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Analysis and optimization of a Puffer-type water heater

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Abstract

In this paper we present the design optimization of a Puffer-type water heater for sanitary water production in residential buildings. The optimization study is performed with the aim to fulfill the design requirements of the heater, i.e. the provided mass flow rate and the outlet water temperature, while minimizing the surface area of the coil characterizing the Puffer and thus reducing the overall cost of the device. The coil diameter and pitch and the diameter of the pipe composing the coil are analyzed in the optimization study. Starting from available correlations providing the heat transfer rate between the sanitary water and the heated water surrounding the coil, a simplified thermal resistance model of the heat transfer process within the heater is developed. The analysis show a significant impact on the overall heat transfer rate between the sanitary water and the heated water when the geometrical parameters of the coil are varied. Using an optimal value of the pitch to pipe diameter of the coil for a given coil diameter in particular, it is possible to maximize the heat transfer rate and thus minimize the size of the coil. As a first step, the optimization study is performed by introducing sizing constraints allowing for sharing the same coil among several types of water heaters. In this case, a reduction as high as 14% of the coil surface area can be obtained, whereas a reduction as high as 23% can be achieved if the optimization is performed free of such constraints.

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Keywords: Puffer; Water heater; Optimization; Helicoid; Thermal model

1. Introduction

The use of renewable resources such as solar radiation and geothermal energy is becoming increasingly popular in the heating of residential buildings, as they allow for a significant reduction of energy consumption costs as well as for a reduction of traditional (fossil) resources usage, thereby limiting the environmental impact.

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In a residential heating system, whether renewable resources are used or not, a water heater is always present. The water heater is used for hot sanitary water production as well as for the heating system of the envelope and they rely on a primary source of energy. Water heaters suitable to the use of renewable energy sources can be classified in plate heat exchangers, shell and tube heat exchangers, "Tank in Tank" and Puffer-type heaters. Among these, the Puffer-type water heater is the one employed most frequently in residential heating systems, thanks to its specific design allowing for accumulating the hot water provided by the energy sources until the sanitary hot water is requested by the users.

In this work, an optimization study of a Puffer-type water heater based on the modeling of the heat transfer process between the hot water provided by the energy sources and the sanitary water is presented. The analysis is focused on the coil-type heat exchanger characterizing the Puffer, where the heat transfer between the hot water and the sanitary water takes place. Aim of the study is to fulfill the design requirements of the heater, minimizing at the same time the volume of the coil, thereby reducing the overall production cost of the heater.

Nomenclature

d_e	Coil pipe outer diameter (m)
d_i	Coil pipe inner diameter (m)
d_b	Coil diameter (m)`
g	Gravitational acceleration (m/s^2)
h_e	Convective heat transfer coefficient between the hot water and the coil (W/m^{2}K)
h_i	Convective heat transfer coefficient between the sanitary water and the coil (W/m^{2}K)
k	Thermal conductivity of the coil pipe (W/mK)
A_e	Heat transfer surface referred to the coil pipe outer diameter (m^2)
D_e	Dean number
F	Correction factor for the heat transfer between the hot and the sanitary water
H	Coil pitch (m)
H_b	Coil height (m)
L	Coil length (m)
Pr	Prandtl number
R_c	Conductive thermal resistance across the coil pipe (K/W)
Ra	Rayleigh number
Re	Reynolds number
R_{he}	Convective thermal resistance between the hot water and the coil (K/W)
R_{hi}	Convective thermal resistance between the sanitary water and the coil (K/W)
U_e	Global heat transfer coefficient referred to the coil outer surface (W/m^{2}K)
α	Thermal diffusivity (m^2/s)
β	Thermal expansion coefficient of water (1/K)
λ	Curvature ratio
ν	Kinematic viscosity (m^2/s)
ΔT_{lm}	Logarithmic temperature difference between the hot and the sanitary water (K)
HRW	Hot Reservoir Water
SHW	Sanitary Hot Water
CFD	Computational Fluid Dynamics

2. Heat transfer inside the Puffer and thermal model

In Fig. 1, the main components of the Puffer and a sketch of the heat transfer process inside the heater are shown. The heater consists of an insulated tank where the hot reservoir water (HRW) provided by the thermal sources (solar panels, boilers, heat pumps, etc.) is accumulated. The coil-shaped heat exchanger can be noted in the section shown

Fig. 1(a). The coil provides the main heat transfer surface between the HRW and the hot sanitary water (HSW). The HSW is heated while flowing through the coil, which is heated in turn on the outer surface by the HRW.

The sketch of the heat transfer process inside the Puffer is shown in Fig. 1(b). The HRW, after being heated up by the energy sources, is accumulated inside the tank via the inlet pipes mounted on the outer surface of the heater. In this case, the HRW exchanges heat with the coil via forced convection. However, once the filling process is completed and if a gradual draining of the HRW from the heating system is considered, it can be assumed that the heat transfer process between the coil and the HRW will turn into natural convection. Moreover, the hot water tank is carefully insulated to minimize heat losses (blue arrows in Fig. 1(b)) from the hot water. Therefore, it can be further assumed that such losses can be neglected in the overall energy balance of the Puffer.

The HSW flows inside the Puffer via inlet/outlet pipes located on top of the heater, consisting in the inlet/outlet sections of the coil. Under continuous request of HSW from the users, conditions for which the heater is designed, the HSW flows regularly inside the coil, thereby inducing forced convection between the water flow and the inner surface of the coil tube.

If steady state conditions are considered and assuming a request of HSW from the users for a time interval such that the re-filling of HRW can be avoided, then the same amount of heat (red arrows in Fig. 1(b)) will be transferred first from the HRW to the coil by natural convection, then transmitted by conduction through the thickness of the coil tube and ultimately transferred to the HSW by forced convection, as shown in the sketch of Fig. 2(a).

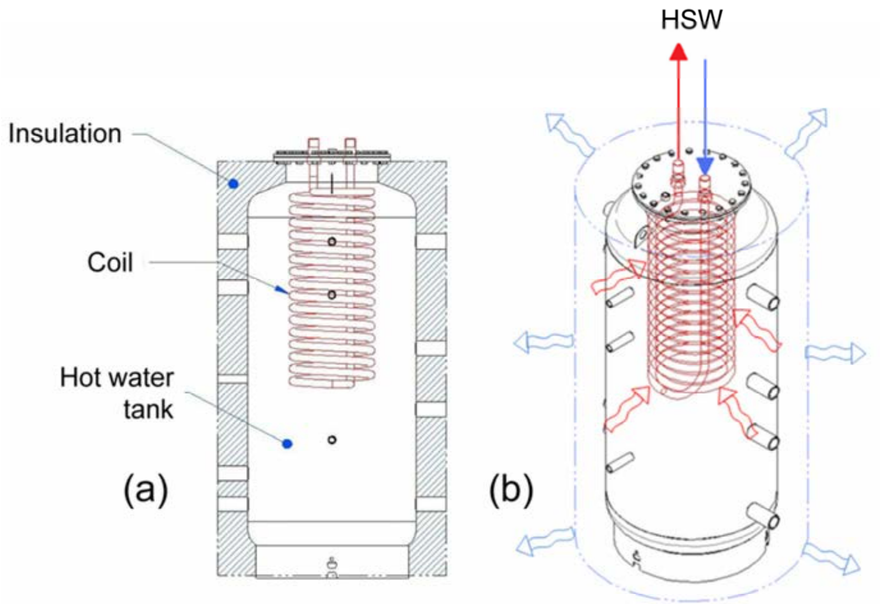

Fig. 1. The Puffer-type water heater: (a) section and main components; (b) sketch of heat fluxes inside the heater.

In such conditions, the heat transfer process between the HRW and the HSW can be modeled via three thermal resistances in series and associated with the three main heat transfer processes [1], as shown in Fig. 2(b). The transmitted heat power can be then calculated as follows:

$$Q = F U_e A_e \Delta T_{lm} \tag{1}$$

where Q is the total heat power exchanged throughout the entire length of the coil, U_e is the global heat transfer coefficient referred to the outer surface of the coil, ΔT_{lm} is the logarithmic temperature difference between the HRW and the HSW, A_e the outer surface area of the coil and F a correction factor.

In the operating conditions assumed in the present study, the global heat transfer coefficient is a function of the three thermal resistances and can be calculated as follows:

$$U_e A_e = \frac{1}{R_t} = \left(\frac{1}{\pi d_e L h_e} + \frac{1}{2\pi k L} \ln \frac{d_e}{d_i} + \frac{1}{\pi d_i L h_i} \right)^{-1} \tag{2}$$

where R_t is the total thermal resistance given by the sum of the thermal resistances associated with the natural convection outside the coil, the conduction across the thickness of the coil tube and the forced convection inside the coil. Likewise, h_e and h_i represent the heat transfer coefficient associated with the natural and forced convection, respectively, k is the thermal conductivity of the coil tube, d_e and d_i are the outer and inner diameter of the coil tube and L the length of the coil.

The logarithmic temperature difference between the HRW and the HSW along the coil is given by:

$$\Delta T_{lm} = \frac{\Delta T_2 - \Delta T_1}{\ln(\Delta T_1 / \Delta T_2)} \tag{3}$$

where ΔT_1 and ΔT_2 represent the temperature difference between the HRW and the HSW at the inlet and outlet sections of the Puffer, respectively. As a first order approximation, it can be assumed that the HRW temperature within the tank is uniform and constant, whereas the HSW inflow and outflow temperature represent two design parameters. In such conditions, the expression (3) can be used explicitly to compute the logarithmic temperature difference.

It should be noted, however, that the logarithmic temperature difference can be strictly obtained by Eq. (3) for an ideal co-flow or counter-flow heat exchanger only, whereas a correction factor should be introduced in case of more complex flow arrangements, as in the case of the Puffer. Nonetheless, due to the lack of reference data for the heat transfer in the coil, a correction factor F=1 is assumed in the present study.

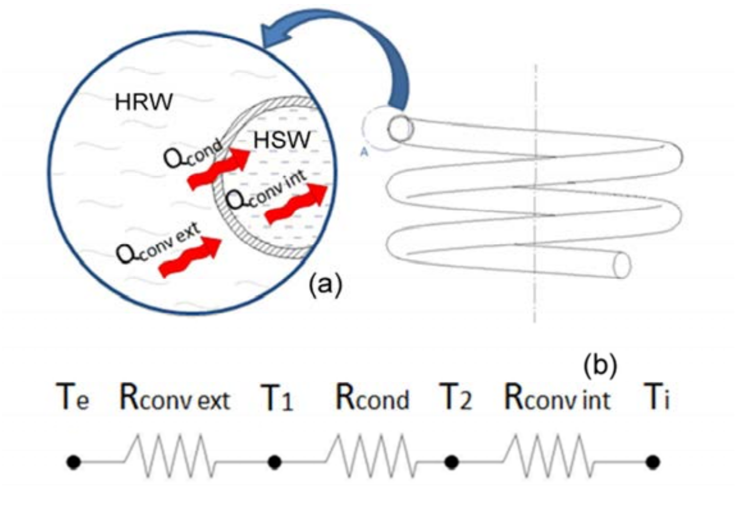

Fig. 2. Heat transfer process in the coil: (a) sketch; (b) thermal resistance model.

2.1 Heat transfer inside the coil

The heat transfer taking place inside the coil is quite complex due to the effect of the significant curvature associated with the flow of the HSW along the tube. The curvature affects the convective heat transfer coefficient inside the coiled tube, which is now a function of the coil geometry. The geometrical parameters characterizing the coil are the inner diameter of the coiled pipe d_i, the diameter of the coil d_b and the coil pitch H. The analysis of research work available in the literature shows that the curvature ratio $\lambda=d_i/d_b$ and the Dean number Dean $\mathrm{De}=\mathrm{Re}(d_i/d_b)^{1/2}$, where Re is the Reynolds number $\mathrm{Re}=U_b d_i/\nu$ associated with the flow inside the coil, represent the parameters dominating the heat transfer in flows within helical pipes. In the present study, the convective heat transfer coefficient inside the coil is obtained via the correlations proposed by Jayakumar et al. [2], established by a CFD analysis of the flow and heat transfer inside helical pipes. The data presented in Tabs. 1, 2 and 3 show the effect of the geometrical parameters of the coil on the average Nusselt number $\mathrm{Nu}=h_i d_i/k$ associated with the turbulent flow inside the pipe, where h_i is the average heat transfer coefficient along the entire length of the coil.

Table 1. Effect of coil diameter on the average Nusselt number inside the coil [2].

	d_b=100mm	d_b=200mm	d_b=300mm	d_b=400mm	D_b=1000mm	d_b=3000mm
Nu	226.33	208.33	197.65	192.04	173.08	157.83

Table 2. Effect of coil pipe diameter on the average Nusselt number inside the coil [2].

	d_i=10mm	d_i=20mm	d_i=30mm	d_i=40mm
Nu	116.39	200.03	289.56	377.38

Table 3. Effect of coil pitch on the average Nusselt number inside the coil [2].

	H=15mm	H=30mm	H=45mm	H=60mm
Nu	191.08	191.75	192.97	192.55

The data reported in Tabs. 1 and 2 show how the reduction of the coil diameter and the increase of the coil tube diameter give rise to a significant increase of the Nusselt number. Such increase is due to the higher curvature of the HSW flow inside the coil, inducing stronger secondary flows able to enhance the heat transfer between the water flow and the tube surface. Tab. 3 shows how the increase of the coil pitch is also able to enhance the heat transfer inside the coil. However, such increase is considerably weaker when compared to the effect of the coil diameter and of the coil tube diameter.

The correlation proposed by Jayakumar et al. [2], where the most significant flow and geometrical parameters only are included, is thus the following:

$$\mathrm{Nu} = 0.116\,\mathrm{Re}^{0.71}\,\mathrm{Pr}^{0.4}\,\lambda^{0.11} \tag{4}$$

where $\mathrm{Pr}=\nu/\alpha$ is the Prandtl number. The correlation (4) is limited to the following flow conditions:

$$\begin{gathered} 1.4\times10^4 \le \mathrm{Re} \le 7.0\times10^4 \\ 3.0\times10^3 \le \mathrm{De} \le 2.2\times10^4 \\ 3.0 \le \mathrm{Pr} \le 5.0 \end{gathered} \tag{5}$$

2.2 Heat transfer outside the coil

In the conditions assumed in the present study, the heat transfer between the HRW and the coil takes place by natural convection. However, the heat transfer is also influenced by the proximity of the coil sections determined by the coil pitch [3]. In order to take into account for such effect in the thermal model of the Puffer, the correlation proposed by Heo & Chung [4] is employed:

$$\mathrm{Nu} = 0.54 Ra^{0.25}\left[1 - N\left(0.072 - 0.065\frac{H}{d_e} + 0.012\left(\frac{H}{d_e}\right)^2\right)\right] \tag{6}$$

which is valid for the following flow conditions:

$$5.5\times10^5 \le \mathrm{Ra} \le 9.4\times10^8$$

$$H / d_e \le 4.0 \tag{7}$$

where N is the *n-th* coil section for which the Nusselt number is calculated and $\mathrm{Ra}=g\beta\Delta T d_e^3/(\nu/\alpha)$ is the Rayleigh number. The analysis of the correlation (6) shows that the Nusselt number is maximum at H/d_e=2.6, a condition for which the heat transfer by natural convection between the HRW and the coil is thus the highest.

3. Results

In this section, the results from the optimization study of the coil characterizing the Puffer-type water heater are presented. The optimization is carried out by satisfying the design constraints given by the flow rate of the HSW through the heater, which is of 24.5 lt/min, and the inlet/outlet temperature of the HSW from the heater, which are of 10°C and 45°C, respectively. The heat power Q needed to heat up the HSW (see Eq. (1)) is then given by the enthalpy balance applied to the flow of the HSW and is equal to about 60kW. The HRW temperature within the tank is 75°C and is assumed uniform and constant, giving a ΔT_{lm} of about 45°C. From this data, the target value of the product U_eA_e given by Eq. (1) is about 1334 W/K.

In the following sections, the optimization study of the coil based on additional geometrical constraints is reported first, whereas an optimal design free of such constraints is presented next.

3.1 Optimal design with geometrical constraints

In the first optimization study, additional geometrical constraints allowing for the use of the coil among a class of Puffer-type water heater are introduced. Such constraints are: a maximum height of the coil H_b of 1600mm, a coil diameter d_b of 510mm and a coil tube diameter d_i =41mm, leaving the pitch as the only parameter that can be varied. The new design is compared to the coil currently employed in the Puffer, characterized by an outer surface area of 2.1m^2.

The analysis performed by fixing the constraints on d_i and d_b and by varying the pitch H shows that the length of the coil able to satisfy the design constraints is minimum for H/d_e=2.6, for which the convective heat transfer coefficient associated with the natural convection around the coil is the highest. The optimal design of the coil obtained in such conditions, shown in Tab. 4, gives a reduction of the coil surface area of about 14%.

Table 4. Optimal design of the coil with geometrical constraints..

H/d_e	L (m)	H_b (m)	A_e (m^2)
2.6	14.11	0.94	1.81

3.2 Optimal design without geometrical constraints

In order to further optimize the size of the coil when the additional geometrical constraints are not taken into account, a sensitivity study towards establish the effect of the coil diameter and the coil tube diameter on the length of the coil needed to fulfill the design requirements is performed first. In both cases, the coil pitch H is varied such that the optimal ratio H/d_e=2.6 for the natural convection outside the coil is maintained.

The results obtained by varying the coil diameter d_b are shown in Tab. 5, where the size of the coil satisfying the design constraints is reported. As it can be noted, the higher mixing rate in the HSW flow associated with the reduction of the coil diameter is such to give a significant reduction of the coil length as well as of the coil outer surface area. However, the reduction of the coil diameter also gives a significant increase of the height of the coil.

Table 5. Effect of coil diameter on other coil dimensions.

	d_b=300mm	d_b=400mm	d_b=500mm	d_b=600mm	d_b=700mm
L (m)	13.88	14	14.09	14.17	14.24
A_e (m2)	1.78	1.8	1.82	1.83	1.84
H_b (m)	1.609	1.22	0.948	0.862	0.71

The effect of the coil tube diameter is reported in Tab. 6. In this case, reducing the coil tube diameter gives an increase of the coil length. Nonetheless, the coil outer surface area and the height of the coil are both reduced due to the smaller diameter of the coil tube.

Table 6. Effect of coil pipe inner diameter on coil dimensions.

	d_i=35mm	d_i=38mm	d_i=41mm	d_i=44mm	d_i=47mm
L (m)	15.01	14.52	14.11	13.77	13.48
A_e (m2)	1.65	1.73	1.81	1.9	1.99
H_b (m)	0.85	0.89	0.94	0.98	1.03

After completing the sensitivity study, an optimal design is achieved by varying both the coil diameter and the coil tube diameter while maintaining the optimal pitch to diameter ratio H/d_e=2.6. The optimal design, presented in Tab. 7, shows that in this case a reduction of the coil outer surface area of about 23% can be achieved.

Table 7. Effect of coil pipe inner diameter on coil dimensions.

d_b (m)	d_i (m)	H/d_e	L (m)	H_b (m)	A_e (m^2)
0.3	0.035	2.6	14.81	1.42	1.62

4. Conclusions

In this work a design optimization study of a Puffer-type water heater has been presented. The optimization study is performed with the aim of satisfying the design requirements of the heater, consisting in the processed flow rate and in the outlet temperature of the sanitary water. Using a simplified thermal resistance model of the heat transfer process between the hot reservoir water and the sanitary water, an optimal design for the coil-shaped heat exchanger characterizing the Puffer is obtained where the heat transfer rate is maximized, thereby reducing the surface area of the coil and the production cost of the heater.

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