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The adoption of pressure independent control valves (PICVs) for the simultaneous optimization of energy consumption and comfort in buildings

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ABSTRACT

The optimization of energy consumption in buildings' HVAC systems plays a crucial role in reducing greenhouse gas emissions worldwide. In new and deeply-renovated buildings, characterized by modulating terminal units and aiming at the maximum exploitation of renewables, an accurate hydraulic balance of the distribution network may become critical, resulting in a significant increase of pumping energy consumptions and a deterioration of indoor thermal comfort conditions. In the present paper, we compare the performance of traditional manual balancing valves and new pressure independent control valves (PICVs), used to balance the hydronic loop of HVAC systems. To perform this analysis, a new MATLAB-Simulink model has been specifically developed to simulate the behavior of PICVs and has been used in an application case study. The efficiency of manual and pressure independent control valves is evaluated numerically by simulating a multi-zone distribution network under variable operating conditions and considering different control strategies of the circulating pump. Results show that in off-design conditions, when some of the branches of the hydraulic network are closed, traditional manual balancing valves are not able to guarantee the nominal mass flow rate in the remaining loops. On the contrary, no significant variations of water mass flow rate are observed when PICVs are adopted, even in partial load conditions. Despite their additional cost, PICVs allow to ensure better indoor thermal comfort sensations for individuals, avoiding under/over-heating of rooms and, moreover, yield a decrease of the pump electric consumption with respect to traditional manual balancing valves. In addition, based on the obtained results, general rules concerning the adoption of PICVs are provided, depending on the extension of the hydraulic loop and the adopted pump control logic. The reported results highlight the role that PICVs can play to ensure energy savings in HVAC systems without penalizing users' indoor comfort conditions.

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1. Introduction

According to the Energy Performance of Buildings Directive (EPBD), revised in 2018 and fast-forwarded via the recent REPowerEU, the building sector is crucial for achieving the EU's energy and environmental goals [1,2]. Europe is activating a complex strategy, based on a wide plan of actions and policies aimed at reducing the energy consumption in this sector and decarbonizing the national building stocks by 2050, to reach a strong decrease of Greenhouse Gases (GHGs) emissions. Among these policies, the development of energy communities [3,4], as well as the near Zero Energy Building (nZEB) concept for new and refurbished buildings, can help to achieve the above-mentioned objectives [5–7]. In addition, innovative configurations of HVAC (Heating, Ventilation and Air Conditioning) systems could significantly enhance the overall energy performance of this kind of systems, allowing a better exploitation of renewable energy sources [8–11]. Furthermore, the adoption of optimized control strategies for HVAC systems has proved to be a good solution to reduce the primary energy consumption of residential and non-residential building sectors with very low investment costs [12].

In order to minimize the overall energy demand linked to space heating/cooling in buildings, an accurate hydraulic balance of heating systems is important to limit undesirable energy consumptions due to distribution network losses and lack of flow rate control along the network during particular operating conditions. A properly balanced hydronic distribution network enables to provide a uniform temperature drop across all terminal units, reducing the supply temperature in the heating system. Unfortunately, a recent

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study across the EU has shown that correct hydraulic balancing is not an established practice both when HVAC systems are installed and when they are maintained, and only 10% of the actual building stock has heating systems properly balanced [13].

Several papers published in the scientific literature report the potential savings achievable with the adoption of dynamically balanced HVAC hydraulic loops. A review on different HVAC regulations is shown in [14], while evaluations on energy efficient HVAC systems are reported in [15]. The potential energy saving linked to a proper hydraulic balancing of a HVAC system distribution network while ensuring thermal comfort of building occupants is well underlined by Cho et al. [16]. These authors proposed a methodology, applied to a group of existing buildings in Geneva during the winter season, demonstrating that there is an energy saving potential between 2% and 14%. Moreover, Cho et al. enriched their analysis studying to which extent hydraulic imbalance affects the thermal energy demand of buildings [17]. Results of this work show that, in a different case study represented by four multi-family buildings in Geneva, hydraulic balancing significantly reduces the room temperature drift, avoiding room overheating and, consequently, ensuring energy savings.

A recent report from ENEA [18] estimates that in Italy 37% of the population lives in two million multi-apartment buildings in which the heating system is based on water-to-air emitters, typically radiators and fan-coils, connected with an extended hydraulic network based on a series of branches operating in series and parallel. For this kind of buildings, a proper balancing of the distribution network is crucial during the commissioning of HVAC systems. Moreover, the heat transfer rate of radiators and fan-coils is usually controlled by modulating the water mass flow rate [19] and, consequently, the nominal conditions occur for a very limited part of the heating season, since the overall mass flow rate changes continuously to manage part load conditions. As distribution network components are generally sized with respect to nominal operating conditions (i.e., water mass flow rates and pressure drops), the network is typically unbalanced during the dynamic operation of the HVAC system. In fact, water mass flow rate and pressure drop depend on one another and variations of mass flow rate within the network due to the modulation of the heat transfer rate of some emitters modify indirectly pressure drops in other loops. Therefore, the water mass flow rate within the hydronic loop is adjusted automatically until a new balance condition is reached. Consequently, over- or under-flows may occur in the network, corresponding to a deviation of the indoor thermal comfort conditions and energy consumptions from the nominal values. This aspect is even more crucial in new and deeply refurbished buildings, that are increasingly thermally insulated and in which the exploitation of solar gains is maximized during the cold season. As a consequence, terminal units need to operate at partial load for a longer time during winter. Furthermore, it is more and more frequent that in multi-zone buildings some terminal units are completely switched off while others need to deliver heat.

Typically, the hydraulic balancing of the distribution network is only guaranteed at nominal conditions by adjusting manually the position of traditional calibration valves during the first start-up of the heating system. As a consequence, the commissioning phase of the hydraulic network involves iterative field tests in order to reach the correct balance of the system, since modifications on a single valve impact on all the network. Conversely, in recent years new typologies of calibrating devices have become more popular within HVAC systems [20,21]. Among them, Pressure Independent Control Valves (PICVs) are one of the most widespread solutions. They differ from traditional manual balancing valves since the water mass flow rate can be stabilized automatically, even if the differential pressure across the valve varies during the HVAC system operation. In addition, their implementation allows to strongly reduce the duration of the commissioning phase, since the network balance is made automatically.

PICVs were first introduced in the last century [22] and their structure and hydraulic behavior have been object of a continuous analysis even in the last years [23–25]. McGowan discussed the importance of the correct selection and sizing of HVAC control valves, paying attention to the most common mistakes concerning, as example, pressure variations [26]. Boldt presented a detailed analysis on several kinds of balancing valves, showing when the adoption of PICVs is appropriate in terms of dependency from pressure drops [27]. More in detail, Boldt explains that PICVs are suitable only when high head losses occur (i.e., extended hydraulic networks with many emitters).

As far as the authors are concerned, the potential benefits on HVAC system energy consumptions and indoor thermal comfort conditions linked to the use of PICVs have not been investigated yet in the scientific literature in a quantitative way. In fact, in many commercial software for dynamic energy modeling of buildings and HVAC systems, such as TRNSYS and EnergyPlus, the calculation of the water mass flow rate in hydronic loops is not automatic in presence of active control devices, since pressure drops are not taken into account and mass flow rate is typically an input variable for mass and energy balances. For this reason, the negative influence of unbalanced distribution networks on the HVAC system performance cannot be evaluated by this kind of software.

In the present paper, the use of PICVs in a HVAC system based on 2-pipe fan-coils as emitters and coupled to a multi-zone residential building is analyzed comparing their performance with those of traditional manual balancing valves. The analysis is performed within the MATLAB-Simulink environment [28], by means of the open-source libraries CARNOT [29] and ALMABuild [30]. In fact, MATLAB-Simulink has proved to be suitable for modelling the complex control systems which are nowadays used in contemporary HVAC plants [31–35]. Nevertheless, such libraries do not include blocks for the simulation of new PICVs. To allow more detailed simulations of this kind of heating systems, in this work a novel specific Simulink block is appositely developed to simulate the behavior of PICVs. This model is consistent with the above-mentioned Simulink tools and is freely available for the scientific community. As far as the authors are concerned, the dynamic modeling of PICVs has never been investigated yet in the literature. Indeed, the novelty of the present paper is the introduction of a new Simulink block by means of which the behavior of PICVs can be considered in numerical simulations. In order to demonstrate the usage of this innovative numerical model, an application case study has been identified. In particular, numerical simulations have been carried out on the distribution network of a reference HVAC system under transient operating conditions, considering different calibrating devices (i.e., manual static valves and PICVs) and control logics of the circulating pump (i.e., fixed-speed, variable-speed with constant-pressure control and variable-speed with proportional-pressure control). The influence of the calibrating valves typology on the hydraulic balancing of the network is assessed, evaluating the achievable savings linked to pumping energy consumptions and the effects on the indoor thermal comfort conditions. Finally, general rules on the adoption of PICVs are provided, outlining their effectiveness in distribution networks on the basis of the total pressure drops and the adopted pump control logic.

Consequently, the presented findings can help quantifying the benefits linked to the use of PICVs in HVAC hydronic networks in terms of energy saving and indoor thermal comfort improvement, underlining in which cases the potential of these devices can be maximized.

2. Numerical model of pressure independent control valves

In this Section, the main technical features of pressure independent control valves and a detailed description of the numerical model developed in this work are presented.

2.1. Description of main technical characteristics of pressure independent control valves

The main feature of PICVs is their capability to stabilize automatically water flow rate in a loop of a distribution network. In fact, as shown in Fig. 1a, these valves are obtained by coupling two components [24,25]: a pressure regulator (component 1 in Fig. 1a) and an actuator (the two-way valve indicated as component 2 in Fig. 1a). The pressure regulator balances the differential pressure thanks to the opposite force of its internal moving spring, which imposes a variable resistance to the water flow, thus keeping the value of the differential pressure across the actuator (p_B – p_C) constant (see Fig. 1a). If the differential pressure across the whole PICV (p_A – p_C) increases due to variations in the operative conditions of the hydraulic circuit (e.g., a terminal unit switched off), the spring automatically moves to close the core and keep the differential pressure (p_B – p_C) as constant. As a consequence, the water flow rate across the actuator does not change. In addition, by varying its bore cross section, the actuator can regulate the desired water flow rate, that can be kept at a constant preset value (e.g., in on–off PICVs for not modulating terminal units) or at the value given by the actuator regulating action (e.g., in modulating PICVs for variable-flow emitters).

The typical characteristic hydraulic curve of a PICV is reported in Fig. 1b, where the volumetric flow rate V is plotted as a function of the total differential pressure across the valve, Δp_{PICV} (equal to p_A – p_C). The constant value V_{nom} of the volumetric flow rate is prescribed, during the commissioning and design phase, equal to the nominal value of the flow rate that must circulate in the specific loop section containing the PICV. The main feature is the presence of a minimum value of the differential pressure across the valve, Δp_{min}, below which the device behaves exactly as a traditional manual calibrating valve. Within this range, the volumetric flow rate V is an increasing function of Δp_{PICV}. Conversely, above this minimum value, a PICV ensures a constant flow rate, independently from the pressure drop. As a consequence, PICVs can be used to balance hydraulically the distribution network of a HVAC system, guaranteeing the nominal water flow rate in each loop of the circuit for almost all operating conditions.

2.2. Description of the Simulink model for pressure independent control valves

A novel Simulink block, implemented within the ALMABuild toolbox and consistent with the CARNOT library, has been developed to simulate the behavior of pressure independent control valves. More in detail, CARNOT is one of the most popular free Simulink libraries focusing on the modeling of HVAC systems [29], whilst ALMABuild is an open-source library of Simulink blocks, developed by the University of Bologna, suitable for dynamic energy simulations of both buildings and HVAC systems [30]. The peculiarity of ALMABuild is its top-down structure, with specific block-sets including sub-blocks and elementary blocks able to model each generic component of a HVAC system.

The scheme of the novel Simulink block is reported in Fig. 2. Most popular Simulink toolboxes dedicated to the dynamic simulation of HVAC systems, such as the above-mentioned ALMABuild and CARNOT libraries, use the THB vector (Thermo-Hydraulic Bus), to exchange data between elementary blocks (representing pipes, valves, pumps, emitters…). This vector contains the values of the main thermo-physical properties of the fluid stream, such as pressure, mass flow rate and temperature, at the inlet/outlet of each block.

For the developed PICV block, data stored in three THB vectors are used as inputs: i. a vector conveying the fluid properties at the PICV inlet (THBin in Fig. 2); ii. a vector conveying the fluid properties at the inlet of the hydraulic branch containing the PICV (THBm in Fig. 2); iii. a vector conveying the fluid properties at the outlet of the hydraulic branch containing the PICV (THBv in Fig. 2). In addition, four parameters must be set to simulate the behavior of a PICV through the proposed Simulink block: the value of the minimum pressure drop across the valve (Δp_{min}), the value of the maximum allowed pressure drop across the valve (Δp_{max}), the value of the nominal water mass flow rate ($\dot{m}_{nom}$) in the hydraulic branch containing the PICV (controlled branch) and the value of the nominal pressure drop of the controlled branch excluding that across the valve itself (Δp_b).

The pressure drop of each controlled branch (i.e., the circuit section whose water flow rate is controlled through a PICV) is evaluated as the difference between the fluid pressure at the branch inlet and outlet, obtained respectively through the bus selectors BS2 and BS3, coupled to the vectors THBm and THBv (see bottom left-hand side of Fig. 2 for reference). Moreover, a memory block (M2 in Fig. 2) is needed because of the algebraic loop involved in the PICV modeling.

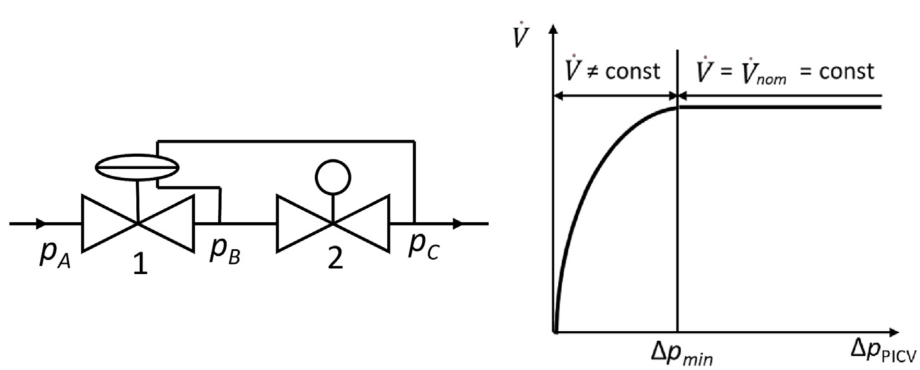


Fig. 1. PICV scheme (a) and characteristic curve (b).

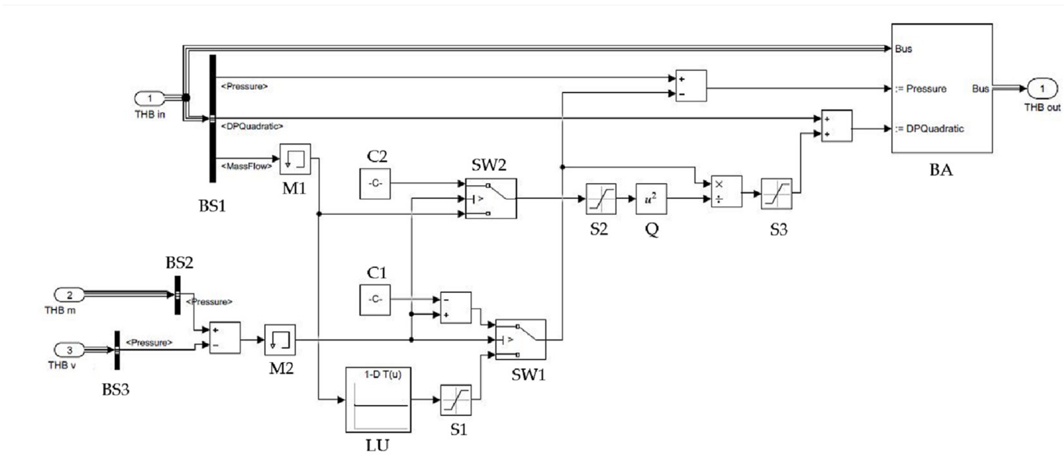


Fig. 2. Scheme of the PICV Simulink block.

A look-up table and a saturation block (LU and S1 blocks in Fig. 2, respectively) are employed to model the PICV characteristic curve for pressure drops across the valve lower than Δp_{min} (i.e., the first part of the valve characteristic curve reported in Fig. 1b). In this case, the valve head loss, Δp_{PICV}, is assumed as a function of the mass flow rate at the valve inlet, obtained through the bus selector BS1 from vector THBin (see Fig. 2). In detail, the first part of the PICV characteristic curve is modeled as a piecewise linear function, obtained by linearly interpolating manufacturer data two by two.

Then, a switch block (SW1 in Fig. 2) evaluates the actual value of Δp_{PICV} as the difference between the overall pressure drop of the controlled branch (output of M2 block) and the nominal pressure drop of the branch excluding the valve, Δp_b (whose value is stored in the constant block C1, see Fig. 2), if the output of M2 block is higher than (Δp_b + Δp_{min}). Otherwise, the value of Δp_{PICV} is obtained from the first part of the valve characteristic curve, that is the output of S1 block.

Finally, the bus assignment block BA is employed to assign the value of the fluid pressure at the valve outlet in the outgoing bus THBout (right-hand side of Fig. 2). In particular, this output is evaluated as the difference between the fluid pressure at the valve inlet (first output of BS1 block) and Δp_{PICV} (output of SW1 block).

The switch block SW2, inserted in the PICV Simulink model, evaluates the mass flow rate at the valve outlet. In detail, when the overall pressure drop of the controlled branch is higher than (Δp_b + Δp_{min}), the nominal fluid mass flow rate $\dot{m}_{nom}$ (whose value is stored in the constant block C2 of Fig. 2) passes through. Otherwise, the mass flow rate value of the previous simulation time step is considered (output of the memory block M1). It is worth mentioning that, for numerical reasons, the output of SW2 block must be limited to positive values by means of the saturation block S2; then it is raised to the square by block Q.

Each TBH vector contains the values of the quadratic, linear and constant coefficients to be used in order to evaluate the head losses of a fluid flowing along a circuit through a second order polynomial expression. In this regard, the behavior of a PICV can be characterized by the following expression:

$$\Delta p_{\mathrm{PICV}} = p\,\dot{m}^2 \tag{1}$$

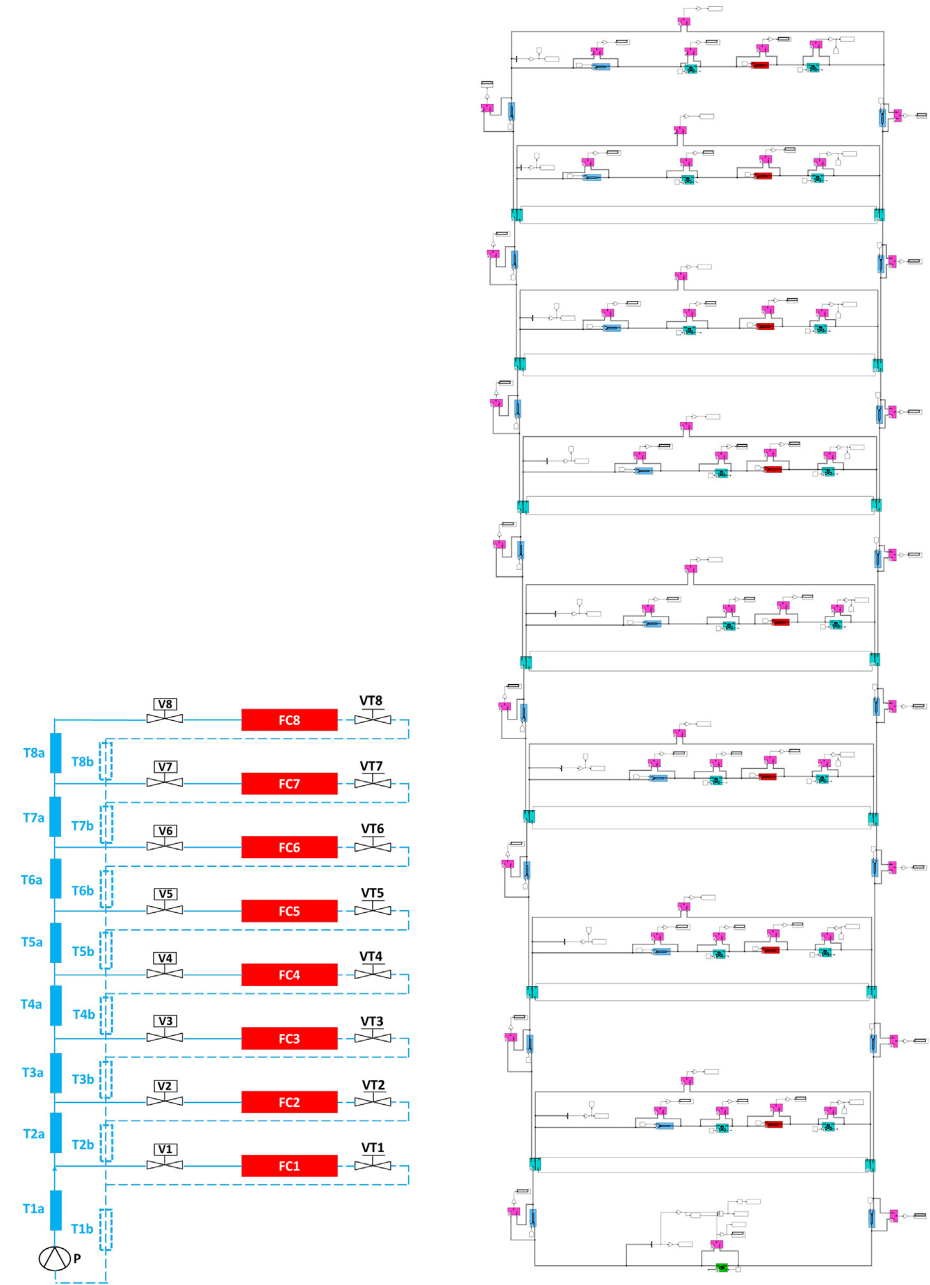
By following Eq. (1), the coefficient p can be evaluated as the ratio between Δp_{PICV} and the square of the mass flow rate (output of Q block). Values of p are limited between 0 and $\Delta p_{max}/\dot{m}_{nom}{}^2$ by means of the saturation block S3 (see Fig. 2). The updated value of the quadratic coefficient for the outgoing bus THBout is then obtained by summing to the corresponding value at the valve inlet (second output of BS1 block) the value of the coefficient p (output of S3 block) and is allocated to the bus THBout by the assignment block BA.

3. Description of the case study

In order to evaluate the performance of pressure independent control valves and compare the influence of different balancing valves typologies on the hydraulic balance of the distribution network of a HVAC system, an application case study has been considered in this work. Numerical simulations have been performed considering traditional manual balancing valves and PICVs, modeled by means of the newly-developed Simulink block described in the previous Section.

The application case study considered in this work is a HVAC system with a vertical-tube distribution network, coupled to a residential building composed of 8 storeys. The network includes 8 loops connected in parallel and a circulating pump (P) placed on the supply line. A sketch of the hydraulic circuit is represented in Fig. 3a. In each parallel branch of the circuit, the same elements are installed: a two-pipe fan-coil (FC), used as emitter to provide for space heating; an on–off control valve (V), used to stop the fluid flow to the fan-coil when no thermal load is present; a balancing valve (VT), used to balance hydraulically the loop during the commissioning phase. Two series of simulations have been performed. In the first run, traditional manual balancing valves have been considered, while in the second series of simulations a PICV has been inserted in each loop of network. For each simulation run, the effective fluid flow rate in the different loops of the hydraulic circuit has been calculated in off-design operating conditions, assessing the influence of balancing valve typology on the hydraulic balance of the network. Moreover, consequent impacts on indoor thermal comfort and pumping energy consumptions have been evaluated numerically within Simulink environment.

The hydronic distribution network considered in this work has been modeled in the Simulink environment by employing blocks from both CARNOT and ALMABuild libraries. The numerical model of the hydraulic circuit is reported in Fig. 3b. It is evident how a direct correspondence can be observed between the blocks of the Simulink model in Fig. 3b and the elements of the HVAC system reported in Fig. 3a. Fig. 4 displays a focus on the Simulink blocks employed to model a generic branch of the network in which a PICV is used as balancing valve, modeled by the newly-developed block.



(a) (b)

Fig. 3. (a) Sketch of the hydronic distribution network of a HVAC system coupled to 8 thermal zones and (b) corresponding Simulink model.

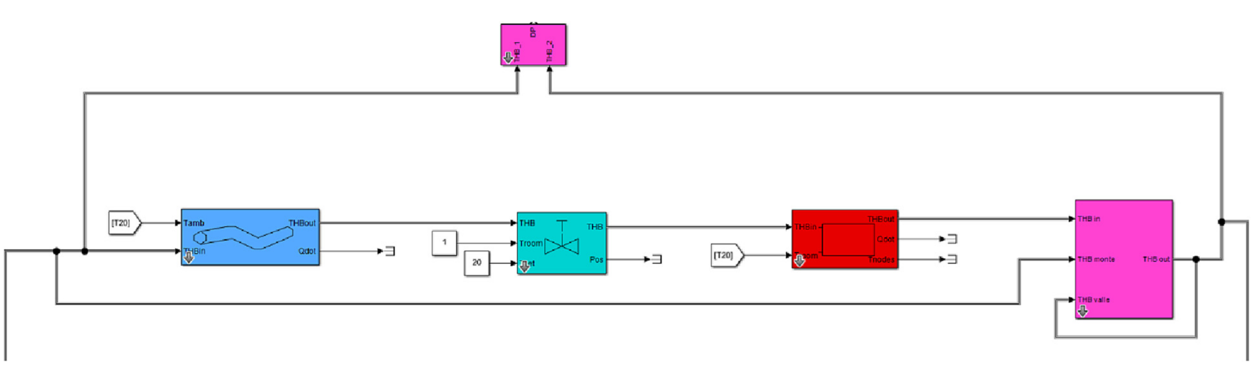


Fig. 4. Simulink blocks for the modeling of a generic branch containing a two-pipe fan-coil (red block), an on–off control valve (teal block) and a PICV (pink block on the right). (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)

As pointed out before, each Simulink block in the model represents an element of the HVAC system, such as tubes, valves and fan-coils. In order to balance hydraulically the network and to size the circulating pump, distributed and concentrated pressure drops have to be determined for each circuit loop. In particular, the total pressure drop of the i-th loop of the network ($\Delta p_{tot,i}$) can be calculated as follows:

$$\Delta p_{tot,i} = \sum_j \Delta p_{T,j,i} + \Delta p_{FC,i} + \Delta p_{V,i} + \Delta p_{VT,i} \tag{2}$$

where $\Delta p_{T,j,i}$ is the distributed head loss along each j-th pipe of the i-th loop, $\Delta p_{FC,i}$ is the concentrated pressure drop linked to the fan-coil installed within the i-th loop, $\Delta p_{V,i}$ and $\Delta p_{VT,i}$ are the concentrated pressure drops due to the on–off control valve and manual balancing valve (if present) of the i-th loop, respectively. In simulations with pressure independent control valves, the term $\Delta p_{VT,i}$ in Eq. (2) is replaced by $\Delta p_{PICV,i}$ (head loss of the PICV of the i-th loop) and the term $\Delta p_{V,i}$ is set equal to zero (the on–off control is included within the PICV itself).

More in detail, the distributed pressure drop along each j-th tube, ($\Delta p_{T,j}$) can be evaluated as a quadratic function of the fluid mass flow rate $\dot{m}_j$ circulating in the pipe:

$$\Delta p_{T,j} = a_j \, \dot{m}_j^2 \tag{3}$$

where a_j is the j-th pipe quadratic coefficient ([Pa•s^2/kg^2]). In this work, for each tube, the coefficient a_j is obtained as the ratio between the nominal distributed pressure drop along the pipe and the corresponding nominal fluid mass flow rate raised to the square (inputs of the Simulink model).

Furthermore, the concentrated pressure loss through the i-th fan-coil, $\Delta p_{FC,i}$, is obtained as:

$$\Delta p_{FC,i} = k_i \, \dot{m}_i^2 \tag{4}$$

where k_i is the i-th fan-coil local coefficient ([Pa•s^2/kg^2]) and $\dot{m}_i$ is the fluid mass flow rate across the i-th fan-coil.

In simulations with traditional manual calibrating valves, the concentrated pressure drop due to the i-th balancing valve, $\Delta p_{VT,i}$ ([Pa]), is calculated as:

$$\Delta p_{VT,i} = \frac{10^5 \, \dot{V}_i^2}{k_{VT,i}^2} \tag{5}$$

where V_i is the volumetric flow rate across the i-th valve ([m^3/h]), and the i-th valve flow coefficient, $k_{VT,i}$ ([m^3/h]), is either equal to $k_{VS,i}$ (flow coefficient of the fully opened valve, reported by the technical datasheet) if the valve is open, or to $0.04 \times k_{VS,i}$ if it is closed. In fact, small flow leakages are present for this kind of valves even at zero opening position and, for this reason, a relative flow coefficient equal to 0.04 has been considered when no fluid flow is required. Similarly, the concentrated pressure drop across each fan-coil on–off control valve, $\Delta p_{V,i}$, is evaluated by means of an expression of the same form as Eq. (5).

For simulations with pressure independent control valves, it is worth reminding that both $\Delta p_{VT,i}$ and $\Delta p_{V,i}$ vanish and the concentrated head loss across each PICV is evaluated by means of the novel Simulink block described in the previous Section.

In addition, flow diverter and flow mixer blocks are employed for each of the seven ramifications and confluences of the network, respectively, in order to consider the mass flow rate changes in the different circuit sections by employing MATLAB S-functions.

The circulating pump elaborates a nominal mass flow rate $\dot{m}_{tot,nom}$ equal to the sum of the nominal mass flow rates of each parallel branch of the network, increasing the fluid pressure by the factor Δp_P, calculated as a function of the actual value of the total fluid mass flow rate, $\dot{m}_{tot}$:

$$\Delta p_P = c \, \dot{m}_{tot}^2 + l \, \dot{m}_{tot} + q \tag{6}$$

where c, l and q are the characteristic quadratic coefficient ([Pa•s^2/kg^2]), linear coefficient ([Pa•s/kg]) and pump head at zero flow ([Pa]), respectively. Values of these coefficients depend on the pump typology and can be determined by the pump technical datasheet. In particular, in case of fixed-speed pump control (pump quadratic characteristic curve), c, l and q coefficients have all non-zero values; in case of constant-pressure pump control (pump constant characteristic curve), the coefficients c and l are equal to zero, while q is equal to the nominal pump head ($\Delta p_{P,nom}$); finally, in case of proportional-pressure pump control (pump linear characteristic curve), the coefficient c is equal to zero, l is set equal to $\Delta p_{P,nom}/(2\dot{m}_{tot,nom})$ and q is set equal to $\Delta p_{P,nom}/2$, considering a recommendable slope of the pump characteristic curve. It is important to observe that, in case of constant- and proportional-pressure pump controls, when the variable-speed pump reaches its maximum speed, the characteristic curve follows the standard quadratic function.

The electric power input of the circulating pump, P_{el}, is evaluated as:

$$P_{el} = \frac{\Delta p_P \, \dot{m}_{tot}}{\rho \, \eta} \tag{7}$$

where ρ is the fluid density and η is the pump efficiency. In the present work, ρ is set equal to 1000 kg/m^3, ignoring the small influence of temperature on the density value as usual for this kind of systems.

For sake of simplicity, each fan-coil installed in the network has the same size (i.e., the thermal load of each thermal zone in the multi-storey building is the same). According to the required load, the commercial fan-coil ESTRO by the manufacturer Galletti [36] has been selected. With reference to the manufacturer technical datasheet, the thermal power of each terminal unit at design operating conditions (i.e., inlet/outlet water temperature = 45/40 °C, room air temperature = 20 °C) is 1910 W, the nominal water flow rate is equal to 330 l/h and the nominal head loss is 14.00 kPa. For each on–off control valve, a flow coefficient equal to 0.8 m^3/h and a nominal pressure drop of 17.22 kPa are assumed, respectively. The hydraulic distribution network has been simulated by considering, for each vertical section of supply and return lines (T1a-T8a and T1b-T8b in Fig. 3a, respectively), the nominal pressure drop values reported in Table 1, selected with reference to typical head losses along multi-storey circuits. In the same table, the nominal pressure drop values of the manual balancing valves (VT1-VT8 in Fig. 3a) are also reported. These values have been calculated according to valves' flow coefficients declared by the manufacturer to balance the distribution network for design operating conditions. As reported previously, PICVs have been considered as alternatives to manual balancing valves. In this case, the Δp_{min} value for each PICV is 15.99 kPa and the consequent authority of the valve in

Table 1
Nominal pressure drops in vertical tubes and manual balancing valves.

Vertical tube	Nominal pressure drop $\Delta p_{T,nom}$ [kPa]	Manual balancing valve	Nominal pressure drop $\Delta p_{VT,nom}$ [kPa]
T1a, T1b	1.42	VT1	10.38
T2a, T2b	0.62	VT2	9.15
T3a, T3b	0.66	VT3	7.84
T4a, T4b	0.64	VT4	6.56
T5a, T5b	0.62	VT5	5.32
T6a, T6b	0.69	VT6	3.95
T7a, T7b	0.65	VT7	2.65
T8a, T8b	0.59	VT8	1.47

the most hydraulically disadvantaged branch (the farthest from the pump) is about 50%, as recommended by technical guidelines.

An inverter-driven commercial pump has been selected to circulate the fluid within the distribution network. Different pump control logics have been considered: the device is able to work both at fixed speed or at variable speed, either with constant-pressure or proportional-pressure control strategies. The pump characteristic curve is reported in Fig. 5. As pointed out by that figure, the duty point is in the central part of the head-flow rate diagram, in correspondence of the best energy efficiency (see red point in Fig. 5). In detail, nominal values of volumetric water flow rate ($V_{tot,nom}$) and pump head ($\Delta p_{tot,nom}$) are equal to 2640 l/h and 44.44 kPa, respectively. The electric power input of the pump is reported as a function of the volumetric flow rate in Fig. 6, for different pumping speeds. The pump efficiency in correspondence of the duty point is equal to 50%.

The Simulink variable-step solver ode 45 (Dormand-Prince) has been selected, with adaptive algorithm, setting automatic absolute tolerance, 10^{-3} as relative tolerance and 2.8422×10^{-13} as time tolerance.

4. Results and discussion

The influence of using traditional manual balancing valves or PICVs on the effective hydraulic balance of the considered distribution network and consequent penalization on indoor thermal comfort conditions and pumping energy consumptions, have been evaluated numerically during off-design operating conditions. In detail, numerical simulations have been performed by considering some fan-coils of the emission sub-system switched off, while the others were delivering heat to the building. As already mentioned, this is a frequent condition in new and retrofitted buildings (especially in high performance thermally insulated buildings), where terminal units in a multi-zone HVAC system can work with variable thermal loads, due to different orientations and/or significant solar and internal gains. Both steady-state and time-dependent operating profiles of the emitters have been considered for the analysis.

4.1. Steady-state analysis

Three different time-constant operating conditions have been selected: i. all circuit branches open; ii. 3 branches off out of 8, namely with fan-coils FC3, FC5 and FC7 switched off (see Fig. 3a); iii. 5 branches off out of 8, namely considering fan-coils FC2, FC3, FC5, FC6 and FC7 switched off (see Fig. 3a). The hydraulic balance of the distribution network has been simulated with the above-mentioned operating conditions, by considering for each case the three circulating pump control logics described in the previous Section (i.e., fixed-speed, variable-speed with constant-pressure and variable-speed with proportional-pressure) and comparing the effectiveness of traditional manual calibrating valves and PICVs.

When manual valves are employed, water circulates in each terminal unit at the nominal fluid flow rate only in design conditions, when all fan-coils in the system are switched on and, consequently, all parallel branches of the distribution network are open (first operating condition). On the contrary, under off-design operations, when some terminal units are switched off (second and third working conditions), over- or under-flows may occur in remaining open branches. In this regard, the relative overflow, defined as the ratio between the effective and the nominal fluid flow rate, has been calculated for each studied case.

In particular, Fig. 7a shows the relative overflow of the whole distribution network, namely the total relative overflow, when manual balancing valves are installed in the system, as a function of the considered fan-coils operating profiles and pump control logics. As expected, the nominal fluid flow rate is observed in correspondence of design conditions (i.e., when all branches are open, blue columns in Fig. 7a), regardless of the pump speed control logic. On the contrary, the water flow rate rises with respect to the nominal value when 3 or 5 circuit branches are closed (orange and yellow columns in Fig. 7a, respectively) when a fixed-speed or a constant-pressure pump control is selected (left-hand and central part of the graph in Fig. 7a). More in detail, the magnitude of overflows increases significantly with a higher number of closed emitters and if the circulating pump works at fixed speed rather than at variable speed with constant-pressure control. Indeed, when 5 branches are closed and a fixed-speed circulating pump is installed in the network, the total relative overflow reaches + 30% compared to design conditions. On the other hand, the overall mass flow rate only increases by 7% when a variable-speed pump controlled at constant pressure is considered in correspondence of the same operating condition (see first and second yellow column in Fig. 7a, respectively).

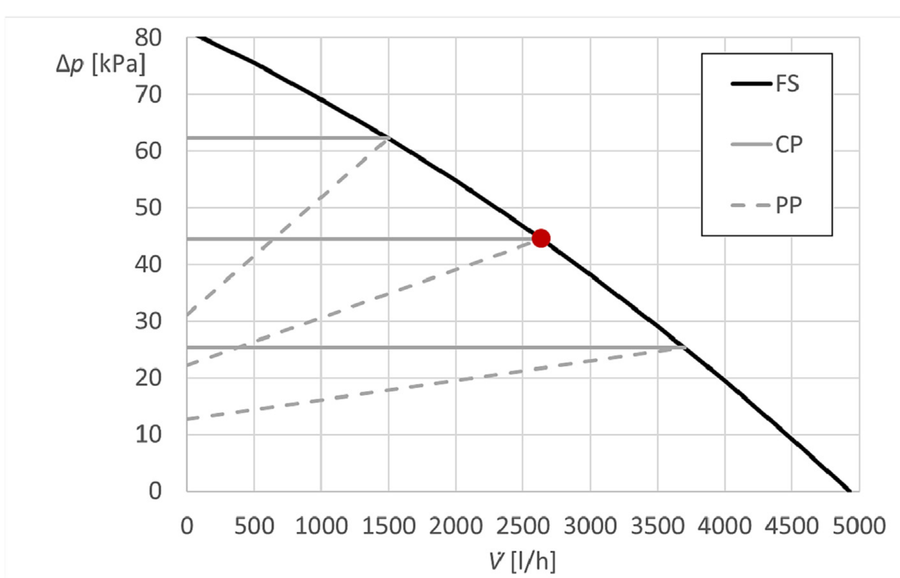


Fig. 5. Characteristic curve of the pump with respect to the adopted control logic: fixed-speed (FS), variable-speed with constant-pressure (CP) and proportional-pressure (PP) control.

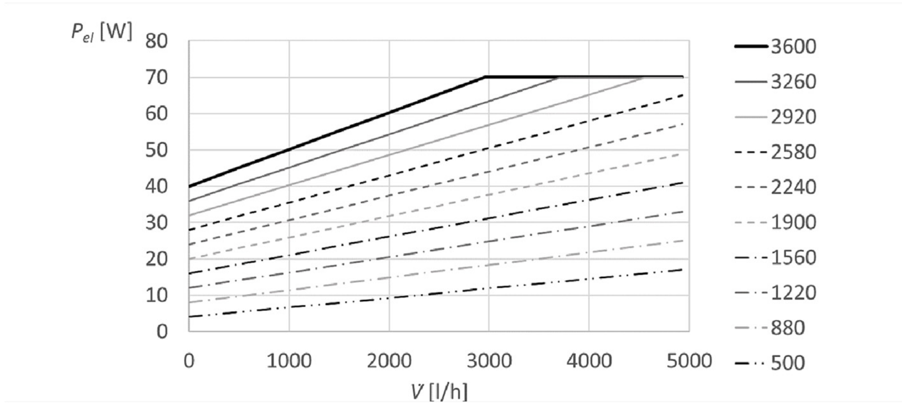


Fig. 6. Pump electric power at different volumetric flow rates and speeds [min^{-1}].

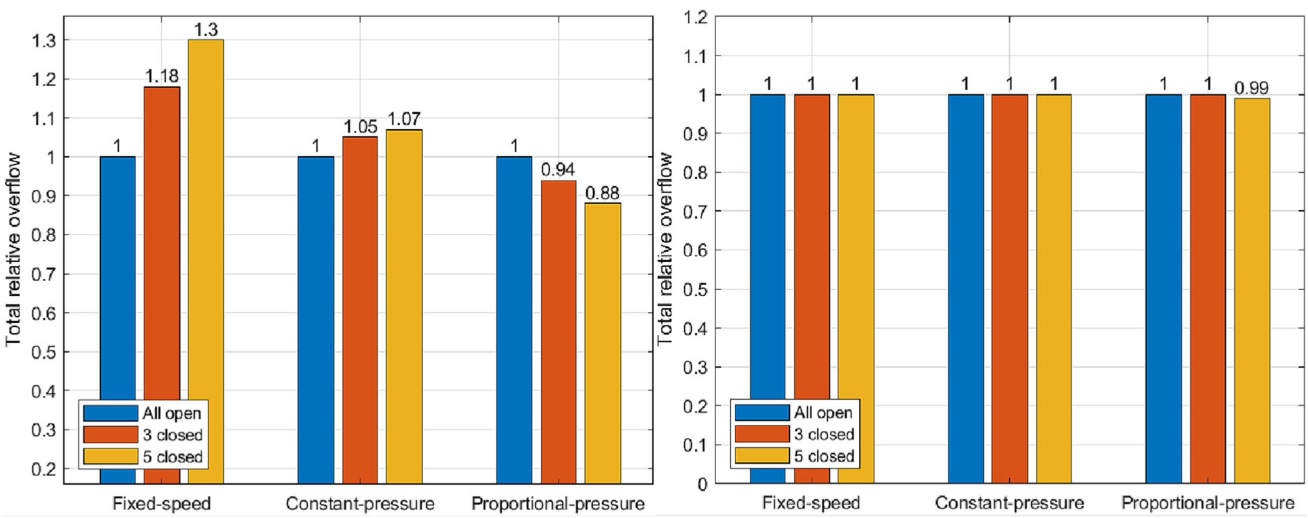


Fig. 7. Total relative overflow with different pump control logics in design and off-design conditions; manual balancing valves (a) and PICVs (b).

On the contrary, when some branches are closed and the circulating pump works with a proportional-pressure control logic, the fluid flow rates in remaining circuits of the network become lower than the nominal values (values of orange and yellow columns in the right-hand side of Fig. 7a lower than 1) and, as a consequence, typical problems linked to underflows can be observed within thermal zones of the building. This effect becomes even more pronounced, up to –12%, if the number of closed branches increases.

It can be concluded that under off-design working conditions, when some branches are closed and a proportional-pressure variable-speed pump is used in the network, nominal values of water flow rate cannot be guaranteed in the remaining part of the system when traditional manual valves are employed to balance hydraulically the network. As will be deeply investigated in following paragraphs, a reduction of water flow rate in some terminal units causes a not negligible worsening of indoor thermal comfort conditions.

The same analysis has been performed by replacing manual balancing valves with PICVs. Obtained results, reported in Fig. 7b, validate the Simulink model, since the mass flow rate through the valves remains constant. In detail, the findings of numerical simulations clearly evidence that with automatic balancing valves the nominal fluid flow rate is always assured, even if some of the fan-coils are switched off, except in the case of 5 closed branches and proportional-pressure pump control (last yellow column on the right-hand side of Fig. 7b). In this case, indeed, a very small underflow is present in the network (–1%), due to the fact that pressure drops on the PICVs fall below the minimum value that allows to keep the fluid flow rate constant and equal to the nominal value (Δp_{min}, see Fig. 1b). Consequently, it is evident how PICVs allow to guarantee optimal indoor comfort conditions in the building also for severe off-design conditions.

In order to identify which branches of the distribution network are subjected to the most severe over- or under-flows during off-design operating conditions, a branch by branch analysis must be performed. In this regard, Fig. 8 shows the relative overflow for each hydraulic branch of the network (namely the ratio between the effective and the nominal fluid flow rate), as a function of the different fan-coils operating profiles and pump control logics. It is worth remembering that branches 3, 5 and 7 are off in the second operating mode, while branches 2, 3, 5, 6 and 7 are closed in the third one.

From Fig. 8a it is evident that when some fan-coils are switched off, with a fixed-speed pump, the most penalized open branch is number 8, namely the farthest from the pump. In fact, this branch displays the major overflows, up to + 22% and + 36% with 3 and 5 fan-coils switched off, respectively (see last orange and yellow columns in Fig. 8a). With this pump control logic, the further the branch is from the pump, the greater the overflow. The same trend is observed with a variable-speed pump controlled by a constant-pressure logic (Fig. 8b). However, in this case smaller overflows (up to + 12% with 5 closed branches, see last yellow column in Fig. 8b)

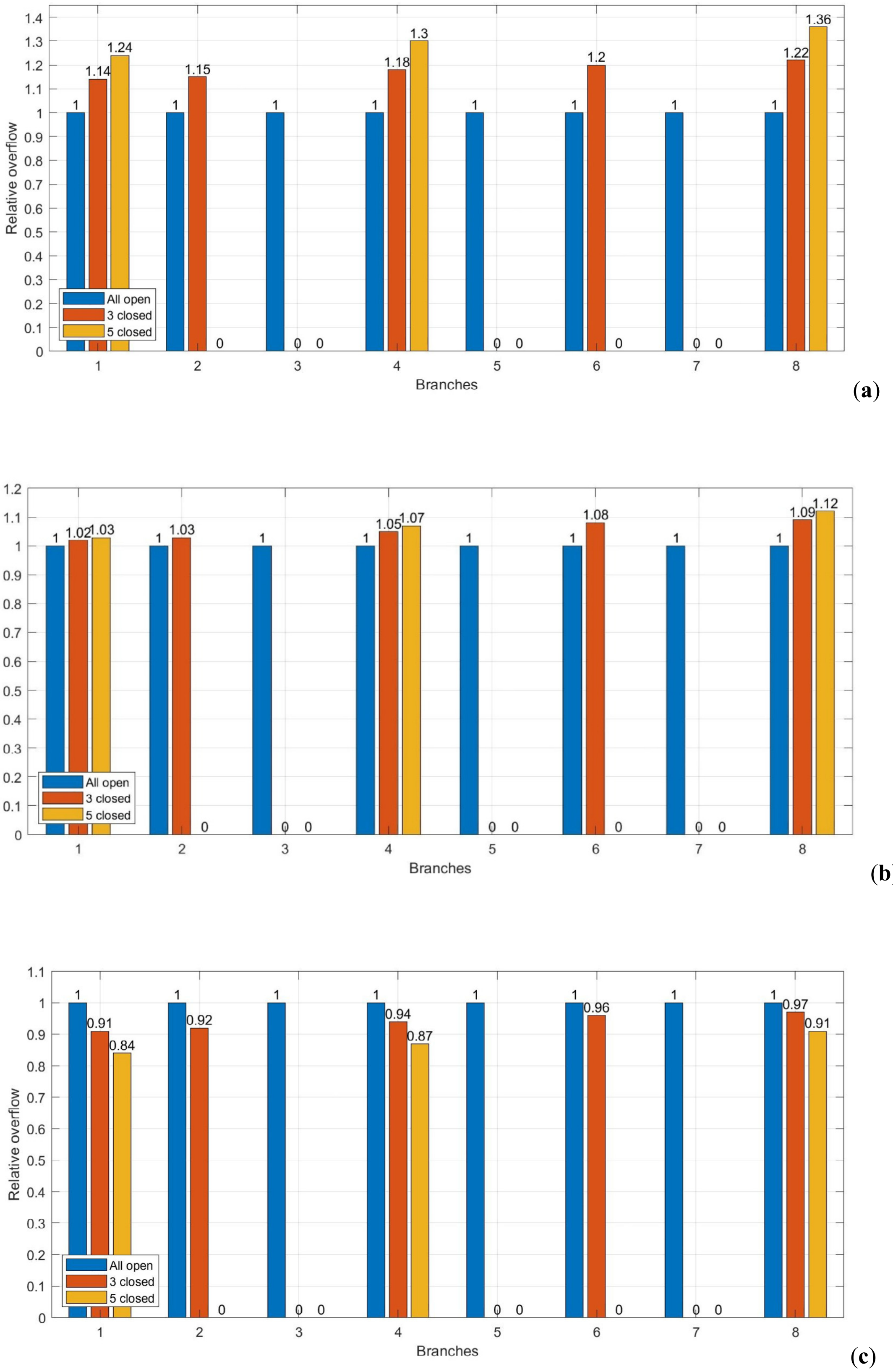


Fig. 8. Relative overflows of single branches using manual balancing valves, as a function of different fan-coils operating profiles and pump control logics: fixed-speed (a), constant-pressure (b) and proportional-pressure (c).

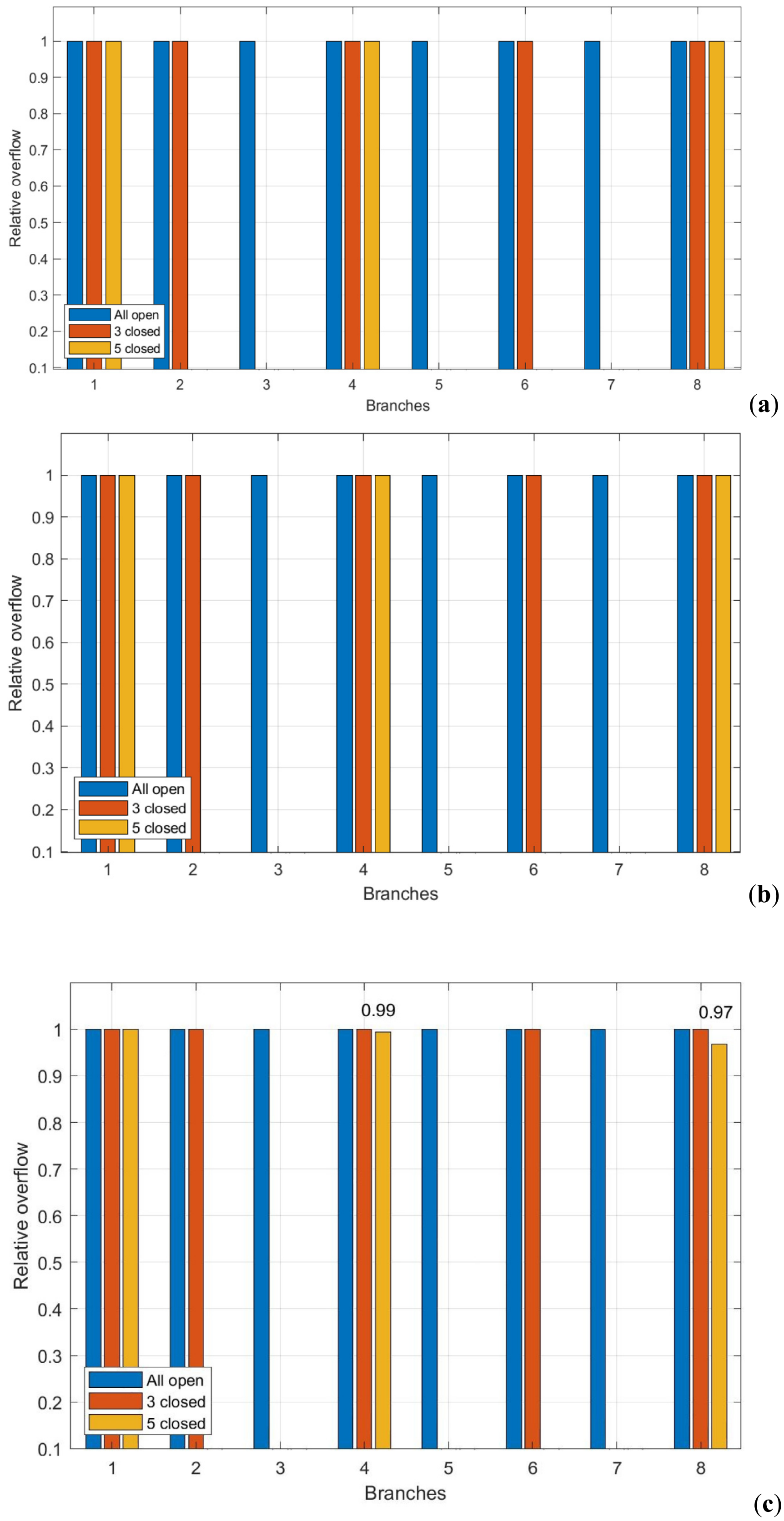


Fig. 9. Relative overflows of single branches using PICVs, as a function of different fan-coils operating profiles and pump control logics: fixed-speed (a), constant-pressure (b) and proportional-pressure (c).

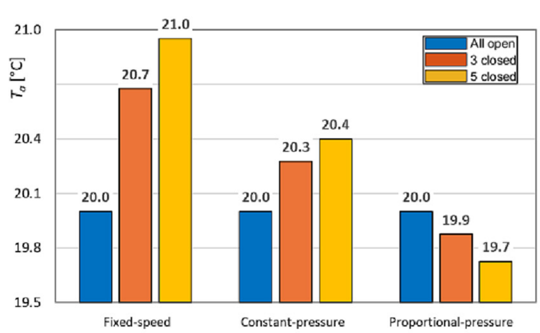
can be observed, according to the better pump control strategy. In contrast and as pointed out before, a proportional-pressure control logic for a variable-speed pump leads to the opposite problem, as evidenced by Fig. 8c: in off-design conditions the distribution network is significantly unbalanced and the fluid flow rate is reduced in open branches. In this case, the most penalized circuit is number 1, i.e., the closest to the pump (relative overflow up to −16% for five closed branches, see first yellow column in Fig. 8c). To summarize, the adoption of variable-speed pumps in heating systems in which the hydraulic balance is guaranteed by manual valves allows to reduce significantly the network unbalance at partial load, even if considerable over- and under-flows occur in the most penalized branches.

Relative overflows obtained in each branch of the circuit by adopting PICVs are plotted in Fig. 9. Results confirm that the distribution network is balanced hydraulically (i.e., the water flow rate is equal to the design value in each branch) even if some branches are closed, when a fixed-speed or a constant-pressure variable-speed pump is adopted (relative overflows always equal to 1 in Fig. 9a, b), thus highlighting the optimal behavior of PICVs in distribution networks which use fixed-speed or constant-pressure variable-speed pumps. On the other hand, with a proportional-pressure variable-speed pump, not negligible underflows may occur in the part of the network far from the pump (i.e., branch 8) when the number of fan-coils switched off is high. This result is caused by the fact that, when some branches are off, the proportional-pressure control yields a decrease of the pump head that mainly affects the farthest part of the circuit, where pressure drops across PICVs risk to fall below Δp_{min} and valves cannot guarantee the nominal fluid flow rate. However, in the considered case study, the obtained underflows do not overcome −3% (see last yellow column in Fig. 9c).

These results highlight that potential benefits achievable by employing PICVs are dependent on:

- a proper pump selection: with fixed-speed and constant-pressure variable-speed pumps, PICVs properly designed assure the nominal fluid flow rate in every branch also during off-design operation;
- a proper PICVs selection: valves with low values of Δp_{min} are more likely to prevent underflows during off-design working conditions, whether a proportional-pressure variable-speed pump is selected to reduce the pump energy consumption.

The problem of over- or under-flows in some terminal units of the hydronic network, in turn, results respectively in over- or under-heating of rooms in which these emitters are located for space heating/cooling, with consequent indoor thermal discomfort. In addition, overheating linked to overflows means also waste of energy. With the aim of quantifying the room temperature difference with respect to the internal set-point, caused by over- or under-flows, the fan-coil heating capacity has been evaluated as a function of the flow rate according to the manufacturer technical data. Consequently, the effective room temperature, T_a, can be simply evaluated as:



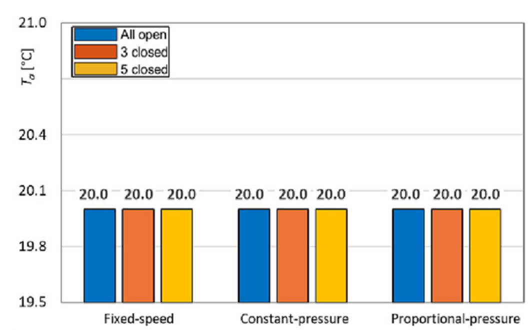


Fig. 10. Room temperature values with different fan-coils operating conditions and pump control logics; manual balancing valves (a) and PICVs (b).

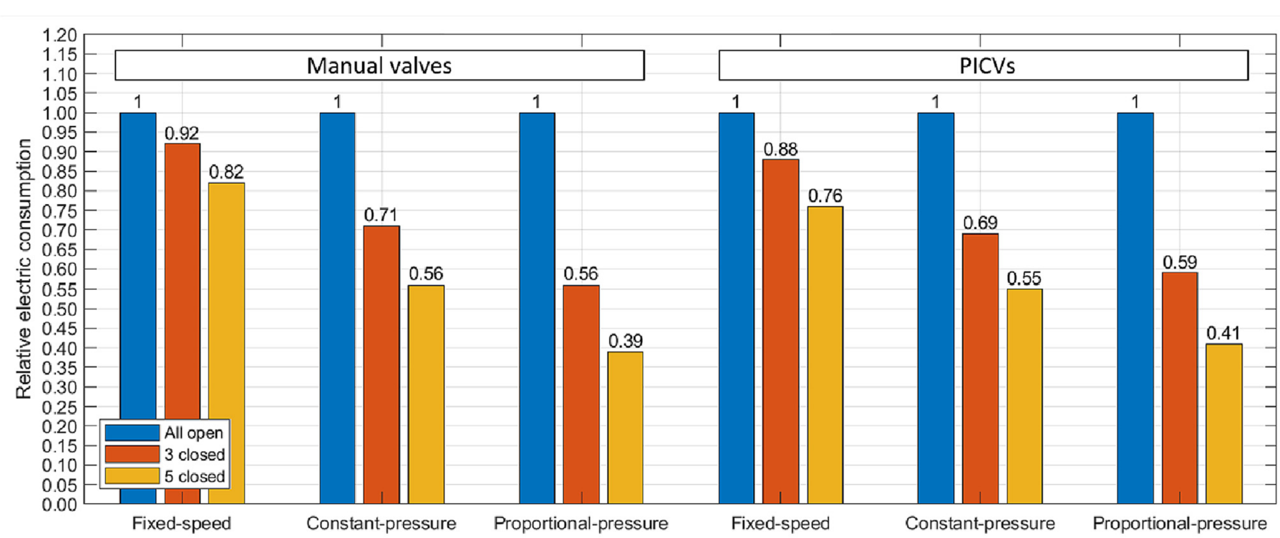


Fig. 11. Relative electric consumption of the pump, with manual balancing valves and PICVs as a function of the pump control logic and fan-coil operating conditions.

Table 2
Electric power input of the pump [W], with manual balancing valves and PICVs as a function of the pump control logic and fan-coil operating conditions.

	Manual valves			PICVs		
	All open	3 closed	5 closed	All open	3 closed	5 closed
Fixed-speed	66.5	61.3	54.6	66.5	58.6	50.4
Constant-pressure	66.5	47.2	37.3	66.5	45.8	36.3
Proportional-pressure	66.5	37.4	25.8	66.5	39.4	27.5

$$T_a = T_d + (T_{set} - T_d)\frac{Q_a}{Q_{nom}} \quad (8)$$

where T_d is the external design temperature, T_{set} is the internal set-point temperature and Q_a and Q_{nom} are the effective (i.e., calculated with the effective water flow rate due to the network unbalance) and nominal fan-coil heating capacity values, respectively.

In the present study, T_{set} has been set equal to 20 °C (as recommended for the winter season by the Italian regulation, namely D.P. R. n. 412/93), T_d has been set equal to –5 °C (external design temperature of Bologna, Northern Italy, according to UNI 10349–1 [37]) and the ratio between Q_a and Q_{nom} has been evaluated through the fan-coil manufacturer technical data by considering the average value of over- or under-flow among the distribution network.

Fig. 10 reports the obtained results in terms of room temperature T_a with manual balancing valves and PICVs (Fig. 10a and Fig. 10b, respectively), as a function of fan-coils operating profiles and pump control strategies considered in this work. It is evident how, with manual valves, overheating in heated thermal zones is a linear function of the total overflow in the distribution network. Indeed, the major room temperature increases (up to + 1 K) are obtained in correspondence of the highest overflows, namely with a large number of closed terminal units and with the circulating pump working at fixed speed (see first yellow column in Fig. 10a). It is worth mentioning that 1 K of overheating corresponds to 6–10% of additional energy consumptions [38,39]. Furthermore, with manual calibrating valves and a proportional-pressure variable-speed pump, the room set-point temperature (i.e., 20 °C) cannot be guaranteed if some emitters in the hydronic network are switched off (compare last columns of Fig. 7a, where underheating values up to –0.3 K are reached).

Unlike manual calibrating valves, PICVs are always able to guarantee the internal comfort conditions, keeping the room temperature at the desired value both in design and off-design operations, as evidenced by Fig. 10b. It is evident that even the small underflow observed with 5 terminal units switched off and a proportional-pressure pump control (see Fig. 7b) does not affect significantly the room temperature.

Fig. 11 shows, for each studied configuration, the relative electric consumption of the circulating pump with respect to the reference case, characterized by all branches open (design condition). In Table 2 the corresponding electric power input of the circulating pump (see Eq. (7)) is reported for each case. As expected, the more terminal units switched off, the lower the pump electric consumption, especially if this device works at variable speed and, in particular, with a proportional-pressure control logic. Indeed, in the latter case, a lower total flow rate corresponds to a lower pump head (see the pump characteristic curve in Fig. 5), with a consequent significant decrease of the pump electric power input, up to 61%.

In general, adopting PICVs in place of traditional manual calibrating valves allows to slightly reduce the pump electric con-

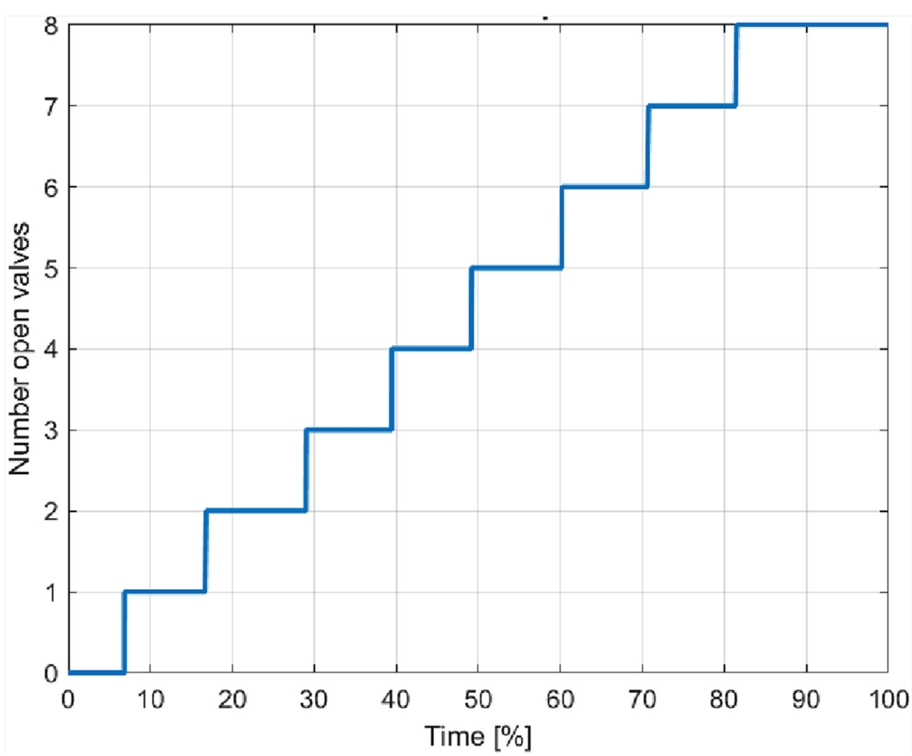


Fig. 12. Cumulative percentage of the number of open valves during the heating season for the selected operating profile.

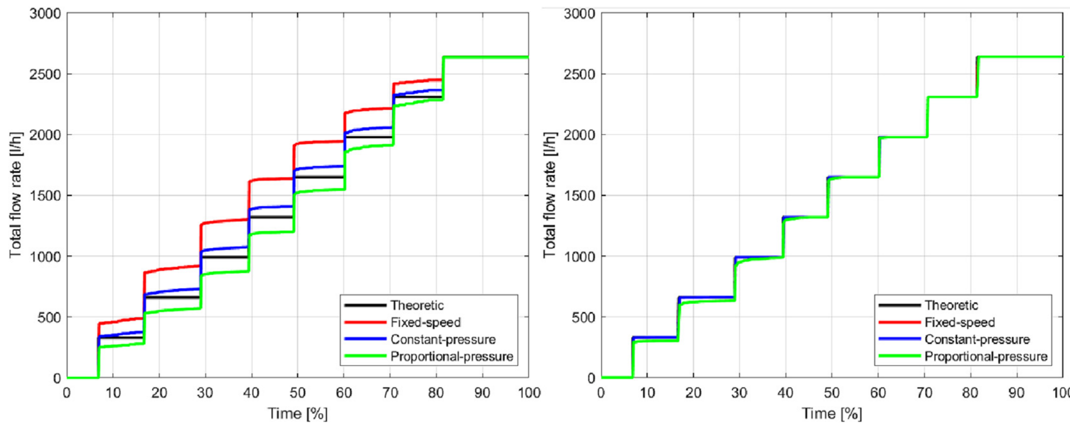


Fig. 13. Cumulative percentage of the total fluid flow rate: theoretical trend and profiles obtained with fixed-speed and variable-speed pumps; manual balancing valves (a) and PICVs (b).

sumption when some emitters are switched off, up to more than 7%, if the pump works at fixed-speed (i.e., by comparing the first yellow column on the left-hand side of Fig. 11 with the corresponding one on the right-hand side). An exception arises if a proportional-pressure pump control is selected, since with PICVs the pump electric consumption shows a very slight increase. As a consequence, given also that with this pump control logic PICVs cannot assure the nominal fluid flow rate in each terminal unit for all operating conditions, a proportional-pressure pump control should not be adopted when PICVs are used in a hydronic network.

However, it is important to highlight that, with manual valves, a lower pump electric consumption due to the closing of some branches is not linked to the same internal comfort conditions, because undesired changes of room temperature occur as a consequence of the hydraulic unbalance of the distribution network. On the contrary, the pump electric power input can be decreased in presence of PICVs by keeping the network balanced and, for this reason, room temperature values are equal to the imposed set-point for all operating conditions.

Finally, an additional benefit worth to be mentioned linked to the use of PICVs is the dramatic reduction of time needed for the hydraulic balancing of distribution networks. In fact, with PICVs the hydraulic balance is made automatically, while with manual valves the correct balance can be guaranteed only after an iterative process which introduces significant delays during the commissioning phase of the heating system (i.e., the adjustment of a single valve opening influences all the network, which has to be balanced again in a continuous process).

4.2. Seasonal analysis

In order to evaluate the benefits of PICVs along the entire heating season, simulations have been performed by considering the dynamic behavior of the hydraulic network during the whole heating period. In particular, a variable profile for the operation of fancoils has been selected. The cumulative percentage of the number of open valves during the heating season is shown in Fig. 12. The chosen operating profile is typical of new and retrofitted buildings, highly insulated, where the contemporary operation of a high number of terminal units is rare, due to different orientations of the building and the strong influence of solar or internal gains. From Fig. 12 it can be noted that for more than 70% of the heating period some branches are open while others are off. As pointed out before, PICVs can provide their beneficial effects for a significant part of the season.

Dynamic simulations have been performed considering both traditional manual calibrating valves and PICVs. In Fig. 13a, the cumulative percentage of the optimal total flow rate (theoretical nominal value) is compared to the corresponding trends obtained with manual valves and the three pumps considered in this work.

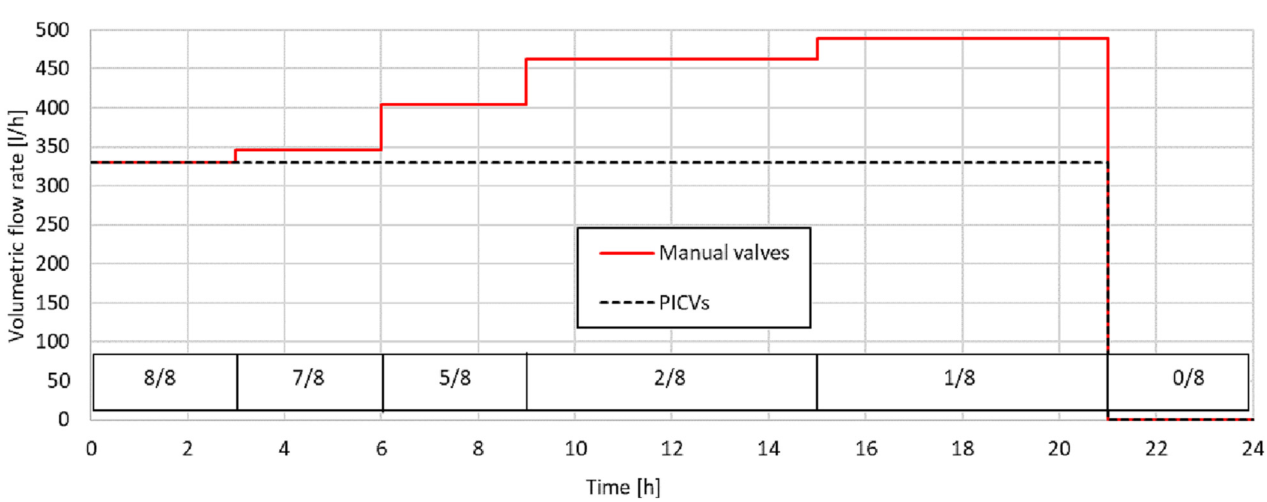


Fig. 14. Volumetric flow rate of the most penalized branch during one day with the fixed-speed pump and the specified number of open valves; trends obtained with manual balancing valves and PICVs.

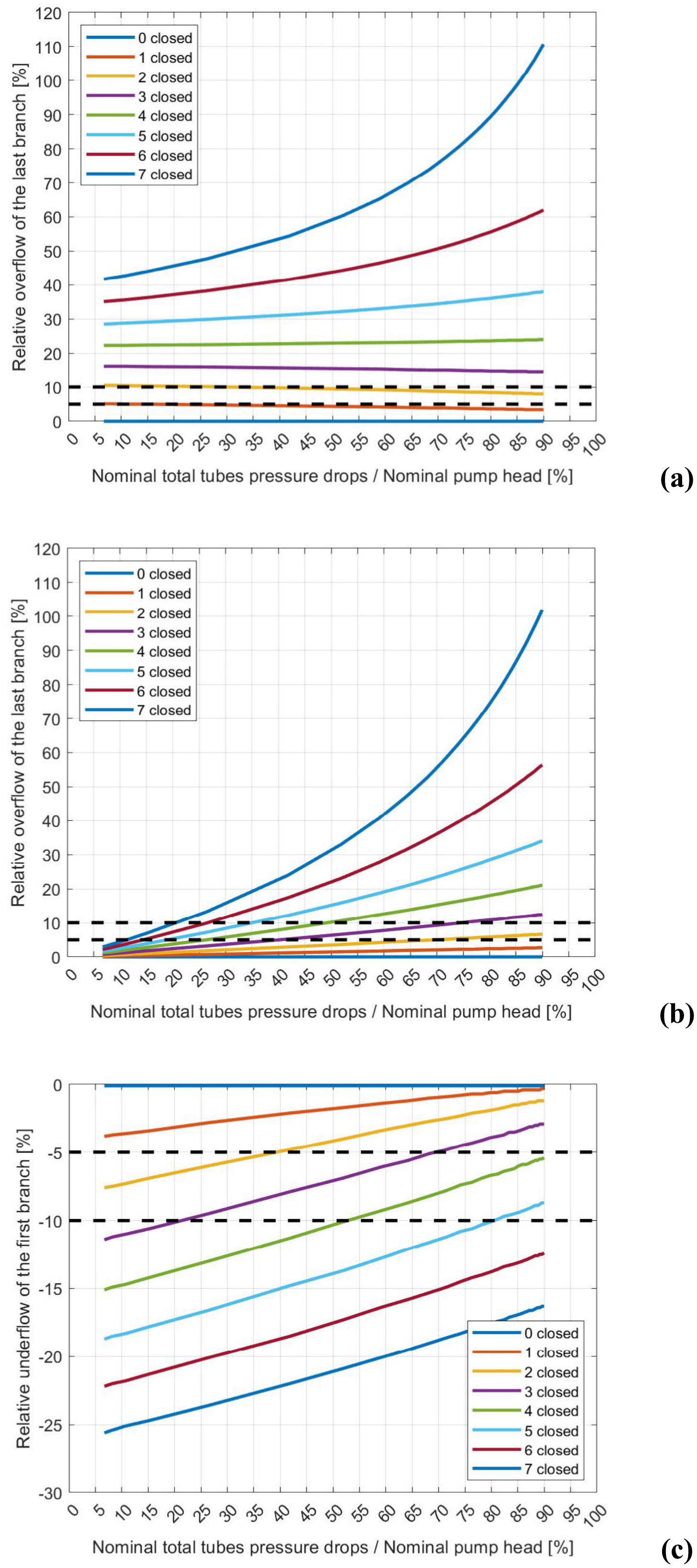


Fig. 15. Relative overflow of the most penalized branch as a function of the ratio between nominal total pressure drops of tubes and nominal pump head; manual balancing valves and fixed-speed pump (a), variable-speed pump with constant-pressure (b) and proportional-pressure (c) control logic.

One can note how the nominal volumetric flow rate (black line in Fig. 13a) cannot be guaranteed in the hydronic network by any of the pump control logics during most part of the season, except for operating conditions with all fan-coils switched on. In detail, significant overflows, up to 300 l/h in the entire network, are present with a fixed-speed circulating pump (red curve in Fig. 13a), while a variable-speed pump with constant-pressure control allows to mitigate the problem, with a total overflow reduced to about 80 l/h (blue line in Fig. 13a). As reported out before for the steady-state off-design analysis, a variable-speed pump with proportional-pressure control yields underflows close to 120 l/h in the network (green curve in Fig. 13a).

On the contrary, simulations carried out by employing PICVs (Fig. 13b) point out that the distribution network is balanced hydraulically along the whole heating season, as highlighted by the negligible difference among theoretical and effective trends of cumulative total flow rate. A very slight underflow, below 30 l/h, is obtained only when a variable-speed pump with proportional-pressure control is adopted, in correspondence of a high number of branches off (see the first part of the green curve in Fig. 13b). This result confirms that the proportional-pressure pump control is not recommended if pressure independent automatic control valves are inserted in a hydraulic circuit.

In order to better understand how the valves operating profile impacts the system behavior, Fig. 14 shows the time dependent volumetric flow rate of the most penalized branch (the farthest from the pump) during one day of the heating period, obtained with the fixed-speed pump and manual balancing valves or PICVs. Due to the variable number of open valves during the day, with manual valves the nominal flow rate of 330 l/h is not guaranteed in the fan-coil for most of the time, whereas PICVs are always able to keep the flow rate at its nominal value (compare red and black curves in Fig. 14). In particular, when the analyzed branch is the only one open, a volumetric flow rate almost 50% higher than the nominal one is obtained with manual valves (see the red curve with 1/8 open valves in Fig. 14).

4.3. Guidelines on the adoption of PICVs

Finally, simulations have been performed to investigate in which hydronic networks the adoption of pressure independent control valves is more convenient. In this regard, the same distribution network, composed by 8 parallel branches, described in previous Sections of this paper has been considered. In this analysis, different operating conditions of the network have been simulated. In detail, the flow rate of the most hydraulically penalized branch has been calculated numerically, by considering different pipes pressure drops and an increasing number of fan-coils switched off, from 0 (design condition) to 7 (only the fan-coil included in the most penalized branch is switched on).

Fig. 15 shows the relative over- or under-flow of the most penalized branch as a function of the ratio between the nominal total pressure drops in the tubes and the nominal pump head, for all simulated conditions. Fig. 15a and 15b refer to a fixed-speed pump and a variable-speed pump with constant-pressure control, respectively, where the most penalized branch is the farthest from the pump (branch 8). Fig. 15c refers to a variable-speed pump with proportional-pressure control, where the most penalized branch is the closest to the pump (branch 1), as demonstrated previously.

Typically, the thermal power supplied by a fan-coil is not very susceptible to variations in the water flow rate, if these are limited within 10% and 5% of the nominal value in the heating season and cooling season, respectively [36]. As a consequence, if the variation of water flow rate at the fan-coil inlet falls within these threshold values (highlighted in Fig. 15 through dashed lines), the adoption of PICVs, that are more expensive than traditional manual valves, is not convenient. On the contrary, if significant variations of the flow rate are obtained in the network, PICVs are crucial to keep the nominal flow rate and, consequently, to assure optimal indoor thermal comfort conditions in all zones.

With a fixed-speed pump (Fig. 15a), the overflow in the most penalized branch tends to increase rapidly when the network passes from a design to an off-design operating condition. In fact, with 3 or more fan-coils switched off in the system, overflows falling outside the tolerance limit for the heating season (i.e., 10% of the nominal value) have been obtained. Moreover, results point out that when a limited number of fan-coils is switched on, overflows rise dramatically when the total pressure drops of pipes represent a significant share of the pump head, higher than 50–60%. Similar conclusions can be outlined also when a variable-speed pump with constant-pressure control is installed in the system (Fig. 15b). However, in this case overflows falling outside the tolerance limit occur only for distribution networks characterized by high distributed and accidental pressure drops in pipes, with a low number of fan-coils switched on. In fact, when tubes pressure losses represent less than 20% of the nominal pump head, the relative overflow of the most penalized branch falls within the tolerance area even with 7 fan-coils switched off out of 8 (blue line in Fig. 15b). Nevertheless, the higher the pressure drops of pipes, the higher the increase of fluid flow rate and the more frequent the operating conditions linked to overflows. In conclusion, in distribution networks characterized by large pressure drops in pipes and based on a fixed-speed or a variable-speed pump with constant-pressure control, PICVs are very useful to replace manual control valves to prevent significant overflows in the most penalized part of the network.

On the other hand, with a proportional-pressure control for a variable-speed pump (Fig. 15c), significant underflows, up to −25% of the nominal value, occur in the most penalized branch (the closest to the pump) for several operating conditions, especially when total pressure drops of pipes account for less than 25% of the pump head.

5. Conclusions

In the present work, a new Simulink block for the simulation of pressure independent control valves (PICVs), used for the dynamic hydraulic balance of hydronic distribution networks in HVAC systems, has been developed. The numerical model is freely available to the scientific community and is compatible with the most popular open-source Simulink libraries CARNOT and ALMABuild.

The adoption of this tool has been demonstrated by simulating an application case study, in which the performance of PICVs has been compared to that of traditional manual balancing valves, with reference to the distribution network of a HVAC system based on fan-coils and coupled to a multi-storey residential building. Simulations of the hydronic network have been performed for off-design operating conditions, evaluating the achievable benefits of PICVs for both a static analysis and on a seasonal basis. Obtained results show that, if some fan-coils of the HVAC system are switched off and, consequently, some branches of the hydraulic circuit are closed, the hydronic distribution network is hydraulically unbalanced when traditional manual balancing valves are inserted in the circuit. Therefore, the nominal flow rate cannot be assured in all branches of the network, with consequent indoor thermal discomfort and energy waste. In particular, overflows occur in branches far from the circulating pump when a fixed-speed pump or a variable-speed pump with constant-pressure control logic are installed in the circuit, whereas in case of a variable-speed pump with proportional-pressure control logic, underflows occur in branches close to the pump for off-design operating conditions. On the contrary, the hydronic network is optimally balanced when PICVs are installed in the circuit, even if a significant part of emitters is switched off. The nominal fluid flow rate can be guaranteed in each branch, avoiding overheating of thermal zones and yielding small reductions in the pump electric consumption, ensuring at the same time optimal indoor thermal comfort conditions for building users. However, a proportional-pressure control logic of the circulating pump yields a risk of going below the minimum value of the differential pressure across the valve that allows a constant flow rate. As a consequence, with the adoption of PICVs the setting of a proportional-pressure control logic for the pumps is not recommended.

Finally, it has been demonstrated how the positive effects of PICVs are emphasized in case of hydronic networks characterized by high pressure drops in pipes (both distributed and accidental) and, especially, if a fixed-speed circulating pump is employed. Since large part of multi-family residential buildings is based on HVAC systems having a distribution network with the above-mentioned features, PICVs can play a significant role in the refurbishment of this kind of systems in the next future, contributing to reduce significantly the environmental impact of heating systems in our cities. Future developments of the present work will encompass the analysis of the integrated building-HVAC system, with different sizes of the terminal units: the operation of a large 6-storey building with a 16 vertical-tubes distribution network will be monitored and studied with the developed numerical tool, by quantifying the savings on pumping energy consumptions linked to the replacement of manual balancing valves with PICVs.

Data availability

The used data are described in the paper and free.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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