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Effect of flow pulsation on energy consumption of a radiator in a centrally heated building

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Abstract

Currently used central heating systems utilise radiators operating on constant flow with on/off control strategy that consumes significant amount of energy. Therefore, enhancing the thermal performance of central heating systems can play a major role in reducing buildings' energy consumption. This work aims to improve the performance of hot water heating system by changing the radiator inlet flow strategy from constant to pulsed flow without changing the installed radiator or compromising the user comfort. Using the Simulink/Matlab software, a mathematical model of a room with single radiator was developed. This model couples the thermal performance of the radiator, air within the heated space, walls and windows. Pulsed flow with amplitudes ranging from 0.024 to 0.048 kg/s, frequencies ranging from 0.0017 to 0.017 Hz and duty cycles ranging from 50 to 80% were investigated and compared with the constant flow. Results showed that up to 22% of the energy consumed for heating can be saved by changing the constant flow to pulsed flow. In addition to the energy saving achieved using this pulsed flow, the indoor temperature response is also shortened from 600 s for the constant flow case to 450 s. Further improvement was achieved by introducing the proportional integral differential (PID) control system with the pulsed flow where the results showed that the fluctuation in the indoor temperature decreased to ± 1 K of the desired temperature of 20°C and energy saving can be increased to 27%.

Keywords: energy saving; pulsed flow; PID control; thermal performance of heated space; Matlab/Simulink

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1 INTRODUCTION

The European Union is demanding for a reduction of energy consumption by 20% before 2020. In order to satisfy such requirements, the UK has targeted the largest single source of energy consumption, residential buildings, to achieve 52% reduction of its total energy consumption [1]. Enhancing the efficiency of domestic central heating systems is an important factor in achieving this target [2]. Many studies have been performed to improve the design and performance of the heating appliances, such as panel radiators, floor heating and wall heating equipment. John *et al.* [2] showed that the heat transfer performance of a panel radiator inside a room is affected by the ventilation system. They found that the panel radiator convection heat transfer output can be improved by increasing air flow on its surfaces. Bangert [3] studied the feasibility of enhancing the heat output from the panel radiator by coating the wall behind the radiator with different

colours. He concluded that the wall coated with high emissivity (smooth black surface) material produced the highest heat output from the panel radiator. Ploskic and Holmberg [4] investigated the use of baseboard device integrated air supply to enhance a hydronic heating system. They aimed to minimise the supply temperature by pre-heating the incoming ventilation airflow and concluded that their proposed system can produce up to 21% more heat output compared with the conventional hydronic baseboard heating system. Myhren and Holmberg [5] carried out a study to enhance the energy efficiency in exhaust-ventilated buildings with warm water heating system using ventilation radiator combined with heat emission device. A CFD model of the proposed system was developed and the results showed that the heat output can be increased by 20% comparing with traditional radiators.

The control strategy used in any domestic central heating system can play a major role in improving its energy consumption and carbon emissions. Current control technology commonly used in

residential homes and offices in the UK are based on switching the radiators on/off at certain intervals using radiator valves, thermostats and timing programmers. Adaptive control was studied by Adolph *et al.* [6] to reduce the energy demand of building based on the user's profile yielding that $\sim 11\%$ of the energy was saved. A wireless control technique for the thermostat was developed by Liu *et al.* [7] and concluded that $\sim 30\%$ of the energy can be saved. Lehmann *et al.* [8] used a model predictive control (MPC) technique to improve the accuracy of the control system and showed promising energy savings. Modulating the air-to-water heat pump system used for residential heating was studied by Verhelsta *et al.* [9]. The study aimed to optimise the performance of a heat pump-based hydronic central heating system with results showing 5% energy saving.

A number of studies have been carried out by various researchers to enhance the heat transfer of various heat-emitting devices. Flow pulsation is a method of heat transfer enhancement studied for various industrial applications including heat exchangers, pulse combustors, electronic cooling devices and cooling systems of nuclear reactors [10, 11]. A study was carried out by Zohir [12] to enhance the heat transfer of the parallel and counter flow heat exchangers using ball valve as pulsar device. Results showed that using pulsed flow the heat transfer can be enhanced by 20% for the parallel flow heat exchangers and 90% for the counter flow heat exchangers. The effect of flow pulsation in heat transfer enhancement of the double pipe steam water heat exchangers using solenoid valve triggered by pressure switch was studied by Lemlich [13]. The test was performed at Reynolds number ranging from 500 to 5000 and frequency of 1.5 Hz and results showed that the overall heat transfer coefficient was increased by 80% depending on the distance between the solenoid valve and the test section. Shuai *et al.* [14] experimentally investigated the effects of pulsed perturbation on convective heat transfer for laminar flow in co-axial cylindrical tube heat exchangers. The tests were performed at low Reynolds number ranging from 150 to 1000 and frequency ranging from 0 to 2 Hz with the

reciprocating pump used as a pulsating device. Results showed that the heat transfer coefficient can be enhanced by up to 300%. A study was carried out to investigate the potential of embedded pulsating heat pipes for space or terrestrial applications [15]. The enhancement was explored based on the Biot number to minimise the conductive resistance of the thermal radiator. They concluded that effective thermal conductivities of 400–2300 W/mK can be achieved due to the applied pulsed flow. Sailor *et al.* [16] carried out a series of experiments to compare the heat transfer enhancement for pulsed air jet and steady flow air jet. Results showed that up to 50% heat transfer enhancement can be achieved by the pulsed jet relative to steady flow jet [16].

Although various studies have been carried out to enhance the heat transfer of heating or cooling devices using flow pulsation for various industrial applications, little attention has been given to the panel radiator-based hydronic heating systems. Therefore, this work aims to investigate the effects of different flow pulsation schemes on the performance of the hydronic radiators of domestic central heating systems and develop a control strategy that can achieve energy saving without compromising the user comfort. This work involves mathematical modelling of various pulsed flow conditions using the Matlab/Simulink software. The pulsed flow with amplitudes ranging from 0.024 to 0.048 kg/s, frequencies ranging from 0.0017 to 0.017 Hz and duty cycles ranging from 50 to 80% were investigated and compared with the constant flow. The energy saving achieved by the pulsed flow to heat the space to the target comfort temperature and the fluctuation of the indoor room temperature around this target temperature were used to determine the best pulsed flow conditions.

2 MATHEMATICAL MODELLING

Figure 1 shows a schematic diagram for the heat transfer processes for a heated space with hydronic radiator including the

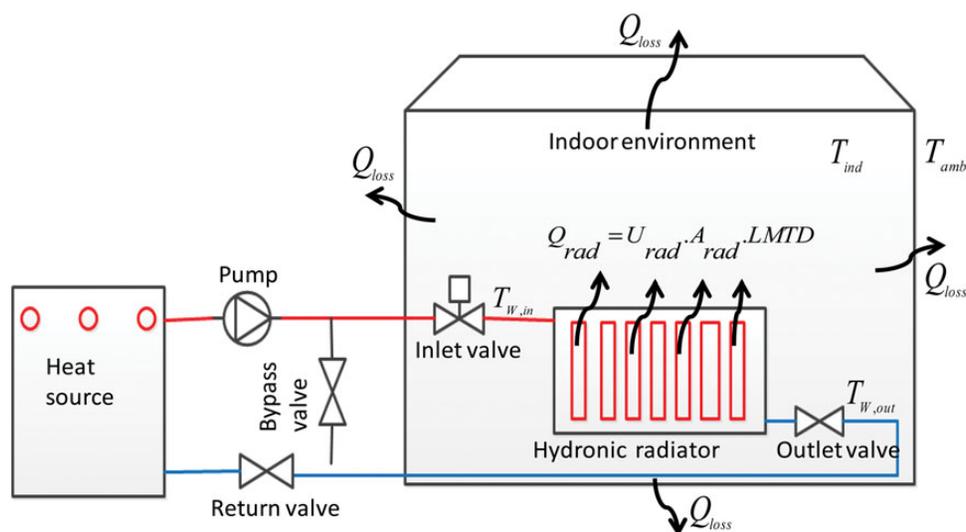


Figure 1. Schematic diagram of heat transfer processes in a space heated with hydronic panel radiators.

heat loss through the walls, ceiling, floor and windows due to the temperature difference between the heated space and the ambient, heat input from the heater radiator to the room and the network of pipes in a hydronic central heating system. The heat transfer processes to/from the radiator are described by the input energy (\dot{Q}_w), output energy (\dot{Q}_{rad}) and dynamic energy balance as shown in the following equations, respectively.

$$\dot{Q}_w = \dot{m}_w \cdot C_w \cdot \Delta T \quad (1)$$

$$\dot{Q}_{rad} = U_{rad} \cdot A_{rad} \cdot LMTD \quad (2)$$

$$(M_{rad} \cdot C_{rad} + M_w \cdot C_w) \cdot \frac{dT_{w,m}}{dt} = \dot{m}_w \cdot C_w \cdot (\Delta T) - U_{rad} \cdot A_{rad} \cdot LMTD$$

where

$$T_{w,m} = \frac{T_{w,in} + T_{w,out}}{2} \quad (3)$$

where ΔT is the difference between the water inlet and outlet temperatures of the radiator; A_{rad} is Radiator heat transfer surface area; C_w the water specific heat capacity; C_{rad} the radiator skin metal-specific heat; \dot{m}_w is the water mass flow rate; M_w the mass of water inside the radiator and M_{rad} the mass of the radiator skin. LMTD is the log mean temperature difference between the variation of water temperature inside the radiator ($T_{w,in}$ and $T_{w,out}$) and the bulk indoor air temperature (T_{ind}) inside the room given by the following equation as

$$LMTD = \frac{T_{w,in} - T_{w,out}}{\ln(T_{w,in} - T_{ind} / T_{w,out} - T_{ind})} \quad (4)$$

The overall heat transfer coefficient of the radiator (U_{rad}) is calculated by the following equation [17]:

$$\frac{1}{U_{rad}} = \frac{1}{\alpha_{ins}} + \frac{l_{rad}}{k_{rad}} + \frac{1}{\alpha_{out}} \quad (5)$$

where α_{ins} is the convective heat transfer coefficient between hot water and the radiator metal skins; k_{rad} the radiator skins conductivity; l_{rad} the thickness of the radiator skins and α_{out} the sum of radiative and convective heat transfer coefficient between the radiator and the air in the room ($\alpha_{out} = \alpha_{conv} + \alpha_{radiative}$). The heat transfer coefficient (U_{rad}) for the panel heater radiator is $\sim 7-10 \text{ W m}^{-2} \text{ K}^{-1}$ [17].

The energy balance of the heated room coupled with hydronic heating radiator is described using the following equation:

$$\dot{Q}_{rad} + \dot{Q}_{int,gain} = \dot{Q}_{trans} + \dot{Q}_{vent} + \dot{Q}_{inf} \quad (6)$$

The left-hand side of Equation (6) represents the heat gain from the heat source (radiator) and internal gains, while the right-hand side represents the heat loss from the room by the building structure (Equation 7); ventilation (Equation 8) and infiltrations

(Equation 9) [18–20].

$$\begin{aligned} \dot{Q}_{trans} &= (U_{win} \cdot A_{win} + U_{wall} \cdot A_{wall} + U_{rf} \cdot A_{rf} + U_{flr} \cdot A_{flr}) \\ &\cdot (T_{ind} - T_{amb}) = U_{tot} A_{tot} (T_{ind} - T_{amb}) \end{aligned} \quad (7)$$

$$\dot{Q}_{vent} = N \cdot \rho_a \cdot V_a \cdot C_{p,a} (T_{ind} - T_{amb}) \quad (8)$$

$$\dot{Q}_{inf} = \dot{V}_{a,inf} \cdot \rho_a \cdot C_{p,a} \cdot (T_{ind} - T_{amb}) \quad (9)$$

where the right-hand term of Equation (7) is the sum of heat losses through the walls, floor, roof and windows of the building envelope. N is number of air changes per hour; ρ_a is the air density; V_a is the air volume; $C_{p,a}$ is the air specific heat capacity; $\dot{V}_{a,inf}$ is the volume flow rate of air infiltration.

Figure 2 shows a schematic diagram for the heat transfer process between the indoor temperature (T_{ind}) and ambient temperature (T_{amb}) through the wall consisting of various layers. The heat transfer through the layers of a composite wall and the overall thermal resistance can be calculated as the sum of individual resistances comprising the assembly of the layers (Kirchhoff current law) as shown in the following equations [20]:

$$\dot{Q} = U \cdot A \cdot \Delta T \quad (10)$$

$$U = \frac{1}{R_{tot}} \quad (11)$$

where

$$\begin{aligned} R_{tot} &= R_{a,ind} + R_1 + R_2 + R_3 + R_4 + R_{a,amb} \\ &= \frac{1}{\alpha_{a,ind}} + \frac{l_1}{k_1} + \frac{l_2}{k_2} + \frac{l_3}{k_3} + \frac{l_4}{k_4} + \frac{1}{\alpha_{a,amb}} \end{aligned} \quad (12)$$

The dynamic energy balance of the heated space (room) is expressed using the following equation:

$$\begin{aligned} \rho_a C_a V_a \frac{dT_{ind}}{dt} &= [U_{rad} A_{rad} LMTD - U_{tot} \cdot A_{tot} (T_{ind} - T_{amb}) \\ &- V_a \cdot N \cdot \rho_a \cdot C_a \cdot (T_{ind} - T_{amb})] \end{aligned} \quad (13)$$

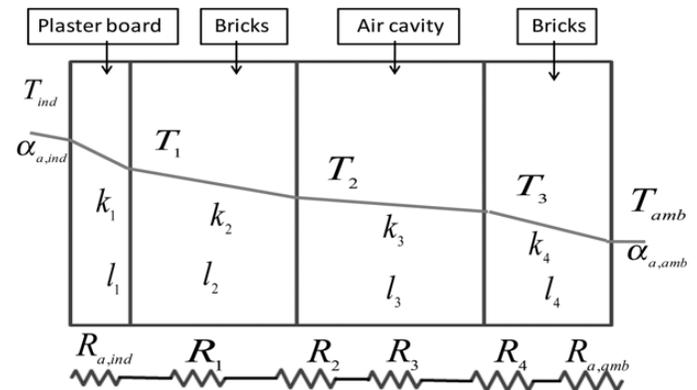


Figure 2. Heat transfer through the wall layers of building and its thermal resistance.

where the left-hand side represents the dynamic heat load of the indoor heated space and the right-hand side represents the net of heat gained from the radiator, the heat loss through the structure of the building and ventilation as a function of time. The infiltration is neglected for this work because the room is assumed to be controlled with no unintentional air leakage openings.

3 MATLAB/SIMULINK MODELLING

The work presented in this section is concerned with modelling the dynamic thermal performance of a single room heated with a hydronic panel radiator using the Matlab/Simulink software. Simulink is a Matlab graphic user interface that can be used for dynamic system simulations [21–23]. The simulation is carried out with the following assumptions:

- The room temperature is assumed to be uniform.
- The only heat source is the radiator; all other heat gains from the Sun, people, light bulbs and other appliances are neglected.

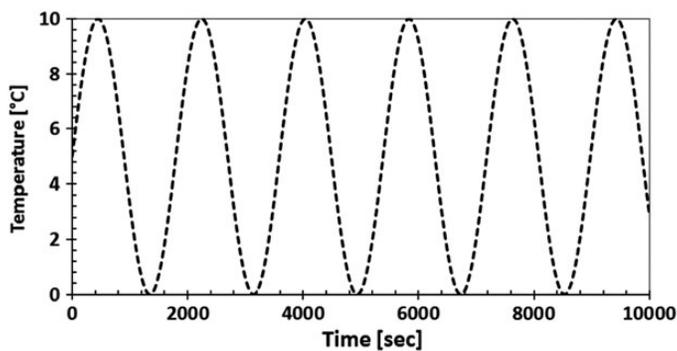


Figure 3. The UK winter ambient temperature fluctuations [24].

- The simulation is carried out using UK winter conditions as shown in Figure 3 [24].
- The room is assumed to be empty and the only heat loss is due to the thermal properties of the building (transmission) and the air ventilations.

The main governing equation used in the Simulink to predict the variation of the heated space temperature with time was derived from Equation (13) and is shown by the following equation:

$$T_{ind} = \int \left[\frac{\dot{m}C_w}{V_a \cdot C_a \rho_a} (T_{w,in} - T_{w,out}) - \frac{U_{tot}A_{tot}}{V_a \cdot C_a \cdot \rho_a} (T_{ind} - T_{amb}) - \frac{V_a \cdot N \cdot \rho_a \cdot C_a}{V_a C_a \rho_a} \cdot (T_{ind} - T_{amb}) \right] dt \quad (14)$$

where the indoor temperature (T_{ind}) is determined by the integral ratio of the heat input to the radiator, heat loss through the building structure and the heat loss by ventilations to the thermal capacity of the heated space ($V_a \cdot C_a \cdot \rho_a$). The radiator inlet and outlet temperatures were taken from the British Standard BS EN 442-2 radiators and convectors Part 2 [25]. The input values of the thermal properties of the building structure and the standard input value of the hydronic panel radiator are shown in Tables 1 and 2, respectively [25, 26]. Figure 4 shows the flow diagram for the thermal modelling process of a single room coupled with heater radiator using the Simulink software. Block 1 indicates the primary procedure of the Simulink simulation, block 2 the constant mass flow input operating condition and block 3 the pulsed flow input operating conditions.

4 SIMULINK SIMULATION RESULTS

Figure 5 shows the Simulink subsystems block diagram including the heater radiator model, the ambient temperature model,

Table 1. The thermal properties of building materials and hydronic panel radiator heater used for this work.

Components	Materials	K [W/m K]	c [J/kg K]	U [W/m ² K]	ρ [kg/m ³]	UA [W/K]
Internal wall	Plasterboard	0.16	840	0.35	950	11.3
External wall	Brick	0.77	800	0.86	1700	3.69
Window	Glass	0.96	750	1.4–3.1	2400	5.20
Floor	Screed	0.41	840	0.35	1200	4.03
Ceiling	Wood wool slab	0.10	1000	0.3	500	3.45
Radiator	Aluminium	205	900	7	2700	7
Room envelope	Air	0.025	1005	–	1.2	–
Door	Wood	0.14	1200	–	650	–
Working fluid	Water	–	4180	–	1000	–

Table 2. The standard heater radiator operating input values.

T_{rad} inlet [°C]	T_{rad} outlet [°C]	T_{amb} [°C] (average)	Mass flow rate [kg/s]	Type of controller	T_{set} point [°C]
75	65	5	0–0.03	PID and ON/OFF	20

the building structure model, the input hot water mass flow port; the indoor temperature output port and the thermostat radiator valve (TRV on/off control relay). The central heating system was investigated for both the constant flow condition typically used in the UK hydronic heating systems (Figure 6a and the proposed pulsed flow strategy investigated in this work, Figure 6b). Figure 7 shows the pulsed radiator flow rate with an amplitude of 0.0384 kg/s, a frequency of 0.0027 Hz and various duty cycles used in this investigation. The duty cycle is defined as the ratio of the on cycle duration to the total cycle period as shown in Figure 6b and the following equation.

$$\text{Duty cycle} = \left(\frac{\text{On cycle period}}{\text{Total cycle period}} \right) \times 100 \quad (15)$$

Figure 8 shows the room temperature variation with time for the pulsed flow input strategies as shown in Figure 7 compared with that of the constant flow shown in Figure 6a. It can be seen that

the room temperature increases to the target temperature of 20°C within the first 700 s (the rising period (RP)) and then it fluctuates ~20°C during the steady-state period by ± 2°C. Also, Figure 8 shows that as the percentage of duty cycle increases the RPs are shortened. The energy consumption of the central heating system was calculated at the RP and at the steady-state period as shown in Figure 8. The rate of energy consumption for the RP and steady-state period at a constant flow scenario (traditional central heating flow strategy) is calculated using Equations (16) and (17), respectively. The rate of energy consumption for the RP and steady-state period at a pulsed flow scenario (proposed flow for this work) is calculated using Equations (18) and (19), respectively. The rate of saved energy for both rising and steady-state periods due to the pulsed flow compared with the constant flow is calculated using Equations (20) and (21), respectively. The rate of saved energy for both RP and steady-state period (Figure 8) is determined by using the difference between the rate of energy consumed by the constant flow and the rate of energy consumed by pulsed flow. The indoor temperature fluctuation also is calculated using Equation (22):

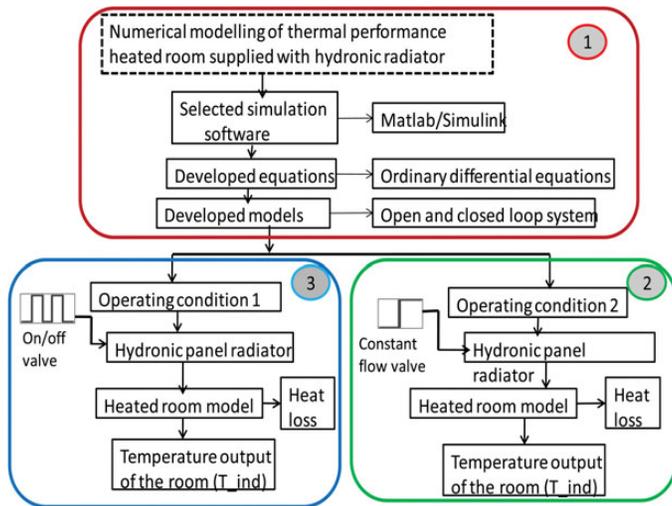


Figure 4. Matlab/Simulink flow diagram of a single room thermal model for both pulsed and constant flow strategies.

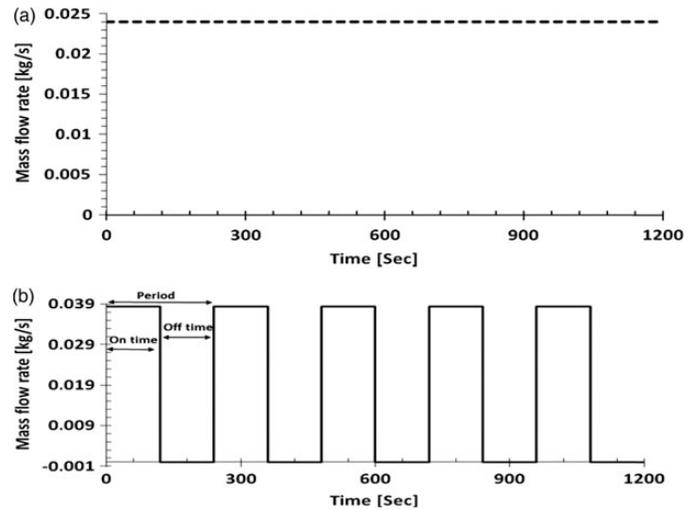


Figure 6. Profiles of the various mass flow input strategies at a constant inlet hot water temperature of 75°C. (a) Constant flow strategy. (b) Pulsed flow strategy.

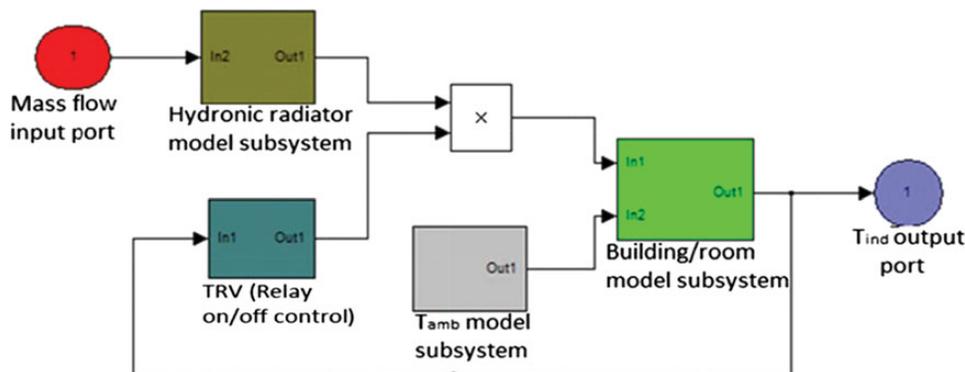


Figure 5. Subsystems of a Simulink block diagram for central heating system.

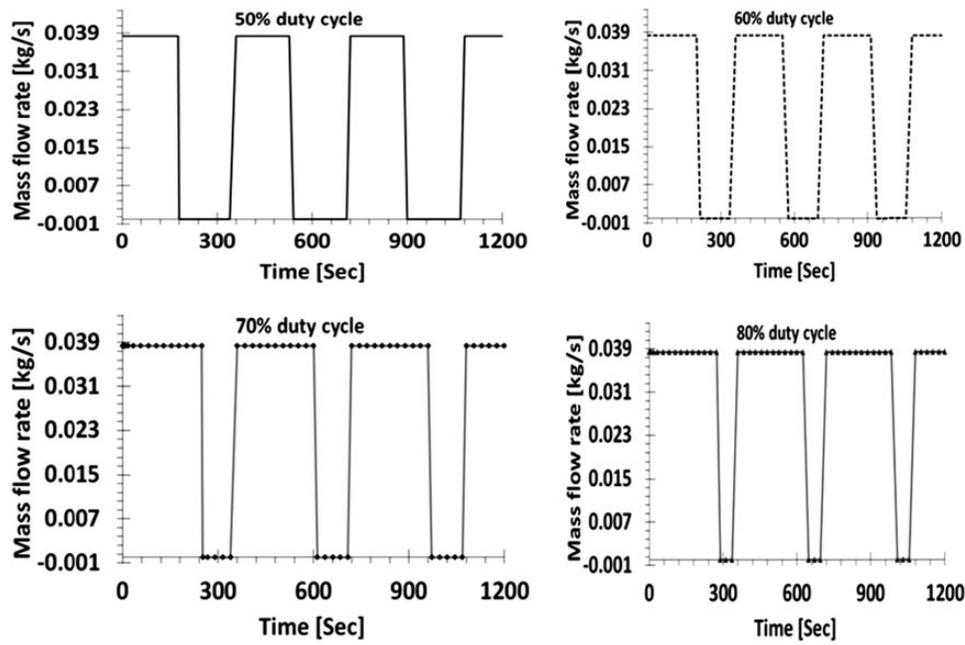


Figure 7. Input radiator pulsed flow at various duty cycles and an amplitude of 0.0384 kg/s.

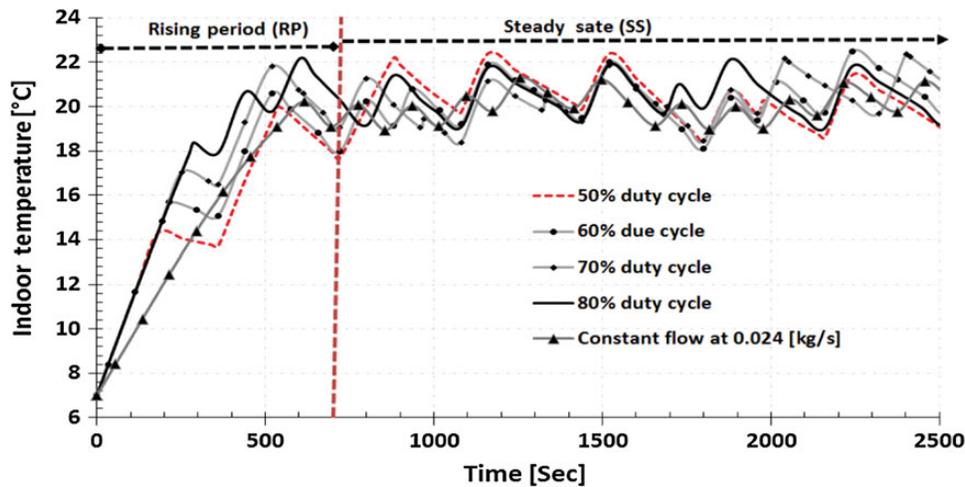


Figure 8. Indoor temperature output for both steady and pulsed flow conditions as shown in Figure 7.

$$q_{\text{cons,CF,RP}} = \dot{m}_w C_w \Delta T \quad (16) \quad \text{where}$$

$$q_{\text{cons,CF,SS}} = (\text{TRV on ratio}) \cdot (\dot{m}_w C_w \Delta T) \quad (17)$$

$$q_{\text{cons,PF,RP}} = (\text{Duty cycle}) \cdot (\dot{m}_w C_w \Delta T) \quad (18)$$

$$q_{\text{cons,PF,SS}} = (\text{Duty cycle} - \text{TRV off ratio}) \cdot (\dot{m}_w C_w \Delta T) \quad (19)$$

$$\text{TRV on ratio} = \frac{\text{On time of relay controller}}{\text{Total time}}$$

$$\text{TRV off ratio} = \frac{\text{Off time of relay controller}}{\text{Total time}}$$

The relay TRV is on for:

$$T_{\text{ind}} - T_{\text{set}} < 0$$

The relay TRV is off for:

$$T_{\text{ind}} - T_{\text{set}} \geq 0$$

$$ES_{\text{RP}} = \frac{q_{\text{cons,CF,RF}} - q_{\text{cons,PF,RP}}}{q_{\text{cons,CF,RP}}} \times 100 \quad (20)$$

$$ES_{\text{SS}} = \frac{q_{\text{cons,CF,SS}} - q_{\text{cons,PF,SS}}}{q_{\text{cons,CF,SS}}} \times 100 \quad (21)$$

$$T_{\text{ind}}(\text{fluctuation}) = T_{\text{ind}}(\text{max}) - T_{\text{set}} \text{ or } T_{\text{set}} - T_{\text{ind}}(\text{min}) \quad (22)$$

where ES is the percentage of energy saved; q_{cons} the rate of energy consumed; CF the constant flow; PF the pulsed flow; RP is the rising period; SS the steady state; T_{set} is the target room temperature taken as 20°C. Table 3 presents the energy saving calculated at the listed operating conditions (at various pulsed flow amplitudes and various pulsed duty cycles) compared with the constant flow case. A positive ES value in Table 3 shows the energy consumed by the pulsed flow is lower than the energy consumed by the constant flow, while a negative value shows the opposite. It can be seen that the maximum energy saving of 20.3% is achieved at 50% duty cycle and flow amplitude of 0.0384 kg/s, which highlights the potential of flow pulsation.

All the above simulations (Table 3) were carried out at a constant frequency of 0.0027 Hz. In order to investigate the effects of cycle frequency on energy saving, the best operating condition shown in Table 3 was used at various cycle frequencies ranging from 0.0017 to 0.017 Hz. Figure 9a shows the indoor temperature simulated at various frequencies but with a constant flow amplitude of 0.0384 kg/s and Figure 9b highlights the advantage of the pulsed flow compared with the constant flow on the predicted indoor temperature response time. It is clear from Figure 9a that as the flow frequency decreases higher fluctuation of the indoor temperature can be observed; for example, as the frequency decreases from 0.017 to 0.0017 Hz the indoor

temperature fluctuation increases from 20 ± 1.5 to 20 ± 3 °C. Therefore, the frequency value of 0.017 Hz can keep the temperature fluctuation in the room within the required comfort level of 20 ± 2 °C. The other advantage of using such a frequency is to shorten the RP from 600 to 450 s as shown in Figure 9b. Table 4 presents the energy saving obtained using the pulsed flow as shown in Figure 9a. It is clear that the highest saving can be achieved at a flow frequency of 0.017 Hz without compromising the comfort temperature of the user as shown in Table 4. Figure 10 shows the average and instantaneous rate of energy consumption of the hydronic heating system for a constant flow scenario and the best selected pulsed flow case at 50% duty cycle, 0.0384 kg/s amplitude and 0.017 Hz. Figure 11 summarises the energy saving for the best pulsating flow strategy compared with the constant flow case.

5 PID CONTROL SYSTEM

To achieve further reduction in energy saving and to decrease the indoor temperature fluctuations, a three-term proportional integral differential (PID) feedback control strategy was investigated. The following equation describes the proposed three-term PID controller.

$$u(t) = K_p e(t) + K_i \int e(t) dt + K_D \frac{de}{dt} \quad (23)$$

where u is the control signal, K_p the proportional gain; K_i the integral gain; K_D the differential gain; e the error signal and t the time.

Figure 12 is the modified Simulink subsystems diagram showing the PID controller replacing the thermostat on/off controller. Figure 13 shows the indoor temperature variation for the

Table 3. Calculated energy saved based on the amplitudes and pulsed % duty cycles.

Flow amps [kg/s]	Duty cycles [%]	Average mass flow rate [kg/s]	Energy cons rate at RP level [W]	Energy cons at SS [W]	E_saved at RP [%]	E_saved at SS [%]	Fluctuation indoor T [°C]
0.048	50	0.024	835	400	16.42	16	± 3
	60	0.0288	860	420	14.0	12.5	± 3.2
	70	0.0338	884	440	11.6	8.3	± 3.1
	80	0.0384	911	450	8.9	6.25	± 3.1
0.0384	50	0.0192	827	375	17.3	20.3	± 2.2
	60	0.023	838	390	16.2	18.75	± 2.8
	70	0.0269	880	400	12.0	16	± 2.8
	80	0.0307	914	420	8.6	12.5	± 3.0
0.0307	50	0.015	1086	450	-8.6	6.25	± 2.2
	60	0.0184	1097	425	-9.7	11.45	± 2
	70	0.0245	975	420	2.5	12.5	± 2.1
	80	0.024	878	400	12.2	16	± 2.2
0.024	50	0.012	1236	490	-23.6	-2.08	± 3.0
	60	0.0144	1186	480	-18.6	0	± 2.6
	70	0.0168	1025	450	-2.5	6.25	± 2.1
	80	0.0192	971	430	2.9	10.41	± 1.8
Constant flow	Constant	0.024	1000	480	—	—	

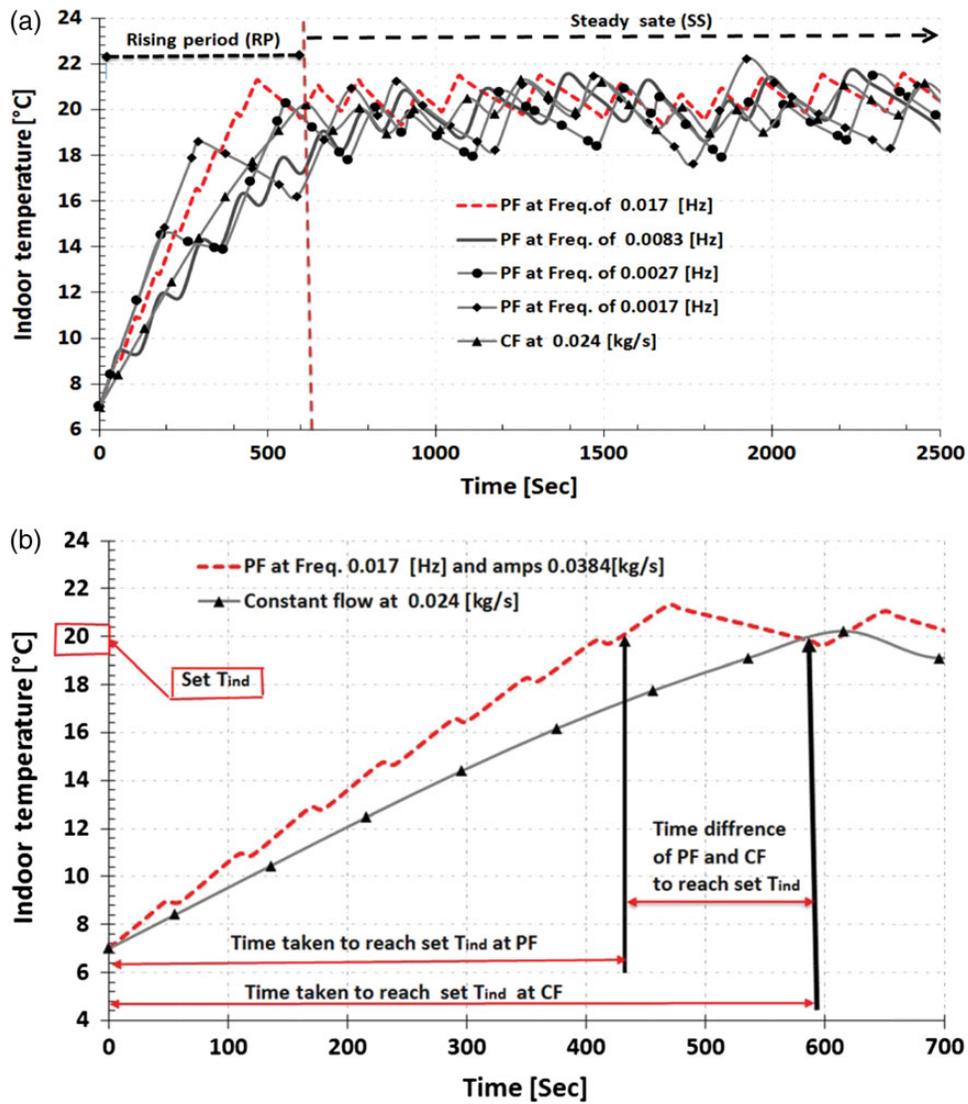


Figure 9. (a) Indoor temperature at various frequencies, 50% on duty cycle; 0.0384 kg/s of flow amplitudes and an average mass flow rate of 0.0192 [kg/s]. (b) Time response of indoor temperature for the best pulsed flow (PF) compared with the constant flow (CF).

Table 4. Calculated energy saved based on the flow frequency at constant amplitudes and selected 50% duty cycles flow pulsation.

Operating f [Hz]	Flow amps [kg/s]	Average mass flow rate [kg/s]	Energy cons rate at TS level [W]	Energy cons rate at SS [W]	E _{saved} at RP [%]	E _{saved} at SS [%]	Fluctuation indoor T [°C]
0.017	0.0384	0.0192	800	375	20	21.9	± 1.5
0.0083	0.0384	0.0192	815	382	18.5	20.3	± 1.5
0.0027	0.0384	0.0192	827	384	17.3	20	± 2.1
0.0017	0.0384	0.0192	845	403	15.5	16	± 3
(steady flow)	Constant	0.024	1000	480			

PID controlled pulsed flow compared with that of the PID controlled constant flow. It is clear from this figure that the fluctuation of the indoor temperature is reduced to $20 \pm 1^\circ\text{C}$, which

is well within the comfort level. As for the energy saving, with this PID control strategy, Figure 14 shows that $\sim 27\%$ of energy can be saved. Clearly, this can add more saving than that of

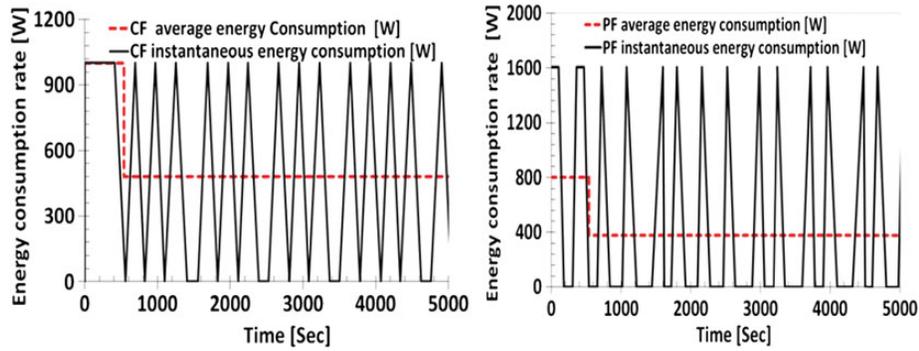


Figure 10. Average and instantaneous energy consumption rate for the constant mass flow strategy and pulsed mass flow strategy.

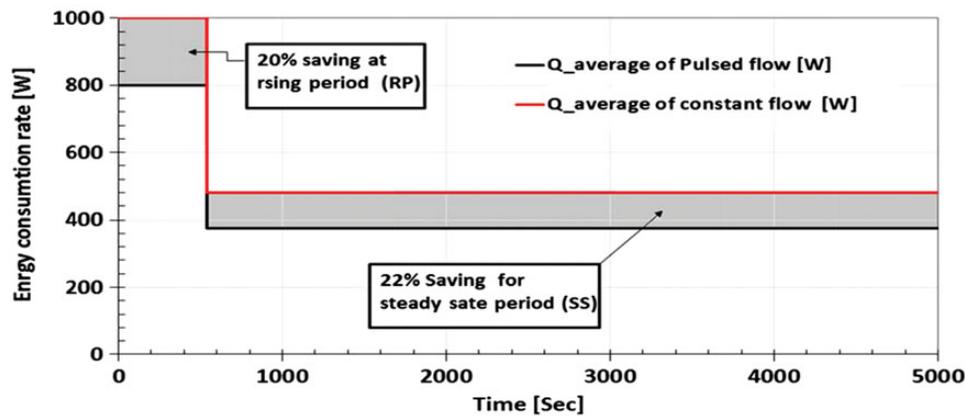


Figure 11. Average percentage energy saving rate at the selected flow pulsation comparing with the constant mass flow case.

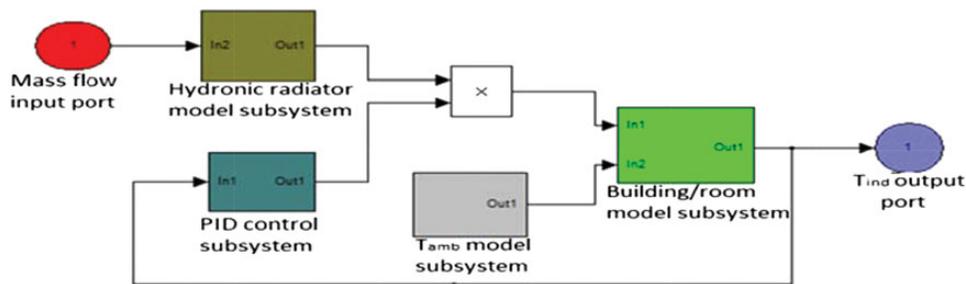


Figure 12. Simulink subsystem block diagram of a domestic central heating system with the PID control feedback system.

using only the pulsed flow strategy which gave 22% energy saving, thus highlighting the potential of using PID control with the pulsed flow strategy.

6 CONCLUSIONS

Thermal simulation was used to investigate the feasibility of enhancing the performance of hot water heating system by changing the flow strategy from constant flow to pulsed flow without

compromising the user comfort. Mathematical modelling for a single room heated using a hydronic heater radiator was developed, this modelling includes coupling of the thermal performance of the radiator, air in the space, walls and windows using the Simulink/Matlab software.

Results showed that 20–22% of the energy consumed using a current constant flow heating method in centrally heated buildings can be saved by changing the flow strategy to pulsed flow. This energy saving was achieved using pulsed flow with an amplitude of 0.0384 kg/s, a frequency of 0.017 Hz and a duty cycle of

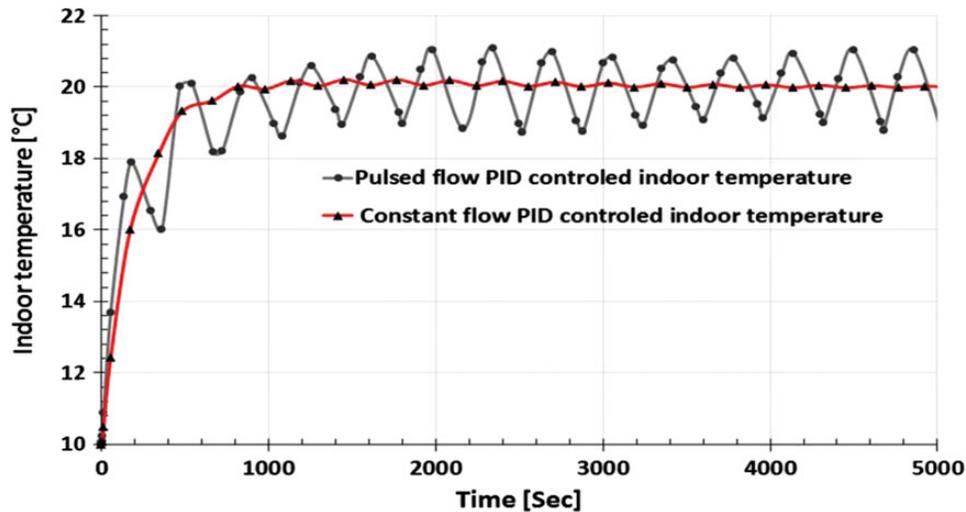


Figure 13. Indoor comfort temperature after the PID control system is applied to the hydronic central heating system.

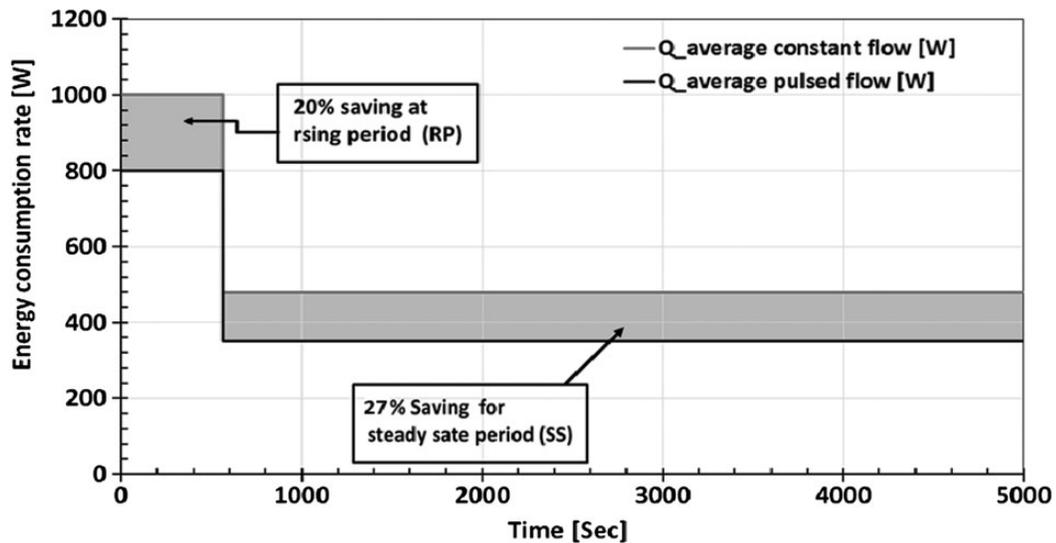


Figure 14. Energy saved after the PID control system was applied to the domestic hydronic heating system.

50%. In addition to the energy saving achieved using this pulsed flow, the indoor temperature response is also shortened from 600 s for the constant flow case to 450 s.

Further improvement was achieved by introducing the PID feedback control with the pulsed flow where the results showed that energy saving can be increased to 27% and the fluctuation in the desired indoor temperature decreased to ± 1 . This saving is achieved with an additional investment of \sim £40 covering the cost of the control system resulting in a payback period of 4 months for three bed room family house. Regarding additional operating energy consumption due to the excitation voltage supplied to the motorised valve, it is estimated to be \sim 0.3% of the overall energy consumption in the room. Such additional operating cost is not considerable.

The results from this work highlight the potential of applying flow pulsation together with PID controllers in hydronic central heating systems for energy saving in buildings.

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