

# Thermal Transfer Performance of Downhole Electric Heaters for In-Situ Pyrolysis in Tar-Rich Coal

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**Abstract.** The performance of a large-size helical baffle heater in an in-situ operating is investigated using a numerical simulation method. It revealed that the fluid in the shell retains a spiral flow, and the output flow velocity is higher than in the surrounding area. However, the pitch design is rather big, resulting in a low-velocity flow zone on the backwind side. At 100 kW and 500 m<sup>3</sup>/h, the fluid flow is turbulent. 50 kW and 200 m<sup>3</sup>/h, the fluid remained laminar. As flow rate rose, the pressure of tar-rich coal formation grew dramatically. The wall temperature exhibited spiral plunger at the inlet, but the bottom temperature was symmetrically distributed. Under low power and flow, Reynolds number change has a greater impact on the combination of Nusselt and Prandtl number. The wellbore experiences higher thermal loads during downhole heating, increasing the possibility of thermal damage dramatically. An increase in the heater shell length improves the total heat transfer performance. Conventional heaters often only heat the bottom formation. Therefore, while optimizing the construction, it is vital to ensure that the weight of heater itself does not exceed the tensile strength of cable, and consider shifting down the perforation outlet or lowering the outlet.

**Key words:** tar-rich coal; in-situ heat injection; downhole heater; thermal performance; well-wall stability

## 1. INTRODUCTION

Non-conventional energy sources are gaining prominence, causing a major shift in the global energy environment[1]. Tar-rich coal, a dual-attribute energy resource with coal and hydrocarbon properties, can be transformed into coal tar, pyrolysis gas, and semi-coke using low and medium-temperature pyrolysis, and its development potential is enormous[2, 3]. However, the inefficiency of heat injection in the in-situ mining process has become an important obstacle to the effective production of tar-rich coal[4-6]. As a result, the present study focuses on improving the heat injection mechanism and increasing the efficiency of heat energy consumption[7, 8].

Downhole electric heaters, as a highly efficient energy-saving solution, considerably minimize heat loss during transmission by reducing the distance between the heat source and the reservoir[9]. The Shell E-ICP trial successfully used a Y-type electric heater to heat the formation via heat transfer, proving for the first time the practical use of downhole heaters in unconventional energy study[10]. However, because to the restricted heat transfer area of a single heater and low thermal conductivity in formation, the heating effect is not optimal. However, because to the restricted heat transfer area of a single heater and the formations' low thermal conductivity, the heating effect is not optimal[11]. To improve the heat transfer performance of downhole electric heaters, researchers began investigating the optimization of the heat transfer structure to increase heat transfer efficiency by causing a change in the fluid flow pattern inside the shell. Traditional bow-shaped baffle plates may provide transverse flow, but it is

accompanied by a flow dead zone, inequality heat transmission, and a shell range pressure decrease[12]. The baffling rod structure may create a longitudinal flow and minimize resistance, but its design is complex and difficult to popularize[13, 14].

In this case, Vukic et al. proposed a helical baffle structure that significantly improves heat transfer capacity by forming an approximate plunger flow and enhancing the shell turbulence effect, while effectively avoiding inherent weaknesses of the transverse flow structure[15, 16]. The helical baffle is classified into two forms based on its structural features: lap and continuous. Although the lap type has made substantial advances in heat transfer performance, there are still limitations such as insufficient local scouring and the formation of blind zones, particularly at high flow rates, which have a major influence on the heater performance[17]. In contrast to the continuous helical baffle plate heat ex-changer structure for electric heating rods to provide a stable continuous support, shell flow uniformity. The structural parameters have little influence on the stability of the heat transfer performance, which provides a high degree of flexibility and engineering adaptability.

Previous research has shown that using downhole heaters in high-temperature condition can cause casing creep and extra thermal loads, reducing casing strength and stability[18]. According to some researches, the axial thermal expansion stress of casing column is a primary cause of casing breakage during the heavy oil thermal recovery process. Almost 85% of casing damage occurs at the coupling[19], which might be attributed to the coupling's material properties, connection method, and insufficient heat exposure[20]. Downhole heating may exacerbate the corrosive effects of fluids in the well-bore, while the high-temperature environment can also

cause scaling and deposition[21]. As a result, the integrity of the well bore is compromised. As the size of thermal mining activities grows, the issue of wellbore integrity failure in geothermal wells owing to thermal stress, corrosion, and clogging becomes more prevalent. To encourage the development of in-situ pyrolysis technology for tar-rich coal re-sources, this research focuses on one of the most significant technical components of in-situ pyrolysis: the downhole heater. To investigate the heat transfer performance and wall stability of heater in the well, based on the helical baffle heat transfer structure, the temperature and pressure distribution of the heater, as well as the formation temperature and pressure evolution regulation, were simulated and analyzed using FLUENT. The heat transfer performance under various flow rates and heating powers was comprehensively evaluated, and the influence of the downhole heater on wall stability during the heating process was comprehensively analyzed, with the goal of verifying the heater feasibility and validity. A strong technical support system for the effective development and usage of tar-rich coal will be implemented.

## 2. MATERIAL AND METHOD

### 2.1. Test area information.

The test area formation is located in the Yangjiapo block of Linxing East, which is located in the northern region of Xingxian County, in the Hedong coal field, Shanxi Province. The target coal seam is No. 8+9, which has a depth of 1000 meters and a thickness of 10-11 meters. As indicated in Figure 1. Table 1 shows the special thermophysical properties of tar-rich coal in the test area.

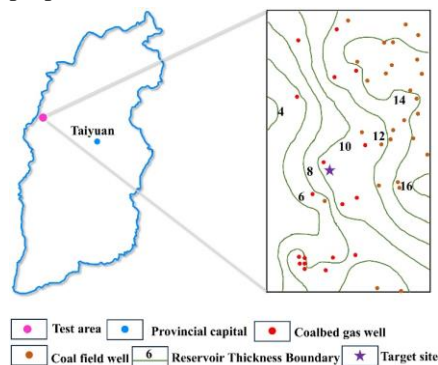


Fig.1. Test area location

TABLE 1. THERMOPHYSICAL PARAMETERS OF TAR-RICH COAL

Object	Density	Thermal conductivity	Specific heat capacity	Thermal diffusion coefficient
Tar-rich coal	1.48 t/m <sup>3</sup>	0.37 W/m · K	1.14 kJ/kg · K	0.43 W/m · °C

### 2.2. Physical model.

This research proposes a large-size helical baffle heater (LSHB) based on the formation characteristic of the test

region. Its primary construction is made up of three parts: a helical baffle, an electric heating rod (heat source), and a

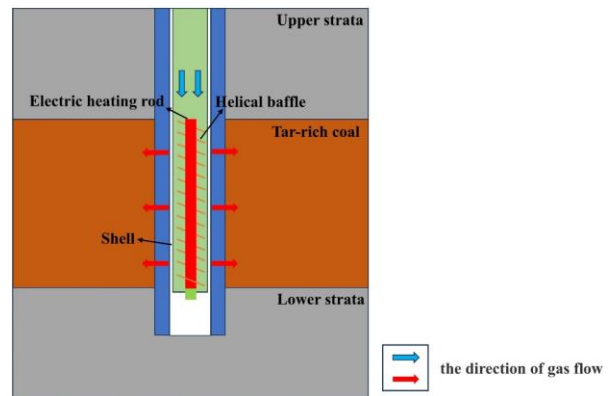


Fig.2. Schematic diagram of tar-rich coal heating in situ using heater shell. The heater is inserted into the well, and gas is pumped into the formation via perforations in the wellbore, which is commonly separated into three layers: upper strata, tar-rich coal formation, and lower strata, as seen in Figure 2. Table 2 shows the specific structure of LSHB.

TABLE 2. MODEL STRUCTURE PARAMETERS

Item	Dimension (m)	Item	Dimension (m)
Shell diameter	0.4	Reservoir thickness	10
Wall inside diameter	0.4	Baffle length	10
Wall outside diameter	0.6	Baffle thickness	2×10 <sup>-3</sup>
Upper strata thickness	3	Helical pitch	1.1
Lower strata thickness	3	Heating tube diameter	0.1

### 2.3. Parameter setting.

In order to appropriately set the simulation starting parameters, the porosity of the tar-rich coal was measured at various temperatures in the test region. In addition, well tests and numerical well tests were carried out on the tar-rich coal reservoir in the test region to determine reservoir pressure and temperature, respectively.

Porosity test samples were acquired using a dry distillation experiment. The heating rate for the dry distillation experiment was 5°C/min, and the temperature was gradually increased from room temperature to 300°C, 400°C, 500°C, 600°C, and 700°C over 30 minutes. Natural cooling was performed for 60 minutes, following which the porosity was determined using the volumetric method. The formation pressure in the test region was then determined using well test analysis. In addition, the reservoir temperature was determined using numerical well test analysis.

Figure 3 depicts the temperature-dependent trend in tar-rich coal porosity. The initial tar-rich coal in the test location had

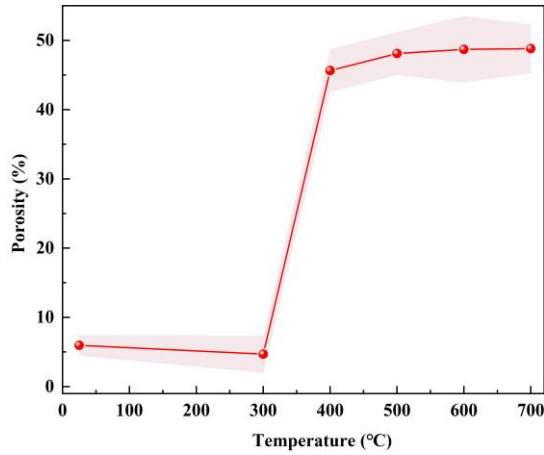


Fig.3. Tar-rich coal porosity evolution with temperature

a porosity of less than 10%, which remained rather stable following pyrolysis at 300°C. This happens given the coal at 300°C only undergoes dewatering, degassing, and carboxylate fracture reaction, and the macroscopic structure of the coal did not change significantly, making the porosity basically constant. However, the porosity increased significantly to close to 50% when the pyrolysis temperature was up to and higher than 400°C.

Coal reservoir pressure, or the fluid pressure acting on the coal pore and fissure space, also known as pore fluid pressure, has a direct impact on model injection pressure. The experimental findings suggest that the reservoir pressure of coal seam No. 8+9 fluctuates between 5.08 to 9.32 MPa, with an average of 7.48 MPa. The pressure gradient ranges from 0.73 to 0.91 MPa/100m, with an average of 0.82 MPa/100m. Furthermore, the reservoir temperature has an immediate impact on the model boundary settings. The coal reservoir has a temperature range of 15°C-31°C, with an average of 22.31°C. The coal seam has a ground temperature gradient of around 2.49°C/100m, as illustrated in Figure 4.

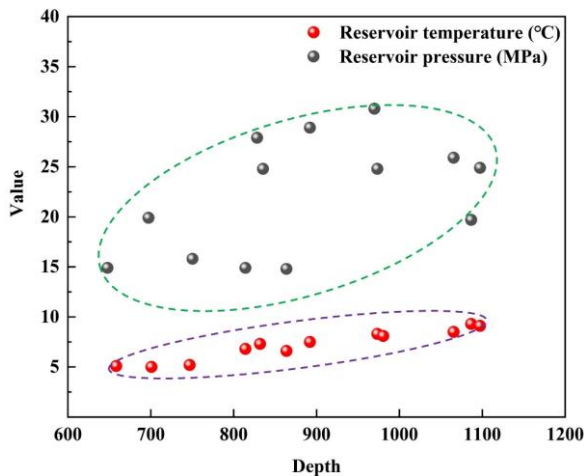


Fig.4. Reservoir temperature, reservoir pressure vs. depth

## 2.4. Simulated condition.

This paper makes the following assumptions for numerical simulations:

- The shell-side fluid was a fully developed turbulent flow in a steady state;
- The fluid in the helical channel is incompressible;
- Heat dissipation on the external wall of the shell cylinder was ignored;
- The heating rod was regarded as wall with a constant heat flux density;
- The gas inlet temperature was 283 K and the outlet was a pressure outlet boundary, which follows the constant temperature boundary condition  $T_w = 873\text{K}$ ;
- Tar-rich coal formation was considered as porous medium.

The heater shell process gas flows according to the following equations[21].

The mass conservation equation:

$$\frac{\partial \bar{u}_i}{\partial x_i} = 0. \quad (1)$$

The momentum conservation equation:

$$\frac{\partial(\rho \bar{u}_i)}{\partial t} + \frac{\partial(\rho \bar{u}_i \bar{u}_j)}{\partial x_j} = -\frac{\partial \bar{p}}{\partial x_i} + \frac{\partial}{\partial x_j} (\mu_{eff} \frac{\partial \bar{u}_i}{\partial x_j} - \rho \bar{u}_i' \cdot \bar{u}_j'). \quad (2)$$

The energy conservation equation:

$$\frac{\partial}{\partial t} (\rho E) + \frac{\partial}{\partial x_i} (u_i (\rho E + p)) = \frac{\partial}{\partial x_i} (k_{eff} \frac{\partial T}{\partial x_i}) - \sum_{j'} h_{j'} J_{j'} + u_j (\tau_{ij})_{eff} + S_h. \quad (3)$$

Where  $E = h - \frac{p}{\rho} + \frac{u_i^2}{2}$ ;  $k_{eff}$  is effective conductivity coefficient;  $J_{j'}$  is component diffusion flow rate;  $S_h$  is volumetric heat source.

The universal governing equation for mass, energy, momentum, and RNG  $k-\varepsilon$  turbulent viscosity is as follows:

$$div(\rho U \Phi) = div(\Gamma_\Phi grad \Phi) + S_\Phi. \quad (4)$$

Where  $U$  is the velocity vector;  $\Phi$  is a universal variable representing;  $u_i$ 、 $T$ 、 $k$ 、 $\varepsilon$  or another variable;  $\Gamma_\Phi$  is a generalized diffusion coefficient, and  $S_\Phi$  is a generalized source term.

## 2.5. Energy correction.

In this simulation, there is no external energy increase in the energy system, hence the energy absorbed per kilogram of air,  $Q$  is defined as[22]:

$$Q = (f_{out} - f_{in}) + \frac{1}{2} (V_{out}^2 - V_{in}^2) + g(Z_{out} - Z_{in}) + \Delta W_t. \quad (5)$$

Where  $f$  is the specific enthalpy of the air;  $V$  is the velocity of the shell-side air;  $g$  is gravity acceleration;  $Z$  is the

relative height of the cross section in the reference system and  $\Delta W_t$  is the heat loss power per kilogram of air.

If we suppose that the air density is constant, because the inlet and outlet have the same area,  $V_{in}$  is equal to  $V_{out}$ . It is assumed that the electric energy is completely converted into heat energy during the steady heating of the air.

The heat loss power  $\Delta W$  per unit time of air is expressed as follows[23]:

$$\Delta W = P_a - (f_{out} - f_{in})M_s. \quad (6)$$

The effective power  $P_e$  is defined as follows[24]:

$$P_e = P_a - \Delta W = (f_{out} - f_{in})M_s. \quad (7)$$

Where  $P_a$  is the actual power;  $M_s$  is the mass flow rate of the air.

To accurately evaluate the heat transfer performance of the SLSHB, this study utilized the total heat transfer coefficient formula from STHX. The total heat transfer coefficient of the SLSHB was determined by refining the concept of heat transfer temperature difference. The validity of this approach has been confirmed in previous investigations.

The overall heat transfer coefficient  $K$  is defined as follows [24]:

$$K = \frac{P_e}{A\Delta t_m}. \quad (8)$$

The logarithmic heat transfer temperature difference  $\Delta t_m$  is defined as follows:

$$\Delta t_m = \frac{\Delta t_{max} - \Delta t_{min}}{\ln\left(\frac{\Delta t_{max}}{\Delta t_{min}}\right)}. \quad (9)$$

Where  $A$  is the total heat transfer area of the heating rod;  $\Delta t_{max}$  is the temperature difference between the temperature on the heating rod surface and the temperature at the top-most fluid outlet and  $\Delta t_{min}$  is the temperature difference between on the heating rod surface and the temperature at the bottom fluid outlet. The surface temperature of the heating rod obtained by linear interpolation of the surface temperature measured at the fully expanded section of the heating rod.

The Reynolds number  $R_e$  of the SLSHB is defined as follows [25]:

$$R_e = \frac{d_e U_s \rho}{\mu}. \quad (10)$$

Where  $d_e$  is the hydraulic diameter of the shell side;  $U_s$  is the velocity of the shell side;  $\rho$  is the characteristic density of the shell-side air and  $\mu$  is the characteristic viscosity of the shell-side air.

The convective heat transfer coefficient  $h$  of the heating rod surface is defined as follows[26]:

$$h = \frac{\lambda N_u}{d_e}. \quad (11)$$

Where  $\lambda$  is the thermal conductivity of the heating rod material.

The shell-side Nusselt number  $N_u$  is defined as follows[27]:

$$N_u = C R_e^m P_r^{\frac{1}{3}}. \quad (12)$$

The shell-side Prandtl number  $P_r$  is defined as follows[27]:

$$P_r = \frac{c_p \mu}{\lambda}. \quad (13)$$

The thermal resistance of an SLSHB is defined as follows[28]:

$$\frac{1}{K} = \frac{1}{h} + R_f. \quad (14)$$

Because the SLSHB is new, the fouling resistance  $R_f$  is not considered, and the following equation can be obtained:

$$K = h. \quad (15)$$

## 2.6. Test scheme.

In this experiment, the heater outlet was set to function in a free-flow state, and a constant heating power scheme was used to thoroughly evaluate the heater's performance. The inlet and output gas temperatures, pressures, and surface temperature of the heater rod were all measured. Furthermore, the shell-side Reynolds number and total heat transfer coefficient were investigated at various heating powers and gas flow rates. Cross-over tests were carried out to determine the heater's performance at two distinct gas flow rates and heating power levels.

## 3. RESULTS AND ANALYSIS

### 3.1. Shell flow characterization.

Figures 5 and 6 show the flow rate distribution and the trajectory of the fluid in the heater during in-situ heating. Inside the LSHB shell, the fluid has a consistent spiral flow pattern, and the flow rate drops gradually and consistently along the direction of flow. This unique flow feature guarantees that the fluid flows uniformly and continuously across the surface of the heater rod, eliminating excessive heat rod concentration, which significantly improves the LSHB heater's stability and dependability in difficult downhole working conditions. This function is especially significant for LSHB heaters with lengths more than 10 meters, since it guarantees that the heater maintains an effective and consistent heating effect along its length. While the fluid flow rate reduces as the shell is expanded, the flow rate increases significantly near the fluid outlet, which is where the perforations are located. The fluid channel is abruptly constricted at the position of the perforations, causing the fluid to flow quicker as it exits the heater. The higher flow rate improves heat exchange efficiency between the fluid and the heater rod to some extent, hence boosting the heater's overall effectiveness.



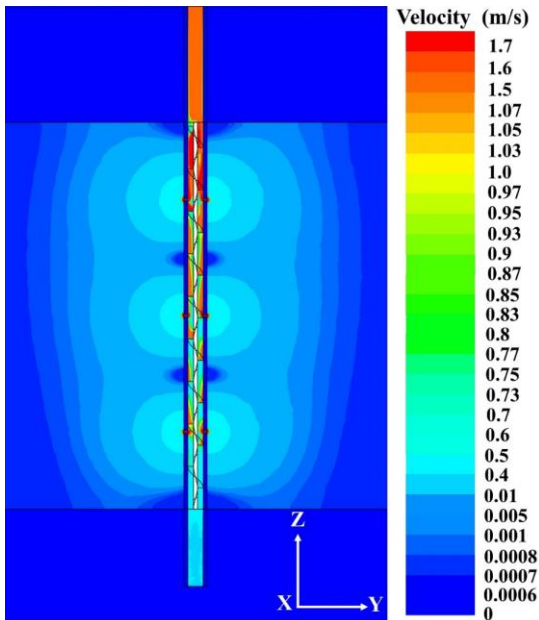


Fig.5. Characterization of the fluid in the shell (500m<sup>3</sup>/h)

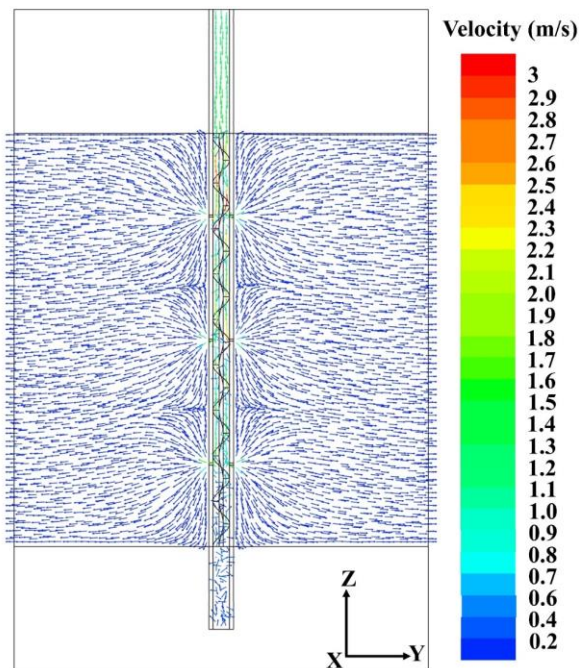


Fig.6. Flow field trajectory (500m<sup>3</sup>/h)

During the extensive inspection, we also discovered a distinct zone of low velocity fluid flow on the backwind side of the baffle. This problem is most likely the result of an extremely large pitch design, which inhibits fluid movement in this location. The presence of this low velocity zone cannot be overlooked, as it may cause the fluid to not touch and cool the heater rod uniformly, resulting in localized overheating of the rod's central piece, posing a real and possible risk to the heater's lifetime. As a result, while optimizing the structure's future design, we must consider the heater's pitch size. By adjusting the pitch appropriately, we can guarantee that the fluid throughout the heating process maintains a consistent and efficient flow

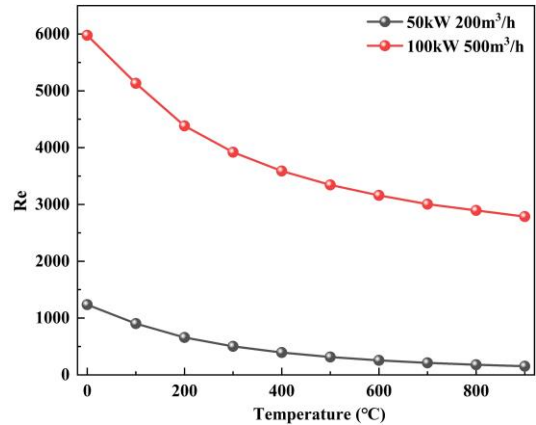


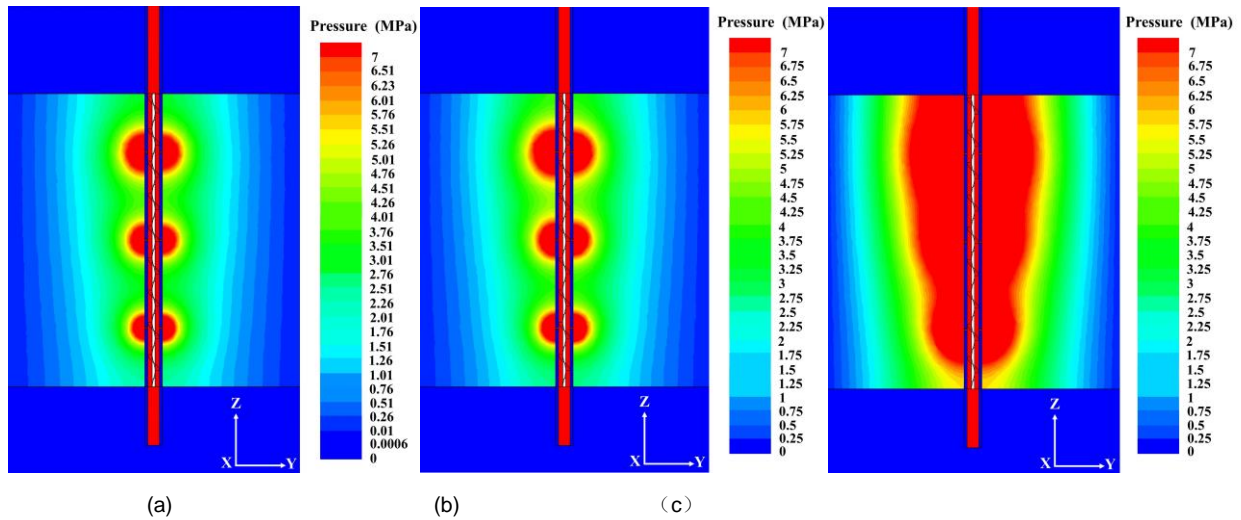
Fig.7. Re number versus temperature

state, which not only extends the heater's utilizable life but also considerably improves its overall performance[2]. Figure 7 demonstrates that the Reynolds number gradually decreases as temperature increases. This is because increasing the temperature causes a rise in fluid viscosity, which increases flow resistance and reduces the Reynolds number. In addition, the heater's top-down perforation design contributes to the Reynolds number decline. At 50kW, the decline in Reynolds number is very gentle, but at 100kW, it becomes more significant. This is because more heating power causes the gas to heat up faster over the same flow distance, resulting in a more dramatic change in the fluid flow condition. In 100 kW, 500m<sup>3</sup>/h, the fluid flow pattern presents a turbulent pattern, which under the heat transfer effect is relatively better. As the flow distance increases, the flow pattern shifts from turbulence to laminar flow, which means that the temperature at the bottom of the heater is the highest, but the heater's heat transfer efficiency may not be the best. In 50 kW, 200m<sup>3</sup>/h, the fluid flow maintains a laminar flow pattern, which is more stable but has lower heat transfer efficiency compared to the former.

### 3.2. Pressure Characterization.

Figure 8 clearly shows that there is no significant loss of pressure in the shell when the number of perforations in the wellbore rises. This phenomenon is mostly linked to the original experimental parameter and the meticulous design of the heater. Throughout the studies, the flow rate of the injected fluid was kept substantially greater than the flow rate of the fluid output, ensuring that there was always enough gas within the heater. This de-sign not only helps to maintain a consistent flow of gas inside the heater, but it also guarantees that the heating process runs constantly and effectively. The bigger volume implies that the heater can handle more gas, which improves heating efficiency and capacity. In reality, this implies that the heater can heat more gas to the desired temperature in less time, giving significant support for in-situ heating.

In addition, the larger injection flow effectively compensates for any pressure loss that may occur during the use of the heater. Even if some pressure loss occurs, it can be quickly replenished by continuous injection of additional gas, thus maintaining a stable pressure inside the heater.



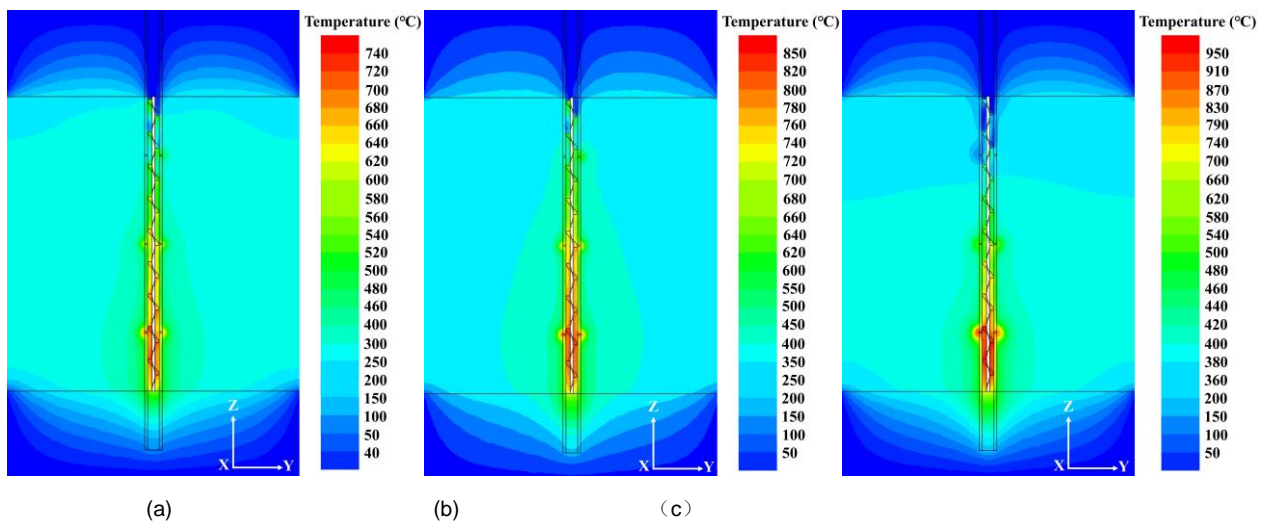
**Fig.8.** Pressure distribution under in-situ operation of the heater. (a) 50kW, 200m<sup>3</sup>/h; (b) 100kW, 200m<sup>3</sup>/h; (c) 100kW, 500m<sup>3</sup>/h

This design not only improves the reliability and stability of the heater, but also ensures a smooth in-situ heating. Figures 8 (a) and (c) indicate that the pressure in the tar-rich coal formation increases significantly as the injection flow increases. This alteration is of critical relevance for in situ heating. The increased pressure in the formation indicates that the convective heat transfer of gas in the formation will be greatly improved. Convective heat transfer is one of the primary ways of heat transmission in the in-situ heating. It ensures uniform heating for the whole formation by transferring heat from the heater to the gas in the formation, which then passes the heat to the surrounding coal body. When formation pressure rises, gas flows through the formation at a higher pace, making convective heat transfer more effective. The quicker the gas flow, the more heat it can take away and transfer to a longer distance, so widening the heating range and enhancing heating efficiency. This gain effect is extremely beneficial to the development of in-situ heating, as it may minimize heating time, energy consumption, and enhance oil recovery. Aside from the gain impact of convective heat transfer, a rise in formation

pressure improves other conditions in in-situ heating. According to previous studies, formation pressure can enhance formation permeability, making it simpler for gases to move and disperse inside the formation. Simultaneously, it can increase the thermal conductivity of tar-rich coal, making heat transmission and diffusion simpler. All of these enhancements contribute to the overall effectiveness of in-situ heating and provide more robust support for the efficient use of tar-rich coal resources[23, 29].

### 3.3. Temperature Characterization.

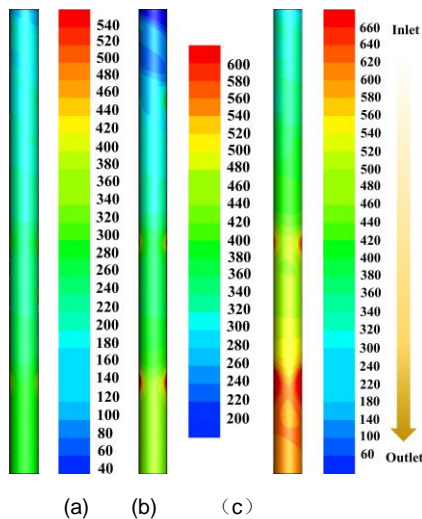
Figure 9 demonstrates that as the length of the heater shell grows, so does the shell temperature, and the formation temperature follows suit. As the fluid goes via various perforated outlets, the flow velocity and scouring of the heater rods reduce, resulting in a steady increase in temperature at the heater bottom. Furthermore, increased fluid viscosity exacerbates the situation. Other helical baffle heaters have the same phenomena of rising temperature at



**Fig.9.** Temperature distribution under in-situ operation of the heater. (a) 50kW, 200m<sup>3</sup>/h; (b) 100kW, 200m<sup>3</sup>/h; (c) 100kW, 500m<sup>3</sup>/h

the bottom as the heater shell climbs. The shell-side perforation has no effect on this phenomenon, which is inherent in helical baffle plates.

When Figures 9 (a) and (b) are compared, the temperature at the outlet of the lowest perforation gradually rises and exceeds that of the surrounding formation, and this warming trend gradually spreads to the location of the middle perforation outlet, which is primarily caused by increasing the heating power of the center heating rod. Figure 9 (c) shows this warming process more clearly. Observing the overall temperature distribution reveals that



**Fig.10.** Wall temperature distribution in heat injection well.

(a) 50kW, 200m<sup>3</sup>/h; (b) 100kW, 200m<sup>3</sup>/h; (c) 100kW, 500m<sup>3</sup>/h

this sort of heater can greatly enhance the temperature of the tar-rich coal formation, particularly the tar-rich coal formation in the lower center. During the heating phase, the tar-rich coal formation warms the surrounding formation via conduction.

### 3.4. Characterization of wall temperatures in heat injection well.

Figure 10 depicts a number of interesting and relevant elements in the wall temperature distribution under various conditions. We can plainly see that the wall temperature steadily increases with the depth of the heat injection, which is quite similar to the heater temperature distribution pattern. This likeness is not coincidence, because the heat created by the heater as a heat source is progressively transmitted to the wall by heat conduction and convective heat transfer, resulting in an increase in wall temperature.

Closer to the inlet, the wall temperature resembles a spiral plunger flow. This is because the gas at the inlet has just entered the heater and has not yet been exposed to the heating rod's high temperature stress. Hence, its flow pattern is symbolized by a spiral plunger flow. This flow pattern enhances heat exchange between the gas and the heater rod, causing the wall temperature at the inlet to rise fast. As the fluid flows and the heating proceeds, the temperature at the perforation outlet may rise above that of the surrounding wall, as seen in Figures 10 (b) and (c). This phenomenon occurs because the flow velocity at the

perforation outlet is higher, resulting in a greater convective heat transfer impact. When the fluid flows out of the heater at a faster rate, it removes a substantial quantity of heat and strongly exchanges heat with the surrounding wall, resulting in a comparatively high wall temperature at the perforation outlet.

As the heater shell length and the number of perforation outlets grow, the convective heat transfer impact of gas rapidly diminishes. This is because increasing the length of the shell and the number of perforated outlets creates a more complicated and convoluted flow path for the gas in the heater, causing the flow velocity to slow down and limiting the effectiveness of convection heat transfer. In addition, as the heating progresses, the temperature difference between the heater and the gas narrows, making convective heat transfer less effective. Because the gas flow rate at the bottom of the heater is sluggish, the wall temperature is often conveyed by heat conduction. In the heat transfer process, the heating rod heats the gas in the shell, which subsequently transmits heat to the well-wall. Furthermore, because the heating rod is positioned in the middle of the heat injection well, the temperature distribution at the wall's bottom will be symmetrical, with the heat injection well serving as the symmetry axis.

This temperature distribution feature is critical for the in-situ exploitation of tar-rich coal reserves. A reasonable use of this temperature distribution function can considerably increase heat transfer efficiency and lifetime. For example, reduce the pitch, increase the shell length, and arrange the location and number of perforation outlets in a reasonable manner, optimize flow path and flow time of the fluid in the heater, so that the fluid can interact more fully with the heating rod for heat exchange, and improve the flow rate of gas and convective heat transfer effect. The above optimization approaches can greatly increase the heater heat transfer efficiency and lifespan, resulting in a more efficient and dependable heating solution for in-situ mining of tar-rich coal reserves.

Wellbore materials (such as steel casing) can produce thermal strains during downhole heating as a result of thermal expansion and contraction. Although this stress may not be substantial in the near term, long-term consequences affect the material micro-structure, increasing the chance of fatigue, cracking, and even fracture. When the heating temperature exceeds the material endurance limit, the mechanical properties of the material are significantly reduced (e.g., lower strength and weaker toughness), making the wellbore more susceptible to deformation or damage when subjected to external pressure or internal fluid impact. Furthermore, the uneven wall temperature distribution induced by downhole heating is a concern that should not be overlooked. The temperature of the wellbore wall may fluctuate depending on the position of the heating source, the heating power, and the thermal conductivity in the formation, resulting in a temperature gradient. This temperature gradient not only concentrates thermal stresses in the wellbore material, but it can also cause the wellbore to deform axially or radially. In particular, in areas with large temperature gradients, such as



near a heating source or where the thermal conductivity of the formation changes abruptly, the concentration of thermal stresses may exceed the material ultimate strength, causing localized damage to the wellbore. Localized high temperatures may develop in some portions of the wellbore, producing thermal damage to the material (e.g., carbonization, melting), which poses a major danger to the wellbore structural integrity. As a result, selecting appropriate high-temperature-resistant materials and a fair range of heating temperatures is critical to assuring the wellbore structural safety.

In conclusion, despite the 50kW, 200m<sup>3</sup>/h, the wall temperature at the inlet remains over 160°C, which is severely detrimental to the well-wall stability. The range of thermal stress impact grows in proportion to the wall temperature gradient. According to field experience[30], the thermal expansion impact of the well-wall is insignificant when the wall temperature is less than 50°C. As a result, given the parameters of this simulation, the wall temperature effect on ranges between 20 and 40 meters.

### 3.5. Thermal characterization.

According to equations (8)-(15), the relationship between the overall heat transfer coefficient and the shell-side flow field can be expressed as follows[30]:

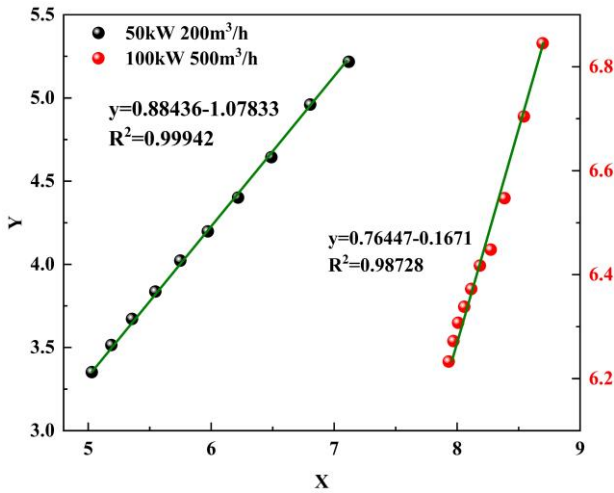


Fig.11. Heat transfer correlation

$$\ln\left(N_u P_r^{-\frac{1}{3}}\right) = D \ln(R_e) + \ln(C). \quad (16)$$

The following equation can be obtained by the linear regression of equation (16):

$$Y = DX + b. \quad (17)$$

Where  $X = \ln(R_e)$ ,  $Y = \ln\left(N_u P_r^{-\frac{1}{3}}\right)$ , and  $b = \ln(C)$ . The regression parameters are listed in Table 3.

TABLE 3. REGRESSION PARAMETERS

Test	C	m	K (W·m <sup>-2</sup> ·K <sup>-1</sup> )
50KW, 200m <sup>3</sup> /h	0.175	0.9213	0.1764
100KW, 200m <sup>3</sup> /h	0.175	0.9213	0.3704
100KW, 500m <sup>3</sup> /h	0.175	0.9213	0.1946

The linear regression equation properly depicts the fundamental relationship between the Nusselt number, Prandtl number, and Reynolds number. Figure 11 clearly shows that the fitting of both regression curves is 0.96 or higher, proving the regression equation's dependability and correctness.

The regression curve shows a slope of around 0.7645 for 100 kW power and 500 m<sup>3</sup>/h flow rate, and 0.8844 for 50

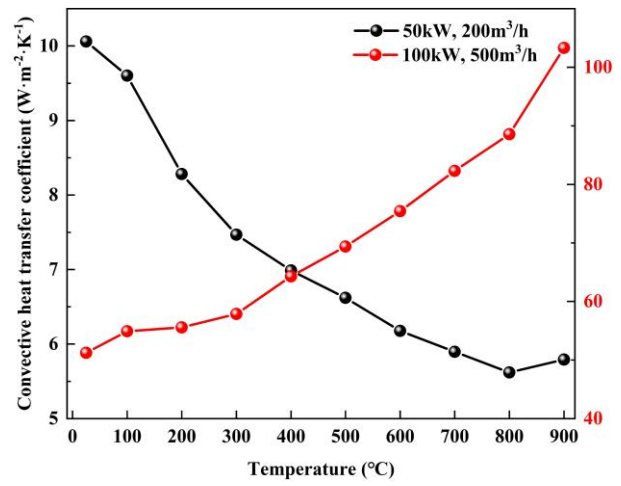


Fig.12. Convective heat transfer coefficient vs. temperature

kW power and 200 m<sup>3</sup>/h flow rate. This disparity suggests that changing the Reynolds number has a greater impact on the combination of Nusselt number and Prandtl number under low power and flow conditions. At other words, when the fluid is at low power and low flow, it transitions from laminar to turbulent flow more quickly, which contributes to increased heat transfer efficiency. In the 50 kW, 200 m<sup>3</sup>/h, the intercept is 1.0783, whereas the 100 kW, 500 m<sup>3</sup>/h has an intercept of 0.1671. This disparity shows a major variance in the heaters' heat exchanger efficiency under various base conditions. Specifically, for low Reynolds numbers, high power and flow rate conditions result in a slight decline in the heat transfer performance. This discovery is a useful guideline for adjusting the heater's operating settings and increasing heat transfer efficiency. The gas temperature rises as the gas flow channel within the heater shell expands. Figure 12 depicts the temperature dependent trend of the convective heat transfer coefficient, which reflects the complexity of fluid movement and heat transmission within the heater. The convective heat transfer coefficient declines with shell length at 50 kW and 200 m<sup>3</sup>/h, but steadily increases at 100 kW and 500 m<sup>3</sup>/h. The low gas injection flow rate of 200 m<sup>3</sup>/h at 50 kW prevents effective convective heat transfer between the gas and heating rod. However, convection is more effective at 100 kW and 500 m<sup>3</sup>/h.



As the gas moves along the heater shell, its flow condition and time, which has a substantial impact on heat transmission. On the one hand, the gas in the flow continues to absorb heat, and the temperature steadily rises, causing changes in the physical properties of gas (such as density, viscosity), influencing the size of the Reynolds number. Figure 7 demonstrates that the Reynolds number gradually decreases as the gas temperature increases. As the Reynolds number decreases, the fluid flow stabilizes and the turbulence degree decreases, affecting the efficiency of convective heat transmission to some amount. The heater, on the other hand, is constructed with a number of perforated outlets in the heater shell, which causes the gas to continually diverge during the flow, so reducing the flow rate. The slower flow rate increases the gas's residence time inside the heater and the contact time with the heating rods, providing more possibilities for heat exchange. The increase in heat transfer coefficient is compounded by the greater heating rate and, as a result, the rate of gas warming in the shell, particularly at 100 kW.

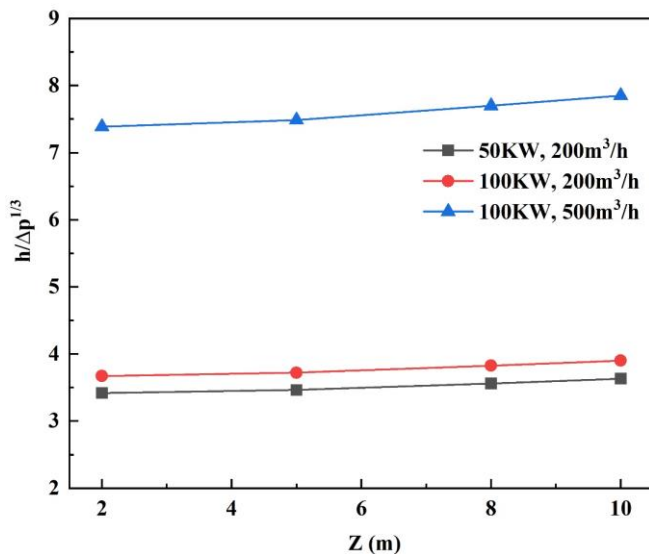


Fig.13. Comprehensive evaluation vs heater route

### 3.6. Comprehensive assessment.

The crucial parameters for evaluating heater performance are  $h/\Delta P$  and  $h/\Delta P^{1/3}$ . This option is meant to more accurately portray the heater's balanced connection between heat transfer efficiency and energy usage[29].

Figure 13 depicts the tight link between the heater shell length and total heat transfer performance. The temperature at the bottom of the heater rises as the shell length grows, which directly affects the gas viscosity during flow. As viscosity rises, the internal friction of the gas flow increases, and the flow rate decreases, lengthening the gas's residence time in the heater and offering a greater chance for heat transfer between the gas and the heating rod. As a result, the heat transfer coefficient increases, improving the heater's

total heat transfer performance. This phenomenon is especially evident in simulation experiments with varying heating powers, where the improvement in the heat transfer coefficient is more pronounced under the high power condition because more heat is generated by the heating rod, the temperature of the gas rises faster, and the viscosity change is greater.

However, when focusing great thermal efficiency, we must not overlook the energy consumed during heater operation. In the same heat transfer structure, raising the injection flow rate can minimize pressure loss in the shell, which looks to benefit the heater's overall heat transfer performance. In reality, this increase is ineffective, particularly at low power levels (for example, 50 kW). This is because, while reduced pressure losses reduce energy consumption, higher injection flow requires more gas to be supplied by surface equipment, increasing the cost of mining and exploiting unconventional energy sources. With the rising cost of energy extraction, this additional input might be a significant obstacle to the heater's overall performance improvement.

As a result, while optimizing heater design and enhancing heat transfer efficiency, we must take into account a wide range of elements. On the one hand, it is required to increase the heat transfer coefficient by adjusting the heater's structural characteristics (e.g., shell length, heating rod arrangement, etc.) so that the heater can fully use thermal energy while working effectively. On the other hand, it is vital to monitor energy usage and lower the heater's running costs through proper flow control and pressure loss management. Furthermore, in order to optimize the heater's overall performance, the long-term stability of its operation, maintenance costs, and environmental friendliness must all be taken into consideration.

Finally, full performance evaluation of heaters is a complicated and diverse task. When picking assessment indexes, we should take into account the heater real working environment and application needs. When optimizing the design, heat transfer efficiency, energy consumption, cost, and environmental impact should all be addressed in order to obtain the optimal balance of the heater's overall performance[31].

## 4. CONCLUSIONS

This work investigates the heat transfer performance of a large-size helical baffle heater with the goal of improving a critical heating step for in-situ pyrolysis of tar-rich coal resources. Fluent numerical simulation was used to thoroughly analyze the heater's temperature and pressure distribution, as well as the wall temperature, in the heat injection well under in-situ conditions, while orthogonal simulation was used to investigate heat transfer and overall performance under various injection flow rates and heating powers. The study initially proved the heater's in-situ viability and provided a foundation for its further structural optimization. The specific conclusions are as follows:

The fluid in the shell program flows in a spiral pattern with a steadily declining flow velocity, the exit flow velocity is

much greater than the surrounding area, and a low velocity flow zone exists on the backwind side of the baffle plate. The increased temperature causes a reduction in the Reynolds number, which is worsened by the sidewall perforations outflow. At high power and flow rates, fluid flow becomes turbulent, but at low power and flow rates, fluid flow remains laminar. The increased openness of the sidewall perforations causes no substantial pressure loss, and the high injection flow rate compensates for the pressure loss, increases formation pressure, and promotes gas convective heat transfer.

- The shell temperature rises with length, and the fluid outlet temperature at the bottom is the greatest, spreading toward the center, while the heating process is accompanied by heat transmission to the surrounding formation. The well-wall temperature rises with the depth of the heat injection well, indicating spiral plunger flow characteristics at the input; the temperature at the outlet may be greater than the wall; and the temperature distribution at the bottom is symmetrical.
- At low power and high flow rate, changing the Reynolds number has a major influence on heat transfer efficiency, as the fluid transitions from laminar to turbulent flow more quickly, hence increasing heat transfer. At low Reynolds numbers, the heater's heat transfer performance is slightly poorer at high power and flow rates.
- Heater shell length rises, as does bottom temperature, gas viscosity, flow rate, heating duration, and heat transfer coefficient, all of which enhance total heat transfer performance. In the same heat transfer structure, raising the injection flow rate has little effect on lowering pressure loss and enhancing total heat transfer performance, particularly under low power.
- Conventional heaters have a restricted heating scope; this heater may increase the heating scope, enhance the convection heat transfer effect, and have a smaller diameter, allowing for a wider range of applications. The pitch is too large, resulting in insufficient limitations on the gas flow pattern, and the position and number of perforations on the side wall significantly affect the heating range. Structural optimization should take into account the weight of the heater in relation to the cable's tensile strength, and it is advised that perforations be moved down or reduced in number.

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