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Evaluation of Dynamic Operation Effects for a Heat Pump in a Solar Combi-Plus System

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Summary

Real-like –dynamic- working conditions shall be taken into account in the assessment of the seasonal performance of thermal system, particularly when these are driven by renewable energy sources. A vapour compression reversible heat pump installed in a solar combi-plus system was studied to verify the influence of variable sources and loads on the overall performance. The machine was tested in laboratory, following a dynamic sequence, which reproduces the seasonal real-like boundary conditions. The results of the dynamic test highlight the non-negligible influence of dynamic working conditions on the seasonal performance of the heat pump. Two unfavourable factors were identified, i.e. the presence of many on-off cycles during cooling mode and fast variations of the boundary condition – i.e. changing of set points in switching the operation from space heating to domestic hot water preparation. Given the importance of correctly evaluating these aspects, a short dynamic test procedure was developed and validated. This allows the evaluation of the seasonal performance of the heat pump, as far as the frequency distribution of the machine performance.

Key words: dynamic test, compression heat pump, seasonal performance, solar combi-plus systems

1. Introduction

In the last decades, heat pumps have become a mature technology and have experienced a large diffusion. Different commercial solutions are available, which integrate heat pumps with solar thermal systems to produce domestic hot water and space heating (in addition to space cooling for reversible components) (Chua et al., 2010), (Lloyd and Kerr, 2008), (Panaras et al., 2014).

The performance of heat pumps, as given by the manufacturers, is assessed following procedures described in international standards, by means of laboratory tests under steady state conditions. However, the actual operating conditions under which heat pumps operate are usually quite variable in time, especially if renewable energy sources are also employed. As a consequence, the performance obtained in real installations can considerably differ from that rated in steady state conditions.

New approaches have been proposed in the open literature, which foresee the dynamic characterization of heat pumps performance through dynamic numerical models or dynamic test methods. Numerical dynamic models validated with monitoring data could be find in (Madonna and Bazzocchi 2013), while dynamic test procedures were used to study the combination of heat pumps with combi-stores by (Haberl et al. 2014) and (Panaras et al., 2014). Dynamic laboratory tests allows the characterization of the component but also the optimization of the control strategies and the evaluation of the component limitations. However, the definition of standardized procedures to perform dynamic tests is still lacking, especially for what concerns the selection of the boundary conditions to be tested and the assessment of the seasonal performance starting from the test results.

This paper presents the results of a dynamic tests campaign run over the entire heating and cooling season for a commercial reversible compression heat pump installed in a solar hybrid system. The goal of the test campaign was the evaluation of the effects of dynamic working conditions, on-off transients and variable loads on the performance of the heat pump. The tests showed that, as expected, that dynamic working conditions

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strongly affect the performance of the heat pump, and should therefore be properly evaluated. To this end, a short dynamic test procedure was defined, which allows the characterization of the seasonal performance of the machine without a whole season test, starting from a statistical selection of the boundary conditions.

2. Test Method

The tests were performed by reproducing the real-like seasonal boundary conditions of the heat pump in the laboratory. As a first step, the system in which the heat pump operates was simulated, including all components and the control logics. The system details and simulation environment are described in (D'Antoni et al. 2011), while the description of the heat pump' operative condition can be seen in section 3. The performance here reported refers to a system located in Bolzano, Italy.

From the results of the system simulation, the instantaneous boundary conditions are extrapolated for the heat pump, as time series. These include the inlet mass-flows and temperatures of evaporator and condenser of the heat pump. Starting from the ON-OFF cycles, a continuous period of ON is identified as an *Event*. The test boundary conditions are then created as a sequence of *Events*, each of which includes the corresponding profiles of temperatures and mass flows. Two consecutive *Events* are spaced out by OFF conditions, in order to also account for the transient phases.

The results presented in this paper are obtained first by testing the series of boundary conditions covering the entire yearly operation, in order to perform a monthly and seasonal performance analysis (around 50 days of continuous testing 24/7). The results are then compared with those obtained out of a reduced set of *Events* selected, which test requires only a few days.

During the tests, the electric consumption, the inlet and outlet temperatures and pressures and the volumetric flows of the two heat pump circuits are measured every 5 seconds. The thermal powers are calculated from the measurements and are used to retrieve the instantaneous COP and EER:

$$COP = \frac{\dot{Q}_{cond}}{\dot{W}_{hp}}$$

(eq. 1)

$$EER = \frac{\dot{Q}_{evap}}{\dot{W}_{hp}}$$

(eq. 2)

where $\dot{Q}_{cond}$, $\dot{Q}_{evap}$ and $\dot{W}_{hp}$ are respectively the condenser, evaporator and electrical powers. By integration of the powers, the SCOP (or SEER) is calculated on monthly and seasonal bases:

$$SCOP_{ON} = \frac{\sum_{i=1}^{n} (\dot{Q}_{cond,i} \cdot \tau_i)}{\sum_{i=1}^{n} (\dot{W}_{el,i} \cdot \tau_i)}$$

(eq. 3)

$$SEER_{ON} = \frac{\sum_{i=1}^{n} (\dot{Q}_{evap,i} \cdot \tau_i)}{\sum_{i=1}^{n} (\dot{W}_{el,i} \cdot \tau_i)}$$

(eq. 4)

where the index *i* and *n* are referred to respective integration period (monthly or seasonal). τ_i is the duration of the i-th condition.

3. Influence of the control strategy on the boundary conditions

The tested heat pump (CLIVET WSHN EE 31) is part of a hybrid solar system and is used in winter for the preparation of domestic hot water (feeding a hot water storage) and for space heating (with a radiant floor distribution), and in summer for space cooling only (again with a radiant system distribution). The water-to-water heat pump is connected to a dry cooler (air source) and to a solar field (solar source) having the possibility of use one of those two sources. To manage these operating modes, different control schemes for the heat pump have been identified:

- SC: space cooling;
- DHW solar: preparation of domestic hot water with solar source;
- DHW air: preparation of domestic hot water with air source;
- SH solar: space heating with solar source;
- SH air: space heating with air source;

The different combinations of schemes throughout the season result in varying boundary conditions for the heat pump, influencing thus the overall performance of the machine.

3.1 Heating Mode

Heat pumps perform better with high evaporation temperatures and low condensation temperatures. When the air source is used, the inlet evaporator temperature is constrained by the external temperature; as a consequence, air source heating schemes result in a better performance during the mid-season months and in worse COPs during the colder months.

With respect to the user side, since in winter the heat pump is used both for space heating and for DHW preparation, two different temperature levels are foreseen. In particular, since the set point for DHW preparation is higher, the performance of the heat pump in the DHW schemes is worse than during space heating schemes.

Table 1 shows the number of *Events*, the number of schemes' activation and their total duration in the whole winter season. The control strategy implemented in the system allows changing from one scheme to another (e.g. from space heating to DHW preparation and back to space heating), without requiring the heat pump to be turned OFF. This means that during one *Event* none, one or more changes of schemes could be done. As a consequence, the number of heat pump activations is independent of the sum of schemes activations. For example, considering the whole season (first row of Tab. 1), the heat pump is activated 554 times while the DHW_schemes are activated 253 times (6+247) and the space heating schemes are activated 558 times (115+443).

Tab. 1: Heating schemes. Activations and durations.

		Scheme Activations [N-times]				**Scheme duration [h]**			
	Nev	DHW_{sol}	DHW_{air}	SH_{sol}	SH_{air}	DHW_{sol}	DHW_{air}	SH_{sol}	SH_{air}
Seasonal	554	6	247	115	443	1	50	90	785
October	4	0	0	1	5	0	0	1	5
November	93	0	32	16	66	0	7	10	129
December	167	1	95	23	141	0.08	17	22	234
January	130	5	67	33	112	1	14	33	239
February	89	0	36	26	73	0	9	17	144
March	68	0	17	16	43	0	3	7	33
April	3	0	0	0	3	0	0	0	1

To understand how the use of the different loads and sources is distributed throughout the season, the data contained in Tab. 1 can be represented as percentage distributions of the number of activations, as showed in Fig. 1a, or as percentage distributions of schemes duration, as showed in Fig. 1b.

From the two figures, it is clear that in October and April the heat pump is not used for the domestic hot water preparation, which is produced directly with the solar energy. The average temperature at the user's side (condenser) is therefore lower than in the other months, and a positive effect on the COP is expected. During the other months, while the percentage of DHW activation schemes ranges between 22% and 37% (the maximum is verified in December), the amount of time in which the heat pump works for the DHW preparation is only 5-6%. This means that DHW schemes are activated frequently and for short periods.

Similar considerations to those done for the loads, can be done also in terms of use of the different sources. The months with the lowest evaporation temperatures are November and December, which also have a low share of solar source use (7% and 8% of the total duration respectively). One particular case is represented by April, where only the air source is employed (but with higher external temperature). The share of solar source for the other months ranges between 10 and 18% of the working time.

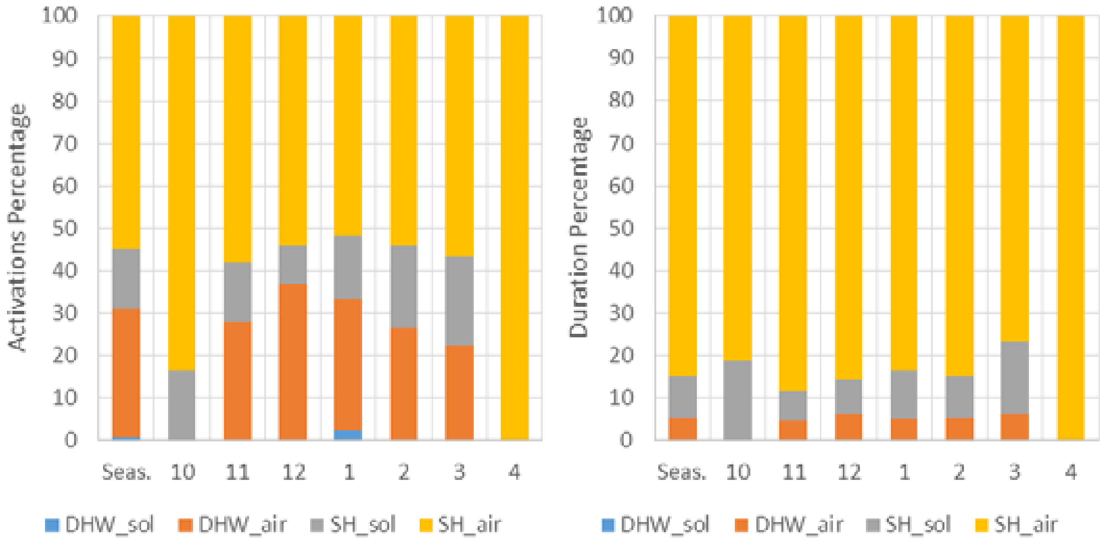

Fig. 1: Heating mode. 1a) Percentage of schemes activation. 1b) Percentage of schemes duration

3.2 Cooling Mode

In summer, the heat pump works from June to September with one single scheme (space cooling). In Tab. 2, the number of activations of the scheme for the entire cooling season and for the single months is reported. With respect to the heating mode, the cooling season is characterized by shorter and more frequent activations of the heat pump, due to the oversizing of the heat pump capacity compared to the building load. Indeed, the cooling scheme is activated 660 times in 4 month for a total of 79 working hours (versus the 554 times in 7 months with 937 working hours for the heating mode). The average *Event* duration is 7 minutes instead of 100 minutes in heating mode.

Tab. 2: Cooling scheme. Activations and durations.

	Nev	Duration τ_{on} [h]
Seasonal	660	79
June	151	16
July	215	26
August	209	29
September	85	8

4. Seasonal Characterization – Results

The characterization of the heat pump performance was separated into the two working modes: heating and cooling.

4.1 Heating Mode

Table 3 presents the seasonal and the monthly results for the heating season in terms of number of events, total duration, average condenser and evaporator temperatures, SCOP, electric energy consumed by the heat pump and exchanged thermal energies at the condenser and the evaporator.

The considerations streamlined with respect to the schemes distribution help understanding the results in Tab. 3. The SCOP was calculate with eq. 3 considering the integration domain on month and seasonal basis. The SCOP varies for the different months between 3.36 and 3.85 while the seasonal value is 3.47. The seasonal value is lower than the mathematical average of the monthly values because the months with a higher SCOP present few working hours. In particular, the months with higher SCOP are, as anticipated, April and October, with one and six working hours respectively. In addition, also March, with a total of 41 working hours, presents a quite high SCOP, because the air source scheme can work with high evaporator temperatures (mild external air temperature). The months with lower SCOP are, as expected, the colder ones, i.e. December, January and February, with respectively 377, 390 and 175 working hours.

Tab. 3: Monthly and seasonal results. Heating Mode.

	N_{ev}	τ_{on} [h]	$\overline{T_{cond}}$ [°C]	$\overline{T_{evap}}$ [°C]	SCOP [-]	W_{hp} [kWh]	Q_{co} [kWh]	Q_{ev} [kWh]
Seasonal	554	937	27.1	1.9	3.47	1873.2	6496.4	4574.2
October	4	6	26.1	4.6	3.85	12.4	47.9	35.1
November	93	147	27.1	1.8	3.59	291.4	1046.5	732.5
December	167	277	27.8	0.4	3.36	558.0	1877.0	1296.1
January	130	290	27.7	-0.2	3.44	580.8	1998.5	1387.6
February	89	175	27.4	0.8	3.49	348.9	1217.9	878.4
March	68	41	25.9	5.6	3.78	80.0	302.2	239.4
April	3	1	25.4	5.6	3.71	1.7	6.4	5.0

Besides the evaluation of the seasonal and monthly SCOPs, dynamic tests also allow a deeper analysis of the behaviour of the heat pump during transients. This can be done by considering a single *Event* as shown in Fig. 2, where an example of temperature and power times-series for winter operation with air source are reported. The *Event* starts with a space heating scheme at minute 4; at minute 17 the scheme switches to DHW preparation until minute 38, when the scheme is switched back to space heating.

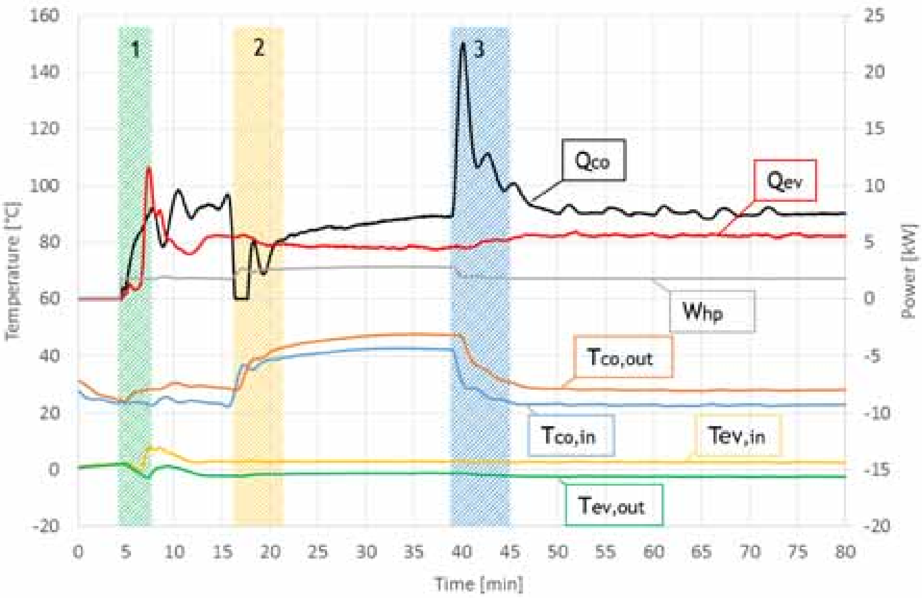

Fig. 2: Temperature and power profile example. Heating Mode.

The electric power consumption (W_{hp}) is stable from the switch ON and throughout the whole *Event*, while the condenser power ($\dot{Q}_{co}$) and, as consequence, the COP vary. In particular, the condenser power presents two transients moving from zero to a stable value. The first corresponds to the switch ON of the heat pump (green rectangle 1 in Fig. 2); the second is localized at the switch from space heating to domestic hot water preparation (yellow rectangle 2 in Fig. 2). In both cases, the outlet condenser temperature ($T_{co,out}$) has to increase and it does so with a certain delay due to the thermal inertia of the machine. Consequently, the instantaneous temperature values are similar (or lower) to the inlet temperature ($T_{co,in}$). This results in a null instantaneous power, which progressively increases towards a stationary value (with a positive temperature difference).

During the switch from domestic hot water to space heating, a third transient phase takes place (blue rectangle 3 in Fig. 2), where the condenser power abruptly increases and then progressively decreases to a new stationary condition. In this case, the inlet condenser temperature decreases, while the outlet temperature follows with some delay caused again by the thermal inertia. The high instantaneous values of the condenser power are a consequence of temperature differences higher than those obtained in stationary operation.

The dynamic behaviour represented in Fig. 2 could be interpreted as a “storage” effect of the heat pump. In the transient phases with increasing temperatures the heat exchanger of the condenser “stores” energy; this is “released” during the transient phases where the temperatures decrease. If an *Event* stops with a domestic hot water scheme, the energy “stored” in the initial transient is lost most of the times.

To understand how these transients affect the seasonal performance of the heat pump, the instantaneous COPs obtained during the dynamic tests for the whole season are reported in Fig. 3, as a function of the condenser temperature. The curves obtained for different evaporator temperatures under steady state conditions are also plotted as a reference. Different working conditions can be identified in the figure:

- Stationary state operation points corresponding to the cloud of points distributed over the stationary curves; the red points in dynamic condition(those at evaporator temperature of 10°C) are obtained for only short time, as consequence the stationary conditions are not reached.
- Initial transient points (indicated with the two green-arrows – 1a/heating and 1b/DHW – in Fig.3). Depending on the scheme with which the heat pump is activated, these points are localized at different condenser temperatures;
- Points corresponding to the switch from space heating to domestic hot water (indicated with the yellow arrow – 2 – in Fig. 3). During these transients the temperature of the condenser is increasing and a “storing” effect occurs;
- Points corresponding to the switch from domestic hot water to space heating (indicated with the blue arrow – 3 – in Fig. 3). During these transients the condenser temperature decreases and an “energy releasing” effect occurs;
- Area without dynamic COP points (indicated by a black ellipse – 4 – in Fig. 3). This zone correspond to the evaporator temperatures between the space heating and the domestic hot water set points (namely 32°C and 40°C for the examined plant). The machine is never working at steady state conditions in this range.

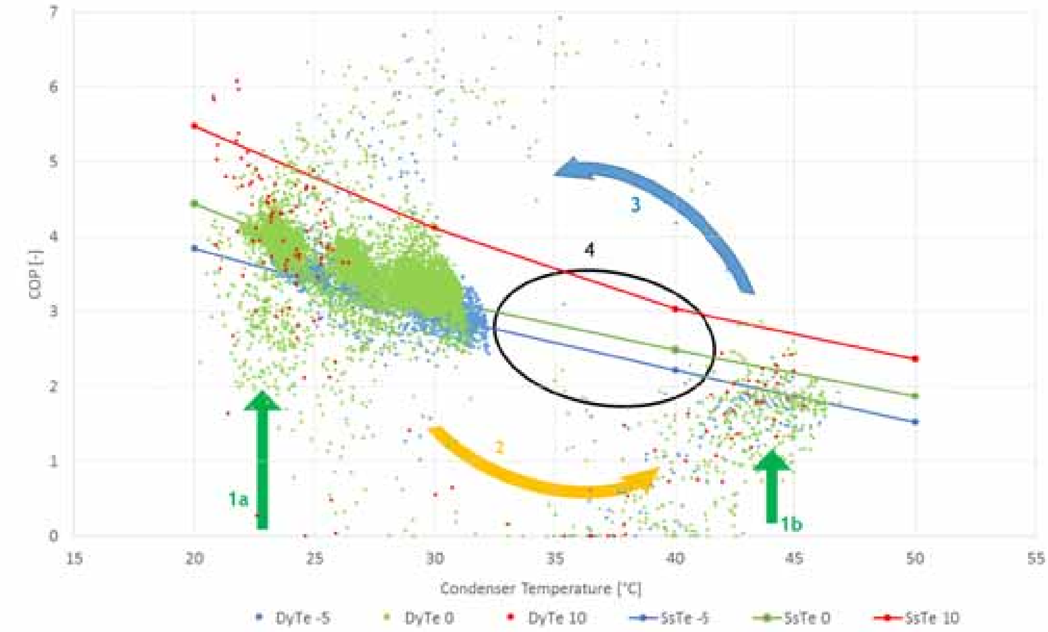

Fig. 3: COP comparison in dynamic conditions and stationary conditions. Heating mode.
The series are divided by evaporator temperature (Te) respectively for the case of dynamic test (Dy) and steady-state test (Ss).
The name of the series indicate the typology of test (dynamic - Dy or stationary Ss) and the evaporator temperature (i.e. Te -5)

4.2 Cooling Mode

Table 4 presents the seasonal and the monthly results for the cooling season, similarly to what already presented for the heating season. The SEER was calculate with eq. 4 considering the integration domain on month and seasonal basis. The monthly SEER is varying from 3.55 to 3.85 while the seasonal value is 3.75. September is the month with the lowest SEER, and it also presents the lowest average duration of *Events* (about 5 minutes). In this case, the presence of initial transients has a stronger effect on the performance.

Tab. 4: Monthly and seasonal results. Cooling Mode.

	Nev	τ_{on} [h]	Tcond [°C]	Tevap [°C]	SEER [-]	W_{hp} [kWh]	Q_{cond} [kWh]	Q_{evap} [kWh]
Seasonal	660	79	28.2	17.0	3.75	163.7	709.1	613.7
June	151	16	27.8	17.1	3.73	32.2	136.6	120.3
July	215	26	28.5	16.9	3.73	55.0	237.6	205.2
August	209	29	28.4	16.9	3.83	60.8	273.0	232.6
September	85	8	27.8	17.1	3.55	15.7	62.0	55.6

A deeper insight on the behaviour of the machine during transients can be obtained by looking at the temperature series recorded during a single *Event* in the cooling season, as reported in Fig. 4. The *Event* starts at minute 1 and ends at minute 19. While the electric consumption (W_{el}) is stable over the whole period, the condenser and evaporator powers present a transient phase of about 3 minutes before reaching the steady state conditions. The temperature difference at the evaporator obtained during the transient is lower than that obtained in steady state and so is the instantaneous EER.

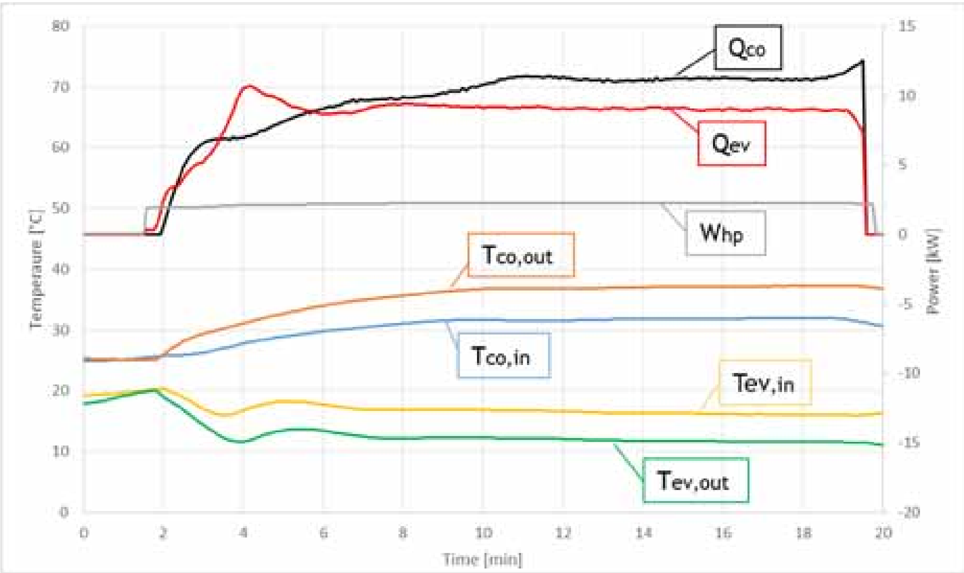

Fig. 4: Temperature profile of one *Event*. Heating mode.

With respect to the winter case, the boundary conditions in cooling mode are less variable (the machine is working with one scheme only) but a larger number of *Events* with short duration is present. For example, the average duration of an *Event* in cooling mode is 7 minutes but many *Events* have a shorter duration. As a consequence, the starting transients have a large impact on the performance and no “storage effect” is occurring. This can be easily observed in Fig. 5, showing the instantaneous EER as a function of the condenser temperature along with the steady state curves obtained for different evaporator temperatures. Two main areas can be identified in Fig. 5. The first one (black rectangle) is located near the steady state curves: it includes the working points obtained after the initial transient phases. The second cloud (indicated with a green arrow) has a larger extension and cover the zone from zero EER to the stationary conditions: these points represent the switch ON transient working conditions.

Fig. 5: EER comparison between dynamic conditions and steady state condition. Cooling mode. The series are divided by evaporator temperature (Te) respectively for the case of dynamic test (Dy) and steady-state test (Ss). The name of the series indicate the typology of test (dynamic - Dy or stationary Ss) and the evaporator temperature (i.e. Te 14)

5. Short Test Sequence

The analysis in the previous chapter highlights how, for a correct evaluation of the seasonal performance of a heat pump, it is necessary to consider its real operation and include also dynamic and transient effects. Of course, the implementation of a seasonal dynamic test is both costly and time consuming, and therefore not an option in a real application. Similar evaluations should be performed with a short test sequence, easily reproducible in a laboratory and, at the same time, capable of capturing all important features of the machine operation. To this end, a short dynamic test procedure has been elaborated and the results obtained for the two operating modes have been compared with the whole season tests.

The short test sequence is defined starting from the *Events* series obtained from the system seasonal simulation. The *Events* are classified into equivalent discrete groups, or *Classes*, i.e. multidimensional vectors defined as a function of the *Event* duration, the average and the amplitude of the machine boundary conditions. For a vapour compression heat pump, the boundary conditions are the evaporator and condenser inlet temperatures and mass flows. For each *Event*, 9 parameters can be identified: duration, evaporator temperature amplitude and average, evaporator mass flow amplitude and average, condenser temperature amplitude and average, condenser mass flow amplitude and average. From the range of variation of these parameters the *Classes* are created identifying intervals of 3 K for temperature parameters, 150 kg/h for mass-flow parameters and 30 minutes for the duration (Menegon et al. 2014).

Considering only those *Classes* containing a number of *Events* higher than a threshold value, a representative part of the seasonal boundary conditions can be selected, and the events that take place with a low frequency –therefore slightly influencing the performance- are neglected. The results obtained with the short test sequence can be extrapolated to the whole season on the basis of a proportion between the test duration and the operation time of the heat pump during the entire season.

5.1 Heating Mode

Table 5 shows the results obtained with the seasonal test and the results extrapolated from the short test sequence. The duration of the short test is 6 days with respect to 43 days needed for the full-length test (and representing the whole winter season). The error on the evaluation of the SCOP is 2.6%.

Besides the evaluation of the seasonal performance figures, the proposed test sequence allows analysing also the frequency distribution of the instantaneous performance figures (COP and powers). In Fig. 6, the COP distribution obtained during the seasonal and short tests are compared. The COP has a typical normal distribution spanning between 0 to 5.5 with a peak around 3.7. As a consequence of the transients, about 8%

of the values are below 2. From Fig. 6, it is clear that the distribution obtained with the short sequence is comparable to the seasonal one. This is possible because the boundary conditions selection takes into account the statistical distribution of the seasonal boundary conditions.

Tab. 5: Seasonal and short test results. Heating Mode.

	N_{ev}	τ_{test} [days]	SCOP [-]	W_{hp} [kWh]	Q_{co} [kWh]	Q_{ev} [kWh]
Seasonal	554	43	3.47	1873	6496	4574
Short Test	161	6	3.39	1899	6442	4795

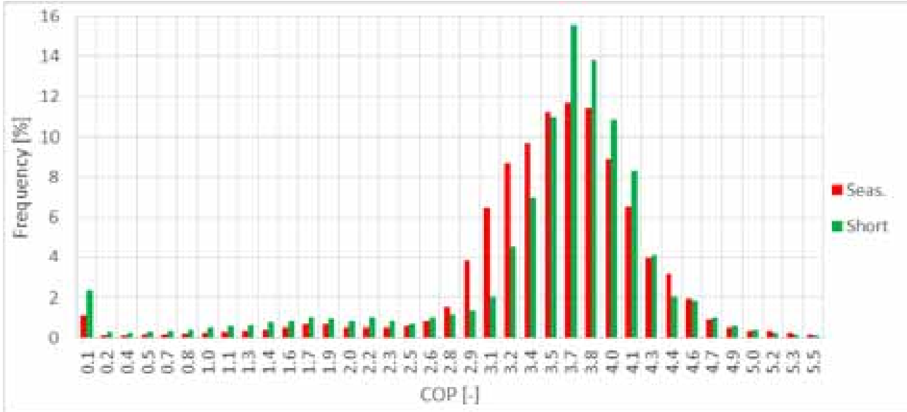

Fig. 6: COP distribution. Heating Mode.

5.2 Cooling Mode

Table 6 shows the results obtained in the seasonal test and the results extrapolated with the short test for the cooling season. The duration of the short test is half of the seasonal test and the SEER is evaluated with an error of 1.6%.

Tab. 6: Seasonal and short test results. Cooling Mode.

	Nev	τ_{test} [days]	SEER [-]	W_{hp} [kWh]	Q_{cond} [kWh]	Q_{evap} [kWh]
Seasonal	660	8	3.75	164	709	614
Short Test	330	4	3.69	163	692	604

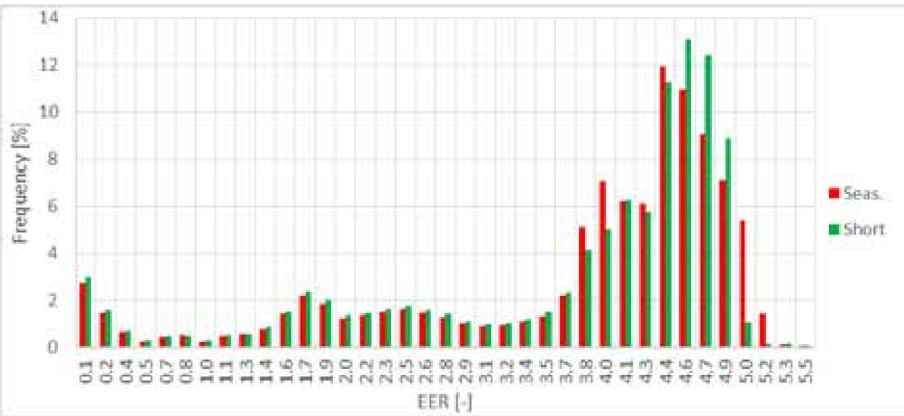

Fig. 7: EER distribution. Cooling mode.

Figure 7 reports the comparison of EER frequency distributions as obtained from the whole season tests and from the short test sequence. The EER varies between 0 and 5.3; in particular, about 14% of the values are lower than 2. The large amount of points with low EER is due to the presence of short events, with an overall duration comparable to the initial transient phase duration. The shape of the short test distribution is close to the seasonal test one, validating again the boundary conditions selection procedure.

6. Conclusions

This paper presented the results obtained from an experimental campaign analysing the dynamic operation of a compression reversible heat pump. The applied test procedure consists of the extraction of the real-like boundary conditions for the machine from a numerical simulation.

The heat pump was studied in both heating and cooling modes. The results of the two seasonal tests highlight different aspects of the machine behaviour under dynamic and varying working conditions. In heating mode and for the considered case-study, the heat pump works for space heating and domestic hot water preparation by using two different sources (air and solar thermal energy). As an effect of the control system, switching between the different operating modes, the boundary conditions of the heat pump are quite variable, with a strong effect on the performance. During the ON transient phase and the switch from space heating to domestic hot water preparation, the instantaneous performance is lower than that obtained under stationary state conditions. When the heat pump is switching from domestic hot water preparation to space heating, higher performance than under stationary state are obtained. The effect of this behaviour is not negligible and affects the seasonal distribution of performance figures. Nonetheless, the majority of working conditions is close to the steady state operation.

In summer, the heat pump works only for space cooling and the boundary conditions are less variable. However, the machine operation is characterized by numerous transients (switch ON and OFF) of short duration. These reflect negatively on the overall seasonal performance and have a stronger impact than in the winter case.

With the aim of providing a tool for the characterization of dynamic effects, without the need of performing long seasonal laboratory tests, a short test procedure was developed and the results were presented. The procedure allows not only a reliable evaluation of the seasonal performance of the machine (within a 2% margin), but also a correct representation of the frequency distribution of the instantaneous performance figures.

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