



Article Analysis of Enhanced Heat Transfer Characteristics of Coaxial Borehole Heat Exchanger

Lin Sun^{1,2}, Biwei Fu^{2,*}, Menghui Wei² and Si Zhang^{1,2}

- ¹ Cooperative Innovation Center of Unconventional Oil and Gas, Yangtze University
- (Ministry of Education & Hubei Province), Wuhan 430100, China
- ² School of Mechanical Engineering, Yangtze University, Jinzhou 434023, China
- * Correspondence: fubiwei@yangtzeu.edu.cn

Abstract: Coaxial borehole heat exchangers provide a practical method for geothermal energy extraction, but heat transfer efficiency is low. In order to address this problem, three coaxial borehole heat exchangers with vortex generators, based on the enhanced heat transfer theory, are proposed in this paper. The author compared and analyzed the heat transfer performance of three coaxial borehole heat exchangers with vortex generators and those of traditional structures, which explains why the new heat exchanger's heat transfer mechanism is enhanced. The results demonstrated that the vortex generator can enhance the fluid flow's turbulent kinetic energy in the coaxial heat exchanger. This generator can also improve the mixing characteristics of the fluid flow and heat transfer. The resultant increase in the inlet flow velocity can decrease the friction coefficient f, increase the Nusselt number and strengthen the coaxial sleeve. As a result, the heat exchange performance of the tubular heat exchanger will also be improved. The thread vortex generator (TVG) heat exchanger outperforms the other three heat exchangers in terms of heat exchange performance, extraction temperature and heat extraction power. The results evidenced that the TVG heat exchanger is better than the smooth tube heat exchanger. The thermal performance coefficient PEC was improved by 1.1 times, and the extraction temperature and heating power were increased by 24.06% and 11.93%, respectively. A solid theoretical foundation is provided by the extracted outcomes for designing and selecting high-efficiency coaxial borehole heat exchangers suitable for geothermal energy extraction.

Keywords: geothermal energy; coaxial heat exchanger; vortex generator; enhanced heat transfer

1. Introduction

Geothermal energy is sustainable and clean energy with an extremely high utilization value. This energy can effectively address the current global energy crisis and achieve "carbon peaking" and "carbon neutrality". The coaxial borehole heat exchanger is regarded as one of the most efficient methods of geothermal energy extraction [1]. The coaxial borehole heat exchanger is a closed-circulation heat exchange system with numerous advantages, including a simple structure, a large heat exchange area and an easy installation. In addition, there is no geothermal fluid production, which saves energy for reinjection and protects the environment. This heat exchanger has a simple structure, a large heat transfer surface area and a convenient installation procedure, among other advantages. Therefore, it has a wide range of engineering applications. The heat transfer performance of the coaxial borehole heat exchanger directly affects the efficiency of geothermal exploitation. Hence, it is crucial to improve the heat transfer performance of the coaxial borehole heat exchanger. At present, the articles in the literature that have dealt with the heat exchange performance of the coaxial borehole heat exchanger mainly focus on the structural size of the heat exchanger, the geological environment where the heat exchanger is located and the working fluid.

By carrying out numerical calculations and tests, the impact of the heat exchanger's structural size on its heat transfer performance has also been examined in several articles



Citation: Sun, L.; Fu, B.; Wei, M.; Zhang, S. Analysis of Enhanced Heat Transfer Characteristics of Coaxial Borehole Heat Exchanger. *Processes* 2022, *10*, 2057. https://doi.org/ 10.3390/pr10102057

Academic Editors: Ferdinando Salata and Virgilio Ciancio

Received: 18 September 2022 Accepted: 9 October 2022 Published: 12 October 2022

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Copyright: © 2022 by the authors. Licensee MDPI, Basel, Switzerland. This article is an open access article distributed under the terms and conditions of the Creative Commons Attribution (CC BY) license (https:// creativecommons.org/licenses/by/ 4.0/). in the literature. Alimonti [2] established a numerical model of wellbore heat exchangers to simulate the effects of different inner diameters of pipes and heat transfer fluids on the performance of heat exchangers. Pan Sheng et al. [3] conducted a sensitivity analysis on the heat exchanger's design parameters, such as the outer diameter, inner diameter, flow rate, outer tube material, grouting material and drilling depth. Consequently, they found that the outer diameter of the pipe, drilling depth and flow rate significantly affected the heat extraction rate. They also proposed an optimization method based on the lowest average energy consumption index. Noorollahi et al. [4] simulated heat transfer between the fluid injected into the well and the surrounding hot rock. Their results demonstrated that thermal gradients, input mass flow, the well casing geometry and size of injection and withdrawal piping are crucial for the thermal recovery output. Song Xianzhi et al. [5] developed the unsteady heat transfer model of a deep coaxial borehole heat exchanger (CBHE) and comprehensively analyzed the essential elements affecting the CBHE's performance.

Simultaneously, some academics have investigated the heat exchanger's geological environment. Renaud et al. [6] used computational fluid dynamics techniques to simulate the 30-year production of a closed-circulation deep borehole heat exchanger (DBHE) and investigated the heat impact and heat recovery associated with drilling into a hypothetical DBHE in the Iceland Deep Drilling Project (IDDP)'s geological environment. Chen Chaofan et al. [7] used OpenGeoSys (OGS) software and the bicontinuous medium method to construct a comprehensive numerical model and examine the effects of pipe material, grouting thermal conductivity, geothermal gradient, soil thermal conductivity and groundwater flow on DBHE. Saedi et al. [8] introduced a novel transient analysis modeling method for geothermal energy production that ensures stable surface fluid temperature by invoking the circulating fluid circulation rate. Simultaneously, geothermal gradients, vertical well depth and thermal inner strength were found to significantly affect the wellhead fluid temperature. Holmberg et al. [1] established the numerical model of the coaxial BHE and conducted a parametric investigation of the coaxial shell with different drilling depths, flow rates and collector characteristics.

The working fluid exerts a significant effect on the heat exchanger's performance. Shi Yan et al. [9] investigated the heat transfer mechanism and optimization method of using CO_2 as a circulating fluid in a mid-depth coaxial heat exchanger geothermal system. They discussed the effects of injection/production pressure differences, mass flow and specific enthalpy changes on the heat production performance. Their results demonstrated that the thermal extraction efficiency of CO_2 -based MDCHEs was increased by 31% compared with water-based MDCHEs. Zhang Yiqun et al. [10] investigated the endothermic performance of nine working fluids in a downhole coaxial heat exchanger (DCHE) geothermal system. They evaluated the applicability of the working fluids under the same conditions and studied the effects of parameters on the wellbore, reservoir and fluid. Luo Yongqiang et al. [11] proposed a new analysis model for CBHE. They studied the model's sensitivity to the fluid's flow direction, geothermal gradient and borehole thermal resistance. Bu Xianbiao et al. [12,13] established an equation describing the heat exchange between fluid and rock. They conducted a parametric investigation to determine the optimal values of the main parameters. Their results showed that the geothermal energy generated by abandoned wells strongly depends on fluid flow rates and geothermal gradients.

To improve heat transfer performance in the heat exchanger, it is crucial to increase the heat transfer area, heat transfer temperature difference and heat transfer coefficient. Increasing the heat transfer area is a common and effective approach for improving the heat transfer process. To this end, scholars usually adopt some passive technical devices, such as the use of groove geometry [14], corrugated surfaces [15], twisted belts [16], fins [17], tapered inserts [18] and other types of vortex generators [19]. These devices can significantly increase flow mixing, boundary layer disruption and turbulence to improve the heat transfer performance. Some scholars have also studied the effects of nanofluids on heat transfer [20–23].

Although some studies in the literature have focused on the optimal design of the structure of the coaxial borehole heat exchanger, some research results have achieved improvements in the (application background) heat transfer of the coaxial tube. The strengthening methods mainly include installing ties, coil springs and fin film, among others. Some research work and foundations can be provided for this study. Hamed Arjmandi et al. [24] performed a numerical simulation of the combination of a vortex generator, twisted ribbon vortex generator and Al₂O₃-H₂O nanofluid. They examined the heat transfer rate and pressure drop characteristics of the double tube heat exchanger. The combiner's geometry is optimized based on the response surface method of the central composite design. Mehedi Hasan Tusar et al. [25] used numerical simulations to investigate the heat transfer performance of the coaxial double-tube heat exchanger. Their results demonstrated that the annulus with the spiral belt has a higher heat transfer coefficient (HTC) and friction coefficient than the ordinary annulus. Rishabh Kumar et al. [26] numerically investigated a novel triangular perforated twisted tape with a V-notch inserted in the inner tube of a double-tube heat exchanger. They discovered that its thermal performance exhibited the best hydraulic performance at a lower pitch value of 50 mm. Gnanavel et al. [27] numerically examined the thermal and flow fields of nanofluidic double tube heat exchangers with different coil spring inserts. Furthermore, Do Huu-Quan et al. [28] numerically investigated the turbulent forced convective heat transfer in a new type of inner flat tube double tube heat exchanger. They studied the effects of the inner tube geometry on flow and heat transfer and found that the effect of the inner tube geometry on the thermal properties strongly depends on the value of Re. Iman Bashtani et al. [29] numerically examined the effects of adding alumina to water in a six-gear disk water-to-water double-tube heat exchanger. Their results showed that the turbulator improved the thermal effect. The fluid's collision with the turbulator's surface and the subsequent rupture of the boundary layer increased the heat transfer and local Nusselt number by approximately 70%. Simultaneously, adding nanoparticles to heat exchangers with turbulators increases the average Nusselt number, efficiency and transfer units. Marwa A.M. Ali et al. [30] improved the heat exchange efficiency of the heat exchanger by introducing the rotation of the inner tube of the heat exchanger and adjusting the tube's eccentricity. Their findings demonstrated that the heat transfer rate was significantly increased by 223% because of the variation in the eccentricity by up to 40 mm and the inner tube rotation speed of 500 rpm.

Osama A. Mohsen et al. [31] experimentally analyzed the heat transfer of the doubletube heat exchanger with different fin geometries on the heat exchanger's surface. Fin geometries include interrupted rectangular fins, circular fins and helical ribs. They experimentally measured the HTCs and pressure drops of hot and cold fluids with different Reynolds numbers. Their findings showed that the heat transfer coefficient could be improved by using different fin geometries to expand the surface. Moreover, the improved heat transfer of rectangular fins is the largest, and the improved heat transfer of circular fins is the smallest. Z. Iqbal et al. [32] used the genetic algorithm as the optimizer and the discontinuous Galerkin finite element method (DG-FEM) as the solver of the governing equations to investigate the optimal design problem of the longitudinal perturbation of the outer surface of the double pipe inner pipe. Their findings revealed that the characteristic length, the number of fins, the electrical conductivity of the heating surface material and the number of control points significantly affect the fins' optimal design.

To summarize, vortex generators, such as fins and spiral ties, positively affect the improvement of heat transfer in coaxial tubes. Therefore, three new coaxial borehole heat exchangers with vortex generators (to form a new special structure) were proposed in this work, namely the impeller vortex generator (IVG), the bump vortex generator (BVG) and the thread vortex generator (TVG), for geothermal energy extraction. In addition, numerical simulation was performed to systematically investigate the impact of the vortex generator on the heat transfer enhancement of the heat exchanger by establishing a three-dimensional heat transfer model of a coaxial borehole heat exchanger with a vortex

generator. Furthermore, the three heat exchangers under different inlet flow rates were compared and analyzed to strengthen their mechanisms and heat transfer performances. Our work paves the way for the improvement of the extraction efficiency of the coaxial borehole heat exchanger.

2. Geometric Model and Evaluation Index of the Coaxial Borehole Heat Exchanger

2.1. Working Principle of Coaxial Borehole Heat Exchanger

Figure 1 depicts the working principle of the coaxial borehole heat exchanger. The heat exchanger consists of a hot rock mass, an injection channel, an extraction channel and an inner pipe. The well wall forms the annular cavity. The inner pipe was used as the injection channel, the inner pipe was used as the production channel and the bottom of the injection channel was closed. The supercooled fluid continuously absorbs heat from the surrounding hot well wall during its downward injection along the injection channel. After passing through the vortex generator, the supercooled fluid improves the fluid flow and mixing characteristics, destroys the boundary layer near the wall and enhances heat transfer. After this fluid reached the bottom of the well, the high-temperature fluid was produced from the production channel because of the bottom hole pressure, resulting in geothermal energy production.



Figure 1. Schematic diagram of working principle of the coaxial borehole heat exchanger.

The coaxial borehole heat exchanger in this study was mainly used for medium and deep geothermal energy extraction, with a well depth from 2000 m to 3000 m. According to the needs of various sites, one or more vortex generators can be added at different well depths to enhance the heat recovery efficiency of coaxial borehole heat exchangers. In order to reveal the enhanced heat transfer mechanism of a single vortex generator, a heat transfer

simulation model with an L = 10 m length was constructed based on the coaxial borehole in the hot rock area at the bottom of the well.

2.2. Geometric Model of Coaxial Borehole Heat Exchanger

This paper proposed three types of vortex generators, namely IVG, BVG and TVG structures, to obtain a coaxial borehole heat exchanger structure with improved heat transfer performance. Simultaneously, this paper developed a geometric model of the coaxial heat exchanger with vortex generators (Figure 2) to analyze the effect of different vortex generator structures on the heat transfer performance of the coaxial borehole heat exchanger. The model considers the heat exchanger's bottom hole L = 10 m ($L = L_1 + L_2 + L_3$) to examine the strengthening mechanism of the vortex generator in the heat exchanger in greater detail. Table 1 presents the specific geometric parameters of the model.



Figure 2. Geometric model of coaxial borehole heat exchanger.

Parameter	Value(mm)	
L_1	2850	
L_2	300	
L_3	7150	
D_1	177.8	
D_2	100	
D ₃	130	
Н	200	

Table 1. Geometrical parameters of coaxial borehole heat exchanger with a vortex generator.

2.3. *Enhanced Heat Transfer Parameters and Heat Transfer Performance Evaluation Index* 2.3.1. Enhanced Heat Transfer Parameters

The local Nusselt number *Nu* can be expressed as follows [15]:

$$Nu = \frac{hD_h}{k} = \frac{q''}{T_w - T_m} \frac{D_h}{k} \tag{1}$$

q'' represents the heat flux, and T_m can be expressed as follows:

$$T_m = \frac{T_{in} + T_{out}}{2} \tag{2}$$

The hydraulic diameter D_h is expressed as follows:

$$D_h = \frac{4A_c}{P_w} \tag{3}$$

The Darcy coefficient of friction in the fully developed water flow is estimated as follows [25]:

$$T = \frac{2D_h}{L} \frac{\Delta P}{\rho \mu^2} \tag{4}$$

where ΔP is the pressure difference between the inlet and outlet. Importantly, the Darcy coefficient of friction determines the pump's power requirements.

An increased flow resistance accompanies an improved heat transfer. Therefore, most studies [15,18,33] use the thermal performance coefficient, PEC, to assess heat exchangers' overall heat transfer performance. Its specific expression is presented as follows:

$$PEC = \frac{(Nu/Nu_s)}{(f/f_s)^{1/3}}$$
(5)

where Nu and Nu_s are the Nusselt numbers of the vortex generator heat exchanger and the smooth tube heat exchanger, respectively.

2.3.2. Heat Transfer Performance Evaluation Index

The extraction temperature T_{out} and the extraction power Q_{out} directly reflect the amount of geothermal energy absorbed by the fluid. These parameters can be used in the evaluation index of the heat exchange performance of the coaxial heat exchanger. The heating power Q_{out} is affected by the fluid's physical parameters, the flow channel's size, the fluid flow rate, and the temperature difference between the inlet and outlet. The heating power is expressed as follows [34]:

$$Q_{out} = c_{\rho}\rho D_2 V_{out} (T_{out} - T_{in}) \tag{6}$$

3. Numerical Analysis Model and Its Validation

3.1. Model Assumptions

Due to the intricacies of the heat exchange process between the heat exchanger and the rock, the following reasonable assumptions were made to comprehensively analyze the heat exchange performance of the heat exchanger:

- (1) The rock and the soil around the geothermal underground coaxial borehole heat exchanger were treated as a homogeneous medium. Moreover, the effect of groundwater seepage is ignored, and the heat transfer in the underground rock and soil is treated as pure heat conduction.
- (2) The temperature at the numerical simulation region's radical boundary is considered constant.
- (3) The bottom hole's heat source and surface temperature are considered constant.
- (4) The temperatures of the rock and the wellbore were considered the same because the wellbore has been attached to the rock for a long time.

3.2. Governing Equations

The (RNG) k- ε turbulence model with good performance in eddies and strong streamline bending was used to solve the flow field. It is based on the above model's assumptions and the Reynolds-averaged Navier–Stokes (RANS) model. The main governing equations for continuity, momentum and energy can be respectively expressed as follows [18]:

$$\frac{\partial}{\partial x_i}(\rho u_i) = 0 \tag{7}$$

$$\frac{\partial}{\partial x_j}(u_j\rho u_i) = -\frac{\partial p}{\partial x} + \frac{\partial}{\partial x_j} \left\{ \mu \left[\frac{\partial u_i}{\partial x_j} - \overline{\rho u_i' u_j'} \right] \right\}$$
(8)

$$\frac{\partial}{\partial x_j}(\rho u_i T) = \frac{\partial}{\partial x_i} \left\{ \left[\frac{\mu}{Pr} + \frac{\mu_t}{Pr_t} \right] \frac{\partial T}{\partial x_i} \right\}$$
(9)

where ρ , p, u and μ are the fluid density, pressure, turbulent fluctuation, pressure and dynamic viscosity of water, respectively. The Reynolds stress term ($\overline{\rho u'_i u'_j}$) can be expressed as follows:

$$\overline{\rho u_i' u_j'} = \mu_t \left[\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right] - \frac{2}{3} \rho k \delta_{ij} - \frac{2}{3} \mu_t \frac{\partial u_k}{\partial x_k} \delta_{ij}$$
(10)

The equations for turbulent kinetic energy (k) and dissipation rate (ε) are respectively expressed as follows [24]:

$$\frac{\partial}{\partial x_i}(\rho u_i k) = \frac{\partial}{\partial x_j} \left\{ \left[\frac{\mu_t}{\sigma_k} + \mu \right] \frac{\partial k}{\partial x_j} \right\} + G_k \rho \varepsilon$$
(11)

$$\frac{\partial}{\partial x_i}(\rho u_i \varepsilon) = \frac{\partial}{\partial x_j} \left\{ \left[\frac{\mu_t}{\sigma_{\varepsilon}} + \mu \right] \frac{\partial \varepsilon}{\partial x_j} \right\} + \frac{\varepsilon}{k} [C_{1\varepsilon} G_k - \rho C_{2\varepsilon} \varepsilon]$$
(12)

In the above equation, $\sigma_k = 1$, $\sigma_{\varepsilon} = 1.3$, $\sigma_{1\varepsilon} = 1.42$ and $C_{2\varepsilon} = 1.68$. k = 2 is a constant for the (RNG) *k*- ε turbulence model.

3.3. Boundary Conditions

The inlet boundary was set as the velocity inlet, and the outlet boundary was set as the pressure outlet. The heat transfer boundary condition between the circulating fluid in the tube and the tube wall was defined as the coupled heat transfer boundary. The surface temperature was 288.15 K [12], the bottom boundary of the heat exchanger was set at a constant temperature of 408.15 K (Formula 13) and the temperature gradient of the borehole wall was $T_g = 4.5$ K/100 m [35,36], which was controlled by a user-defined function.

To obtain more accurate numerical calculation results, the rock's temperature and the pressure at the outlet can be expressed as follows:

$$T_W = T_{sur} + T_g z \tag{13}$$

$$P_{out} = \rho g z \tag{14}$$

The fluid medium in the calculation is water, and the casing and inner pipe are steel. The specific physical parameters are shown in Table 2.

Table 2. Physical parameters.

Parameter/Unit	Water	Inner Pipe and Well Wall
Density $\rho/\text{kg}\cdot\text{m}^{-3}$	998.2	8030
Specific heat $c_{\rho}/J \cdot (\text{kg} \cdot \text{k})^{-1}$	4182	502.48
Thermal conductivity $k/W \cdot (m \cdot k)^{-1}$	0.6	16.27
Viscosity $\mu/\text{kg}\cdot(\text{m}\cdot\text{s})^{-1}$	0.001003	0

3.4. Mesh and Model Validation

3.4.1. Meshing and Independence Verification

Figure 3 shows the meshing scheme of the coaxial borehole heat exchanger. Notably, a mixed mesh of structured hexahedrons and unstructured tetrahedrons is used. By considering the impact of the boundary layer near the wall area of heat transfer, the boundary layer and mesh refinements were performed at the interface between the tube wall and the fluid and the surface of the vortex generator to effectively capture the turbulent flow characteristics and address the viscous sublayer effect near the tube wall in question. Moreover,



Table 3 presents the mesh quality parameters for ST and TVG, which are consistent with the literature [25].

Figure 3. Mesh generation of coaxial borehole heat exchangers.

Table 3. Mesh metrics.

Geometry	Nodes	Average Skewness	Average Orthogonal Quality	Average Aspect Ratio
ST	485,750	0.078	0.985	5.309
TVG	538,195	0.178	0.927	2.108

In order to eliminate the influence of the mesh on the acquired numerical calculation results, the mesh independence verification of the numerical model is performed. As shown in Figure 4, when the inlet speed was $V_{in} = 0.1 \text{ m/s}$, the Nusselt number Nu, the friction coefficient of the ST heat exchanger and the TVG heat exchanger changed as a function of the number of meshes. Additionally, it can be seen that the Nu and friction factors were almost the same for different mesh nodes. Thus, it can be concluded that the simulations were mesh independent.

3.4.2. Numerical Simulation Model Validation

To verify the extracted numerical simulation results, the *Nu* and friction factor coefficients of the ST heat exchangers were compared with those obtained by other works in the literature [25,37].

The Dittus–Boelter correlation for *Nu* is as follows:

$$Nu = 0.023 \mathrm{Re}^{0.8} \mathrm{Pr}^n \tag{15}$$

where Pr stands for the Prandtl number, n = 0.4 for heating (applicable in this study) and n = 0.3 for cooling.

$$\Pr = \frac{c_p \mu}{\lambda} \tag{16}$$



Figure 4. Grid independency verification; (**a**) grid independency verification of *Nu*; (**b**) grid independency verification of *f*.

The Gneilski correlation for *Nu* is as follows [38]:

$$Nu = \frac{(f/8)(\text{Re} - 1000)\text{Pr}}{1 + 12.7(f/8)^{\frac{1}{2}}(\text{Pr}^{\frac{2}{3}} - 1)}$$
(17)

where $0.5 \le Pr \le 2000$ and $3000 \le Re \le 10^6$, while the friction factor correlation of Petukhov can be expressed as follows [39]:

$$f = \frac{1}{\left(-1.64 + 0.79Ln(\text{Re})\right)^2}, \ 3000 \le \text{Re} \le 10^6$$
(18)

Friction factor correlation for Blasius is in the form,

$$= 0.3164 \operatorname{Re}^{-0.25} 4 \times 10^3 < \operatorname{Re} < 10^5 \tag{19}$$

Figure 5 displays a comparison of the simulation results of the ST heat exchanger and the empirical formula. As can be ascertained, the error between the simulation results and the empirical correlation formula is less than $\pm 6\%$, which verifies the rationality of the introduced numerical model.



Figure 5. Validation of the numerical analysis model. (a) Comparison of Nu and (b) f between the current work and other reports in the literature.

4. Results and Discussions

4.1. Analysis of the Enhanced Heat Transfer Mechanism of Vortex Generator

4.1.1. The Effect of Vortex Generators on the Turbulent Kinetic Energy

Figure 6 shows the turbulent kinetic energy distributions of the four kinds of coaxial heat exchangers in the vortex generator area. As can be observed, the turbulent kinetic energy of the three coaxial heat exchangers with vortex generators is more significant than that of the smooth tube coaxial heat exchanger. Furthermore, the turbulent kinetic energy of the TVG heat exchanger is much larger than that of the other three exchangers. Physically speaking, the vortex generator reduces the cross-sectional area of the fluid flow and increases the average velocity of the flow cross-sectional area as well as Reynolds number, which increases turbulent energy. The guiding and shearing effects of the vortex generator on the fluid led to the following changes: the flow direction and velocity of the fluid changed, the turbulent kinetic energy of the fluid was enhanced and the destructive effect of the thermal boundary layer was enhanced. As a result, the heat transfer between the fluid and the hot rock was enhanced, leading to an improved efficiency in geothermal energy extraction.



Figure 6. Distributions of the turbulent kinetic energy in the vortex generator area of the four heat exchangers.

Figure 7 shows the distributions of the turbulent energy and velocity vectors at the bottom of the well for the different longitudinal sections of the heat exchanger. This figure thoroughly demonstrates the difference between the turbulent kinetic energy and the velocity vector at the bottom of the well for different heat exchangers. Compared with other heat exchanger structures, the IVG and TVG heat exchangers significantly impact the flow state of the bottom hole fluid. The velocity vector shows a low-velocity zone in the bottom hole cavity of the smooth tube (ST) and BVG heat exchangers. The fluid has more of a slow flow, and the central fluid in the injection channel directly flows into the inner pipe from the end of the inner pipe, resulting in a lower heat exchange effect. The higher turbulent kinetic energy of the IVG and TVG heat exchangers destroys the low-velocity zone in the bottom hole cavity and weakens the boundary layer. This higher energy also allows more slow-flowing high-temperature fluids to mix with the central fluid, thus, improving the heat exchange process. The turbulent kinetic energy is the strongest and has the most substantial effect on the slow-flow region. Simultaneously, a high turbulence area was observed at the inlet of the inner pipe because of the change in the flow channel's diameter, which is conducive to the rapid outflow of the fluid through the inner pipe. It is also conducive to enhancing the heat exchange efficiency of the heat exchanger.



Figure 7. Distributions of turbulent energy and velocity vector at the bottom hole of the different longitudinal sections of the four heat exchangers.

4.1.2. Effect of Vortex Generator on the Velocity Field

Figure 8 depicts the velocity distributions of the vortex generation region along the longitudinal and cross-section Plane1. More specifically, Figure 8a presents the velocity distribution of the Plane1 section. This figure shows that when the fluid passes through the vortex generator of the coaxial heat exchanger, the fluid's velocity increases because of the reduction in the flow cross-sectional area. Figure 8b shows that the fluid velocity first increases and decreases along the injection direction because the fluid enters the vortex generation area, leading to a decrease in the flow cross-sectional area and an increase in the fluid velocity. While flowing out from the vortex generation area, the velocity decreases rapidly because of the increase in the flow cross-sectional area. The massive change in velocity causes an increase in the turbulent kinetic energy of the fluid. As a result, the damage to the thermal boundary layer increases, improving the heat exchange of the coaxial heat exchanger. At the same time, the vortex generator changes the fluid flow

from one-dimension to three-dimension. It increases the radial and tangential velocities of the fluid, which destroys the thermal boundary layer and improves the heat exchange performance of the coaxial heat exchanger. Among them, the velocity change in the fluid in the TVG is the most evident, and the fluid's thermal boundary layer destruction effect is the best. To summarize, the TVG heat exchanger exhibited a good heat exchange performance. This structure (TVG), as the vortex generator of the coaxial heat exchanger, is beneficial in improving the mining efficiency of geothermal heat.



Figure 8. Velocity distributions in the vortex generator area of the four heat exchangers.

Figure 9 presents the velocity variation of fluid in the injection and production channels of different heat exchangers along the well's depth. This figure shows that after the fluid passes through the vortex generator, the fluid velocity increases due to the reduced cross-sectional area of the runner. The increasing velocity strengthens the heat exchange between the fluid and the hot wall. Among them, the TVG heat exchanger exerts the most obvious acceleration effect. Simultaneously, under the disturbance and swirl effect of the vortex generator, the fluid velocity changes from one-dimensional motion to three-dimensional motion. This change increases the time for heat exchange between the fluid and the hot wall and enhances the destructive effect of the boundary layer, thereby improving the heat recovery energy. After the fluid passes through the bottom of the well, the fluid velocity in the inner tube increases because of the reduction in the inner tube's cross-sectional area.

Figure 10 illustrates the velocity variation curve of the cross-section Plane1 along the radial direction. The fluid's velocity in the near-wall boundary layer was relatively low due to fluid viscosity, weakening the heat transfer between the fluid and the hot rock wall. This figure shows that the fluid's flow velocity in the injection channel increases significantly after passing through the vortex generator, and the flow velocity in the area close to the hot rock wall surface increases greatly, which is conducive to enhancing the turbulent flow characteristics of the fluid in the area near the hot rock wall surface, destroying the boundary layer fluid and improving the flow rate of the heat transfer energy between the fluid and the hot rock wall. Compared with the other three heat exchangers, the fluid's flow velocity in the TVG heat exchanger on the Plane1 cross-section fluctuates significantly

under the influence of the asymmetric threaded vortex generator's structure. Moreover, the flow velocity in the area near the hot rock wall increases significantly. This increase can effectively reduce the effects of the boundary layer on heat exchange, improve the heat exchange of the fluid diameter in the radial direction and improve the heat recovery effect of the coaxial heat exchanger.



Figure 9. Velocity variation curves along the well's depth in the injection channel and production channel of the four heat exchangers.



Figure 10. Velocity variation curves along the radial direction of cross-section Plane1 of four heat exchangers.

Figure 11 shows the velocity variation curves of the fluid in the cross-section area injected into the channel at the vortex generator in the *X*, *Y* and *Z* directions, respectively. As can be observed from Figure 11a, the *X*-direction velocity V_x of the ST heat exchanger

approaches the value of 0, while V_x of the heat exchangers with vortex generator fluctuates and causes some fluid disturbance in the X-direction. Moreover, the velocity fluctuation of the IVG is the most obvious since the flow direction of the fluid is changed by the vortex generator structure. Figure 11b shows that the Y-direction velocity V_y of the IVG and TVG heat exchangers was significantly improved compared with the ST heat exchangers. The disturbance and shear effects of the vortex generator structure on the fluid increase the value of V_y , so that the high-speed fluid near the wall can effectively help the fluid to mix. It will destroy the boundary, helping the central cold fluid and the peripheral hot fluid to mix, as well as improving the heat transmission efficiency. The Y-direction velocity V_y of the heat exchanger is arranged in order from high to low as follows: TVG > IVG > BVG > ST. As can be ascertained from Figure 11c, the vortex generator also has a promoting impact on the Z-direction velocity V_z , whereas the increase in TVG is the greatest. It is conducive to improving the heat transmission effect of the fluid and wall surface.



Figure 11. Velocity variation curves of fluid in the injection channel of the cross-section at the vortex generator of the different heat exchangers. (**a**) Variation curve of the velocity in the *X*-direction of the four heat exchangers; (**b**) variation curve of the velocity in the *Y*-direction of the four heat exchangers; (**c**) variation curve of the velocity in the *Z*-direction of the four heat exchangers.

4.1.3. Impact of Vortex Generator on the Temperature Field

Figure 12 shows the cross-sectional temperature distributions of the four coaxial heat exchangers in the vortex generation area. The cross-sectional temperature distribution of the ST heat exchanger reveals that when the low-temperature fluid was injected into the channel, it continued to absorb heat from the hot rock wall, further increasing the fluid temperature in the area close to the hot rock wall. The temperature of the central area of the fluid increases because of the heat transfer inside the fluid in the radial direction. The fluid's temperature near the insulated pipe's wall was still low due to the small thermal conductivity of water. Compared with the ST heat exchanger, the fluid's turbulent flow characteristics are enhanced after it passes through the three vortex generators. Furthermore, the heat exchange between the fluid and the surface of the hot rock was strengthened and the low-temperature area of the fluid was significantly reduced. The low-temperature area basically disappeared in IVG and TVG heat exchangers. These findings demonstrate that the IVG and TVG heat exchangers have a better heat recovery effect.



Figure 12. Temperature distributions in the vortex generator area of the four heat exchangers.

Figure 13 presents the average temperature variation curve of the fluid in the injection channel and the production channel of the four heat exchangers along the cross—section of the well's depth. This figure shows that the temperature of the fluid injected into the annulus increases gradually with the well's depth and decreases locally in the region where the vortex occurs. This is because the fluid's velocity is faster in the vortex region and the turbulent characteristics of the fluid are enhanced. In addition, the local high-temperature boundary layer was destroyed and the heat transfer time between the fluid and the hot rock wall was reduced, thus, decreasing the average fluid temperature. Simultaneously, Figure 13 shows that, in the near-bottom region, the high-temperature boundary layer in the injection channel cannot be directly produced from the production channel because of



its low velocity. Most of this fluid flows into the bottom hole, and the fluid temperature in the injection channel is higher than that in the production channel.

Figure 13. Temperature change curves of fluids in different heat exchanger injection channels and production channels along well depth. (a) Inject channel; (b) production channel.

Figure 14 shows the temperature distribution curves of the four heat exchangers along the radial direction in the region of the vortex generator. This figure shows that the very high fluid temperature of the boundary layer is attributed to the relatively small fluid velocity of the boundary layer near the hot rock wall surface. Furthermore, more energy is absorbed by the hot rock. The thermal conductivity of water is lower; therefore, the fluid's thermal conductivity in the radial direction is poor and the temperature of the fluid injected into the channel decreases rapidly in the radial direction. According to this figure, the fluid temperature in the production channel is presented as high to low: TVG > IVG > BVG > ST. This arrangement shows that the heat recovery of the TVG heat exchanger effect is the best, and the heat recovery efficiency of the coaxial heat exchanger is improved.



Figure 14. Radial temperature variation curve at the cross-section of the vortex generator region.

4.2. Influence of Vortex Generator on Enhanced Heat Transfer Parameters

Figure 15a shows the variation curve of the friction coefficient of the hot rock wall surface of the four heat exchangers with the inlet flow velocity. The figure shows that the friction coefficient f decreases rapidly when the inlet velocity increases from 0.1 m/s to 0.3 m/s. However, the friction coefficient decreased gradually when the inlet velocity decreased from 0.3 m/s to 1 m/s. Formula (6) shows that reducing the friction coefficient can effectively improve the thermal performance coefficient PEC of the coaxial heat exchanger. Therefore, increasing the inlet flow rate can improve the thermal performance coefficient of the coaxial heat exchanger. When the inlet flow rate is lower than 0.3 m/s, the friction coefficient is higher due to fluid viscosity and the thermal performance coefficient PEC of the coaxial heat exchanger is lower. Simultaneously, Figure 15a shows that under the same flow rate, the friction coefficient f of TVG is significantly higher than that of the other three heat exchangers, which is not conducive to improving the heat exchange performance of the coaxial heat exchanger. Therefore, increasing the inlet flow rate is beneficial in improving the heat exchange performance of the coaxial heat exchanger. Figure 15b shows the variation of the Nusselt number Nu with the inlet flow velocity for four different heat exchangers. The figure shows that the Nusselt number Nu increases linearly with the increase in the flow rate. The increasing speed is presented in descending order as follows: TVG > IVG > BVG > ST. Combined with Formula (6), the thermal performance coefficient PEC of the TVG heat exchanger increases the most with the increase in the inlet velocity. Simultaneously, under the same inlet flow rate, the Nusselt number Nu from large to small is as follows: TVG > IVG > BVG > ST. Compared with the ST heat exchanger, when the inlet velocity is 1 m/s, the Nusselt number Nu in the BVG, IVG and TVG heat exchangers increases by 4.08%, 9.05% and 20.89%, respectively, indicating that the TVG heat exchanger has the best heat transfer performance. These findings show that the TVG heat exchanger exhibits the best heat transfer performance under the same conditions. In addition, increasing the inlet flow rate improves the heat transfer performance of the TVG heat exchanger.



Figure 15. Influence of inlet velocity on strengthening parameters. (**a**) Influence of the inlet velocity on the friction coefficient f; (**b**) influence of the inlet velocity on the Nusselt number Nu.

Figure 16 shows the variation curve of the thermal performance coefficient PEC against the inlet velocity of three kinds of coaxial heat exchangers with vortex generators. The figure shows that the value of PEC is greater than 1.0, which indicates that the proposed new coaxial borehole heat exchanger-enhanced heat transfer design is feasible. The PEC value of TVG increases rapidly with the increase in the inlet velocity. The PEC value increases slowly when the inlet velocity exceeds 0.3 m/s and gradually becomes stable. This result shows that the inlet flow velocity of over 0.3 m/s exerts a minor effect on the heat transfer performance of the TVG heat exchanger. When the inlet velocity is 0.1~0.2 m/s, the PEC values of the IVG and BVG heat exchangers decrease rapidly. Moreover, when the inlet velocity exceeds 0.3 m/s, the PEC values increase slowly and gradually become stable. Simultaneously, under the same conditions, the PEC value of the TVG is much larger than that of the BVG and IVG heat exchangers. When the inlet flow rate was 1 m/s, the PEC of the TVG heat exchangers reached 1.1, whereas that of the BVG and IVG heat exchangers were only 1.03 and 1.06, respectively. This result shows that TVG has a better heat transfer performance.



Figure 16. Influence of inlet flow rate on PEC of heat exchangers with vortex generators.

4.3. *Heat Transfer Performance Analysis of the Coaxial Borehole Heat Exchanger* 4.3.1. Effect of Inlet Flow Rate on Outlet Temperature

Figure 17 shows the variation curves of the production temperature of the four heat exchangers with the inlet flow rate. The figure shows that the production temperature gradually decreases with the increase in the inlet flow rate. When the inlet velocity increased from 0.1 m/s to 1 m/s, the production temperatures of ST, BVG, IVG and TVG heat exchangers decreased by 17.0 K, 17.6 K, 17.3 K and 15.4 K, respectively. Simultaneously, under the same inlet flow rate, the outlet temperature of the heat exchanger is presented as follows: TVG > IVG > BVG > ST. The arrangement shows that the vortex generator can enhance the heat recovery capacity of the coaxial heat exchanger. When the inlet flow rate was 0.5 m/s and the inlet temperature was 310 K, the production temperatures of ST, BVG, IVG and TVG heat exchangers increased by 16.5 K, 17.0 K, 17.8 K and 20.5 K, respectively, compared with the inlet temperature. The production temperature of the TVG heat exchanger is 24.06% higher than that of the ST heat exchanger.

4.3.2. Influence of Inlet Flow Rate on Production Power

Figure 18 shows the variation curves of the heat recovery power of the four heat exchangers with the inlet flow rate. The figure shows that the heat recovery power increases with the increase in the inlet flow rate. When the inlet temperature was 310 K and the inlet velocity increased from 0.1 m/s to 1 m/s, the heat recovery power of ST, BVG, IVG and TVG heat exchangers increased by 6.73 times, 6.68 times, 6.76 times and 7.14 times, respectively. In addition, the heat recovery power of the vortex generator heat exchanger

has been improved compared with the ST heat exchanger. When the inlet flow rate was 1 m/s, the heat recovery power of the BVG, IVG and TVG heat exchangers increased by 38.33 KW, 85.28 KW and 241.96 KW, respectively, compared with the ST heat exchanger. Among them, the output power of the TVG heat exchanger increased by 11.93% compared with the ST heat exchanger. In conclusion, the heat recovery effect of the TVG heat exchanger is the highest and the increase in the inlet flow rate is beneficial in improving the heat recovery power of the TVG. However, the fluid flow rate increases with the flow rate, resulting in an increase in the required power of the injection pump, which has higher requirements for the pump's performance. Therefore, the selection of the water injection flow rate in the project can comprehensively consider the amount of heat production, the production temperature and the pump's power consumption.



Figure 17. Variation curves of the production temperature of the four heat exchangers with inlet flow velocity.



Figure 18. Variation curves of the heat recovery power of the four heat exchangers with inlet flow velocity.

5. Conclusions

Three new structures of the coaxial borehole heat exchanger with vortex generators were proposed in this work to improve the heat transfer efficiency of the coaxial borehole heat exchanger. In addition, the researchers used numerical methods to study the enhanced heat transfer mechanism with vortex generators. Next, the researchers compared and analyzed the heat transfer performance and heat recovery capacity of the four heat exchangers. The conclusions are presented as follows:

- (1) The investigation of the enhanced heat transfer mechanism of the vortex generator revealed that the vortex generator can improve the turbulent kinetic energy of the fluid flow, increase the injected fluid velocity and flow velocity in the boundary layer region of the hot rock wall. Similarly, the fluid's velocity in the radial direction fluctuates up and down, thereby destroying the high-temperature boundary layer, strengthening the heat exchange inside the fluid and strengthening heat transfer.
- (2) From the investigation of the heat transfer performance of four kinds of coaxial borehole heat exchangers, it was found that increasing the inlet flow rate decreases the friction coefficient and increases the Nusselt number. Compared with the ST heat exchanger, when the inlet velocity was increased by 1 m/s, the Nusselt number in BVG, IVG and TVG heat exchangers increased by 4.08%, 9.05% and 20.89%, respectively. The PEC of the TVG heat exchanger reached 1.1, which is favorable for improving the heat transfer performance of the coaxial heat exchanger. The impact of the pressure drop on the heat exchanger performance needs to be reduced.
- (3) The analysis of the heat recovery capacity of four kinds of coaxial borehole heat exchangers showed that the heat recovery power increases linearly with the increase in the inlet flow rate. Increasing the inlet flow rate is beneficial in improving the heat recovery capacity of the coaxial heat exchanger; however, it is necessary to comprehensively consider the working performance of the pump to propose the optimal inlet flow rate.

By comparing the strengthening mechanism, heat exchange performance and heat recovery capacity of the four coaxial borehole heat exchangers, the order of the performance of the four heat exchangers from good to poor is presented as follows: TVG > IVG > BVG > ST. Compared with the ST heat exchanger, the production temperature of the TVG heat exchanger increased by 24.06% and the heating power increased by 11.93%, which is more suitable for geothermal mining.

Studies have shown that vortex generators can greatly improve the heat transfer performance of coaxial borehole heat exchangers, but it will result in pressure loss. Further research on the vortex generator structural parameters, the coaxial borehole structural parameters, process parameters and cycle fluid can be carried out to optimize geothermal energy extraction. It will provide a theoretical basis for the design of a coaxial borehole heat exchanger with vortex generation that has low flow friction and high heat transfer capacity.

Author Contributions: Conceptualization, methodology and validation, B.F.; formal analysis and writing—original draft preparation, L.S.; analysis and writing—part numerical analysis, M.W.; writing—review and editing, S.Z. All authors have read and agreed to the published version of the manuscript.

Funding: This project was supported by the Natural Science Foundation of China (No. 51974036 and No. 51604039), the Yangtze Fund for Youth Teams of Science and Technology Innovation (No. 2016cqt01) and the Open Foundation of Cooperative Innovation Center of Unconventional Oil and Gas, Yangtze University (Ministry of Education & Hubei Province) (No. UOG2022-29).

Data Availability Statement: The data that support the findings of this study are available within the article.

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

Nomenclature

- *T_{in}* Inlet temperature, K
- *T*_{out} Outlet temperature, K
- T_g The geothermal gradient, K/km
- V_{in} Inlet velocity, m/s
- *V*_{out} Outlet velocity, m/s
- c_{ρ} Specific heat capacity of water, J/(kg·k)
- ρ Density of water, kg/m³
- *K* Thermal conductivity of water, $W \cdot (m \cdot k)^{-1}$
- *h* Convective heat transfer coefficient of water
- *T_{sur}* Surface temperature, K
- T_w Local rock temperature, K
- *T_m* Local fluid temperature, K
- *H* The distance between the lower end of the inner pipe and the bottom of the well, mm
- z Well depth, m
- *P* Fluid pressure, Pa
- A_c cross-sectional area, m²
- P_w Wet perimeter of cross-section, m
- *g* Gravitational acceleration, m/s^2
- μ Dynamic viscosity of water, kg·(m·s)⁻¹
- *P*out Outlet pressure, Pa
- *D* Radial dimensions of the heat exchanger, mm
- *D*₁ Heat exchanger's outer tube diameter, mm
- *D*₂ Inner pipe's inner diameter, mm
- *D*₃ Vortex generator's diameter, mm
- *L*₁ Vortex generator's distance from wellhead, mm
- *L*₂ Vortex generator's length, mm
- L_3 The distance from the vortex generator to the bottom of the well, mm

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