Dynamic Simulation as a Tool for the Analysis of the Interactions Among the Controllers of HVAC Systems

Mara Magni – University of Bologna – mara.magni2@unibo.it
Jean Pierre Campana – University of Bologna - jeanpierre.campana2@unibo.it
Gian Luca Morini – University of Bologna – gianluca.morini3@unibo.it

Abstract

During the last decade, the application of electronics to the different components of HVAC systems has offered new and sophisticated control systems capable of adapting the behaviour of each single device and of the whole plant to design specifications. These sophisticated finely-tuned control systems are, in principle, able to play an important role in reducing energy consumption and in improving thermal comfort. At the same time, there has been a marked increase in the complexity of the HVAC plant layout. By combining the complexity of the plants with that of the control systems and by taking into account the possibility of equipping each component of the plant with its own control system, the result is a new generation of energy systems with many possible interactions between controllers. Hence, it becomes impossible for a designer to evaluate, in an easy way, the effects of such interactions. As a consequence, it is difficult to know, at the design-stage, how the overall control system will operate. Nowadays, the dynamic simulation of the complete HVAC system makes it possible to emulate the system in which the controllers operate. In this paper, the dynamic model of a basic HVAC system involving a condensing boiler, radiators with thermostatic valves and an inverter driven hydraulic pump is presented. Each element of the circuit is equipped with its own control. The model of the system was built by using a custom-made library of Simulink blocks specifically created for the dynamic simulation of controlled HVAC systems. The dynamic model will be used in order to underline the strong influence of the control system on the HVAC energy efficiency and thermal comfort conditions. Specific design rules limiting the negative interaction among the activated control systems are inferred by the results shown in this paper.

1. Introduction

Dynamic simulation is an important tool for designers of hydronic systems. Satyavada and Baldi (2016) highlighted the importance of an effective control strategy for HVAC system in order to reduce energy consumption and to avoid instabilities. As shown by Morini and Piva (2007; 2008), SIMULINK is a suitable framework for the analysis of the behaviour of a controlled HVAC system. CARNOT (Conventional And Renewable eNergy Optimization Toolbox) is one of the most popular SIMULINK block sets for dynamic modelling of heating equipment (Wemhöner et al., 2000), developed with the financial support of Viessmann GmbH. One of the main advantages in adopting a MATLAB/SIMULINK framework is that this platform is widely used both in academic and professional environments, and it becomes easy for users to model new HVAC devices and control logic. In this way, each user can easily add new components to the library either by designing directly new graphical Simulink models or by using C-, Fortran- or MATLAB M-scripting languages.
Since the authors are convinced of the potential of this approach, for this study a SIMULINK library named ALMAHVAC, fully compatible with the CARNOT block set, was used for modelling of heating plants coupled with ALMABuild library for the description of the building thermal characteristics. A description of ALMABuild can be found in Campana et al. (2017). In this paper, ALMAHVAC was used in order to analyse the impact on the energy and indoor comfort performance of some design choices regarding the supply temperature setpoint, the continuous or intermittent operating mode of the heating plant, as well as the pump sizing and its control mode, and the valve sizing.

2. Description of the Model

ALMAHVAC is used in order to model the heating plant of a building (modelled with ALMABuild) by using a series of blocks able to simulate boilers, thermostatic valves, pumps, and radiators. The hydraulic loops are built in SIMULINK by simply connecting the blocks to thermal bus lines (defined in CARNOT), able to exchange data among the blocks. The building is a one floor detached house located in Bologna and composed of four thermal zones (kitchen, living room, bathroom, bedroom). A description of the envelope components of the building is given in Campana et al. (2017). A room temperature equal to 24 °C is set for the bathroom, and 20 °C for the other rooms.

Winter simulations were carried out considering the typical winter period of Northern Italian sites from September 15 to April 30 (5448 hours). The complete set of METEONORM climatic hourly data for Bologna was considered in the simulations. The heating system was based on a condensing boiler; radiators were used as terminals and each radiator was controlled by means of a thermostatic valve according to the operative temperature. Fig. 1 shows the layout of the hydraulic loop. Different types of circulation pumps were considered in this work (i.e. constant or variable speed circulators). The performances of the variable speed circulators were calculated following the method presented in Ahonen et al. (2010). The condensing boiler used in the heating plant is a VITODENS 222-W B2LB, with a nominal thermal power equal to 19 kW and a large power modulation ratio (1:10).

A SIMULINK model was created on the main technical data of this boiler. The model needed as inputs only the values which are available from the boiler technical sheet.

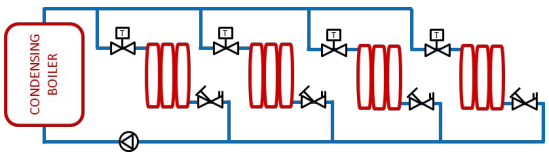

Fig. 1 – Hydraulic system scheme

Starting from a first guess, exhaust gas temperature, the chimney and envelope losses were calculated on the knowledge of both the exhaust gas and water return temperature. Then, the power released to the water was calculated. Finally, the exhaust gas temperature was recalculated by means of a thermal power balance between the exhaust gas and water.

The condensing boiler was controlled by a PI controller on the supply water temperature; its parameters were determined by the Ziegler-Nichols loop tuning method for open loop test. The PI control implements a back calculation for anti-windup. The condensing boiler control implements two additional logics based on the water mass flow rate and on the minimum switch off time. When the water mass flow rate across the boiler is lower than 1/12 of its nominal value, the control switches off the boiler. This kind of control is mandatory especially for boilers having low water content in order to avoid boiling conditions within the device. In addition, in order to reduce the on-off cycles, it is generally imposed that a minimum time interval equal to $\Delta\tau$ (i.e. 15 minutes) must be guaranteed between two consecutive boiler switches. The radiator was modelled by using an RC scheme with 7 thermal resistances and 7 capacities (7R7C). The whole radiator was divided into 7 nodes with the same thermal capacity. The power exchanged with the room air and with the water was calculated in each radiator node. The total thermal power delivered from the radiator to the room was divided between the radiative node and the convective node of the thermal zone on the basis of the radiator characteristics. In this case, 30 % of the heat exchanged with the room was charged on the convective node (low temperature radiators). The thermostatic valve block calculated the pressure drop over the valve body by considering the valve flow coefficient K_v as a function of the valve position, both for linear and equal percentage valves. In this work, only equal percentage valves were considered. Reference thermostatic valves with a rangeability of 40, and the appropriate flow coefficient values for each room ($K_{v,0\,Kitchen} = 0.64$, $K_{v,0\,Livingroom} = 0.72$, $K_{v,0\,WC} = 0.48$ and $K_{v,0\,Bedroom} = 0.47$), were considered in the simulations. In order to simulate oversized valves, in some cases the reference $K_{v,0}$ values were increased by a C_{Kv} factor (as indicated in Table 1) larger than 1.

Table 1 – A summary of the different cases simulated (A: fixed speed pump, B: variable speed pump with controlled constant pressure head, C: variable speed pump with controlled decreasing pressure head)

Case	Valve sizing (C_{Kv})	Pump type	Pump sizing (C_H)	$\vartheta_{supply,set}$ [°C]
(1)	1	A	1	60
(2)	1	A	1	70
(3)	1	A	1	80
(4)	1	B	1	70
(5)	1	B	1.2	70
(6)	1	B	1.5	70
(7)	1	B	2	70
(8)	1	B	3	70
(9)	1	C	1	70
(10)	1.2	B	1	70
(11)	1.5	B	1	70
(12)	2	B	1	70
(13)	3	B	1	70
(14)	1	B	1	70

The reference pump had a hydraulic head of 18 kPa for a volume flow rate of 0.6 m^3/h. In order to simulate oversized pumps, in some cases the pump hydraulic head was increased by a C_H factor (as reported in Table 1) larger than 1. A complete overview of the different scenarios analysed in this paper is given in Table 1; for each case the values of C_H and C_{Kv}, the pump type and the water supply temperature setpoint used in each simulation are indicated. Fourteen cases were simulated to gain information about the role played by the pump type and sizing, valve sizing, and water temperature supply on the seasonal performance of the heating plant.

3. Influence of Supply Temperature

The interaction between the imposed boiler setpoint temperature (water supply temperature) and the thermostatic valves was analysed first, by considering the layout of the heating system analysed in this paper. The thermostatic valve controls the water mass flow rate across the radiators with the goal to maintain a constant room temperature. Since the power exchanged between the radiators and the room depends on the mean radiator temperature, if the room load is fixed the mean radiator temperature is also fixed. If the water supply temperature increases, the thermostatic valve reduces the water mass flow rate and, as a consequence, the water temperature at the radiator outlet, is also reduced. The lower the water return temperature, the higher the condensing boiler efficiency is. However, by increasing the water supply temperature, the water mass flow rate is progressively reduced, and lower mass flow rates can become a problem for the boiler and the system stability.

In fact, boilers with low water content cannot operate with low water mass flow rates in order to avoid the risk of localized water boiling. For this reason, the interaction between the water supply temperature setpoint and the thermostatic valves becomes so important for variable mass flow rate hydraulic loops, as demonstrated by Lazzarin (2012 and 2014).

The results of Cases (1), (2) and (3) are now analysed (see Table 1). Fig. 2 shows the trend of the yearly distribution of the total water mass flow rate ($m_{w,tot}$), of the water supply temperature (ϑ_{supply}), of the water return temperature (ϑ_{ret}) and of the load factor (ϕ). By comparing the different charts, it is possible to highlight the percentage of winter time in which the heating system works under defined operating conditions. It becomes evident how, when reducing the water supply temperature, the time during which the boiler is switched on and the thermostatic valves are opened, increases; at the same time, the water return temperature is larger when the water supply is reduced.

Table 2 summarizes the most important results carried out from the yearly simulations by varying the water supply temperature setpoint. Table 2 shows that by adopting a supply temperature of 80 °C (Case (3)) the boiler works in condensing regime and the amount of condensate per hour is 142 % higher when compared to Case (1) and 81 % higher than Case (2).

This is due to the lower water return temperature obtained through a larger water supply temperature.

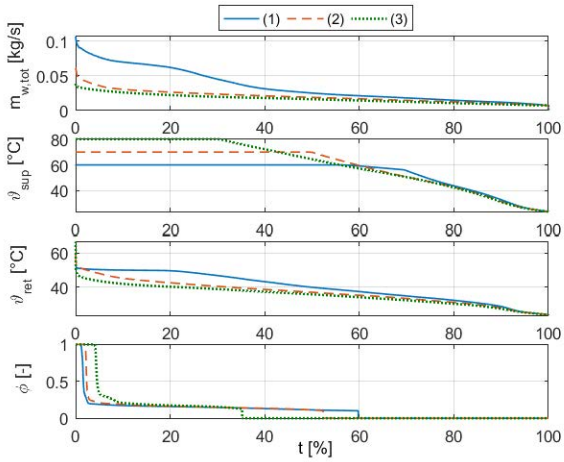

Fig. 2 – Yearly simulation results with different supply water temperature setpoints at full load: ((1) $\vartheta_{supply} = 60$ °C, (2) $\vartheta_{supply} = 70$ °C, (3) $\vartheta_{supply} = 80$ °C)

However, more condensation doesn't always correspond to higher boiler seasonal efficiency. In fact, it can be noticed that the seasonal efficiency of the boiler, by neglecting all the losses ($\eta_{g,no\ losses}$), increases when the supply temperature is increased; on the contrary, the boiler seasonal efficiency η_g (which takes into account the losses) decreases when the supply temperature goes from 70 °C (Case(2)) to 80 °C (Case(3)). The opposite trend of $\eta_{g,no\ losses}$ and η_g highlights that from Case (2) and Case (3) the boiler losses are increased. This is due to the increase of the boiler on-off cycles from Case (2) to Case (3). In fact, Case (3) is characterized by a higher number of on-off cycles because when the supply temperature is increased, the water mass flow rate becomes lower and its value, especially during the warmest months, can become lower than the value in correspondence of which the boiler with a low water content, must be switched off. In Case (1) the water mass flow is higher than in the other cases because of the low supply temperature; in this case the condensing boiler operation is rarely stopped due to the large value of the mass flow rate. This means that during the warmer periods when the building thermal loads are lower and the condensing boiler works with low load factors (close to the minimum of its modulation range) in Case (1), the boiler on-off cycles have a frequency limited only by the minimum time interval between two consecutive switches (i.e. 15 min). The net energy delivered to the water increases with the supply temperature setpoint, but it is necessary to notice that also the comfort level increases. In fact, the number of hours in which the rooms are under their setpoint temperature decreases when the supply temperature increases; this means that the radiators were designed to work with a supply temperature higher than 70 °C and in the colder periods a supply temperature of 60 °C is not enough to cover the building demands. However, the heating plant is not able to satisfy the bathroom setup temperature during the entire colder season, even in Case (3). The energy demand of the pump decreases if the supply temperature increases because the water flow rate is reduced.

Table 2 – Seasonal results reached by varying the water supply temperatures setpoint

	(1)	(2)	(3)
E_{fuel} [kWh/y]	9785	9907	10040
E_w [kWh/y]	9705	9826	9898
η_g	99.18	99.18	98.59
$\eta_{g,no\ losses}$	100.66	101.06	101.79
t_{on} [h/y]	3258	2854	1919
$\%t_{cond}$	98	98	100.0
Cond [kg_{H2O}/h]	0.12	0.16	0.29
Nr. on/off cycles	1355	668	1016
E_{pump} [kWh/y]	17	13	12
$\vartheta_{kitchen}$<19.9 °C [h]	335	0	0
ϑ_{living}<19.9 °C [h]	0	0	0
ϑ_{wc}<23.9 °C [h]	3144	1300	873
$\vartheta_{bedroom}$<19.9 °C [h]	0	0	0

4. Influence of Pump Type and Size

4.1 Pump size

In this section, the effects generated by the adoption of an oversized circulation pump are analysed. The interactions between circulation pump and thermostatic valves have an influence on the correct behaviour of the whole circuit. The pump must be chosen in order to avoid that thermostatic valves operate at the limits of their operative field. Since the thermostatic valves are not able to close their cross section area (A) completely, a minimum mass flow rate is always present, and its amount depends on the pressure drop over the valve for a fixed value of the valve rangeability (R) and K_v as indicated by the following relationship:

$$\dot{V} = \sqrt{\frac{DP_{valve}}{10^5}}\ K_{v,0}\ [R^{(A_{\%}-1)}] \qquad (1)$$

By selecting a more powerful pump, the mass flow rate increases and the thermostatic valves tend to operate closer to the final part of their stroke (small values of A). In this way, they lose their modulating capacity completely and become on-off controls. The thermal indoor comfort can be affected by this behaviour. In order to investigate this effect, different simulations by adopting different variable speed pump sizes were carried out. Five different cases (Cases (4), (5), (6), (7) and (8)) were considered (see Table 1).

Table 3 – Seasonal results obtained by adopting pumps of different sizes

	(4)	(5)	(6)	(7)	(8)
E_{fuel} [kWh/y]	9857	9905	9975	10081	10300
Ew [kWh/y]	9773	9824	9902	10019	10274
η_g	99.14	99.19	99.27	99.38	99.74
$\eta_{g,\text{no losses}}$	101.1	101.1	101.1	101.1	101.1
t_{on} [h/y]	2853	2863	2906	2993	3279
$\%t_{cond}$	98.4	98.4	98.1	97.9	97.8
E_{pump} [kWh/y]	10.1	12.2	15.3	20.6	31.0
On Off/h [n/h]	0.23	0.23	0.22	0.20	0.14
Cond [kg_{H2O}/h]	0.16	0.16	0.16	0.16	0.15
$\vartheta_{kitchen}$> 20.8 °C [h]	1206	1304	1416	1645	2234
ϑ_{living} >20.8 °C [h]	2313	2469	2881	3498	4358
ϑ_{wc} >24.8 °C [h]	7	5	7	14	21
$\vartheta_{bedroom}$ >20.8 °C [h]	1865	2181	2421	2857	3616

Table 3 shows that when the pump hydraulic head is increased, the thermostatic valves are no longer able to control the ambient temperature and the overheated periods are longer.

The bathroom is the room where overheating is less important, due to the higher thermal loads which allow the thermostatic valve to work in its whole operative range.

Along with the indoor comfort issue, also energy consumption supports the conclusion that the adoption of an oversized pump is always a bad design choice. Table 3 shows that with an oversized pump the heating system delivers more energy to the building, but unfortunately, this additional heat is mainly used to overheat the rooms. The condensing boiler efficiency is not influenced by the pump size, but the pump energy consumption increases proportionally to the pump hydraulic head.

4.2 Pump Type

In this section, three different pump types are compared (i.e. Cases (2), (4) and (9), see Table 1). Variable speed circulation pumps are generally coupled to thermostatic valves in order to minimize the pressure drop across the valve when the load factor goes down. In this way noise problems due to high-pressure drops across the valve can be overcome and the pump energy consumptions can be reduced.

In Table 4 the main results obtained by adopting constant speed and variable speed pumps are reported. It is evident that the condensing boiler efficiency and its operating time are not influenced by the pump type. It can be noticed that adopting a variable speed pump with a decreasing pressure head (Case (9)) less energy is delivered to the water, and this is underlined by longer periods in which the rooms are underheated. The overheating period is strongly reduced in Case (9) because the pump reduces its hydraulic head when the valves are closing. Therefore, the water mass flow rate along the hydraulic loop is lower than in the other cases. On the other hand, the underheating periods are longer in Case (9) with respect to the other cases because the mass flow rate moved by the pump is slightly lower than in Cases (2) and (4). The minimum operative temperature guaranteed in the rooms is similar in the three cases; it can thus be assessed that the indoor comfort level is quite good, independently from the adopted pump type. The results reported in Table 4, put into evidence that the pump energy consumptions are considerably reduced in Case (9) (-51 % with respect to Case (4), -62 % when compared to Case (2)). In large hydraulic networks important energy savings are expected by the use of

variable speed pumps with a decreasing pressure head coupled to thermostatic valves.

Table 4 – Seasonal results obtained by using different pump types

	(2)	(4)	(9)
E_{fuel} [kWh/y]	9907	9857	9723
Ew [kWh/y]	9826	9773	9630
η_g	99.18	99.14	99.04
$\eta_{g,no\ losses}$	101.06	101.06	101.06
E_{pump} [kWh/y]	13	10	5
t_{on} [h/y]	2854	2853	2805
$\vartheta_{kitchen}$<19.9 °C [h]	0	2	45
ϑ_{living} <19.9 °C [h]	0	0	0
ϑ_{wc}<23.9 °C [h]	1300	1665	3000
$\vartheta_{bedroom}$<19.9 °C [h]	0	0	0
$\vartheta_{min,kitchen}$ [°C]	19.93	19.90	19.84
$\vartheta_{min,livingroom}$ [°C]	20.22	20.19	20.14
$\vartheta_{min,wc}$ [°C]	23.61	23.57	23.50
$\vartheta_{min,bedroom}$ [°C]	20.18	20.15	20.09
$\vartheta_{kitchen}$ >20.8 °C [h]	1300	1206	1020
ϑ_{living} >20.8 °C [h]	2479	2313	1668
ϑ_{wc} >24.8 °C [h]	5	7	4
$\vartheta_{bedroom}$ >20.8 °C [h]	2190	1865	1526

5. Valve Sizes

Thermostatic valve sizing is important because it is correlated to its authority. The authority is the ratio between the pressure drop over the open valve to the total pressure drop over the controlled branch of the hydraulic circuit in which the valve is inserted. A rule of thumb for the valve sizing is to select the valve size in order to obtain a valve authority of around 0.5. In fact, if the valve is oversized its authority is low and the valve is no longer able to reduce the mass flow rate in the controlled branch, if needed. In order to show the effects of the adoption of oversized valves on the performances of the heating plant five yearly simulations were carried out in order to compare the seasonal results obtained with different valve sizing (Cases (4),(10),(11),(12),(13)), by changing the C_{kv} factor as described in Table 1.
Table 5 summarizes the main results. By increasing the valve size, the water mass flow rate increases because the valves are characterized by lower authority; this means that, even in the warm period, the radiators receive larger water flow rates with high temperatures that contribute to the room overheating. Since the minimum mass flow rate increases with the valves size, the boiler is switched on for longer periods when larger valves are adopted. The water return temperature increases and the flow rate of condensed water decreases. The number of the boiler on-off cycles per hour is lower in the Cases (4), (10), and (11) because in the warm periods due to the low mass flow rates the boiler remains switched off. This parameter is 0 in Case (13) because the mass flow rate is always large enough to avoid the boiler switch off, and it has a peak in Case (12).

Table 5 – Seasonal results obtained by using different valve sizes

	(4)	(10)	(11)	(12)	(13)
E_{fuel} [kWh/y]	9857	9957	10129	12004	13553
Ew [kWh/y]	9773	9883	10075	12069	13567
η_g	99.1	99.3	99.5	100.5	100.1
$\eta_{g,no\ losses}$	101.1	101.1	101.1	101.6	100.1
t_{on} [h/y]	2853	2899	3049	4417	5448
On Off/h [n/h]	0.23	0.22	0.19	0.47	0.00
Cond [kg_{H2O}/h]	0.16	0.16	0.16	0.16	0.08
E_{pump} [kWh/y]	10.1	10.2	10.3	9.8	11.0
$\vartheta_{kitchen}$ >20.8 °C [h]	1206	1390	1787	3225	4306
ϑ_{living} >20.8 °C [h]	2313	2822	3624	5088	5448
ϑ_{wc} >24.8 °C [h]	7	7	17	2089	2912
$\vartheta_{bedroom}$ >20.8 °C [h]	1865	2380	3040	4268	5437

In Case (12), the water mass flow rate is always larger than the minimum value needed in order to keep the boiler switched on but the load factor is often close to the boiler minimum modulation level. This leads the boiler to work in on-off cycles with a frequency imposed by the minimum switch off time of the boiler (i.e. 15 min). The energy required by the pump is almost constant because when the water mass flow rate increases, the pressure drop over the valves decreases if larger valves are adopted. The most important aspect linked to the adoption of valves having different sizes, is related to the room

comfort. From Table 5 it is evident that oversized valves can have problems in controlling the room temperature: important overheating periods are present especially in Cases (12) and (13). This leads to an increase of energy consumptions (see E_{fuel} in Table 5), too.

6. Intermittent Operation Mode

It often happens that, during the nighttime, the users decide to reduce the setup value of the room temperature or to switch the heating plant off with the aim to reduce energy consumption. If the heating plant is switched off, room temperature can decrease 2-4 K during the nighttime, depending on the building thermal insulation level. This leads the thermostatic valves to be open completely when the heating system restarts in the morning.

Table 6 – Seasonal results obtained under continuous (case (4)) or intermittent (Case (14)) operation mode

	(4)	(14)
E_{fuel} [kWh/y]	9857	9664
Ew [kWh/y]	9773	9524
η_g	99.14	98.56
$\eta_{g,no\ losses}$	101.06	100.41
t_{on} [h/y]	2853	2358
$\%t_{cond}$	98	55
Cond [kg_{H2O}/h]	0.16	0.14
E_{pump} [kWh/y]	10.1	15.6
$\vartheta_{kitchen}$<19.9 °C [h]	0	1185
ϑ_{living} <19.9 °C [h]	0	480
ϑ_{wc}<23.9 °C [h]	1665	3544
$\vartheta_{bedroom}$<19.9 °C [h]	0	521

The boiler will work at the maximum load factor and also the water mass flow rate will be at its maximum.

This makes the system work with larger water return temperature, which can reduce the boiler seasonal efficiency. In Table 6 the main results of the simulations in continuous operation mode (Case (4)) and intermittent regime (Case (14) are reported: the same input parameters of Case (4) but the boiler is switched off every day from 11pm to 6 am during the whole winter). From Table 6, it becomes evident how the adoption of an intermittent regime reduces the boiler seasonal efficiency. Since the boiler has good performance even when it works without condensation, the seasonal efficiency is always quite large.

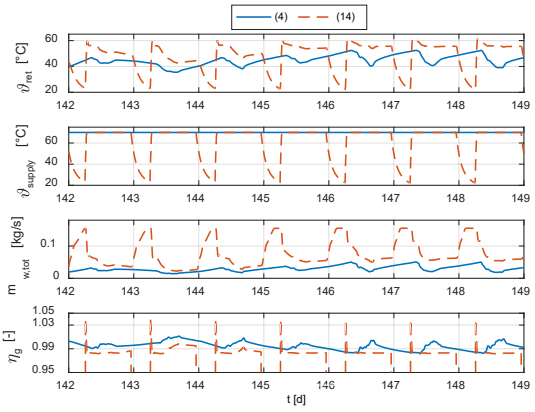

Fig. 3 – Supply temperature, return temperature, water mass flow rate and boiler efficiency under continuous (Case (4) and intermit-tent (Case (14)) operation mode

The energy saving is only 1.95 % in Case (14) with respect to Case (4) even if the average room temperatures are much lower. This result can be explained if one considers that the number of hours in which the boiler works in condensation regime is reduced to 50 % by adopting an intermittent regime.

Fig. 3 shows a period of seven days in wintertime starting from December 19. The trends of the water mass flow rate, of the water return temperature and of the boiler efficiency confirm that the intermittent regime is characterised by lower efficiency due to larger water flow rates and higher return temperatures.

7. Conclusion

In this work the capability of a new Simulink block set named ALMAHVAC (coupled with ALMA-Build) as tools for the dynamic simulations of hydronic systems was tested. The Simulink blocks used to model pumps, boilers, radiators and thermostatic valves are fully compatible with the CARNOT library. A series of numerical results are shown, analysing the influence of the pump sizing rules, the pump control type, the valves sizing and user behaviour (by varying the water supply temperature setpoint and by selecting the intermittent use of the heating system) on the seasonal efficiency,

and on the thermal indoor comfort. A higher supply temperature leads to a lower return temperature, which is a good condition for the condensing boiler, and to a lower water mass flow rate which can increase the number of on-off cycles especially during the warmest winter periods. This means that a trade-off supply temperature value exists in order to maximize the seasonal efficiency of the system. Oversized valves and pumps make the system less efficient in terms of capacity, to guarantee both ideal indoor thermal conditions and low energy consumption. It is also proven that the adoption of an intermittent regime for the heating plant can reduce significantly the indoor comfort and the boiler efficiency.

Nomenclature

Symbols

η	Efficiency [-]
ϕ	Load factor [-]
ϑ	Temperature (°C)
E	Energy [kWh]
T	Time [h]
A	Valve position [%]
K	Flow coefficient [h^{-1} m^3 b $ar^{-1/2}$]
R	Rangeability [-]
DP	Pressure difference [Pa]
$\dot{V}$	Volume flow [m^3/h]
Cond	Condensation
m	Mass flow rate [kg/s]
C	Multiplication factor

Subscripts/Superscripts

g	Generation
g,no losses	Generation without the boiler losses
g,ist	Instantaneous generation
ret	Return
w	Water

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