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Study on Heat Transfer Characteristics and Performance of the Full Premixed Cauldron Stove with Porous Media

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Abstract: The cauldron stoves used in restaurants and canteens usually adopt the combustion mode of blast diffusion. Low combustion efficiency leads to low thermal efficiency and high CO and NOx emissions. To address these problems, a 52 kW fully premixed stove with porous media is designed, and the heat transfer characteristics of the stove are analyzed by theoretical analysis and numerical simulation. The results show that under the rated power, the thermal efficiency of the stove reaches 68.55%, which is more than twice the thermal efficiency of the traditional blast diffusion stove. Among them, the radiant heat efficiency of the stove reaches 47.16%; thus, radiation heat transfer has become an important way of heat transfer of the porous media stove. Moreover, increasing the diameter and emissivity of porous media will increase the radiant thermal efficiency of the stove, but it will significantly reduce the flame temperature. In addition, the influence of the diameter is greater than the emissivity. The increase of the thickness of porous media can significantly improve the preheating temperature of the premixed gas, thus improving the ignition performance of the stove. Additionally, the stove has an appropriate thickness (approximately 3 mm), which not only ensures the preheating temperature but also does not easily allow for breakage and damage of porous media. Increasing the pore density or reducing the porosity of porous media can enhance the ignition performance of the stove. Moreover, the results of numerical simulation verify the theoretical results to a certain extent and shows that there is an optimal flue position as well.

Keywords: fully premixed combustion; porous media; heat transfer characteristics; thermal efficiency; numerical simulation

Citation: Zheng, D.; Su, L.; Ou, H.; Ruan, S. Study on Heat Transfer Characteristics and Performance of the Full Premixed Cauldron Stove with Porous Media. *Energies* **2022**, *15*, 9523. https://doi.org/10.3390/en15249523

Academic Editors: Li Chen, Xiaoying Zhang and Feifei Qin

Received: 9 November 2022
Accepted: 8 December 2022
Published: 15 December 2022

1. Introduction

Gas cookers are common cooking equipment in kitchens. Their large retention and usage rates make their gas consumption huge. The output of commercial gas stoves in China was 39.464 million in 2018, and the number of gas stoves in China increased at a rate of approximately 7% per year [1]. The traditional commercial cauldron stove adopts mainly the combustion method of blast diffusion, which has low combustion efficiency and high CO and NOx emissions. In addition, the thermal efficiency is generally not higher than 35% [2]. This leads to great energy waste and air pollution. Although there are some low-power porous medium stoves on the market, the application of porous medium combustion technology in the field of gas stoves has not been popularized, and high-power commercial caldron stoves still need to be developed. Therefore, the traditional commercial cauldron stove used in canteens, hotels, and restaurants requires more efficient and cleaner combustion technologies.

Full premixed combustion [3] is a complete combustion method with low CO and NO_X emissions. The fuel is fully mixed with air equal to or greater than all the air required for combustion in advance, and then it is sent to the combustion chamber for combustion. The research shows that [4] fully premixed combustion technology has high combustion intensity and relatively short flame. In addition, under the fully premixed combustion technology, the flame surface is relatively close to the heat exchanger, which can greatly improve the heat transfer coefficient and reduce the ratio of CO and NOx in the combustion process. However, the flame stability of fully premixed combustion is poor and is prone to tempering and de-flaming.

The technology of using porous media (PM) materials involved in combustion is called porous media combustion (PMC) technology [5]. It has a wide combustion area and uniform temperature distribution [6], so the combustion flame can be stabilized inside or on the surface of porous media. Meanwhile, the wider combustion reaction zone extends the residence time of the gas in the combustion zone [7], resulting in more complete combustion and lower CO production. This provides a new way for the improvement of the cauldron stove. Moreover, many scholars have studied the flow and heat transfer properties of porous media and porous media combustion technology, and it has been applied in many fields [8–12].

Scholars have designed different types of energy-saving cookers on the basis of different ways to save energy, enhance heat transferring, improve combustion efficiency, etc. Pankaj P. Gohil [13] et al. designed a new inverted conical flame heat shield. The heat insulation material was used to reduce heat loss from fuel combustion and improve the thermal efficiency of the stove. It was found that the maximum thermal efficiency could reach 74.7%, which was approximately 8% higher than that of domestic liquefied petroleum gas (LPG) stoves without heat shields. Wang [14] et al. enhanced the heat transfer between the stove and the bottom of the pan by installing fins on the outer wall of the pan, increasing the heat transfer area and flame disturbance. Through simulation and experimental research, it was found that with the addition of fins, the peak thermal efficiency of the LPG stoves reached 60.28%, which was approximately 8.2% higher than the LPG stoves without fins.

Meanwhile, some scholars have designed various porous media gas stoves on the basis of PMC technology. V.K. Pantangi [15] et al. designed a porous radiant burner used for liquefied petroleum gas domestic cooking stoves. The burner was composed of two-layer porous media. The combustion zone consisted of silicon carbide, and alumina balls formed the preheating zone. The experimental results showed that the maximum thermal efficiency of this gas stove was 68%, which was 3% higher than the conventional domestic LPG cooking stoves. Lav Kumar Kaushik [16] et al. presented the economic and environmental impact of a double porous media gas cooker (Mishra [17]) rated at 10 kW. The results showed that gas cookers with porous radiant burners have higher thermal efficiency, longer life cycle, and lower CO and NOX emissions, making them a better choice to replace conventional gas cookers.

In this paper, in order to improve the thermal efficiency of traditional caldron stoves, a 52 kW cauldron stove is designed on the basis of full premixed combustion and porous media combustion technology. Compared with the traditional caldron stove, its combustion mode is changed from diffusion combustion to full premixed combustion; the main heat transfer mode is changed from convection heat transfer to convection and radiation heat transfer. To solve the proposed model, theoretical analysis and numerical simulation methods are used. A theoretical model on the basis of heat transfer and thermodynamics is established, and the full premixed combustion with porous media combustion is theoretically coupled. The performance of the cooker is analyzed; the influence of the porous media, chamber structure, and other factors on the combustion heat transfer performance of the stove are also analyzed by theoretical analysis and numerical simulation. Various conclusions obtained from the theoretical analysis are qualitatively and quantitatively compared with the experimental results in the literature, which are in good agreement. Meanwhile, the results of numerical simulations are close to the theoretical results. Thus, a theoretical basis for the design of other power cauldron stoves in the future can be provided.

2. Theoretical Model

2.1. Structure and Working Principle of Fully Premixed Porous Media Cauldron Stove

The structure of the full premixed porous media cauldron stove is shown in Figure 1. The working principle of the stove is that the fully premixed gas is sent to the flow equalizing chamber through the gas transmission pipeline and then enters the porous media for convective heat exchange with the porous media skeleton and is preheated. Finally, it is ignited and burned at the flame surface on the upper surface of the porous media. The high-temperature flue gas generated by combustion carries out convective heat exchange with the bottom of the pot and is discharged from the stove through the smoke extraction channel.

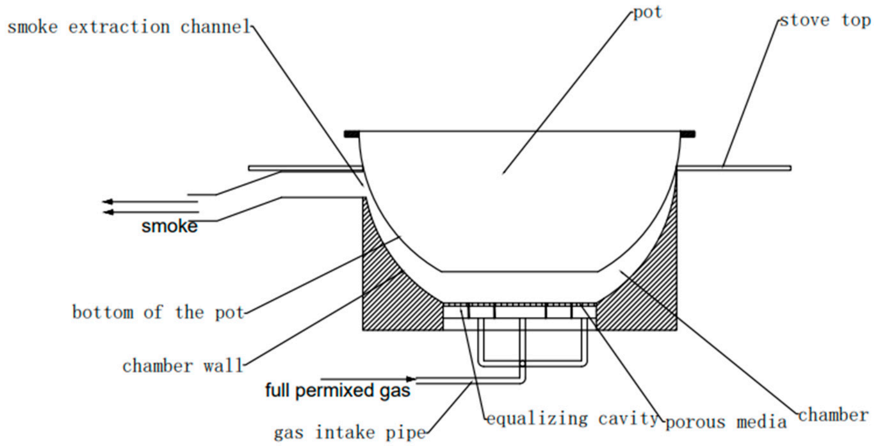


Figure 1. Structure of fully premixed porous media cauldron stove.

Figure 2 shows the structure of porous media, which is composed of porous media, equalizing cavity, and partitions.

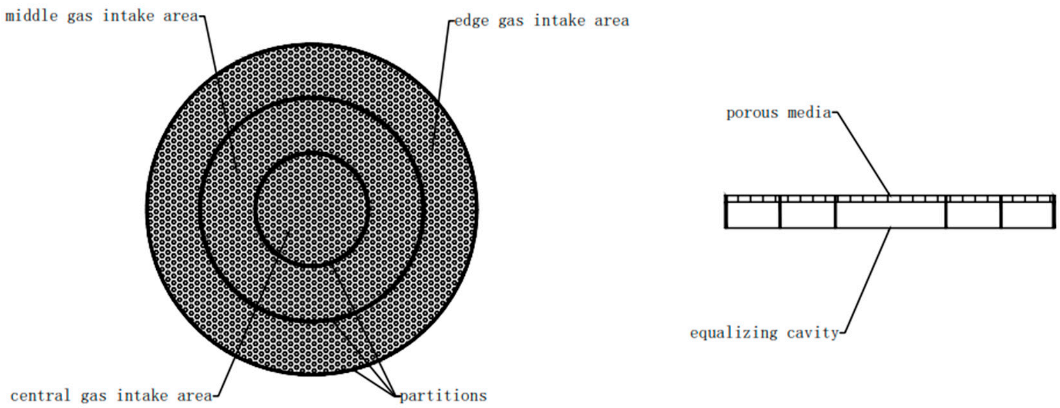


Figure 2. Structure of porous media.

2.2. Theoretical Model

The combustion heat transfer theoretical model of fully premixed gas in the stove is established, including the heat balance equation in the stove chamber, the heat balance equation of the flame surface, the heat transfer model inside the porous media, and the radiation heat transfer model of the closed stove chamber. Through the correlation and coupling of heat between models, the flame temperature, chamber wall temperature, and exhaust gas temperature of the stove, the preheating temperature of premixed gas inside the porous media, as well as the radiant and convective heat transfer of the stove, are obtained through iterative calculation. Finally, the thermal efficiency and heat losses of the stove are obtained. The following assumptions are made in the theoretical model: ① The temperature at the bottom of the pot is uniform; ② Since the high-temperature plane flame is formed close to the upper surface of porous media, it is assumed that the upper surface temperature of the porous media is equal to the flame temperature; ③ The heat dissipation of the stove wall is small, so the chamber wall is regarded as an adiabatic surface in the radiation heat transfer model; and ④ The natural gas is fully premixed with air, and the equivalence ratio of the mixed gas is 0.95.

2.2.1. Heat Balance Equation of the Stove Chamber

Heat balance equation in stove chamber: The heat released by the fuel in the combustion is equal to the sum of the radiant heat of porous media, the convective heat of flue gas, the heat dissipated by the stove wall, and the heat discharged by flue gas, as shown in Figure 3.

$$Q_Z = Q_f + Q_d + Q_s + Q_y \quad (1)$$

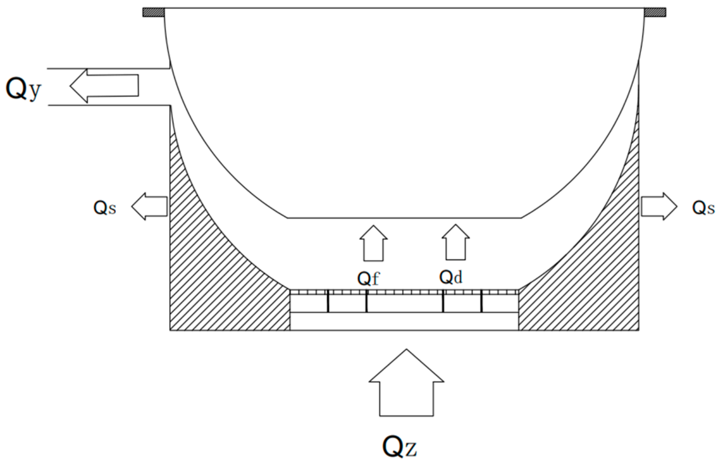


Figure 3. Heat distribution of stove.

Heat balance equation of flame surface: At the flame surface of the stove, the heat released by the combustion of fuel in the stove is equal to the sum of the radiant heat of the PM and the heat absorption of the high-temperature flue gas on the flame surface.

$$Q_Z = Q_f + Q_w \quad (2)$$

$$Q_Z = Q_c \cdot q_v \quad (3)$$

where Q_Z is the heat release of gas combustion; Q_f is the radiant heat of the stove; Q_d is flue gas convection heat; Q_s is the heat dissipation of the chamber wall; Q_y is the heat emitted by flue gas; Q_w is the heat absorption of high-temperature flue gas at flame surface, kW; Q_c is the low calorific value of gas, kJ/kg; and q_v is the volume flow of premixed gas, Nm^3/s.

(1) Radiant heat equation of porous media

Radiant heat of the porous media [18]:

$$Q_f = \frac{\sigma({T_h}^4 - {T_g}^4)}{R_z} \quad (4)$$

The stove chamber is a closed stove chamber, which is composed of a porous media, a stove chamber wall, and a pot bottom surface. The radiation process is the radiation heat transfer from the porous media to the pot bottom surface. Since the stove wall is an adiabatic wall, the radiation model adopts the three-surface double radiation surface model. Among them, Points 1, 2, and 3 correspond to the three radiation surfaces of the porous media, the chamber wall, and the pot bottom, respectively. The radiation network diagram is shown in Figure 4:

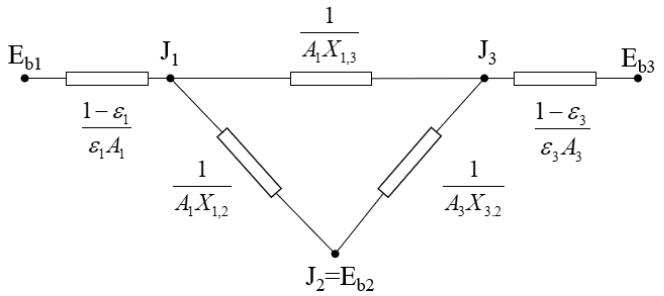


Figure 4. Radiation network diagram.

Radiant thermal resistance:

$$R_Z = \frac{1}{\frac{1}{\frac{1}{A_1X_{1,2}}+\frac{1}{A_3X_{3,2}}}+\frac{1}{A_1X_{1,3}}} + \frac{1-\varepsilon_1}{\varepsilon_1 A_1} + \frac{1-\varepsilon_3}{\varepsilon_3 A_3} \quad (5)$$

where R_Z is the total thermal resistance of space radiation, $m^2 \cdot K/W$; σ is the blackbody radiation constant, $\sigma = 5.67 \times 10^{-8} W/(m^2 \cdot K^4)$; T_g is the pot bottom temperature, °C; T_h is the flame temperature, K; ε_1 and ε_3 are the equivalent emissivity of the stove and the emissivity of the lower surface of the pot bottom, respectively; $X_{1,2}$, $X_{3,2}$, and $X_{1,3}$ are the angle coefficients between the three radiation surfaces in the stove; A_1 and A_3 are the surface areas of porous media and pot bottom, m^2, respectively.

(2) Convection heat transfer equation of high temperature flue gas on boiler bottom

This includes the heat transfer between the flue gas and the central area of the pot bottom, the heat transfer between the flue gas and marginal zone of the pot bottom, and the radiant heat transfer of the flue gas to the pot bottom.

$$Q_d = Q_{d1} + Q_{d2} + Q_{f2} \quad (6)$$

where Q_{d1} is the convective heat transfer between the flue gas and the central area of the pot bottom; Q_{d2} is the convective heat transfer between the flue gas and marginal zone of the pot bottom; and Q_{f2} is the radiant heat transfer of the flue gas to the pot bottom, kW.

Heat transfer between the flue gas and the central area of the pot bottom:

$$Nu = 1.29\mathrm{Pr}^{0.4}\mathrm{Re}^{0.5} \quad (7)$$

$$Q_{d1} = \frac{Nu \cdot \lambda}{h_z} S_{d1}(T_h - T_g) \quad (8)$$

where λ is the thermal conductivity of flue gas, $(W/m^2 \cdot K^{-1})$; h_z is the distance from the porous media to the bottom of the pot, m; and S_{d1} is the area of the impinging jet area at the center of the pot bottom, m^2.

Heat transfer between the flue gas and marginal zone of the pot bottom [19]:

$$Nu = 2 + (0.4Re^{0.5} + 0.06Re^{\frac{2}{3}})Pr^{0.4}\frac{\eta_q}{\eta_p} \quad (9)$$

$$Q_{d2} = \frac{Nu \cdot \lambda}{h_z} S_{d2}(T_y - T_g) \quad (10)$$

where η_q is the flue gas dynamic viscosity calculated by the flue gas temperature and η_p is the flue gas dynamic viscosity calculated by the boiler bottom temperature, Pa·s; S_{d2} is the heat exchange area between flue gas and pot bottom, m^2; and T_y is the average temperature of flue gas in the stove, K.

Radiant heat transfer of the flue gas to the pot bottom [20]:

$$Q_{f2} = S_{d2} \cdot \frac{a(a_g + 1)}{2} \cdot \sigma({T_y}^4 - {T_g}^4) \quad (11)$$

$$a = 1 - 2.718^{-kP\frac{3.6V}{F}} \quad (12)$$

$$k = (\frac{(7.8 + 16r_h)}{\sqrt{3.16Pr \cdot \frac{3.6V}{F}}} - 1)(1 - \frac{0.37T_y}{1000}) \cdot r \quad (13)$$

where F is the surface area of the chamber wall, m^2; a and a_g are smoke blackness and pot bottom blackness, respectively, $W/(m^2 \cdot K^4)$; T_g is the bottom temperature of the pot; k is the flue gas attenuation coefficient, $1/(m \cdot MPa)$; r and r_h are the volume share of triatomic gas and the volume share of water vapor, respectively; V is the effective volume of the area in the stove, except the flame jet, m^3; and P is the pressure in the stove, P = 0.101325 MPa.

(3) Thermal efficiency equation of stove

$$k = (\frac{(7.8 + 16r_h)}{\sqrt{3.16Pr \cdot \frac{3.6V}{F}}} - 1)(1 - \frac{0.37T_y}{1000}) \cdot r \quad (14)$$

where η is the total thermal efficiency of stove.

2.2.2. Heat Transfer Model Inside the Porous Media:

In the porous media, the heat is transferred downward along the skeleton. After the premixed gas enters the porous media, it carries out convective heat exchange with the skeleton and is preheated.

$$Q_x = Q_{d3} = Q_r \quad (15)$$

where Q_x is the downward thermal conductivity of the stove along the skeleton; Q_{d3} is the convective heat transfer between the premixed gas and the skeleton of the porous media; and Q_r is the heat absorption of premixed gas inside the porous media, kW.

Downward thermal conductivity of the stove along the skeleton:

$$Q_{dr} = \lambda \cdot \frac{(A_1 - S_k)(T_h - T_x)}{\delta} \quad (16)$$

Convective heat transfer equation between the premixed gas and the skeleton of porous media:

$$Q_{d3} = h \cdot S_n \cdot \left(\frac{(T_h + T_x) - (T_0 + T_r)}{2} \right) \quad (17)$$

$$h = \frac{Nu \cdot \lambda}{d_k} \quad (18)$$

$$Nu = 3.6 \times 10^{-3} \mathrm{Re}^{1.4} \quad 0.1 \le \mathrm{Re} \le 20 \quad (19)$$

$$S_n = \pi d_k l \cdot n \quad (20)$$

Heat absorption of premixed gas inside the porous media:

$$Q_{yr} = c_{pm} \cdot q_m(T_r - T_0) \quad (21)$$

where δ is the thickness of porous media, m; T_x is the lower surface temperature of the porous media, °C; S_n is the total inner surface area of the pores; d_k is the diameter of the pores; l is the thickness of the stove, m; n is the number of pores; c_{pm} is the average constant pressure specific heat of premixed gas, $kJ/(kg \cdot K)$; and T_{yr} is the preheating temperature, °C.

2.3. Numerical Simulation of Heat Transfer Characteristics of the Stove

2.3.1. Physical Model

The physical model of the stove is established, as shown in Figure 5; the symmetry function is adopted for the cooktop model in order to reduce the number of grids and simplify the calculation. The diameter of the porous media is 0.54 m, the diameter of the stove is 1.1 m, the height of the stove is 0.49 m, the height of the stove in the central area is 0.05 m, and the shape of the flue is a square with a side length of 0.11 m, which is arranged on the top edge of the stove.

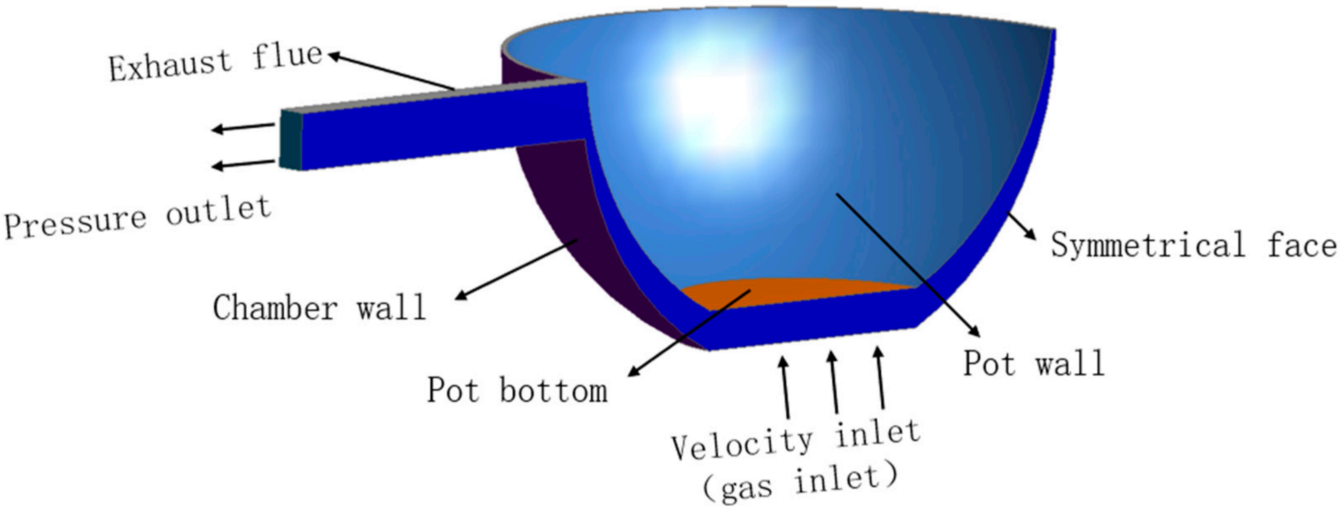


Figure 5. Physical model of stove chamber.

2.3.2. Model Assumptions

When the cooker is running stably, the mass flow rate of the inlet and outlet gas of the cooktop remains constant, and the physical quantity of flue gas (such as flow rate, temperature, pressure, etc.) at each section of the cooker in the direction of flue gas flow does not change with time, so the process can be steady-state flow. To simplify the model, the following assumptions about the simulation have been made:

(1) The temperature of the bottom of the pot is uniform and kept constant;

(2) The heat dissipation of the stove wall is quite small; thus, the chamber wall is regarded as an insulated surface;

(3) The flow rate of flue gas in the stove is fast and the density changes little, and the fluid density is assumed to be a constant value.

2.3.3. Control Equations and Boundary Conditions

Fluid flow must satisfy the laws of conservation of mass, momentum, and energy. In computational fluid dynamics modeling, the continuity equation, momentum equation, and energy equation are solved separately to simulate the flow approximation of the fluid.

Equation for conservation of mass:

$$\frac{\partial(\rho u)}{\partial x} + \frac{\partial(\rho v)}{\partial y} + \frac{\partial(\rho w)}{\partial z} = 0 \tag{22}$$

Equation for conservation of momentum:

$$\frac{\partial(\rho u)}{\partial t} + div(\rho u u) = div(\mu grad u) - \frac{\partial p}{\partial x} + F_x \tag{23}$$

$$\frac{\partial(\rho v)}{\partial t} + div(\rho v u) = div(\mu grad v) - \frac{\partial p}{\partial y} + F_y \tag{24}$$

$$\frac{\partial(\rho w)}{\partial t} + div(\rho w u) = div(\mu grad w) - \frac{\partial p}{\partial z} + F_z \tag{25}$$

where p is the pressure on the fluid microsome, and F_x and F_y and F_z are the physical forces of the microsomes x, y, and z.

In this paper, the model includes gas and solid, and the heat transfer methods are convective and radiation heat transfer between each other, so the energy conservation equation of gas and solid is established, respectively. Among them, the gas is high-temperature flue gas, and the solid is the porous media, the chamber wall, and the bottom of the pot.

Equation for conservation of energy in the solid domain:

$$\frac{\partial}{\partial t}(\rho h) = \nabla \cdot (k \nabla T) + S_h \tag{26}$$

where ρ is density; h is enthalpy; k is thermal conductivity; T is the temperature; and S_h is a volumetric heat source.

Equation for conservation of energy in the gas domain:

$$\frac{\partial}{\partial t}(\rho E) + \nabla \cdot (\vec{v}(\rho E + p)) = \nabla \cdot ((\overline{\overline{\tau}}_{eff} \cdot \vec{v}) - \sum h_j \vec{J}_j) + S_h \tag{27}$$

where $\vec{J_j}$ is the diffusion flux of the component; $\sum h_j \vec{J}_j$ is the energy of the diffusion of the components; $\overline{\overline{\tau}}_{eff} \cdot \vec{v}$ is the energy dissipated by viscosity; and S_h is the heat source.

In this article, the energy equation, radiation model, and flow equation are opened during the simulation. In this paper, the radiation model in the stove is mainly the radiation of the chamber wall and the high-temperature flue gas to the bottom of the pot, so it is more appropriate to choose the discrete coordinate radiation (DO) model. The flow equation adopts the k-e turbulence model. In the numerical solve residual control, the solve parameter of the momentum equation and the continuity equation are set to 10^{-4}, and the solution parameter of the energy equation to 10^{-6}. Under the above conditions, the system of equations is solved separately and implicitly to obtain convergence.

The steady-state model is used for the simulation of the stove chamber, the SIMPLE algorithm is used for the coupling solution of velocity and pressure, and the momentum equation and energy equation are discretized in the second-order style. In terms of boundary conditions, the stove wall is an insulated wall, and the adiabatic boundary condition is adopted; the bottom of the pot adopts a constant wall temperature boundary condition; the radiant surface of the stove adopts the velocity inlet boundary condition, and the flue gas inlet temperature is constant; and the outlet boundary conditions use the pressure outlet provided by Fluent.

2.3.4. Meshing and Grid Independence Verification

Meshing is accomplished using a structured grid, as shown in Figure 6. By comparing the average Nu number of high-temperature flue gas and pot bottom, the grid sensitivity analysis is carried out, and the model is simulated with 20,000, 40,000, 60,000, 80,000, and 100,000 meshes, respectively, and the results are shown in Figure 7. When the total number of meshes is 60,000, 80,000, and 100,000, the average Nu number error is within 0.5%, which satisfies the mesh sensitivity verification, so the model with a total mesh of 80,000 is ultimately selected for calculation and analysis.

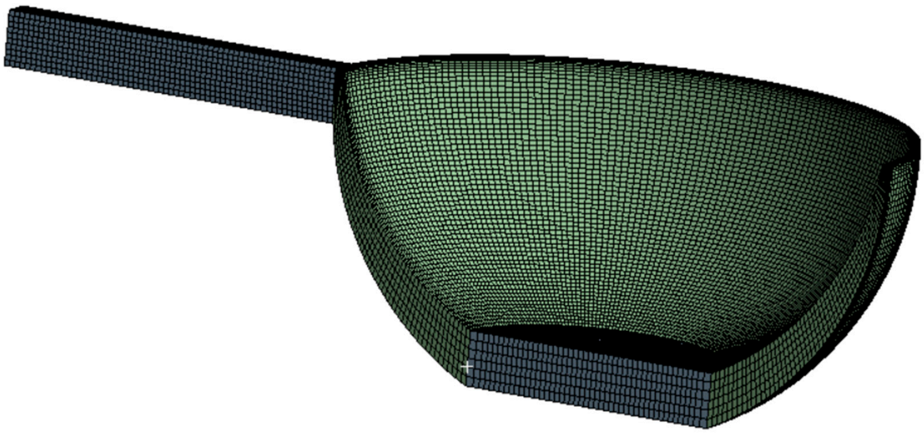


Figure 6. Mesh division.

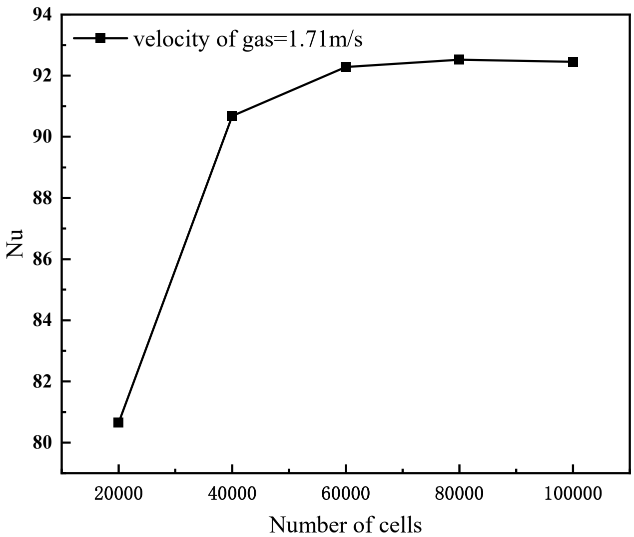


Figure 7. Mesh independence verification.

3. Theoretical Analysis

3.1. Study on Performance of Fully Premixed Porous Stove at Rated Power

For the 52 kw natural gas porous media stove, the structure of porous media and stove chamber is designed, as shown in Table 1. The equivalence ratio is 0.95 at rated power. According to the calculation of the combustion and heat transfer performance of the porous media stove, the thermal efficiency and heat loss of the stove are obtained. The results are shown in Tables 1 and 2.

Table 1. Basic parameters of the stove.

Parameters of PM		Parameters of Stove Chamber	
material	Cordierite	rated power	52 kW
emissivity	0.7	chamber depth	0.48 m
porosity	60%	height between PM and pot bottom	0.05 m
pore density	8 PPC		
average pore diameter	0.0011 m	chamber volume	0.0458 m^3
diameter of PM	0.54 m	pot bottom temperature	210 °C
thickness of PM	0.003 m	ambient temperature	20 °C

Table 2. Thermal performance of stove underrated power.

Item			Thermal Power, kW		Proportion, %
Effective heat	Radiant heat	Radiant heat of PM skeleton	20.362	24.513	47.14
		Radiant heat of the flame surface	4.151		
	Convective heat	Convective heat in the central region	1.542	11.133	21.41
		Convective heat in the edge zone	9.591		
		Total effective heat	35.646		68.55
Heat loss item		Heat loss of exhaust gas	16.229		31.21
		Heat loss of chamber wall	0.171		0.24
		Total heat loss	16.4		31.45

3.2. Analysis on Influencing Factors and Characteristics of the Fully Premixed Porous Stove

The combustion and heat transfer performance of the stove during operation are affected by many factors, including the structure of the porous media and the stove chamber. Table 3 shows the factors affecting the performance of the stove and their value ranges. On the basis of the original value, the influence law of each influencing factor on the performance of the stove is analyzed. Among them, the area of the stove corresponding to the bottom of the pot is defined as the central area, and other area is defined as the edge area.

Table 3. Influencing factors and value ranges.

Item	Influencing Factor	Original Value	Value Range
Influencing factors of PM	porosity	60%	40–90%
	pore density	8 PPC	4–14 PPC
	emissivity	0.7	0.5–0.9
	thickness	0.003 m	0.001–0.005 m
Influencing factors of chamber	diameter	0.54 m	0.3–0.7 m

3.3. Performance Analysis

The main parameters for evaluating the performance of the stove include the total thermal efficiency, flame temperature, exhaust gas temperature, and premixed gas preheating temperature (the higher the preheating temperature, the better the ignition performance of the stove).

3.3.1. Influence of the Porous Media Emissivity on Stove Performance

While the emissivity of the porous media is changed from 0.5 to 0.9, the influence of the porous media emissivity on the thermal efficiency and combustion heat transfer performance of stove are analyzed. The results are shown in Figure 8.

It can be seen from Figure 8a that as the porous media emissivity increases from 0.5 to 0.9, the total thermal efficiency of the stove increases from 67.92% to 69.06%. Among them, the radiant heat efficiency raises from 44.32% to 48.9%, while the convective heat efficiency decreases from 23.6% to 20.17%. Figure 8b shows that with the increase of the emissivity, the flame surface temperature of the stove decreases from 1097 °C to 1009 °C. The exhaust gas temperature decreases from 758 °C to 732 °C. This is because the heat released by the gas is partly radiated to the bottom of the pot by the porous media, and the rest is used to heat the premixed gas to the combustion temperature. Therefore, the flame temperature on the upper surface of the stove depends on the heat radiated from the porous media to the bottom of the pot. The higher the radiant heat, the lower the heat absorption of the premixed gas from the normal temperature to the combustion temperature, and the lower the flame temperature.

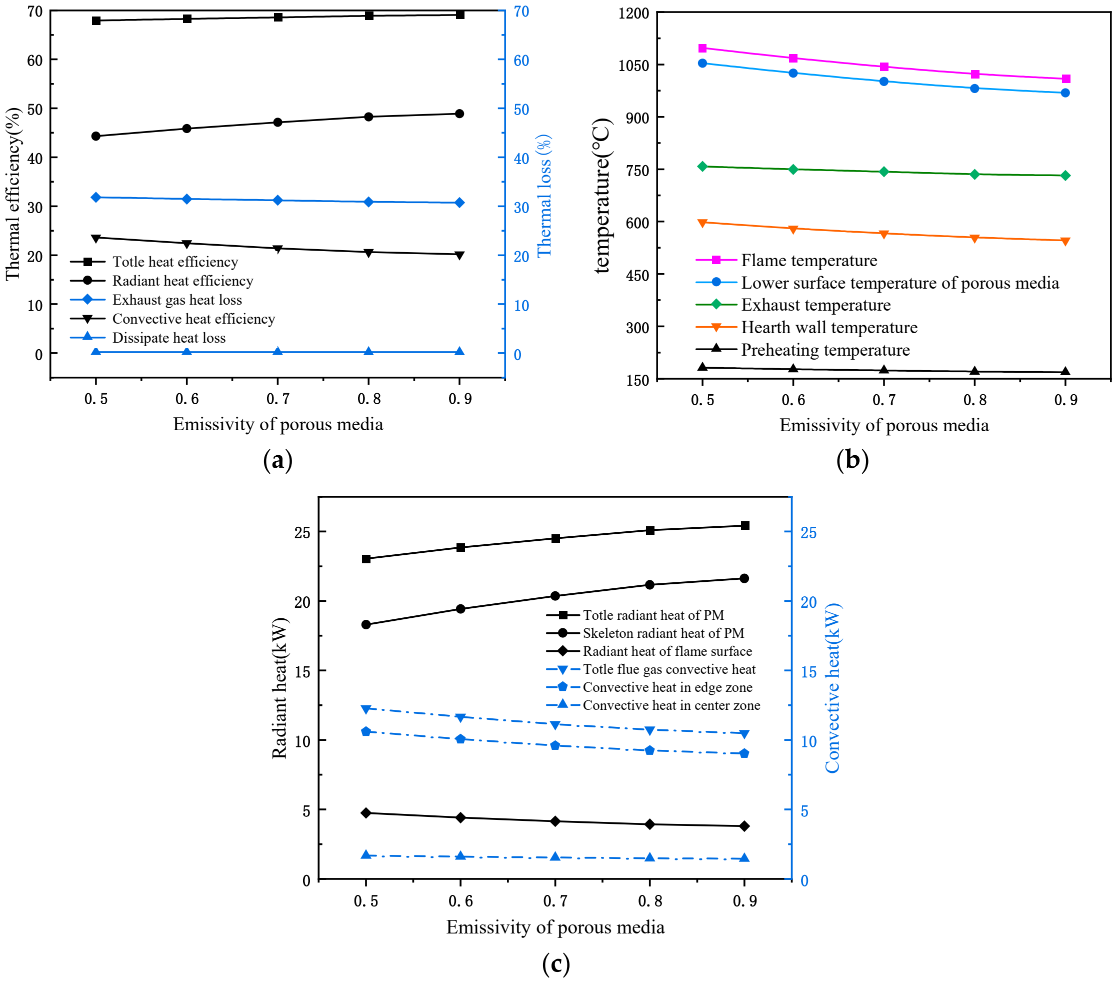


Figure 8. Effect of porous media emissivity on the stove performance. (**a**) Effect of porous media emissivity on the thermal efficiency of the stove. (**b**) Effect of porous media emissivity on the temperature of the stove. (**c**) Effect of porous media emissivity on the heat transfer ratio of the stove.

Figure 8b also shows that the temperature of the upper and lower surfaces of the porous media is significantly reduced. In addition, the heat conduction temperature difference between the upper and lower surfaces of the porous media is reduced by 43 °C to 40 °C, the heat conduction is reduced, and the preheating temperature rise of the premixed gas is slightly reduced, from 181 °C to 168 °C, which is reduced by 15 °C.

According to the variation curve of radiant heat in Figure 8c, when the porous media emissivity is changed from 0.5 to 0.9, the radiant heat of the porous media skeleton increases, while the radiant heat of the stove flame surface decreases. The reason is that the radiant heat of the stove is affected by the emissivity and the temperature of the porous media at the same time. From the results, the effect of the temperature on the radiant heat is less than that of the emissivity on the radiant heat, which causes the radiant heat to increase. The radiant heat of the flame surface is affected by the flame temperature and the emissivity of the three atomic gas components in the flame. The positive effect of the flame temperature on the radiant heat of the flame is more obvious. Therefore, the flame temperature decreases and the radiant heat of the flame surface decreases.

According to the variation curve of convective heat in Figure 8c, the convective heat of the stove all decreases gradually. The reason is that the flame temperature decreases, so the average temperature of the flue gas in the stove decreases. Then, the heat transfer temperature difference between the flue gas and the bottom of the pot decreases, resulting in lower convective heat transfer.

In short, increasing the porous media emissivity will reduce the flame surface temperature and the porous media surface temperature, which is conducive to reducing the production of NOx. At the same time, this will strengthen the thermal radiation performance of the stove and improve its thermal efficiency slightly, but it will weaken the convective heat transfer in the stove chamber so as to reduce the ignition performance of the stove.

3.3.2. Influence of the Porosity on Stove Performance

The porosity of the stove increases from 40% to 90% by changing the average pore diameter (from 0.89 mm to 1.34 mm). Figure 9 shows the influence of the change of the porosity on the thermal efficiency and combustion heat transfer performance of the stove.

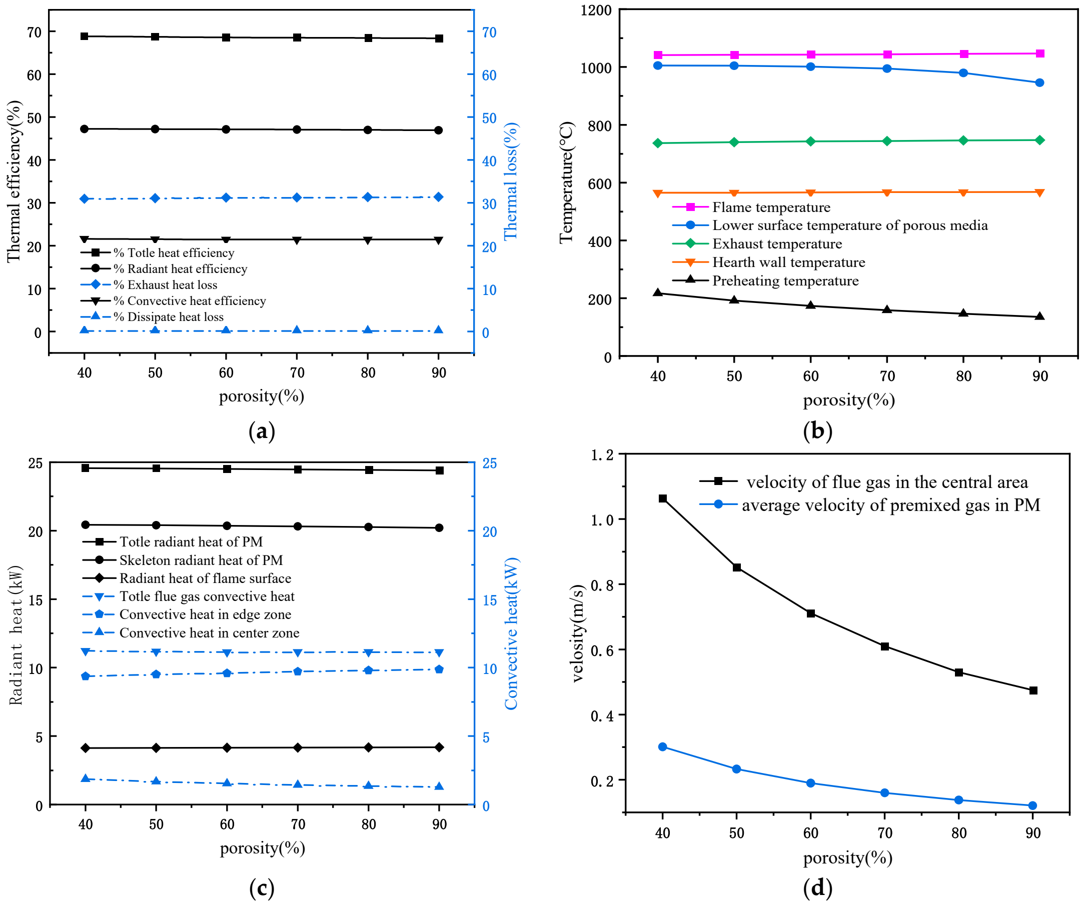


Figure 9. Effect of porosity on stove performance. (**a**) Effect of porous media porosity on the thermal efficiency of the stove. (**b**) Effect of porous media porosity on the temperature of the stove. (**c**) Effect of porous media porosity on the heat transfer ratio of the stove. (**d**) Effect of porous media porosity on the gas velosity of the stove.

Figure 9a shows that as the porosity increases from 40% to 90%, the total thermal efficiency of the stove is almost unchanged, approximately 68.5%. The radiant heat efficiency and the convective heat efficiency remain unchanged as well. Figure 9b shows that the flame temperature increases from 1041 °C to 1047 °C, and the exhaust gas temperature increases from 737 °C to 747 °C.

Figure 9b also shows that the lower surfaces of the porous media decreases from 1005 °C to 946 °C, so the temperature difference between the upper and lower surfaces of the porous media increases from 36 °C to 101 °C; the preheating temperature of premixed gas is reduced from 217 °C to 135 °C. This is because, with the increase of porosity, the surface area of the porous media decreases gradually, and the pore surface area increases gradually. On the one hand, as seen in Figure 9d, the average velocity of premixed gas in the stove decreases from 0.3 m/s to 0.12 m/s, resulting in the decrease of the convective heat transfer coefficient between the premixed gas and the porous media skeleton. On the other hand, the heat conduction area of the porous media skeleton is greatly reduced, and the temperature of the lower surface of porous media gradually decreases, so the heat transfer temperature difference between the premixed gas and the skeleton is reduced. The convective heat transfer decreases gradually, and the preheating temperature rise of premixed gas also decreases gradually. At the same time, because the flow rate of premixed gas in the stove is greatly reduced, the flame stability of the stove is greatly reduced. According to the literature [21], under the condition that methane is mixed with air and the equivalence ratio is within the range of 0.45–1.7, when the flow rate of premixed gas is within the range of 0.25–0.7 m/s, the flame forms a stable propagating combustion wave in the porous media. If the flow rate of premixed gas in the stove exceeds this flow rate range, the flame stability of the stove will be greatly reduced, and tempering or extinguishing of the fire will occur.

Figure 9c shows that with the increase of porosity, the flue gas convective heat shows a decreasing trend, in which the convective heat in the central area of the boiler bottom gradually decreases and the convective heat in the edge area gradually increases. The reason is that the convective heat of flue gas in the central area is affected by the flame temperature and flue gas flow rate at the same time. As the porosity increases from 40% to 90%, the surface area of the porous media skeleton decreases by nearly twofold, and the pore surface area increases by nearly twofold. Thus, the flue gas velocity in the central area decreases from 1.06 m/s to 0.47 m/s, and the convective heat transfer coefficient between flue gas and pot bottom decreases greatly (Figure 9d), while the flame temperature increases slightly, so the convective heat of flue gas in the central area decreases. This leads to a decrease in the temperature drop of the flue gas from the central area to the edge area, and the heat transfer temperature difference between the flue gas in the edge area and the bottom of the boiler increases gradually, so the convective heat in the edge area increases.

In short, increasing the porosity will significantly reduce the ignition performance of the stove, rapidly reduce the flame stability of the premixed gas, and slightly reduce the thermal efficiency and radiant and convective heat power of the stove.

3.3.3. Influence of the Pore Density on Stove Performance

It is important to ensure that the porosity remains unchanged. When the pore density of the porous media is increased from 4 PPI to 14 PPI, the influence of the pore density of the stove on the thermal efficiency and combustion heat transfer performance of the stove is analyzed. The results are shown in Figure 10:

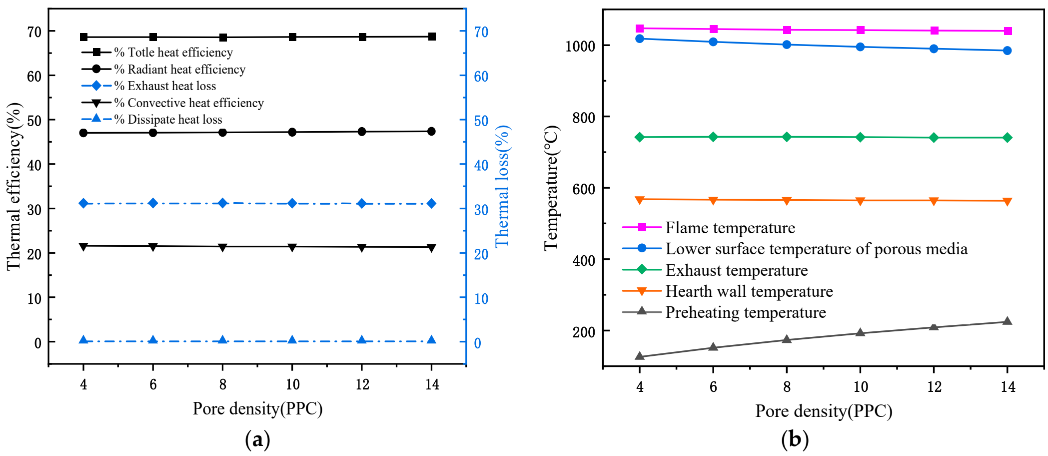


Figure 10. Influence trend of pore density on stove performance. (**a**) Effect of pore density on the thermal efficiency of the stove. (**b**) Effect of pore density on the temperature of the stove.

Figure 10a,b show that when the pore density increases, the efficiency, heat loss, as well as the flame temperature remain unchanged. The heat conduction temperature difference between the upper and lower surfaces of the porous media increases from 29 °C to 55 °C; the preheating temperature of premixed gas increased from 126 °C to 224 °C, increasing by 98 °C. The reason is that the surface area of the porous media skeleton remains unchanged, and the temperature difference between the upper and lower surfaces of the porous media skeleton and the downward thermal conductivity of the stove increases, so the preheating temperature increases gradually.

Therefore, when the porosity remains unchanged, reducing the pore diameter and increasing the pore density will increase the preheating temperature of the premixed gas and improve the ignition performance of the stove, but will not change the flow rate of the premixed gas in the stove, which has little impact on the flame stability.

3.3.4. Influence of Porous Media Thickness on Stove Performance

When the thickness of the porous media is increased from 0.001 m to 0.005 m, the influence of the thickness of the porous media on the thermal efficiency and combustion heat transfer performance of the stove is analyzed, as shown in Figure 11.

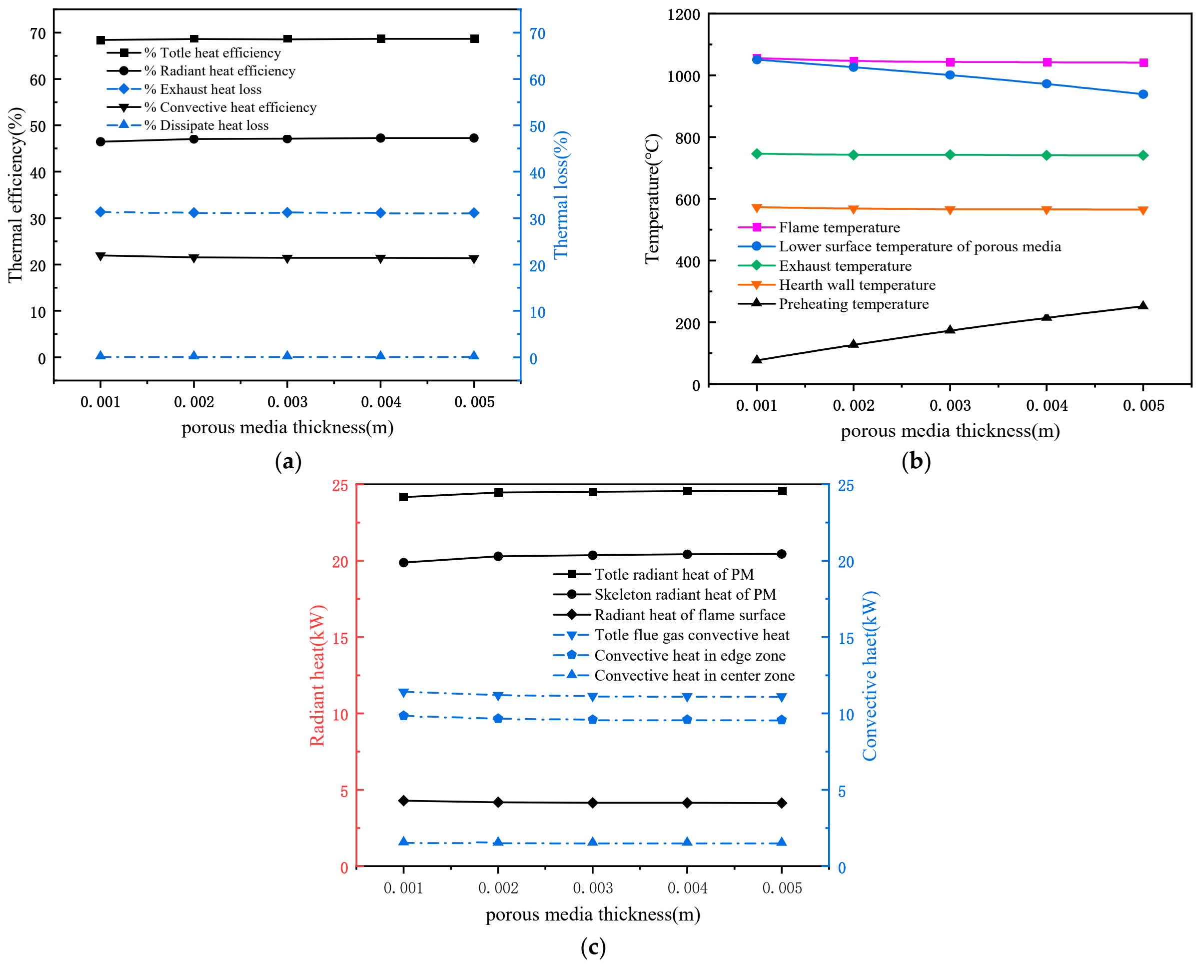


Figure 11. Influence of porous media thickness on stove performance. (**a**) Effect of porous media thickness on the thermal efficiency of the stove. (**b**) Effect of porous media thickness on the temperature of the stove. (**c**) Effect of porous media thickness on the heat transfer ratio of the stove.

Figure 11a shows that when the thickness of the stove ranges from 0.001 m to 0.005 m, the total thermal efficiency of the stove is almost unchanged, approximately 68.5%, where the radiant heat efficiency increased by 0.5%, and the convective heat efficiency reduced by

0.67%. Figure 11b shows that the flame temperature decreased from 1056 °C to 1046 °C. The exhaust gas temperature and the temperature of the chamber wall are basically unchanged. The temperature difference between the upper and lower surfaces of the porous media increased from 5 °C to 102 °C, an increase of nearly 20-fold. The preheating temperature of premixed gas increases from 76 °C to 252 °C. The reason is that with the increase of the thickness of the porous media, the surface area in the hole gradually increases, the convective heat exchange area between the premixed gas and the porous media skeleton gradually increases, and the heat exchange also increases gradually. Therefore, with the increase of premixed gas thickness, the preheating temperature of the premixed gas increases gradually, and the ignition performance of the stove is greatly improved. The thickness of the stove should not be too thin (less than 2 mm), which has poor preheating effect on the premixed gas; it should also not be too thick (more than 4 mm), which will lead to a large temperature difference between the upper and lower surfaces, causing the stove to be broken and damaged. This result is consistent with the conclusions in the literature [22].

It can be seen from Figure 11c that when the thickness of the stove ranges from 0.001 m to 0.005 m, at 0.002 m, the total radiant heat increased by 0.308 kW. Among them, the radiant heat of the skeleton increases by 0.415 kw, and the radiant heat of the flame surface is reduced by 0.107 kW. The reason is that the stove is thin, and when the thickness is doubled, the converted radiation area of the porous media increases, which leads to the increase of the total radiant heat and reduction of the flame temperature. When the thickness of the stove changes from 0.002 m to 0.005 m, the influence of thickness on radiant heat is not obvious. In addition, the total convective heat decreases slightly because, when the flame temperature decreases slightly, the heat transfer temperature difference between the flue gas and the bottom of the boiler decreases, reducing the heat transfer. The convective heat is basically unchanged.

In short, changing the thickness of the porous media affects mainly the ignition performance of the stove. Increasing the thickness of the porous media can enhance the ignition performance of the stove. A suitable thickness can not only preheat the premixed gas but also result in a long service life for the stove.

3.3.5. Influence of the Porous Media Diameter on Stove Performance

When the porous media diameter is changed from 0.3 m to 0.7 m, the influence of porous media diameter on the thermal efficiency and combustion heat transfer performance of the stove is analyzed. The results are shown in Figure 12.

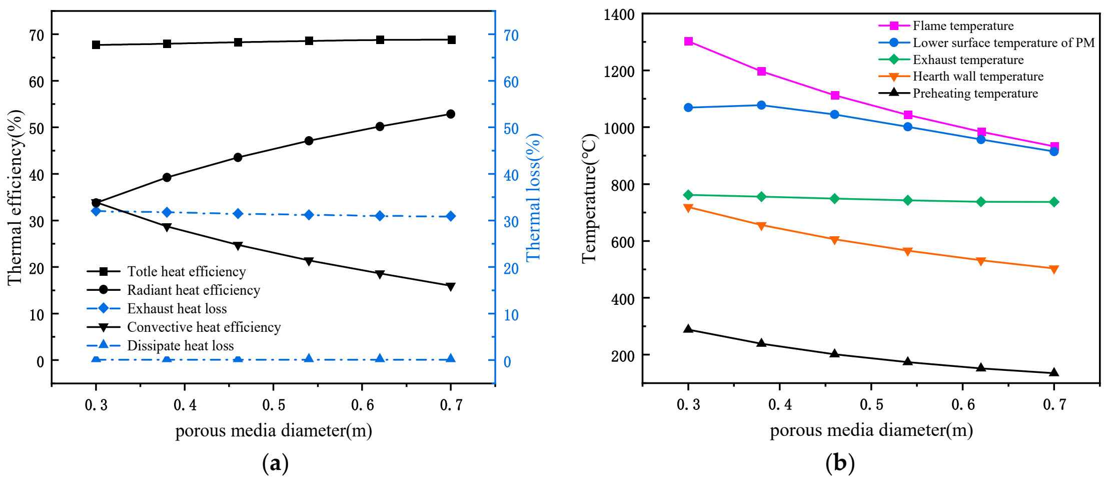


Figure 12. *Cont.*

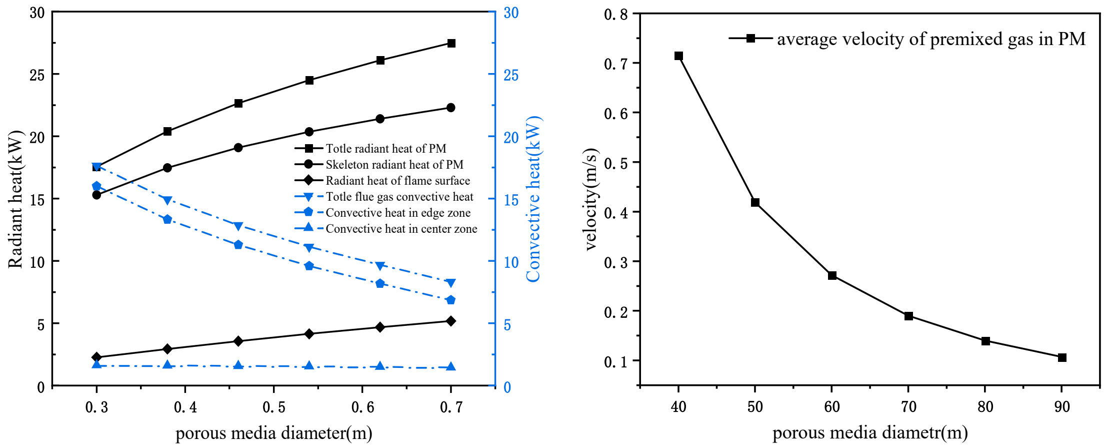


Figure 12. Influence of porous media diameter on stove performance. (**a**) Effect of porous media diameter on the thermal efficiency of the stove. (**b**) Effect of porous media diameter on the temperature of the stove. (**c**) Effect of porous media diameter on the heat transfer ratio of the stove. (**d**) Effect of porous media diameter on the gas velosity of the stove.

As can be seen from Figure 12a, as the diameter of the porous media increases from 0.3 m to 0.7 m, the total thermal efficiency of the stove increases from 67.69% to 68.84%. The radiant heat efficiency increases from 33.77% to 52.85%, and the convective heat efficiency decreases from 33.92% to 15.99%.

Figure 12b shows that the flame surface temperature decreases from 1302 °C to 933 °C with the increase of the diameter. The temperature of the chamber wall is reduced by 216 °C, and the exhaust gas temperature is reduced by 25 °C. Thus, the exhaust gas heat loss decreases. The results show that increasing the diameter can slightly reduce the heat loss, but the control of flame temperature and stove temperature is significant. Selecting the appropriate porous media diameter can control the flame temperature within the appropriate range (below 1200 °C) and greatly reduce the generation of NOx [23].

Figure 12b also shows that with the increase of the diameter, the heat conduction temperature difference between the upper and lower surfaces of the porous media decreases rapidly, from 233 °C to 19 °C. In addition, the preheating temperature of premixed gas decreases from 288 °C to 135 °C. The increase of the flow area of the pores in the porous media leads to the average velocity of the premixed gas in the stove, increasing from 0.715 m/s to 0.107 m/s, so the convective heat transfer coefficient between the premixed gas and the porous media skeleton is significantly reduced, the convective heat transfer is reduced, and the preheating temperature is greatly reduced. Therefore, increasing the diameter of the stove will deteriorate the ignition performance of the stove and reduce the flame stability of the stove.

According to the change curve of Figure 12c, the total radiant heat of the stove, the radiant heat of the porous media skeleton, and the radiant heat of the flame surface all show an obvious increasing trend. The reason is that the hole area of the porous media increases, the radiant area of the porous media skeleton and the radiant area of the flame surface increase as well, but the total radiant heat is also affected by the surface temperature of the porous media. The flame surface temperature decreases obviously, but the effect of the flame temperature on the radiant heat is less than that of the radiant area. Therefore, the total radiant heat of the stove increases with the increase of the diameter of porous media.

In short, increasing the diameter of the porous media will significantly reduce the flame temperature but greatly enhance the radiant heat transfer performance of the stove, reduce the convective heat transfer performance of the stove, and increase the total thermal efficiency of the stove slightly. It will significantly deteriorate the ignition performance of the stove as well as the flame stability.

3.4. Simulation Results

The flow field and temperature field in the cooker chamber under 100% combustion load were simulated and compared with the theoretical results, and the results are shown in Figure 13:

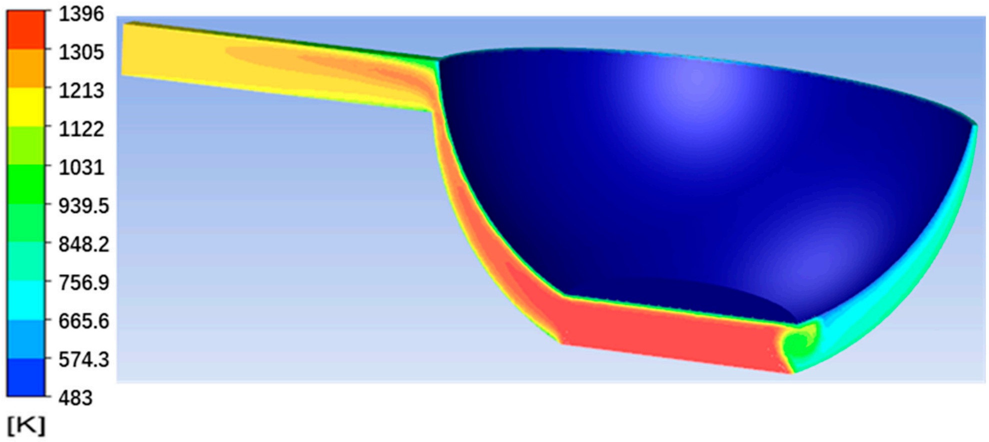
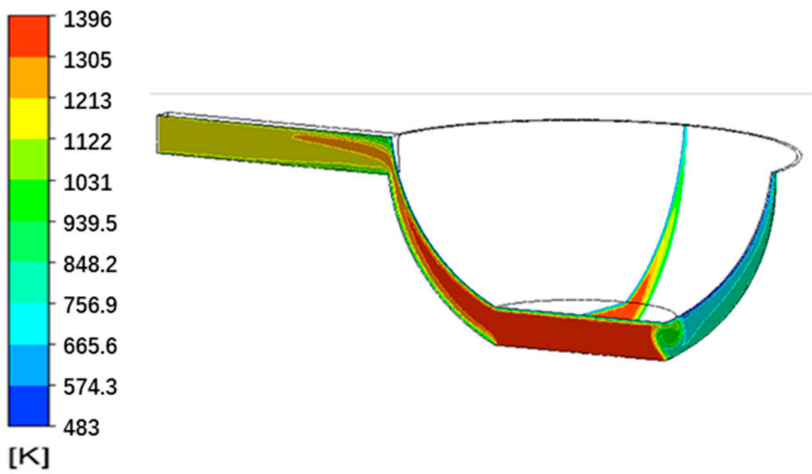


Figure 13. Temperature distribution in stove chamber.

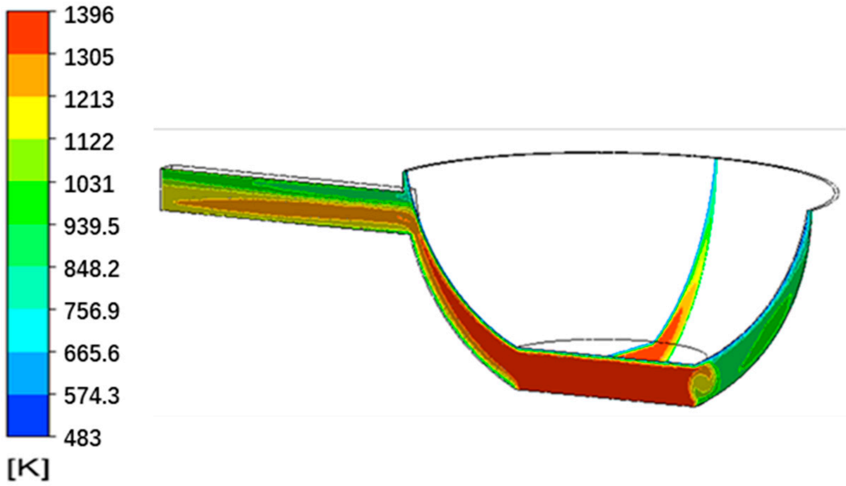
It can be seen from Figure 13 that when the smoke outlet is arranged at a height of 0 cm from the surface of the stove outlet of the stove, the distribution of the simulated temperature in the stove is not uniform, and the flue gas temperature on the side near the flue is much higher than the flue gas temperature on the side away from the flue, indicating that the flow of flue gas in the stove is uneven, the filling degree of the flue gas in the stove is not high, the flushing of the flue gas to the bottom of the pot is uneven, and a large amount of flue gas flows directly from the bottom of the pot to one side of the smoke outlet arrangement, while the other side of the flue gas has a retention zone. Therefore, the flue gas is not sufficiently heated with the entire bottom of the pot. In addition, the average exhaust temperature obtained by the simulation is 1172 K (899 °C), which is 182 °C higher than the theoretical value.

Therefore, four height positions of 0 cm, 5 cm, 10 cm, and 15 cm from the surface of the cooktop are selected to arrange the flue and simulated to observe the flue gas flow, temperature distribution, and average exhaust temperature in the chamber. In this way, the most suitable flue placement position is selected, and the results are shown in Figure 14.

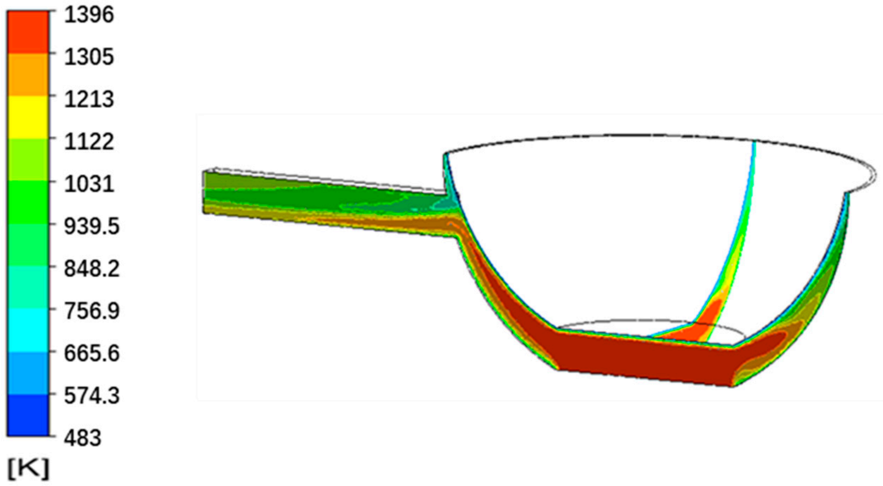


(**a**) 0 cm

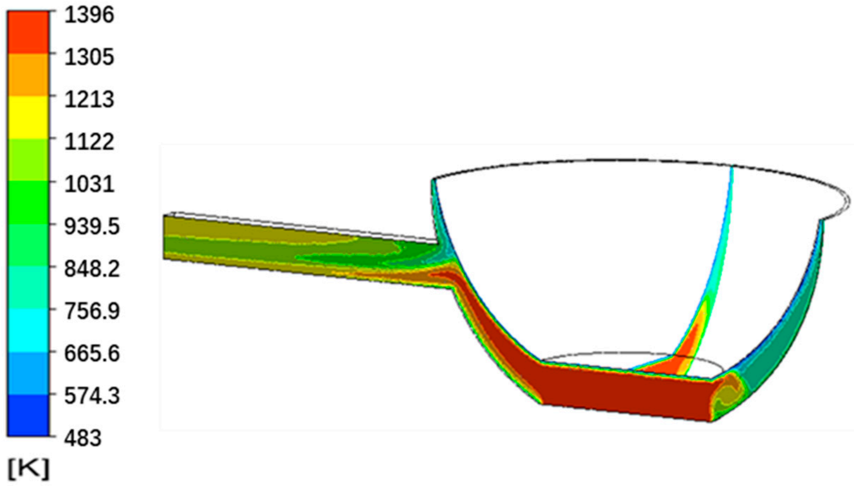
Figure 14. *Cont.*



(**b**) 5 cm



(**c**) 10 cm



(**d**) 15 cm

Figure 14. The temperature field in the hearth of the stove with different positions of the smoke outlet.

As can be seen from Figure 14, the temperature field in the stove changes significantly as the position of the smoke outlet gradually decreases. Additionally, the temperature field in the stove is basically uniform when the distance between the smoke outlet and the upper surface of the stove increases from 0 to 10 cm, which indicates that the heat exchange between the flue gas and the bottom of the pot becomes better. This is because the flue gas will be concentrated at the position of the smoke outlet. The higher the position of the smoke flue, the smaller the gap between the stove wall and the bottom of the pot. The flow rate of the flue gas in this area will be faster. However, the resistance loss of the fluid will be greater, and it may become more difficult for the flue gas to flow upward. Thus, the heat exchange area between the flue gas and the bottom of the pot becomes smaller. On the other hand, the gap between the stove wall and the bottom of the pot gradually increases with the reduction of the position of the flue. The main flow channel of the flue gas is widened. Thus, the flow rate is reduced as well as the flow resistance loss. The heat exchange area between the flue gas and the bottom of the pot is larger, and the heat exchange effect is better. Nevertheless, there is a limitation on lowering the position of the smoke outlet. Specifically, the temperature field in the stove is unevenly changed when the smoke outlet continues to decrease from 10 cm to 15 cm. This is because the flue gas channel is further widened, and the flue gas flow is concentrated mainly in the lower part of the stove. A large amount of flue gas is discharged from the stove directly instead of circulating in the stove. The area of the flue gas sweeping through the bottom of the pot is greatly reduced, and the heat exchange with the bottom of the pot is also greatly reduced. However, the temperature field distribution in the stove under this arrangement is still better than that at the initial position.

Additionally, the average flue gas outlet temperature of each position is obtained by setting the outlet smoke temperature monitoring point, and the results of the last three positions are 1152 K (879 °C), 1103 K (830 °C), and 1155 K (882 °C), respectively. In general, the flow and heat transfer in the stove are relatively good when the flue is placed at the optimum value, namely at 10 cm.

Figure 15 shows the velocity field and flue gas flow line of the flue gas in the stove under the four smoke outlet positions. As can be seen from Figure 15, the flue gas flow velocity of the smoke outlet gradually tends to be uniform, and the average outlet flow rate gradually decreases when the position of the smoke outlet gradually decreases. This indicates that the circulation area of flue gas in the stove gradually increases, and the flow resistance loss gradually decreases. However, the flue gas has a tendency to be directly discharged from the stove chamber when the flow resistance loss is reduced to a certain extent. Thus, the heat transfer efficiency will first increase and then decrease with the decrease of the position of the flue. In addition, the flue gas has a rotary vortex in the stove far from the side of the smoke exhaust. This is because there is less high-temperature flue gas circulation in the upper part of the stove at the beginning of heat exchange, while the low-temperature air will flow downward and mix with high-temperature flue gas, forming a vortex. The vortex impedes the upward flow of high-temperature flue gas, forcing it to flow to the exhaust outlet. As the position of the flue is reduced from 0 cm to 10 cm, the vortex gradually weakens, and increasingly more high-temperature flue gas flows to the upper part of the stove. When the position of the flue is further reduced from 10 cm to 15 cm, the vortex is enhanced. These results are consistent with the analysis in the previous paragraph.

According to the above results, the flue position is placed 10 cm away from the upper surface of the stove, and the simulation under variable load conditions is carried out to observe the error between the simulated exhaust temperature and the theoretical exhaust temperature; the results are shown in Figure 16:

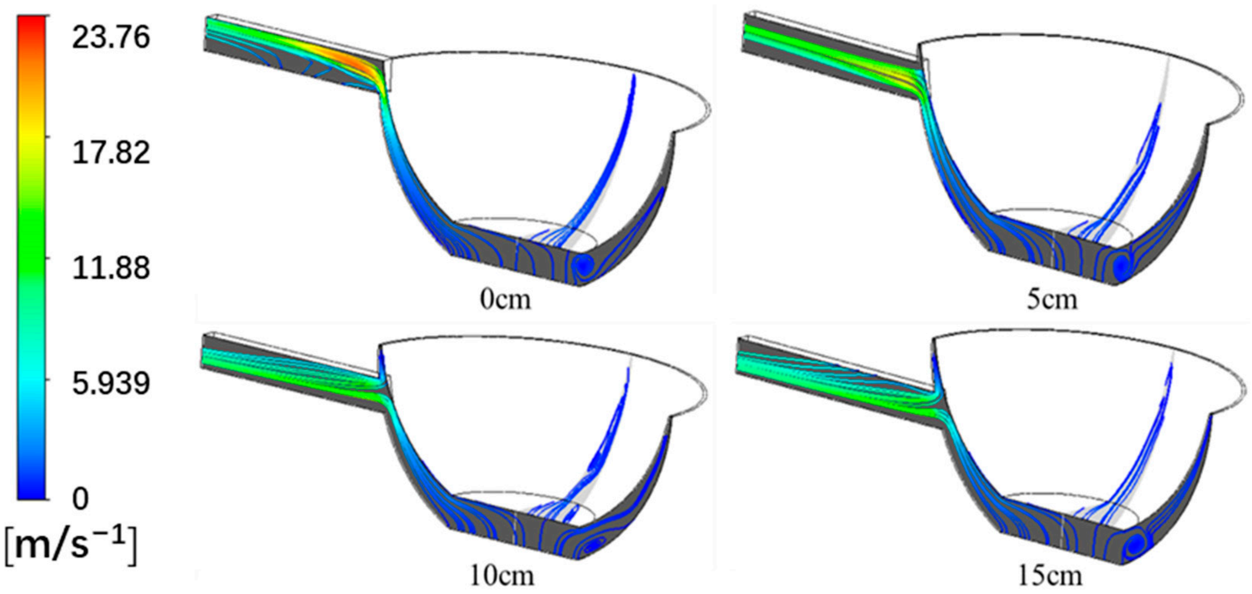


Figure 15. Velocity distribution in stove chamber at different vent locations.

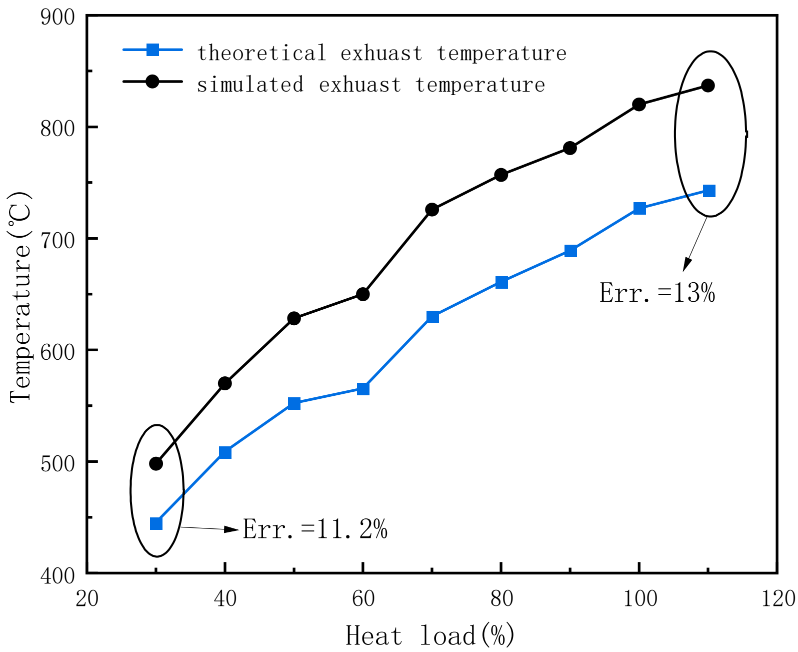


Figure 16. Comparison of theoretical smoke exhaust temperature and simulated smoke exhaust temperature under variable load conditions.

The simulation results are shown in Figure 16; the simulated exhaust temperature is higher than the theoretical exhaust temperature, but the overall change trend is similar, and the error between the simulated smoke exhaust temperature and the theoretical exhaust temperature is less than 15%, of which the minimum error is 11.2% and the maximum error is 13%, which is within the scope of engineering error. On the other hand, as the heat load increases gradually, the difference between the simulated exhaust gas temperature and the theoretical exhaust gas temperature increases gradually, which indicates that the theoretical value is closer to the simulated value under low load.

4. Conclusions

In this paper, a theoretical model is established for the 52 kW fully premixed porous media cauldron stove. The performance study under rated operating conditions and the influence of porous media and cooker structure parameters of the stove on the stove performance are conducted, and the results showed that:

(1) Under the rated working conditions, the flame temperature of the stove is 1043 °C, and the thermal efficiency of the cooker reaches 68.55%, which is consistent with the experimental results in the literature [24]. Among them, the radiative heat efficiency of the cooker reaches 47.16%. Meanwhile, the thermal efficiency of the cooker is higher than 67% under variable factors. The flame temperature mainly ranges from 1000 to 1300 °C under different working conditions, which is consistent with the experimental results in the literature [25].

(2) Under the rated combustion power, the biggest factor affecting the thermal efficiency of the stove is the diameter of porous media. Increasing the diameter will improve the thermal efficiency of the stove.

(3) The main factors affecting the heat transfer performance are the emissivity and the diameter of the porous media. Increasing the emissivity and diameter of the porous media will strengthen the radiant heat transfer performance of the stove and reduce the convective heat transfer performance.

(4) The main factors affecting the ignition performance of the stove are the thickness, porosity, pore density, and diameter of the porous media. Increasing the thickness and pore density can improve the ignition performance of the stove, while increasing the porosity, and the diameter will make the ignition performance of the stove worse.

(5) The numerical simulation results show that there is an optimal height at the exhaust outlet, that is, 10 cm below the outlet surface of the stove. In this place, the flow uniformity of flue gas in the stove is the best, the scouring on the bottom of the pot is more uniform, and the smoke exhaust temperature is lower.

(6) The numerical simulation results show that the simulated exhaust temperature is higher than the theoretical exhaust temperature, but the overall change trend is similar, and the error between the simulated smoke exhaust temperature and the theoretical exhaust temperature is less than 15%, which indicates that the theoretical value is closer to the simulated value under low load.

The study shows that the stove has good economy and energy-saving properties. After the pilot test is completed in the later stage, the initial product promotion of cauldron stoves can be carried out to restaurants, hotels, and university canteens in the catering industry. At the same time, this work is not only applicable to cauldron stoves but also widely applicable to various forms of commercial stoves.

Author Contributions: Conceptualization, L.S.; data curation, D.Z.; formal analysis, D.Z.; funding acquisition, D.Z.; investigation, D.Z.; software, H.O. and S.R.; writing—review and editing, L.S. All authors have read and agreed to the published version of the manuscript.

Funding: This research received no external funding.

Institutional Review Board Statement: Not applicable.

Informed Consent Statement: Not applicable.

Data Availability Statement: Not applicable.

Acknowledgments: I would like to give my heartfelt thanks to all the people who have ever helped me in this paper. We gratefully acknowledge the assistance of Shi and Shen for their valuable comments in revising the content of this paper.

Conflicts of Interest: The authors declare no conflict of interest.

Nomenclature

a	smoke blackness	$W/(m^2 \cdot K^4)$
ag	blackness of pot bottom	$W/(m^2 \cdot K^4)$
c_{pm}	average constant pressure specific heat of premixed gas	$kJ/(kg \cdot K)$
d_k	diameter of the pores	m
h	convection heat transfer coefficient	$W/(m^2 \cdot K)$
h_z	distance from the porous media to the bottom of the pot	m
k	flue gas attenuation coefficient	$1/(m \cdot MPa)$
l	thickness of the stove	m
n	number of pores	
q_v	volume flow of premixed gas	Nm^3/s
r	volume share of triatomic gas	
r_h	volume share of water vapor	
A_1	surface area of porous media	m^2
A_3	surface area of pot bottom	m^2
F	surface area of the chamber wall	m^2
P	pressure in the stove	Pa
Q_c	low calorific value of natural gas	kJ/kg
Q_d	flue gas convection heat	kW
Q_{d1}	convective heat transfer between the flue gas and the central area of the pot bottom	kW
Q_{d2}	convective heat transfer between the flue gas and marginal zone of the pot bottom	kW
Q_f	radiant heat of the stove	kW
Q_{f2}	radiant heat transfer of the flue gas to the pot bottom	kW
Q_s	heat dissipation of the chamber wall	kW
Q_w	heat absorption of high-temperature flue gas at flame surface	kW
Q_y	heat emitted by flue gas	kW
Q_Z	heat release of gas combustion	kW
Rz	total thermal resistance of space radiation	$m^2 \cdot K/W$
S_{d1}	area of the impinging jet area at the center of the pot bottom	m^2
S_{d2}	heat exchange area between flue gas and pot bottom	m^2
S_n	total inner surface area of the pores	m^2
T_g	pot bottom temperature	°C
T_h	flame temperature	°C
T_x	lower surface temperature of the porous media	°C
T_y	average temperature of flue gas in the stove	°C
T_{yr}	preheating temperature of premixed gas	°C
V	effective volume of the area in the stove except the flame jet	m^3
$X_{1,2}$	angle coefficients between porous media and chamber wall	
$X_{1,3}$	angle coefficients between porous media and pot bottom	
$X_{3,2}$	angle coefficients between pot bottom and chamber wall	
σ	blackbody radiation constant	$W/(m^2 \cdot K^{4)}$
ε_1	equivalent emissivity of the porous media	
ε_3	emissivity of the lower surface of the pot bottom	
η_q	flue gas dynamic viscosity calculated by the flue gas temperature	$Pa \cdot s$
η_p	flue gas dynamic viscosity calculated by the boiler bottom temperature	$Pa \cdot s$
η	total thermal efficiency of stove	
δ	thickness of porous media	m
λ	thermal conductivity of flue gas	$W/m^2 \cdot K^{-1}$

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