

# Thermal performance of hydronic radiator with flow pulsation – Numerical investigation

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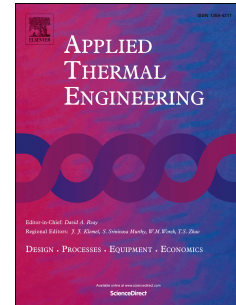
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# Thermal Performance of Hydronic Radiator with Flow Pulsation – Numerical Investigation

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## ABSTRACT

Improving the heat output of hydronic central heating system in buildings can play a major role in energy saving. Current panel radiators of central heating systems are operating at constant flow strategy with thermostat control device. Such operating mode is not efficient in terms of energy consumption; therefore an alternative operating scenario is required to enhance the heat output of the panel radiator. The main aim of this research is to investigate the effect of pulsed flow input on the energy consumption of panel radiators while maintaining the target panel surface temperature. CFD modelling of two hydronic panel radiators with constant and pulsating flows were developed using the conjugate heat transfer module in COMSOL Multiphysics software. The radiators used were one with single finned surface (type 11) and the second is without fins (type 10), both with the dimensions of 500mm long and 300mm high. The CFD results of the constant flow conditions were compared to published experimental work showing good agreement with maximum deviation of 2.4% in the heat output. To investigate the effects of pulsating flow on the performance of the two panel radiators, a wide range of input pulsating flows with amplitude ranging from 0.027m/s to 0.051 m/s and frequency ranging from 0.0523 rad/s to 0.209rad/s while the flow supply temperature remained constant at 75° were simulated. The simulation results showed that using pulsed flow can reduce the energy consumption of panel radiators by up to 20% compared to constant flow operating condition while maintaining the same radiator surface temperature of 50°C. Such results highlight the potential of using pulsed flow to reduce the energy consumption of central heating systems in buildings without compromising the user comfort.

**Keywords:** Energy saving, CFD modelling, COMSOL Multiphysics, Flow pulsation, Constant flow

## 1. Introduction

Building energy utilization has important impact on energy consumption and greenhouse gas emission all over the world. The energy statistical data indicates that around 50% of the total energy consumed in the developed countries is for buildings usage [1]. This produces significant amounts of CO<sub>2</sub> emission and other environmental pollutants which contribute to global warming [2]. This has led the European Union to legislate for reducing its energy consumption by 20% before 2020 which was adopted by the member states [3]. Enhancing the efficiency of domestic central heating system is an important factor in achieving this target.

Considerable research has been carried out in order to improve the domestic central heating systems. The studies were carried out based on the heat sources and heating appliances. Heat sources include Heat pump, District heating, Boiler (Combi boiler, Condensing boiler), and CHP (Combined Heat and Power). Heating appliances include panel radiators with convective fins and without convective fins, floor heating and air heating, skirting's (baseboard). Hydronic (water based) heating radiators are predominantly used for internal heat emission within the residential sector in Europe [4]. The advantages of the heating panel radiator as a heat emitter are: compact design, less space requirement in the rooms and ease of installation to new buildings or retrofits. Various research works have been carried out to enhance the central heating system performance based on design optimization of the panel radiator. Myhren and Halberg [5] achieved improvement in the heat transfer output from panel radiators by increasing the air flow on its heat transferring surfaces. They concluded that, using ventilation-radiators the desired thermal climate could be achieved with a radiator surface temperature as much as 7.8°K lower than standard values. The possibility of enhancing the heat output from the panel radiator by coating the wall behind the radiator in different colours with different emissivity values was also studied by [6]. The study concluded that, the wall coated with higher emissivity (smooth black surface) material can improve the heat output of the heating panel radiator by up to 26%. Beck et al [7] investigated the heat transfer enhancement of panel radiations by placing one or two high emissivity metal sheets between the interior surfaces of double panel radiators. They concluded that placing two high emissivity metals in the interior side of the double

panel radiator can produce an improvement of about 88% compared to the double panel radiator. Enhancement of hydronic heating system using baseboard device integrated air supply was reported by Ploskic and Holmberg [4, 8]. The aim of their work was to minimize the supply temperature by pre-heating the incoming ventilation airflow and concluded that the heat output of their proposed system can produce about 21% more heat compared to the traditional hydronic baseboard heating system [4, 8]. Myhren and Holmberg [9] 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 results showed that the heat output can be increased by 20% compared to traditional radiators.

The hydronic baseboard heating system was studied by [10]. The aim of the study was to investigate the capability of the baseboard supplied at low temperature ranges  $40^{\circ}\text{C}$  -  $45^{\circ}\text{C}$  to suppress strong downdraughts. They concluded that the baseboard at these range of temperature is unable to suppress the strong downdraught instead they have used a supply temperature of  $55^{\circ}\text{C}$  and that was able to overcome the downdraughts to comfort level. Optimization of heat output of the ventilation radiator was studied by varying the distribution of the vertical longitudinal convection fins using computational fluid dynamics (CFD) [11]. They showed that heat transfer can be enhanced by up to 17% by changing the geometrical design of the fins, decreasing the fin to fin distance and cutting the middle section of the fin array. Sanjay and Avanic [12] investigated the use of phase change material to optimize the heat output from the hydronic panel radiator and concluded that 20-25% of energy can be saved compared to the traditional radiators. Kerrigan et al [13] investigated the use of heat pipes to improve the performance of panel radiators at supply temperature as low as  $55^{\circ}\text{C}$ . The power density of the tested panel radiator fitted with heat pipes was nearly tripled compared to the traditional panel radiator. Therefore the researchers concluded that the heat pipe based naturally aspirated radiator is a possible alternative to replace traditional panel radiators particularly when low temperature water heating systems such as heat pumps are used.

Extensive study has 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 applied in various industrial applications including heat exchangers, pulse combustors, electronic cooling devices and cooling of nuclear reactors [14]. A study was

carried out by [15] 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 ones. 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 [16]. The test was performed at Reynolds number ranges from 500 to 5000; and frequency of 1.5 Hz and concluded that the overall heat transfer coefficient was increased by 80% depending on closeness of the solenoid valve to the test section. Effect of pulsed perturbation on convective heat transfer for laminar flow on co-axial cylindrical tube heat exchangers was experimentally investigated by [17]. The test was performed at lower Reynolds number ranging from 150 to 1000; frequency of 0 to 2Hz with reciprocating pump was used as pulsating device. Based on the results they concluded that the heat transfer coefficient can be enhanced by 300% using the strong pulsed perturbations. Experimental study was carried out to explore the convective heat transfer enhancement from heated cylinder in a pulsating flow. A series of experiments was conducted to compare the heat transfer enhancement for pulsed air jet and steady flow air jet and results showed that about 50% heat transfer enhancement was achieved with the pulsed jet compared to the steady flow jet [18]. Sailoret al [19] investigated the potential of embedding pulsating heat pipes for space or terrestrial applications. The enhancement technique was based on the selection of Biot number that minimizes the conductive resistance of the thermal radiator. They concluded that effective thermal conductivities of 400W/mK- 2300W/mK can be achieved due to the applied pulsed flow.

Despite the number of studies reported for using pulsating flow to enhance heat transfer of heating or cooling devices, there are limited studies regarding its use to improve panel radiators in hydronic heating systems. Therefore, this work aims to investigate the heat transfer enhancement due to flow pulsation in panel radiator based hydronic central heating system using computational fluid dynamics modelling with COMSOL multi-physics software. The work investigates the energy saving by using flow pulsation in two types of panel radiators namely; type 10 without fins and type 11 single fined panel radiators.

## 2. CFD Simulation

CFD three dimensional simulations of panel radiators type 10 and 11 were carried out using conjugate heat transfer physics within COMSOL multi-physics [20]. The Navier Stokes equations (Eqs. (1)-(3)) of continuity, momentum and energy used to simulate the heat transfer and fluid flow within the panel radiators and the surroundings [21].

$$\rho \frac{\partial u}{\partial t} + \rho(u \cdot \nabla)u = \nabla \cdot [-pI + (\mu + \mu_T)(\nabla u + (\nabla u)^T) - \frac{2}{3}(\mu + \mu_T)(\nabla \cdot u)I - \frac{2}{3}\rho\kappa I] + F \quad (1)$$

$$\text{Where: } \mu_T = \rho C_\mu \frac{\kappa^2}{\varepsilon}$$

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho V) = 0 \quad (2)$$

$$\rho C_p \frac{dT}{dt} + \rho C_p V \cdot \nabla T = \nabla \cdot (k \nabla T) + Q \quad (3)$$

The k-ε Reynolds Average Navier Stokes (RANS) turbulent model was used to model the turbulent flow inside the radiator channels [20, 21]. The values for the turbulence model of κ (turbulent kinetic energy), ε (turbulent dissipation rate), and the pressure are calculated using Eqs. (4), (5) and (6) respectively.

$$\rho \frac{\partial \kappa}{\partial t} + \rho(u \cdot \nabla)\kappa = \nabla \cdot \left[ \left( \mu + \frac{\mu_T}{\sigma_\kappa} \right) \nabla \kappa \right] + p_\kappa - \rho \varepsilon \quad (4)$$

$$\rho \frac{\partial \varepsilon}{\partial t} + \rho(u \cdot \nabla)\varepsilon = \nabla \cdot \left[ \left( \mu + \frac{\mu_T}{\sigma_\varepsilon} \right) \nabla \varepsilon \right] + C_{\varepsilon 1} \frac{\varepsilon}{\kappa} p_\kappa - C_{\varepsilon 2} \rho \frac{\varepsilon^2}{\kappa} \quad (5)$$

$$p_\kappa = \mu_T [\nabla u : (\nabla u + (\nabla u)^T) - \frac{2}{3}(\nabla \cdot u)^2] - \frac{2}{3}\rho\kappa \nabla \cdot u \quad (6)$$

Where:  $\rho$ ,  $C_p$ ,  $T$ ,  $u$ ,  $p$ ,  $Q$ ,  $I$ ,  $F$ ,  $\varepsilon$ ,  $\mu_T$ ,  $\mu$ ,  $p_k$ ,  $K$ ,  $V$  and  $k$  are density, specific heat capacity, temperature, velocity vector, pressure, heat source other than viscous, identity matrix of [3\*3], total force acting per volume, the turbulence dissipation rate, turbulence dynamic viscosity, dynamic viscosity, turbulent kinetic pressure, heat conductivity, volume and the turbulence kinetic energy respectively.

Fig. 1 shows the 3D model of panel radiator (type11) with the selected mesh concentration for the water flow domain and the fins. Table 1 presents the thermal properties of the materials used in the panel radiator. The panel radiator model includes the rectangular ducts, 15mm side with thickness of 1mm, and fins 270 mm high with thickness of 0.5mm. The radiator model was meshed based on the geometrical structure of each component; swept rectangular mesh was generated for the fins (478080 elements) and tetrahedral mesh was selected for the water domain and the flow channels (1759121 elements). Assumptions used to simulate the panel radiators in both pulsed and constant flow conditions include:

- Natural convection cooling was applied on the air side surface of the radiator
- Constant hot water supply temperature of 75°C was considered
- Heat transfer due to radiation was applied at constant emissivity value of 0.85.

The heat transfer coefficient of the air side surface of the radiator was considered as natural convection and was calculated using the well-known empirical correlation given in Eq. (7) [4].

$$Nu = \frac{hL}{k} = 0.59(Ra)^{1/4}$$

where :

$$Ra = Gr.Pr = \left( \frac{g \cdot \beta (T_s - T_{amb}) L^3}{\nu^3} \right) \cdot Pr \quad (7)$$

$$h = 0.59 \cdot \left( \frac{g \cdot \beta (T_s - T_{amb}) \cdot L^3 \cdot Pr}{\nu^3} \right)^{1/4} \cdot \left( \frac{k}{L} \right)$$

Where:  $Nu$ ,  $Gr$ ,  $h$ ,  $L$ ,  $Ra$ ,  $T_s$ ,  $T_{amb}$ ,  $g$ ,  $Pr$ ,  $k$ , and  $\beta$  are the Nusselt number, Grashof number, convection heat transfer coefficient, characteristics height, Rayleigh number, surface temperature, ambient temperature, gravitational acceleration, Prandtl number (0.71 for air), thermal conductivity, and expansion coefficient (which is  $1/T_{average}$ ) respectively.



### 3. CFD Simulation of the radiators at constant flow rate

In this section, numerical simulation of two panel radiators (type 10 and type 11) at constant flow rate was carried out to predict the radiators temperature distribution and the heat output. Such prediction will be used as a reference case for the proposed pulsed flow analysis and for validation reasons since experimental data is available for constant flow scenario. The radiators were numerically simulated at constant hot water inlet mass flow rate of 0.00434kg/s for type 10 and 0.00571kg/s for type 11, constant inlet temperature of 75°C and ambient air temperature of 20°C. Such flow rates were determined to produce water outlet temperature of 65°C and radiators average surface temperature of 50°C as recommended by BS EN 442 [22-25]. Fig. 2 shows the radiators local surface temperature distribution at constant hot water inlet flow rate for both radiators. Fig. 3 shows the variation of flow outlet temperature, radiator average surface temperature and the rate of heat output from the radiators with time for type 10 (Fig. 3(a)) and type 11 (Fig. 3(b)). It is clear from Fig. 3 that the average surface temperature for both radiators achieved the target average surface temperature of 50°C and water outlet temperature of 65°C as recommended by BS EN 442. However, radiator type 11 produced higher heat output rate due to the increased surface area produced by the fins.

The CFD results of both type 10 and type 11 radiators were validated based on the experimental results reported in [23-25]. Validation was carried out based on the radiator hot water outlet temperature, the average surface temperature and the rate of heat output according to the BS EN 442. The rate of heat output from the radiator can be calculated using Eq. (8).

$$Q_{rad,out} = U_{rad} \cdot A_{rad} \cdot LMTD \quad (8)$$

Where  $U_{rad}$  is overall heat transfer coefficient of the panel radiator determined using Eq. (9) as:

$$U_{rad} = \frac{q_{rad}}{LMTD} \quad (9)$$

and LMTD is the log mean temperature difference given in Eq. (10) as:

$$LMTD = \frac{T_{w,in} - T_{w,out}}{\ln\left(\frac{T_{w,in} - T_{ind}}{T_{w,out} - T_{ind}}\right)} \quad (10)$$

$q_{\text{rad}}$  is the total heat flux obtained from the surface of the panel radiator.

Tables 2 and 3 compare the simulation results to the experimental values [24] in terms of the water outlet temperature, the radiator average surface temperature and the rate of heat output for both radiators. The deviation between the experimental and simulation results is determined using Eq. (11).

$$Dev\% = \left| \frac{Exp - CFD}{Exp} \right| * 100 \quad (11)$$

As shown in Tables 2 and 3 the maximum deviation were 0.27% for the water outlet temperature, 1.20% for the radiator average surface temperature and 2.44% for the rate of heat output. Such low deviation values indicate good agreement between the experimental and the CFD analysis which highlights the validation of the numerical simulation approach.

#### 4. CFD simulation of pulsed flow radiator

This section presents the CFD investigation of the two panel radiators using pulsed flow input conditions. Fig. 4 shows the general profile of the pulsed flow used in this investigation where it is governed by two major parameters namely; the amplitude (amp [m/s]) and frequency ( $freq$  [Hz]). Strouhal number is a dimensionless parameter that describes pulsating flow mechanisms and relates the frequency of pulsation to the velocity of the flow as shown in Eq. (12).

$$St = \frac{freq.L}{V} \quad (12)$$

For heat transfer enhancement applications, Strouhal number values between 0 to 1 are recommended to achieve significant heat transfer enhancement due to flow pulsation [26]. For each radiator, the maximum flow rate is specified by its manufacturer depending on the heat output, size and type. For type10, the maximum mass flow rate is 0.0043 kg/s while for type11 it is 0.005710 kg/s [22, 27]. Such maximum flow rates limit the value of the flow pulsation amplitude. Also, there are limitations on the flow frequency that can be used due to noise and vibration considerations for human threshold of hearing [28]. Based on these velocity amplitude and frequency limitations, the range of Strouhal number that can be used for radiators type10 and 11 are 0.343 to 0.89 and 0.23 to 0.65 respectively. Also, the Reynolds number (Eq. (13)) associated with these flow conditions for both types of radiators

is below 2300 indicating that the flow is laminar. Finally, the inlet hot water temperature and ambient temperature were fixed at 75°C and 20°C respectively for both types of radiators.

$$Re = \frac{\rho V D_h}{\mu} \quad (13)$$

where;  $D_h = \frac{4A}{Per}$

Table 4 shows the various flow amplitudes and average hot water mass flow rate at constant frequency of 0.209rad/s corresponding to maximum Strouhal number. Fig. 5 shows the specific heat output of the two types of radiators at various pulsating flow amplitudes shown in Table 4. The specific heat output is the ratio of heat output rate to the supplied mass flow rate based on Eq. (14). It can be seen from Fig. 5 that the pulsating flow with amplitude of 0.031m/s produces the highest specific heat output for radiator type 10 and the flow with amplitude of 0.041m/s gave the highest specific heat output for radiator type 11.

$$Q_{SP.output} = \frac{Q_{output}}{\dot{m}_{input}} \quad (14)$$

Where:  $Q_{SP.output}$  is the specific heat output (J/kg),  $Q_{output}$  is the rate of heat output from the radiator (W) and  $\dot{m}$  is the inlet mass flow rate (kg/s).

Fig. 6 shows the specific heat output of the best pulsed flow amplitude while varying the flow frequencies for both radiators (a) type10 and (b) type11. As shown in Fig. 6 the highest specific heat output was produced at operating frequency of 0.209rad/s for both type10 and type11 radiators. Fig. 7 shows the local surface temperature contours, the average surface temperatures, the outlet water temperature and the heat output of both type 10 and type 11 radiators at the selected operating flow amplitude and frequency. It is clear from this figure that target temperatures were achieved with panel radiator surface temperature of 50°C and hot water outlet temperature of 65°C. The energy saving due to flow pulsation can be calculated using Eq. (15).

$$\% ES = \left( \frac{\dot{m}_{CF} - \dot{m}_{PF}}{\dot{m}_{CF}} \right) * 100 \quad (15)$$

Where ES is the energy saving; subscripts CF and PF are for constant and pulsed flow respectively. In this analysis, the energy saving is related to the mass flow rate because the hot water inlet and outlet temperatures are constants.

Fig. 8 compares the specific heat output of the two types of radiators for constant and pulsed flow conditions. The shaded areas in Fig. 8 represent the amount of enhancement in specific heat output achieved due to using flow pulsation. Using Eq. (15), such enhancement leads to energy saving of 17% for type 10 radiator and of 20% for radiator type 11 which is due to the reduction in the mass flow rate of the pulsed flow. The higher energy saving achieved in type 11 radiator is due to the extra surface area of the attached fins which highlight the advantages of using pulsed flow with finned radiators.

In this investigation, it is envisaged that the energy saving achieved in the pulsed flow is due to the heat transfer enhancement caused by the changes in the velocity distribution inside the flow channels of the radiators. Fig. 9 shows the velocity contours of radiator type 11 and the velocity distribution at various selected channels. Fig. 9a shows the velocity contours for the pulsed flow condition and the location of the selected channels. Figs. 9b to 9d compares the velocity distribution for pulsed and constant flow scenarios at the selected channels shown in Fig. 9a. These channels are channel 1 at 90mm, channel 2 at 196mm, channel 3 at 302mm and channel 4 at 408mm. The shaded part of Figs. 9b to 9d show the improvement in the average velocity of the pulsed flow compared to the constant flow. The higher flow velocity in the channels due the pulsed flow leads to higher Reynolds number that creates flow turbulence and breaks the boundary layers along the water side surface of the radiator channels. Producing higher velocity in the radiator channels results in higher heat transfer coefficient as shown in Fig. 10 leading to higher heat transfer rate. The convective heat transfer coefficient shown in Fig. 10 was calculated using Eq. (16)

$$h = \frac{q_{conv}}{T_{\infty} - T_{sur}} \quad (16)$$

Where; h is the convection heat transfer coefficient,  $q_{conv}$  is the convective heat flux,  $T_{\infty}$  is the average bulk hot water temperature and  $T_{sur}$  is the average internal surface temperature of the radiator channels. Fig. 10 shows an enhancement in the heat transfer coefficient due to pulsation of up to 50% can be achieved at the steady state condition.

## 5. Conclusions

In this work, flow pulsation technique was investigated for enhancing the heat transfer performance of two types of panel radiators used in central heating systems through CFD modelling. The simulation results for constant flow input case were compared to published experimental work showing good agreement with maximum deviation of 2.44% in the heat output.

The radiators were simulated using wide range of input pulsating flow conditions with velocity amplitude ranging from 0.027 m/s to 0.057 m/s and frequency ranging from 0.052 rad/s to 0.209 rad/s. Results showed higher average velocity was achieved in the channels of the radiators due to the pulsed flow compared to the constant flow velocity resulting in high convective heat transfer coefficient at the water side and higher specific heat output of the radiator. Energy saving of up to 17% (for type10 radiator) and 20% (for type11 radiator) can be achieved without affecting the radiator surface temperature. This overall energy saving can be attributed to the increase in the flow velocity values inside the radiator channels leading to high heat transfer coefficient (up to 50%) and reduced flow rate. The results from this work highlight the potential of using pulsed flow to enhance the heat transfer of panel radiators and reduce the energy consumption of central heating systems in buildings without compromising the user comfort.

## Nomenclature

### Symbols

A	area [m <sup>2</sup> ]
D <sub>h</sub>	hydraulic diameter [m]
ES	energy saved [%]
Eq.	equation
freq	frequency [Hz]

$g$	gravity [ $\text{m/s}^2$ ]
$h$	convective transfer coefficient [ $\text{W}/(\text{m}^2\cdot\text{K})$ ]
$C_p$	specific heat capacity [ $\text{J}/(\text{kg}\cdot\text{K})$ ]
$K$	thermal conductivity [ $\text{W}/(\text{m}\cdot\text{K})$ ]
$L$	characteristics length [ $\text{m}$ ]
LMTD	Log mean temperature difference [ $\text{K}$ ]
$\dot{m}$	mass flow rate [ $\text{kg/s}$ ]
Per	perimeter [ $\text{m}$ ]
$Q$	heat energy [ $\text{W}$ ]
$q$	heat flux [ $\text{W}/\text{m}^2$ ]
Re	Reynolds number [-]
St	Strouhal number [-]
$s$	seconds
$T$	temperature [ $\text{K}$ ]
$t$	time [ $\text{s}$ ]
$U$	Total coefficient of heat transfer [ $\text{W}/(\text{m}^2\text{K})$ ]
$V$	velocity [ $\text{m/s}$ ]

#### Greek symbols

$\rho$	density [ $\text{kg}/\text{m}^3$ ]
$\nu$	kinetic viscosity
$\mu$	dynamic viscosity
$\Delta$	difference

#### Subscripts

CF	constant flow
conv	convective
in	inlet
out	outlet
PF	pulsed flow
rad	radiator
sp.	specific heat output

sur surface area  
w water

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Table 1 the material properties hydronic panel radiator model and water

Domain type	Materials type	K [W/m.K]	C <sub>p</sub> [J/kg.m]	ρ [kg/m <sup>3</sup> ]
Fluid medium	incompressible Water	0.58	4180	1000
Channels	Steel (AISI 4340)	44.5	475	7850
Fins	Steel(AISI 4340)	44.5	475	7850

Table 2 the CFD and experimental results and input value of type10 radiator

Specifications	CFD	Exp	Diff %
Flow rate (kg/s)	0.0174	0.0174	
Ambient temperature (°C)	19.89	19.89	
Inlet temperature (°C)	75.23	75.23	
Outlet temperature (°C)	65.28	65.10	0.27
Mean temperature (°C)	70.255	70.17	0.12
LMTD (°C)	50.5	50.28	0.7
Radiator heat output (W)	720.09	736.51	2.23

Table 3 the CFD and experimental results and input value of the single panel radiator with attached fins (type11)

Specifications	CFD	Exp	Diff %
Flow rate (kg/s)	0.023	0.0228	
Ambient temperature (°C)	19.89	19.89	
Inlet temperature (°C)	74.76	74.76	
Outlet temperature (°C)	64.83	64.71	0.15
Mean temperature (°C)	69.79	69.76	0.043
LMTD (°C)	49.25	49.76	1.20
Radiator heat output (W)	992.1	968.42	2.44

Table 4 Pulsed flow amplitudes and average mass flow rate used at constant frequency of 0.209rad/s

Type 11 radiator		Type 10 radiator	
Amplitudes [m/s]	Average mass [kg/s]	Amplitudes [m/s]	Average mass [kg/s]
0.03552	0.003997	0.027	0.0030
0.0406	0.004568	0.031	0.0037
0.04568	0.005138	0.035	0.0039
0.05074	0.005710	0.039	0.0043

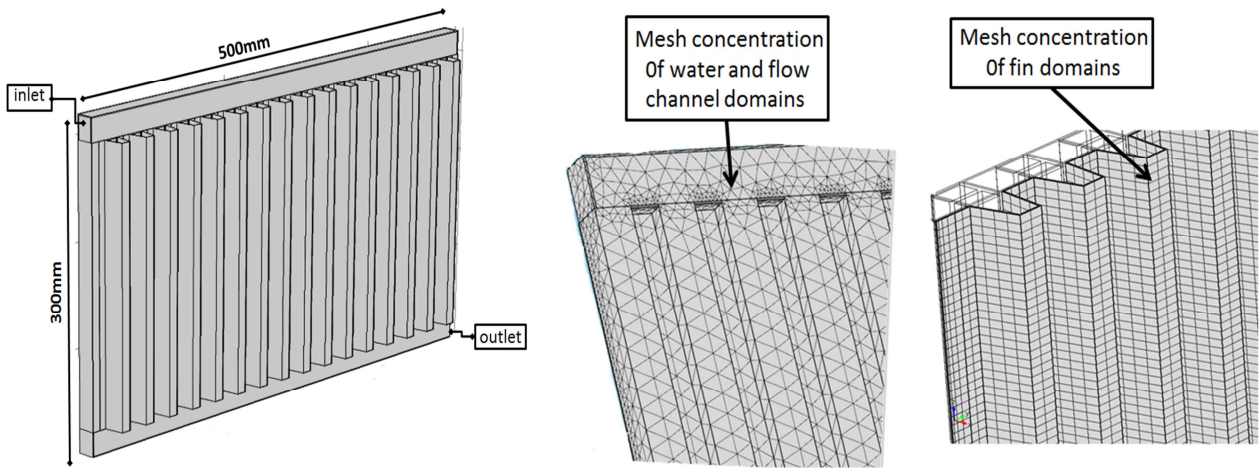


Figure 1 dimensions of the panel radiator geometry and the mesh concentration of the designed radiators

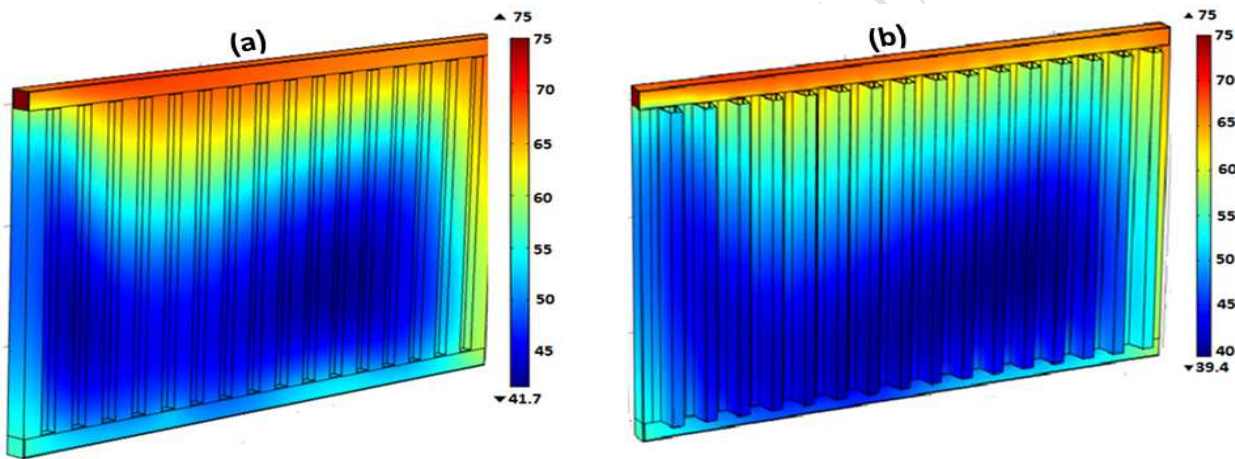


Figure 2 local surface temperature distributions of the radiator (a) type10 and (b) type11 at constant flow condition

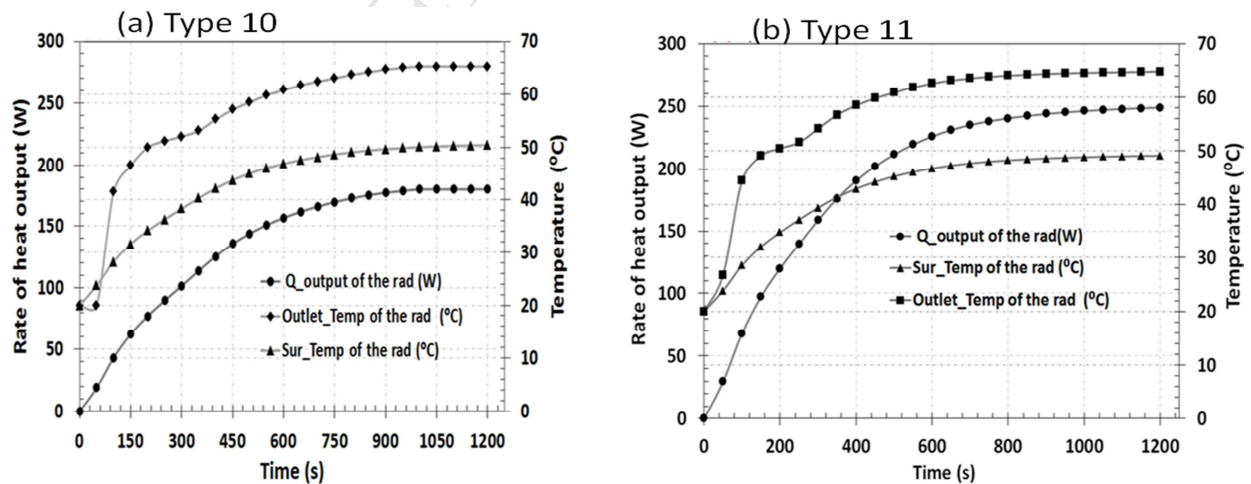


Figure 3 time variation of outlet hot water temperature, rate of heat output and average surface temperature of the radiators (a) type10 and (b) type11 at constant flow condition

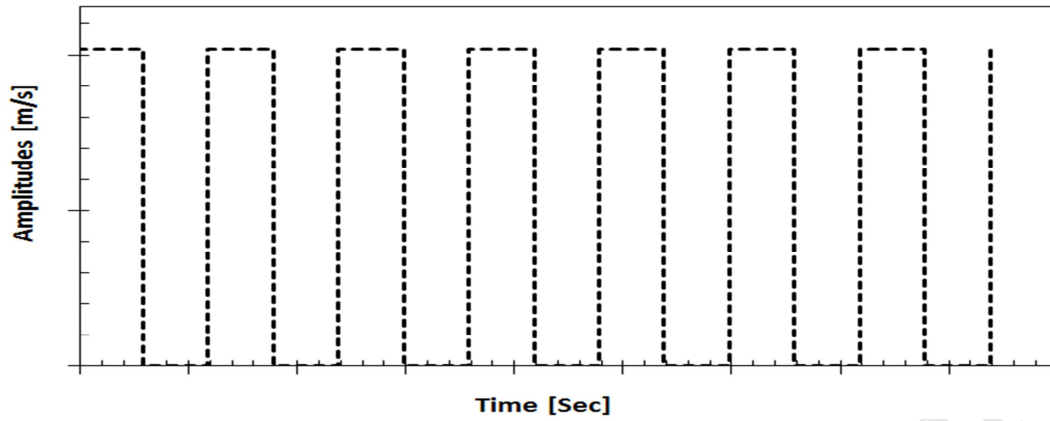


Figure 4 flow input profile of the proposed pulsating flow strategy

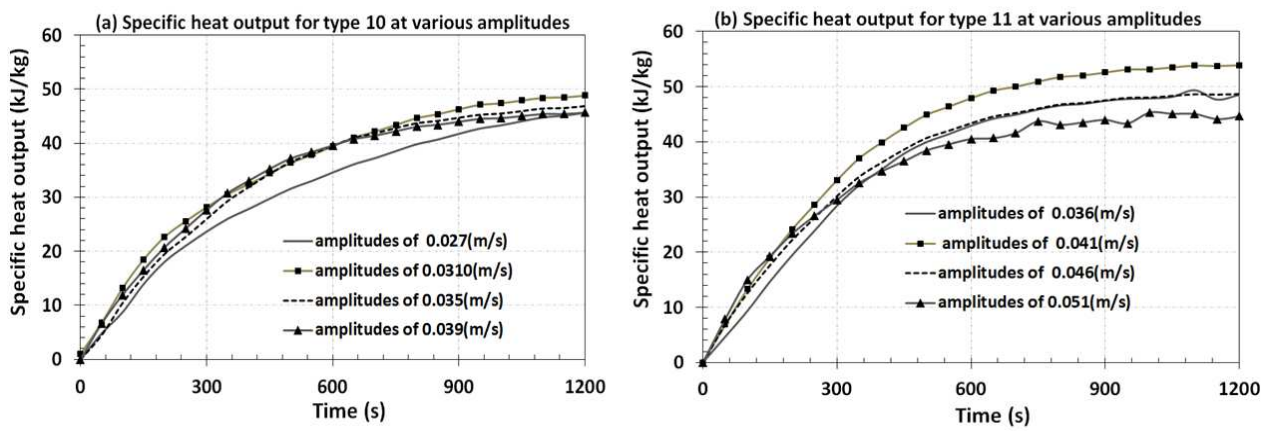


Figure 5 the rate of heat outputs and specific heat outputs of both type11 and typ10 radiators using various flow pulsation amplitudes while maintaining constant frequency of 0.209 rad/s

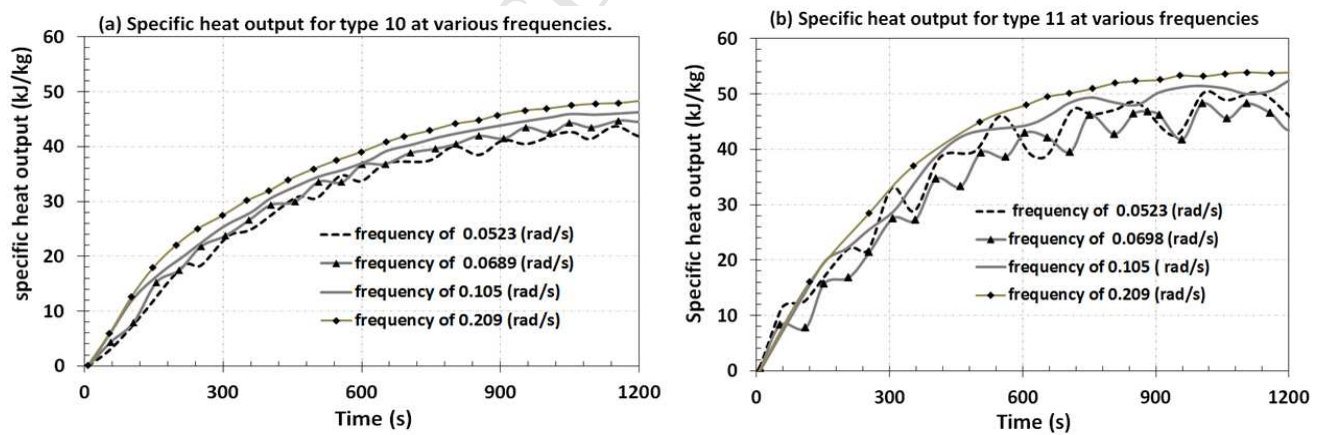
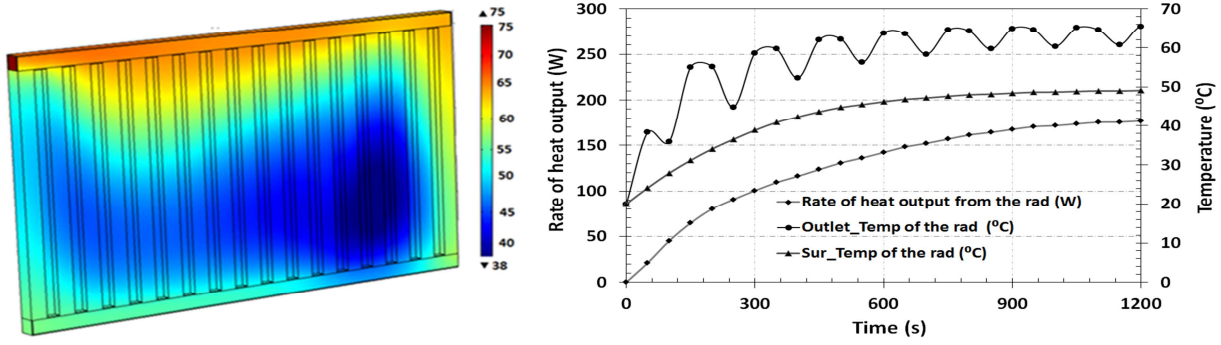


Figure 6 specific heat outputs of (a) type 10 and (b) type 11 radiators using various frequencies while maintaining constant amplitudes of 0.031m/s and 0.0406m/s respectively

(a) Type 10 at flow freq. of 0.209 rad/s and amp of 0.031 m/s



(b) Type 11 at flow freq. of 0.209 rad/s and amp of 0.0406 m/s

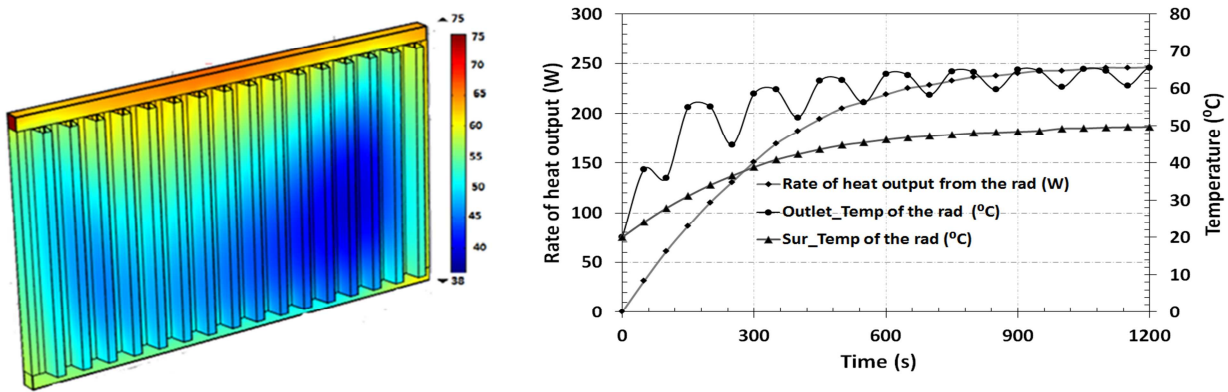


Figure 7 contour of local surface temperature distribution ( $^{\circ}\text{C}$ ); average surface temperature; water outlet temperature and rate of heat output at best pulsation frequencies and amplitudes for both radiators of (a) type 10 and (b) type 11

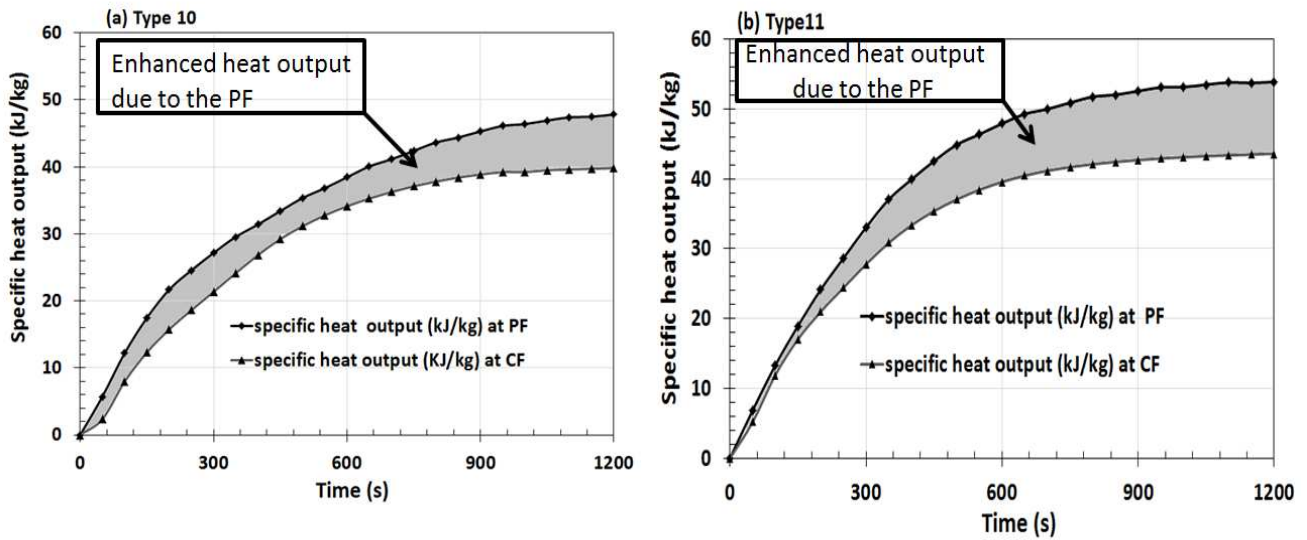


Figure 8 specific heat output and enhanced heat output due to the pulsed flow case compared to the constant flow case of radiators (a) type 10 and (b) type 11



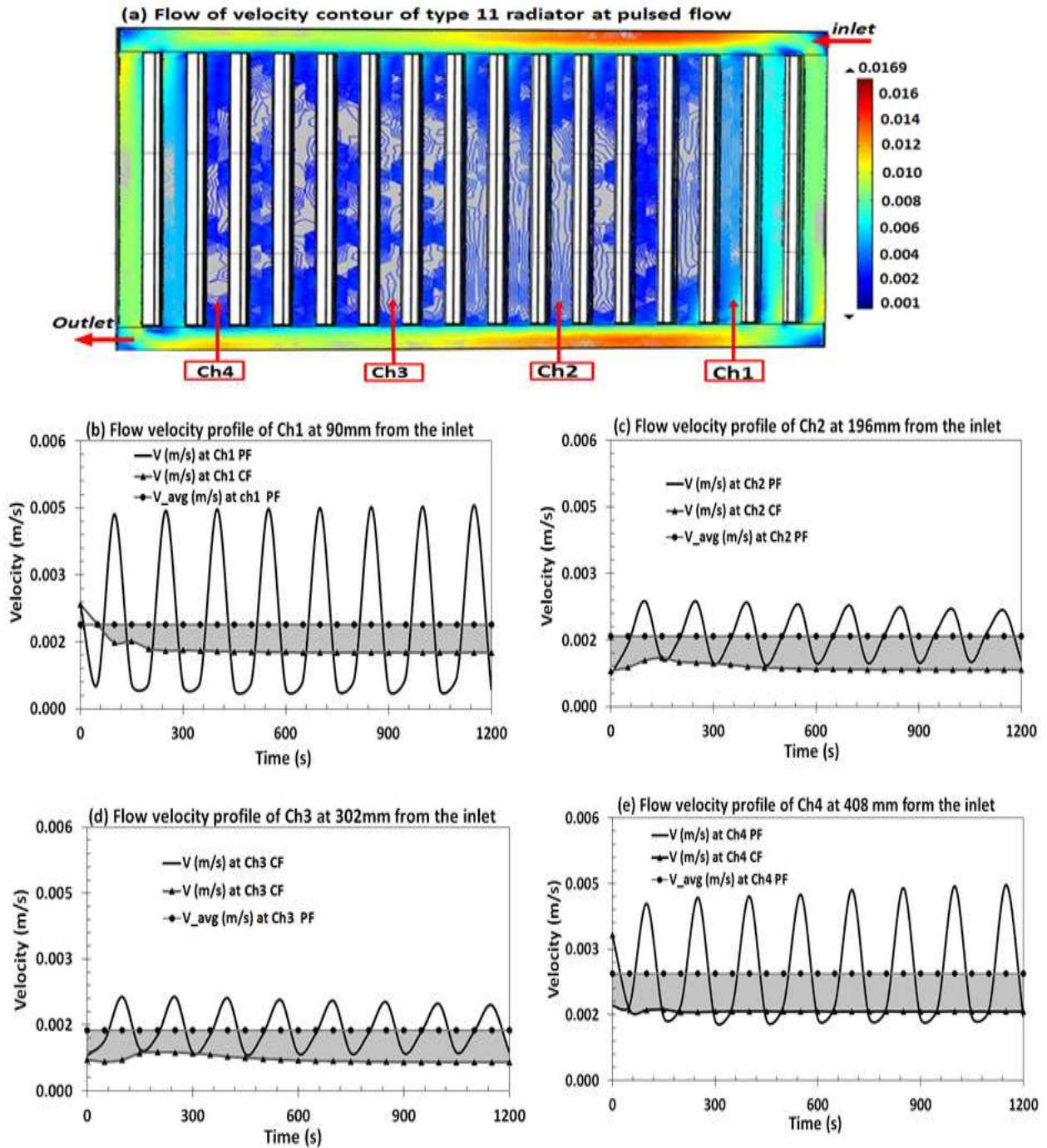


Figure 9 velocity distribution profiles and the enhanced velocity of the proposed pulsed flow compared to the constant flow along the waterside internal skin of the sampled channels indicated in figure9(a)

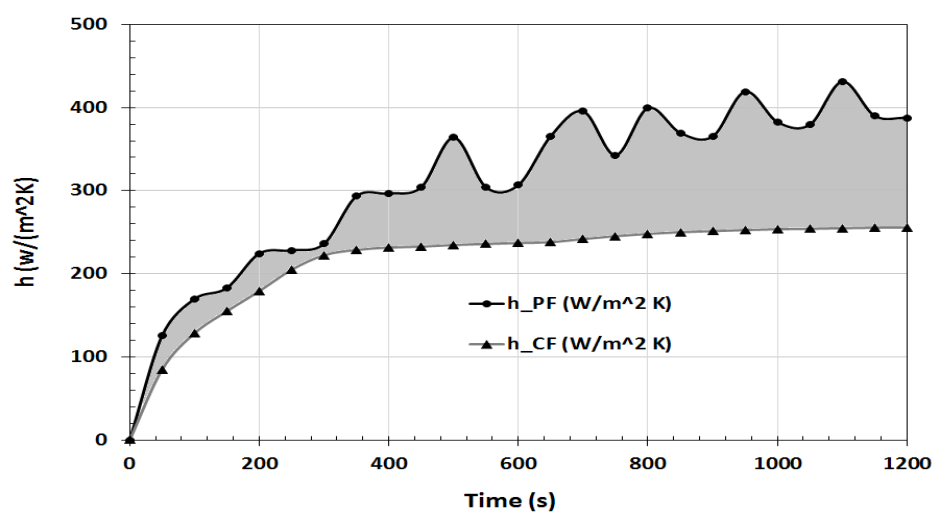


Figure10 the enhanced convective heat transfer coefficient on the waterside (internal) channel walls due to the pulsed flow compared to constant flow

#### Paper Highlights

- 1- CFD simulation of type 10 and type 11 panel radiators with constant and pulsed flow conditions.
- 2- Pulsating the flow enhances the heat transfer performance of panel radiators.
- 3- As flow pulsating frequency increases, the specific heat output of panel radiators increases.
- 4- Pulsating the flow resulted in energy saving of up to 17% for type 10 and 20% for type 11 radiators.