in Fig. 1).

2.3. Seasonal performance evaluation

In the generic *i*-th bin the building can require at the same time cooling energy for air-conditioning ($E_{b,cool}$) and thermal energy for DHW production ($E_{b,DHW}$). By considering the case of a MCHP with two compressors, the virtual activation time of the heat pump with both the compressors on in heat recovery mode ($t_{2/2,rec,virt}$) is equal to the bin duration, t_{bin}, if $E_{b,cool}$ is higher than the product of the maximum heat pump capacity ($P_{2/2,rec}$) and t_{bin}; the corresponding activation time of only one compressor on, $t_{1/2,rec,virt}$, is equal to zero. If $E_{b,cool}$ is lower than the product of the minimum heat pump capacity ($P_{1/2,rec}$) and t_{bin}, $t_{2/2,rec,virt}$ is equal to zero, while $t_{1/2,rec,virt}$ is equal to the ratio between $E_{b,cool}$ and $P_{1/2,rec}$. Else, if $E_{b,cool}$ is intermediate between the energy delivered with one and two compressors on, the activation times are calculated as weighted averages on the compressors energy. The thermal energy available at the heat pump condenser for DHW production, $E_{avail,cond}$, is thus calculated through Eq. (2):

$$E_{avail,cond}(i) = t_{2/2,rec,virt}\left[P_{2/2,rec}(i) + \frac{P_{2/2,rec}(i)}{EER_{2/2,rec}(i)}\right] + t_{1/2,rec,virt}\left[P_{1/2,rec}(i) + \frac{P_{1/2,rec}(i)}{EER_{1/2,rec}(i)}\right] \tag{2}$$

If $E_{b,DHW}$ in the *i*-th bin is equal to – or higher than – $E_{avail,cond}$, the effective activation times of the heat pump with two ($t_{2/2,rec,eff}$) and one ($t_{1/2,rec,eff}$) compressors activated, in heat recovery mode, are equal to the corresponding virtual durations. Otherwise, the heat pump provides cooling energy to the building in heat recovery mode only until the DHW thermal demand is satisfied. The cooling energy $E_{HP,rec}$, supplied by the heat pump in heat recovery mode in the *i*-th bin, is given by Eq. (3).

$$E_{HP,rec}(i) = P_{2/2,rec}(i)\; t_{2/2,rec,eff} + P_{1/2,rec}(i)\; t_{1/2,rec,eff} \tag{3}$$

If $E_{HP,rec}$ is lower than $E_{b,cool}$, in the residual time (t_{res}) of the bin the heat pump delivers cooling energy releasing the condenser heat to the external air. t_{res} is evaluated according to Eq. (4), which takes into account also the time lost in case of heat pump on-off cycles.

$$t_{res}(i) = \frac{t_{bin}(i)}{t_{2/2,rec,virt} + t_{1/2,rec,virt}}\left[\left(t_{2/2,rec,virt} + t_{1/2,rec,virt}\right) - \left(t_{2/2,rec,eff} + t_{1/2,rec,eff}\right)\right] \tag{4}$$

The activation times of the heat pump compressors in cooling mode, $t_{2/2,cool}$ and $t_{1/2,cool}$, are obviously equal to 0 if $E_{b,DHW}$ is higher than $E_{avail,cond}$. If the residual cooling energy required by the building (difference between $E_{b,cool}$ and $E_{HP,rec}$) is higher than the energy the heat pump would deliver with both the compressors on, $t_{2/2,cool}$ coincides with t_{res} and $t_{1/2,cool}$ is equal to zero. If this residual energy is lower than the energy the heat pump would deliver with only one compressor on, $t_{2/2,cool}$ is equal to zero and $t_{1/2,cool}$ is given by the ratio between the residual energy and the heat pump capacity with one compressor on. Else, if the residual energy is intermediate between the energy of the two compressors, the activation times in cooling mode are calculated as weighted averages on the compressors energy.

In the *i*-th bin, the energy delivered to the building for air-conditioning ($E_{HP,rec,cool}$) and the thermal energy recovered at the condenser for DHW ($E_{HP,DHW}$) are given by Eq. (5) and Eq. (6), respectively. The corresponding electric energy $E_{HP,us}$ used by the heat pump is calculated by Eq. (7), where $f_{EER,rec}$ and $f_{EER,cool}$ are the *EER* correction factors for on-off cycles, evaluated through Eq. (8) ([3,4]).

$$E_{HP,rec,cool}(i) = E_{HP,rec}(i) + P_{2/2,cool}(i)\ t_{2/2,cool}(i) + P_{1/2,cool}(i)\ t_{1/2,cool}(i) \tag{5}$$

$$E_{HP,DHW}(i) = \min\left[E_{avail,cond}(i); E_{b,DHW}(i)\right] \tag{6}$$

$$E_{HP,us}(i) = \left[\frac{P_{2/2,rec}(i)}{EER_{2/2,rec}(i)} t_{2/2,rec,eff}(i) + \frac{P_{1/2,rec}(i)}{EER_{1/2,rec}(i)\, f_{EER,rec}(i)} t_{1/2,rec,eff}(i)\right] + \left[\frac{P_{2/2,cool}(i)}{EER_{2/2,cool}(i)} t_{2/2,cool}(i) + \frac{P_{1/2,cool}(i)}{EER_{1/2,cool}(i)\, f_{EER,cool}(i)} t_{1/2,cool}(i)\right] \tag{7}$$

$$f_{EER}(i) = \frac{CR(i)}{1 - C_c + C_c\ CR(i)} \tag{8}$$

In Eq. (8) C_c is the degradation coefficient, set by the standards [3,4] equal to 0.9 in absence of manufacturer specific instructions, and CR is the capacity ratio (the ratio between the building power required and the heat pump capacity with only one compressor on, for MCHPs, or at the minimum frequency, for IDHPs). *CR* can range from 0 to 1.

If in the *i*-th bin $E_{b,DHW}$ is higher than $E_{HP,DHW}$, the back-up system for DHW must be activated in order to supply the missing thermal energy, E_{bk}, with consequent energy consumption, $E_{bk,us}$.

A similar procedure can be adopted in order to model an IDHP.

The seasonal performance coefficients of the system are finally evaluated: the Seasonal Energy Efficiency Ratio, *SEER*, is defined as the ratio between the total cooling energy provided and the corresponding electric energy used by the heat pump during summer ([3]); the Fuel Utilization Efficiency, *FUE*, is defined as the ratio between the total energy delivered to the building by the heat pump and back-up system (for air-conditioning and DHW production) and the corresponding primary energy used. *SEER* and *FUE* can be calculated as follows:

$$SEER = \frac{\sum_i E_{HP,rec,cool}(i)}{\sum_i E_{HP,us}(i)}\ ; \qquad FUE = \frac{\sum_i \left[E_{HP,rec,cool}(i) + E_{HP,DHW}(i) + E_{bk}(i)\right]}{\sum_i \left[\frac{E_{HP,us}(i)}{f_{en}} + E_{bk,us,prim}(i)\right]} \tag{9}$$

f_{en} in Eq. (9) is the thermodynamic efficiency of the electricity system of the country, equal to 0.46 for Italy.

3. Case study

3.1. Heat pumps technical characteristics

The mathematical model presented is here applied to evaluate the seasonal performances of two commercial air-to-water reversible heat pumps, with similar full-load capacity (one MCHP with two compressors and one IDHP), integrated by electric heaters as back-up system for DHW and placed at the service of several buildings in Palermo (Southern Italy), considering a cooling season from May 15th to September 15th. Table 1 shows the heat pumps technical data declared by the manufacturer, for the maximum and minimum capacity, in cooling mode ($T_{w,cool}$=7°C) and in heat recovery mode for DHW production ($T_{w,DHW}$=55°C).

Table 1. Heat pumps technical data at maximum and minimum capacity ($T_{w,cool}$=7°C).

	MCHP power [kW] and (*EER*)		**IDHP power [kW] and (*EER*)**	
T_{ext} [°C]	2 compressors on	1 compressor on	Maximum frequency	Minimum frequency
20	25.10 (4.49)	14.40 (4.81)	26.80 (4.41)	10.3 (4.88)
25	23.90 (4.06)	13.20 (4.19)	25.80 (3.88)	9.89 (4.31)
30	22.70 (3.55)	12.50 (3.69)	24.60 (3.38)	9.49 (3.83)
35	21.40 (3.04)	11.80 (3.17)	23.30 (2.93)	9.04 (3.37)
$T_{w,DHW}$ = 55°C	17.90 (2.10)	9.70 (2.37)	20.40 (2.22)	7.51 (2.60)

The effect of the building cooling load on the seasonal efficiency is analyzed taking into account different Building Energy Signatures, in which T_{zl} is set equal to 16°C ([3]) and $P_{b,max}$ in correspondence of $T_{ext,max}$ (35°C) is varied. As no back-up system for air-conditioning is present, the choice of the building-heat pump combinations has been made in order to have the building cooling demand fully covered at the highest outdoor temperature.

Different building loads for DHW production are also considered, by varying the ratio between the building total DHW demand, $E_{b,DHW,tot}$, and total cooling demand, $E_{b,cool,tot}$.

3.2. Results

Fig. 2. (a) shows *SEER* as a function of the ratio between $E_{b,DHW,tot}$ and $E_{b,cool,tot}$, obtained with the selected heat pumps with several buildings cooling loads. The obtained *SEER* ranges from 2.80 (MCHP, $P_{b,max}$= 10 kW, $E_{b,DHW,tot}$=50% $E_{b,cool,tot}$) to 3.96 (IDHP, $P_{b,max}$=20 kW, $E_{b,DHW,tot}$=5% $E_{b,cool,tot}$) and it decreases with the increase of the building DHW demand, because of the increase of time in heat recovery mode, where the heat pump releases the condenser heat at higher temperature ($T_{w,DHW}$=55°C versus $T_{ext,max}$=35°C). In addition, for a selected heat pump and fraction of DHW building demand, worse seasonal performances are obtained with lower $P_{b,max}$: in this case the heat pump is oversized with respect to the building cooling demand and this fact increases the heat pump on-off cycles.

On the same conditions, the IDHP (blue curves in Fig. 2. (a)) reaches better *SEER* with respect to the MCHP (red curves in Fig. 2. (a)), thanks to higher *EER* values at partial load and to a lower number of on-off cycles (the IDHP is able to reach a minimum capacity lower than that of the MCHP). This result is confirmed by Fig. 2 (b), where the correction factors f_{EER} are reported as functions of the outdoor temperature for $P_{b,max}$=20 kW and $E_{b,DHW,tot}/E_{b,cool,tot}$=15%. f_{EER} is equal to 1 (no on-off cycles) for large T_{ext}, while it decreases when T_{ext} decreases, since the number of on-off cycles increases, due to the decrease of both the building cooling demand (low *BES* values) and DHW demand (low number of bin hours). The results shown in Fig. 2.(b) highlight that the IDHP is characterized by f_{EER} higher than those of the MCHP, both in cooling mode and in heat recovery mode.

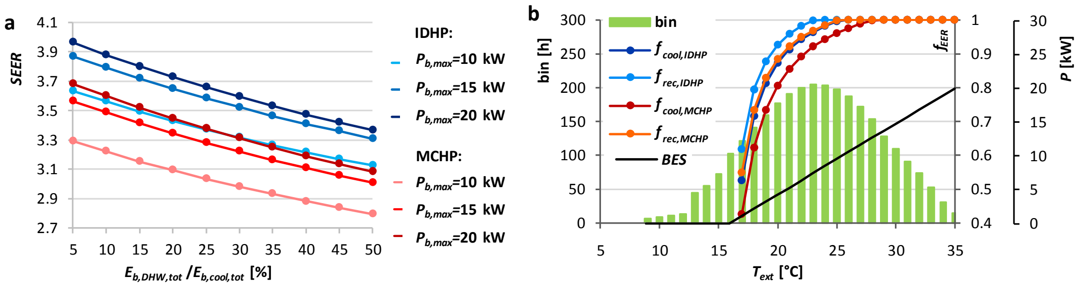

Fig. 2. (a) *SEER* with different building loads; (b) f_{EER} in cooling and heat recovery mode ($P_{b,max}$=20 kW; $E_{b,DHW,tot}/E_{b,cool,tot}$=15%), *BES*, bin trend.

Fig. 3. (a) shows the trend of *FUE* as function of the building cooling and DHW demand, with the selected IDHP integrated by electric back-up and with a traditional system, in which the reversible heat pump only provides to air-conditioning and DHW is entirely produced by a gas boiler. The comparison has been made using for the traditional system the same IDHP (but without heat recovery) and a gas boiler with an efficiency of 0.98. On the same conditions, *FUE* of the studied system is higher than that of the traditional system and this gap is enhanced for high DHW demand (2.03 vs 1.45 for $P_{b,max}$=20 kW, $E_{b,DHW,tot}$=50% $E_{b,cool,tot}$). In fact, even if the primary energy used for cooling in the traditional system is lower (heat released to the outdoor air, better *EER*), the primary energy used by the gas boiler (which provides to all DHW) is higher than that used by the electric back-up (that only provides the energy for DHW which the chiller cannot satisfy in heat recovery mode). Fig. 3 (b) plots the primary energy saving of the studied system with respect to the traditional solution. The energy saving is higher for higher DHW fractions and with the IDHP than with the MCHP, and can reach 31% (IDHP, $P_{b,max}$=20 kW, $E_{b,DHW,tot}$=50% $E_{b,cool,tot}$).

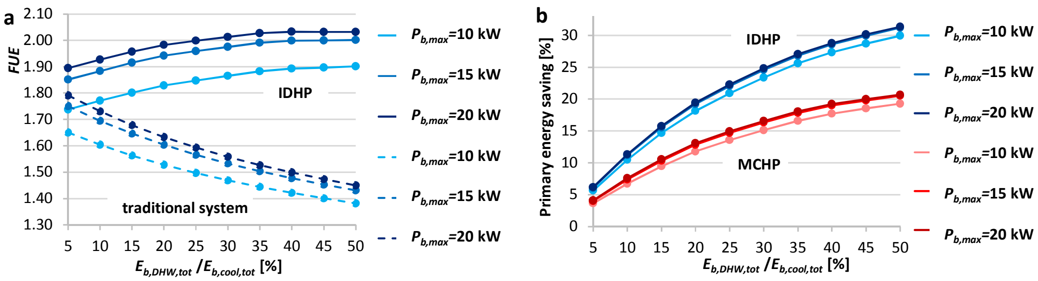

Fig. 3. (a) *FUE* of IDHP and traditional system; (b) primary energy saving with respect to traditional system.

4. Conclusions

A numerical model is presented, to evaluate through the bin-method the seasonal performances of air-source reversible heat pumps, able to produce domestic hot water (DHW) through condensation heat recovery. The model is used to evaluate the seasonal performances of a multi-compressor (MCHP) and an inverter-driven (IDHP) heat pump. The numerical results show that better *SEER* are obtained with the IDHP for high building cooling demand and low DHW demand, while the Fuel Utilization Efficiency increases with the DHW need. The primary energy consumption of the system is lower than that of traditional cooling and DHW devices (even more than 30%).

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