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Improvement and optimization of the convective heat transfer in the polymer pipes with internal surface modifications

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Abstract:

The objective of the work was to improve convective heat transfer intensity in the polymer pipes, e.g., made of high-density polyethylene or cross-linked polyethylene, by applying and optimizing internal pipe surface modifications in the form of small helical ribs. These helical ribs act as turbulence promoters and additionally induce angular velocity in the flowing medium, increasing the flow path length. Such pipes with the modified inner surface may be applied in the vertical and horizontal ground heat exchangers, which are one of the most expensive elements of ground-source heat pump systems. Improving heat transfer intensity between ground and working fluid in pipes may reduce the required number or length of boreholes and, therefore, decrease the overall installation costs of the heat pump system. This will allow more heat delivery to the working fluid per one running meter of the borehole. However, due to the increased turbulence in pipes with the modified internal surface, the pressure losses will also increase, which will increase the power consumed by the pump. Therefore, finding the balance between heat transfer gains and pumping losses is necessary. This paper further numerically analyzes previously selected modifications with rectangular and equilateral triangular cross-sections. Factors such as the height of ribs (vortex generators) and helix pitch were studied in terms of pressure drop and convective heat transfer intensity. Optimal geometrical parameters of ribs were found. Compared to the smooth pipes, optimized pipes with rectangular turbulators had Nusselt numbers 47-58% higher for the considered Reynolds number range, while pipes with triangular turbulators 21-42%. However, optimized pipes also had 92-126% and 31-39% higher pressure drops for rectangular and triangular turbulators, respectively, than the smooth pipe in the same Reynolds number range.

Keywords:

Convective heat transfer intensification; Ground heat exchanger; Numerical simulations; Internal surface modification; Polymer pipe.

1. Introduction

Recently, the share of renewable energy sources in total power and heat generation significantly increases as society focuses more and more on ecological and renewable energy sources. This is due to high greenhouse gas emissions by conventional power and heat sources, an increase in global warming, and high prices of fossil fuels. Renewable energy sources have been replacing fossil fuels-based power sources for years. The current social pressure and political decisions are speeding up this trend. In the European Union, the power sector must quickly adapt to the assumed emission targets by replacing coal-fired power plants, heating plants, and combined heat and power plants with new gas/hydrogen turbines, nuclear reactors, and renewable energy sources. However, in order to achieve the very ambitious climate neutrality targets established by the European Union, it is necessary to dynamically develop technologies allowing more efficient utilization of renewable power and heat sources. This should be done on a macro-scale (i.e., on the level of the power generation system) and on a micro-scale (i.e., on the local level by power and heat consumers). One of the solutions which can be applied on the micro-scale level is the application of heat pumps for space heating and domestic hot water production. Among them, ground-source heat pumps are characterized by the highest efficiency. Therefore, such heat pumps should especially be promoted.

Ground-source heat pumps are used to transfer the heat from the ground to the buildings. Their investment costs are usually higher than in the case of air-to-air or air-to-water heat pumps, but the exploitation costs are lower as they have a higher average coefficient of performance than other types of heat pumps. They work in

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combination with the ground heat exchangers, one of the system's most expensive elements. The pipes for ground heat exchangers are usually made of polymers, e.g., high-density polyethylene (HDPE) or cross-linked polyethylene (PEX). One of the ways to decrease the overall cost of the ground-source heat pumps might be by increasing heat transfer intensity in the ground heat exchangers. This operation may reduce the required number or length of boreholes and, therefore, the system installation's overall costs. The higher effectiveness of ground heat exchangers can be achieved, e.g., by increasing the heat transfer rates inside pipes due to the introduction of modifications in the form of small helical ribs or fins on the internal surface of the pipes used in the ground heat exchanger. The purpose of the modifications on the pipe's internal surface is to increase the intensity of heat transfer between the soil and the working medium flowing through the pipe. The modifications act as turbulence promoters, increase flow turbulisation, force the formation of angular velocity in the flowing medium, and increase the flow path length. This will allow more heat delivery to the working fluid per one running meter of the borehole. However, due to the increased turbulence, the pressure losses will also rise in polymer pipes with internal surface modifications. That will increase power consumption by the pump, which circulates the working medium in the lower heat source. Therefore, the modifications allowing for the highest heat transfer intensity increase at the lowest pressure drop should be designed.

The studies on increasing heat pump systems efficiency were focused, among others, on improving the efficiency of the thermodynamic heat pump cycle itself [1], improving control systems and optimizing the heat pump system configuration and operation [2,3], or using hybrid heat pumps systems in combination with other systems [4-6]. There are only a few efforts to increase the heat transfer efficiency in the ground heat exchangers applied in the heat pump systems. The factors affecting vertical ground heat exchanger efficiency were summarised in [7]. The working fluid mass flow rate, thermal properties of the pipe material, thermal properties of the grout material and soil, soil density and moisture level in the ground, the geological structure of the earth crust in implementation location, groundwater presence, borehole diameter and depth, inlet and outlet pipe diameters, and the pipes' configuration (e.g., coaxial, single/double U-tube, spiral-tube, and multi-tube) are the ground heat exchanger parameters which have the greatest impacts on the efficiency of ground-source heat pump systems. Some of these parameters are related to the ground heat exchanger location and ground parameters. Therefore, the engineers have no influence on them. But there is a group of factors, e.g., the level of flow turbulization in the pipe, which may be modified and optimized to improve heat transfer rates in the ground heat exchanger.

The internal surface modifications, e.g., in the form of grooved, riffled, or corrugated surfaces, in pipes made of metals, were studied numerically and experimentally in many works [8-10]. However, this problem was not investigated thoroughly in the case of polymer pipes. There are only a few papers in which this problem was undertaken. The pressure drop in polyethylene (PE) pipes with internal micro-fins was numerically investigated in [11]. But the simulations were carried out for only one shape of internal micro-fins, and for pipe configurations and sections typical for vertical ground heat exchangers, i.e., entrance, U-turn, and downward sections, were considered. Łapka and Wachnicki [12] recently simulated fluid flow and heat transfer in PEX pipes with internal helical turbulence promoters in the configurations typical for ground heat exchangers. The internal pipe modifications were in the form of rectangles, trapezoids, triangles, and semicircles. They calculated the dependencies between the shape of the internal embossment and the Nusselt number (Nu) and pressure drop in relation to the smooth pipe.

This short state-of-the-art shows that there is a lack of knowledge related to the heat transfer intensity and pressure drop in the polymer pipes with internal modifications. Moreover, the polymer pipes' properties and manufacturing methods are significantly different from pipes made of metals. This means that the knowledge about heat transfer intensification in metal pipes cannot be directly applied in the case of plastic pipes. Therefore, this problem is further analyzed in this paper. Previous studies in [12] showed that heat transfer intensity in polymer pipes might be significantly improved, i.e., in the Reynolds number (Re) range of 3000-8000, depending on the shape of modification, the Nu rose up to 25% in relation to the smooth pipe. But the penalty was a significant increase in the pressure drop, which also depended on the modification shape. Among the studied shapes, Łapka and Wachnicki [12] choose the triangular vortex generators as having the most promising pressure drop and heat transfer characteristics. The second shape selected in this paper was rectangular due to its possible manufacturing easiness. However, Łapka and Wachnicki [12] emphasized that before considering these shapes in practical applications, they have to be optimized in terms of pressured drop and heat transfer intensity, i.e., by finding their optimal geometrical parameters. Therefore, the optimization problem of polymer pipes' internal modifications is undertaken in this paper.

The paper is organized as follows. At first, considered internal modifications of the polymer pipes are presented. Then, the simulation methodology is described. Next, selected shapes of turbulence promoters and the way of their optimization are described. After that, the results of the simulations are shown and discussed in terms of Nu and pressure drop characteristics, and then the work is concluded.

2. Considered pipe models and initial studies

In the previous work [12], six internal modifications were tested, i.e., their influence on the heat transfer intensity and pressure drop were evaluated in the Re range of 3000-8000 and in reference to the smooth pipe. The

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considered pipes had an outer diameter of 32 mm. The minimum wall thickness was 2.9 mm, so the inner diameter of the reference smooth pipe and also the base of the modification was 26.2 mm. The internal surface modifications were based on extruding certain shapes along the helix. In [12], this helix had a pitch of 300 mm. The considered geometries are shown in Fig. 1. For all cases, the embossings were 1.4 mm high. The first case had modification prepared by embossing a rectangle 2.8 mm wide (Fig. 1a). The second one had the shape of a trapezoid with arms at an angle of 15° and the distance between the arms centers of 2.8 mm (Fig. 1b). In the third, the vortex generators were equilateral triangles (Fig. 1c). In the fourth, the vortex generators were semicircles with a radius of 1.4 mm (Fig. 1d). In the fifth and sixth (Fig. 1e and f), the turbulators had the shape of rectangular triangles with an angle at the base of 60°, and with the right angle located on the left- and right-hand side, respectively. Moreover, the reference pipe with a smooth surface was also prepared to obtain reference results.

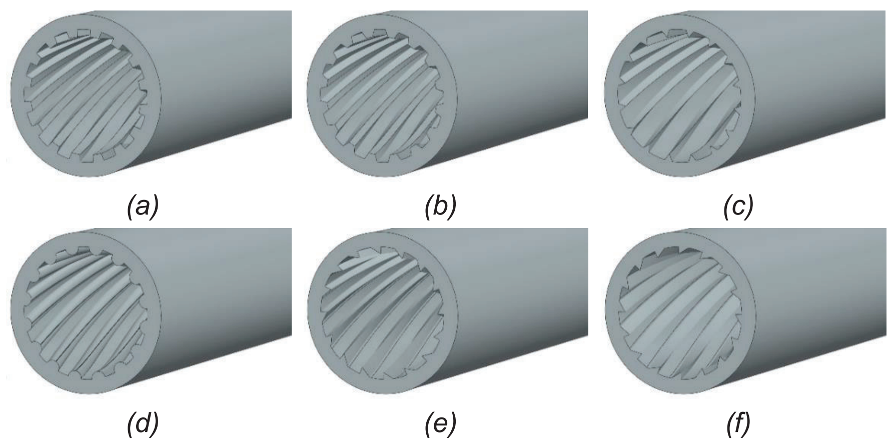

Figure 1. Shapes of turbulence promoters: a) rectangular, b) trapezoidal, c) equilateral triangle, d) semicircular, e) rectangular triangle with a right angle on the left-hand side, f) rectangular triangle with a right angle on the right-hand side.

The simulations conducted in [12] showed that triangular vortex generators had the most promising pressure drop and heat transfer characteristics. They had the lowest pressure drop increase compared to the reference pipe, i.e., they attained a pressure drop 95-125% higher than the reference pipe. The convective heat transfer was most intense for pipes with vortex generators in the shape of equilateral triangles. Moreover, the pipe with rectangle embossing performed well in terms of heat transfer intensity. In the whole considered Re range, these two types of modifications, i.e., equilateral triangle and rectangle, attained c.a. 21-23% higher Nu in relation to the smooth pipe. However, the pipe with rectangle modification had the most significant pressure drop increase, i.e., 170-190% in reference to the smooth pipe. Despite this, considering the results obtained in [12], and also manufacturing limitations, i.e., that modifications in the form of a right triangle might be difficult to manufacture and that the rectangular vortex promoters seem to be the easiest to manufacture, the optimization in terms of heat transfer and fluid flow characteristics was carried out for the embossing having shapes of a rectangle (Figure 1a) and equilateral triangle (Figure 1c). It is expected that further optimization of these geometries might significantly improve their heat transfer efficiency and reduce the pressure drop.

3. Optimization strategy

The vortex generators' optimization process was based on parameterizing some of their geometrical features (i.e., dimensions). The parameters for the optimization process were chosen so that it was possible to obtain great changes in the heat transfer and pressure drop characteristics in the pipe. The goal was to capture geometric features that can most significantly affect the pipes' heat transfer and fluid flow performances. For both selected models, the influence of the pitch of the helix was investigated. This parameter can potentially greatly impact variations in the pressure drop and heat transfer characteristics in the flow. In the base geometries, the pitch of the helix was set to 300 mm [12]. In this paper, additional calculations have been carried out for the pitch values of 150, 450, and 600 mm. The second examined parameter was the size of the turbulators, which was defined as their height. The turbulators were properly scaled to maintain the turbulence shape itself. The basic height of the vortex generators was 1.4 mm [12]. The current calculations have been performed for the following embossing heights: 0.6, 1, and 1.8 mm. Based on the optimization of these two parameters, pipe internal modifications that will potentially achieve the best heat transfer and pressure drop performances were selected. When creating optimized geometries, other geometrical parameters, e.g., the number of turbulators on the perimeter and the width of the embossing, were left unchanged. It was assessed that their influence is not as significant as the influence of helix pitch and embossing height.

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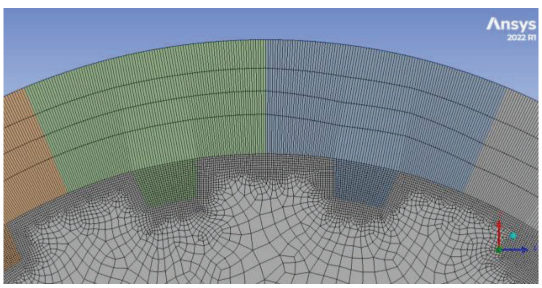
4. Simulation methodology

4.1. Discretisation

The computational meshes were generated for the optimized models presented in Figures 1a and b. Due to the necessity for the correct prediction of convective heat transfer at the pipe's internal wall, very fine meshes in the wall region had to be generated. The goal was to create as structured meshes as possible. Therefore, the sweep method available in the software ANSYS Meshing was used for this purpose, together with significant mesh refining to the wall. The length of the division along the axis was 1 mm. Additionally, the mesh size at the pipe walls was set to 0.00005 m to keep *y*+ as low as possible for the whole considered Re range. These allowed for keeping the same meshes at the inlet and outlet from the computational domain (necessary due to the periodic flow assumption in the first phase of the solution) and obtaining a satisfactorily dense mesh close to the wall. The generated meshes consisted mainly of very good quality hexahedral elements. For a few geometries, triangular prism elements were also generated. An exemplary mesh for a pipe with a rectangular turbulator with a height of 1 mm is shown in Figure 2, while the parameters of the generated meshes are presented in Tables 1 and 2 for pipes with rectangular and triangular vortex generators, respectively.

The meshes' qualities shown in Tables 1 and 2 were within acceptable ranges. However, to obtain these parameters' values, the models with modified internal surfaces had to have a very large number of elements due to the necessity to resolve near-wall regions accurately (see fine mesh at pipe boundaries in Figure 2). Moreover, the aspect ratio was very high to keep a suitable value of *y*+ at the wall, i.e., the *k*-ω shear stress transport (SST) turbulence model was applied in numerical simulations and required *y*+ to close or be below 4-5.



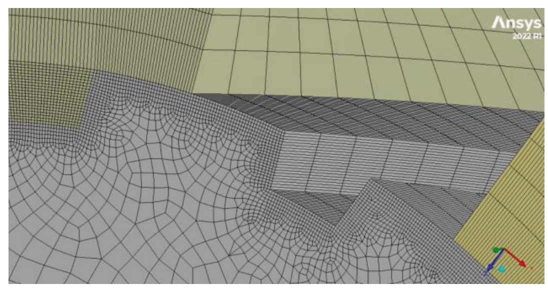


Figure 2. An exemplary mesh for a pipe with rectangular turbulators with a height of 1 mm.

Table 1. Parameters of meshes for pipes with rectangular turbulators.

Turbulator geometry	Number of elements	Max skewness	Max aspect ratio
Height = 0.6 mm	26 633 700	0.86	86.1
Height = 1.0 mm	32 958 900	0.86	84.7
Height = 1.4 mm (base pipe)	91 407 600	0.74	316.59
Height = 1.8 mm	52 866 000	0.86	76.7
Pitch = 150 mm	28 716 300	0.89	95.5
Pitch = 300 mm (base pipe)	91 407 600	0.74	316.59
Pitch = 450 mm	27 999 000	0.89	97.7
Pitch = 600 mm	30 336 360	0.87	240.4

Table 2. Parameters of meshes for pipes with triangular turbulators.

Turbulator geometry	Number of elements for single	Max skewness	Max aspect ratio
Height = 0.6 mm	26 028 900	0.89	285.7
Height = 1.0 mm	27 901 800	0.86	229.2
Height = 1.4 mm (base pipe)	117 933 300	0.74	198.51
Height = 1.8 mm	33 593 400	0.90	181.8
Pitch = 150 mm	28 716 300	0.89	204.9
Pitch = 300 mm (base pipe)	117 933 300	0.74	198.51
Pitch = 450 mm	19 101 600	0.88	230.6
Pitch = 600 mm	17 353 600	0.90	250.5

Compared to the calculations made in [12], the computational grids are characterized by much higher values of the aspect ratio parameter because longer pipes were simulated (0.3 m in [12] vs. 0.9 and 1.2 m in this work) to ensure thermally and hydrodynamically fully developed flow in the rear part of the pipe. This positively

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affects the accuracy of calculations and mapping changes in thermal and flow parameters along the pipe axis (i.e., the fully developed flow was obtained). On the other hand, skewness has slightly increased. But elements with poorer skewness are mainly located in the solid region. Only the energy equation that accounts for heat conduction in solid elements is solved there, which solution is immune to poorer quality elements. In the fluid region, the maximum skewness never exceeds 0.72 in pipes with rectangular vortex generators and 0.81 with triangular vortex generators. The $y+$ values also slightly increased compared to the results from [12], ranging from 1 to 2 for the maximal considered Re of 8000. However, the SST k-ω model allows accurate calculations for $y+$ up to 4-5.

4.2. Solution strategy

In the beginning, it was necessary to consider the stabilization of the thermal boundary layer in the pipe. This was required to find heat transfer rates for fully developed flow in the rear part of the pipes, in which the calculation results were averaged. Therefore, it was necessary to determine the optimal length of the tested pipe sections. It turned out that very good results were obtained for a pipe with a length of 0.9 m for 150, 300, and 450 mm pitches and 1.2 m for a pitch of 600 mm. The results for averaging were picked from the pipe section of length between 0.8 and 0.9 m.

Performed initial verification for the model of smooth pipe using Nu number correlations showed that the differences between correlations and numerical results were in the range of 3-5% for considered Re values. This analysis proves the good accuracy of the developed model. Moreover, mesh sensitivity analyses were performed for the smooth pipe and selected pipes with ribs, i.e., refined meshes were generated with smaller elements close to the walls. For refined grids, the $y+$ was below 1. These simulations' results were compared with those obtained for meshes used in this paper. The relative differences in calculated Nu numbers between current and fine meshes were below 1%, which showed the grid size independence of the developed model. The model validation was not performed at the current state, but the experimental studies are ongoing.

The solution strategy was divided into two phases. In the first phase, the periodic fluid flow problem in pipes without heat transfer was considered to find hydrodynamically fully developed velocity and turbulence fields in the computational domains. The fluid flow in the pipes was solved using the k-ω SST turbulence model implemented in the commercial engineering software ANSYS Fluent. In the second phase, the energy equation was only solved based on the fluid velocity and turbulence fields in the pipes obtained in the previous phase, but the problem was not further assumed periodic. In this phase, the momentum and turbulence equations were not solved. This strategy allowed for simulations of short repeatable segments of the pipes. The heat transfer problem in the pipes was solved based on the hydrodynamically fully developed flow and turbulence fields. As fluid flow and heat transfer equations were decoupled (i.e., fluid properties were assumed temperature-independent), the applied solution strategy did not influence the accuracy of the results. But it significantly speeded up the simulations and reduced the required length of pipe necessary to be simulated to obtain developed flow and thermal profiles.

Table 3. Properties of PEX and 24% wt. ethylene glycol solution.

Property	PEX	Ethylene glycol solution
Density (kg/m^3)	940	1049.56
Specific heat (J/kg/K)	2302.3	3852.09
Thermal conductivity (W/m/K)	0.46	0.258
Dynamic viscosity (kg/m/s)	-	0.00387

Table 4. The hydraulic diameters, cross-sectional areas, and mass flow rates of the working medium for assumed pipes geometries.

Turbulator geometry	Hydraulic diameter (m·10^3)	Cross-sectional area (m^2·10^4)	Mass flow rate (kg/s)		
			Reynolds number		
			3000	5500	8000
Smooth pipe	26.20	5.350	0.233	0.427	0.622
Rectangular height 0.6 mm	22.09	5.179	0.265	0.487	0.708
Rectangular height 1.0 mm	19.91	5.048	0.287	0.527	0.767
Rectangular height 1.4 mm	16.63	4.929	0.336	0.616	0.896
Rectangular height 1.8 mm	14.41	4.640	0.365	0.669	0.974
Triangular height 0.6 mm	22.97	5.358	0.264	0.484	0.705
Triangular height 1.0 mm	21.08	5.299	0.285	0.523	0.760
Triangular height 1.4 mm	19.35	5.212	0.306	0.559	0.814
Triangular height 1.8 mm	18.61	5.096	0.325	0.596	0.868

4.3. Material properties

The pipes were assumed to be made of PEX, while the working medium was 24% wt. aqueous solution of ethylene glycol. Their properties assumed in the simulations are given in Table 3.

4.4. Boundary conditions

In the first phase of the calculations, a periodic boundary condition at the inlet and outlet from the pipe with a given mass flow rate was applied (see Table 4, which contains mass flow rates for different Re, hydraulic diameters, and cross-sectional areas of pipes). In each case, the hydraulic diameter-based Re values were assumed to be the same and equal to 3000, 5500, and 8000. These values corresponded to smooth pipe glycol velocities of 0.335, 0.614, and 0.893 m/s, respectively. However, differences in the values of hydraulic diameters caused the mass flow to vary for each pipe with different vortex generator shapes, as shown in Table 4. In the case of variable helix pitch, mass flow rates were the same for each helix pitch and corresponded to the values for the respective pipes with a height of the turbulators equal to 1.4 mm. In the case of the variable height of the turbulator, mass flow rates varied, as shown in Table 4. In the second phase of the calculations, the fixed glycol temperature of 275.15 K was set at the pipe's inlet and 323.15 K at the external pipe surface. The rest of the surfaces were assumed to be adiabatic. The pipe outer wall temperature was significantly higher than the usual ground temperature, i.e., 281.15 K. However, this boundary condition allowed for obtaining a measurable working fluid temperature increase along the pipe section. The results were read from the pipe section of length between 0.8 and 0.9 m. Obtaining larger temperature increases in the pipe also eliminated the impact of possible numerical errors.

5. Results

5.1. Parametric simulations

In [12], the results of pressure drops and temperature increase in the pipe, heat flux density on the pipe wall, and average Nu were presented. However, to determine the effectiveness of the tested pipes, it is enough to look at the pressure drops and Nu values. In order to show the performances of the modeled pipes, these parameters are shown relative to the smooth pipe in Figures 3 and 4.

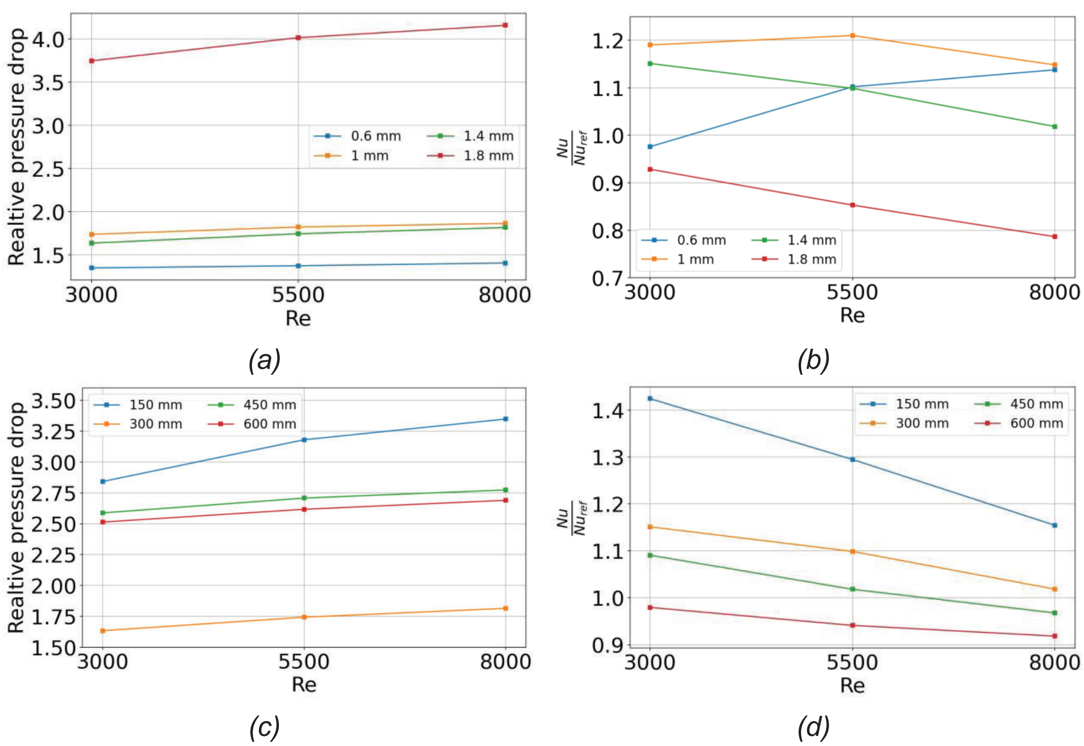

Figure 3. Parametrization analysis results for the model with rectangular turbulators: a) pressure drop depending on turbulator height, b) Nu depending on turbulator height, c) pressure drop depending on the pitch of the helix, d) Nu depending on the pitch of the helix.

In the first step, the pipes with rectangular turbulators were taken into account. Figure 3a shows that the size of the turbulators has a significant impact on the values of the obtained pressure drops. Generally, the larger

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the turbulator, the greater the pressure drop. It can be seen that for turbulators with a height of 1.8 mm, the pressure drop is very large and is over 250-300% of the value for a smooth pipe. Thus, heights in the range of 0.6-1.4 mm, which are characterized by much lower pressure drop increase than the height of 1.8 mm, have the potential to improve the performance of the rectangular turbulators. For a height of 0.6 mm, the increase of the pressure drop is about 35-40%, while for 1 mm and 1.4 mm, the pressure drop rises by 60-80%. Figure 3b also shows that very large turbulators do not improve convective heat transfer. Pipes with heights of 0.6 mm and 1.4 mm achieve very different results depending on the Re. Pipes with vortex generators of a height of 1 mm turned out to be the best, characterized by an increase of the Nu in the range of 15-21% in the entire range of Re. Considering the pitch of the helix, it can be seen in Figure 3c that a pitch of 150 mm gives by far the greatest pressure drop, up to 230% more than for the smooth pipe. The pipe with a helix pitch of 300 mm turned out to be the best, which had the smallest increase in the pressure drop in the range of 60-85%. However, the results of the intensity of convective heat transfer indicate that the model with the smallest pitch of the helix gives the best results. It records an increase in the Nu by 15.5-43% compared to the smooth pipe. The model with a pitch of 300 mm already gives an increase of only 2-15%. The other two pipes perform worse both in terms of pressure drop and intensity of convective heat transfer than the base case. Based on these results, an optimized pipe was proposed. Its turbulators height is assumed to be 1 mm, and the pitch of the helix of 150 mm is selected.

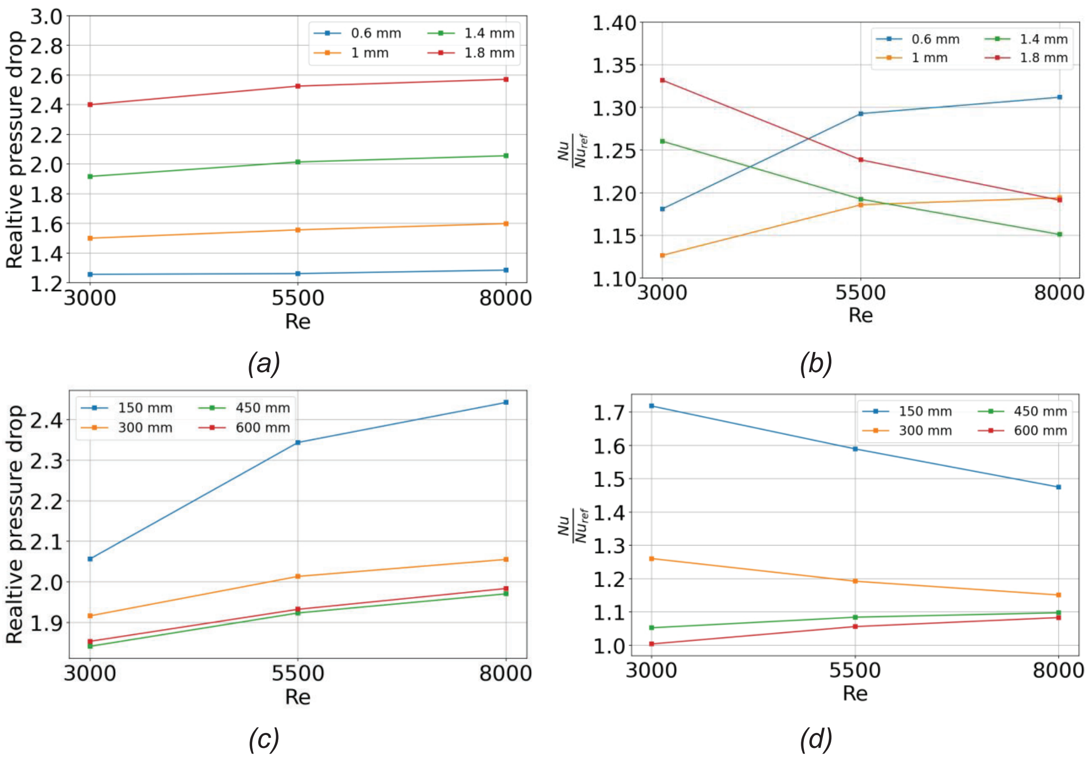

Figure 4. Parametrization analysis results for the model with triangular turbulators: a) pressure drop depending on turbulator height, b) Nu depending on turbulator height, c) pressure drop depending on the pitch of the helix, d) Nu depending on the pitch of the helix.

In the second step, the pipes with triangular turbulators were considered. Again, the increase in the size of the turbulators causes an increase in the pressure drop (Figure 4a), but this time the relationship is much more clear and linear. Of course, the best results are obtained for the pipe with 0.6 mm high turbulators, for which the increase is only 23%. For a height of 1 mm, it is already 50-60% depending on Re value. The thermal performance of all pipes is similar. The increase in the Nu is in the range of 12-33%. Larger turbulators (i.e., 1.4 and 1.8 mm) show better properties at lower Re numbers, while smaller turbulators (i.e., 0.6 and 1.0 mm) are more effective at higher Re. Considering the pitch of the helix, again, the pitch of 150 mm gives the greatest increase in pressure drop of 105-143% (Figure 4c) and the greatest increase in the Nu of 47-72% (Figure 4d). For other pipes, the increase in the pressure drop is significantly lower and amounts to 83-105%. But the increase in the Nu is also much smaller for these pipes and is in the range of 1-26%. Based on these results, the selected optimal pipe with triangular turbulators has a turbulators height of 0.6 mm and a helix pitch of 150 mm.

5.2. Optimized models

In this section, proposed optimized pipes are investigated. Information on computational meshes for optimized pipes is presented in Table 5, while mass flow rates for different Re, hydraulic diameters, and cross-sectional areas of these pipes are presented in Table 6.

Table 5. Parameters of meshes for optimized pipes.

Turbulator geometry	Number of elements	Max skewness	Max aspect ratio
Rectangular	24 056 100	0.86	277.58
Triangular	37 247 400	0.79	119.80

Table 6. The hydraulic diameters, cross-sectional areas, and mass flow rates of the working medium for optimized models.

Turbulator geometry	Hydraulic diameter ($m \cdot 10^3$)	Cross-sectional area ($m^2 \cdot 10^4$)	Mass flow rate (kg/s)		
			Reynolds number		
			3000	5500	8000
Rectangular	19.04	5.155	0.307	0.562	0.818
Triangular	22.97	5.358	0.264	0.484	0.705

The results obtained for the optimized pipes were compared with the base pipes calculated, among others, in [12]. Relative pressure drops and Nu were again compared in Figure 6.

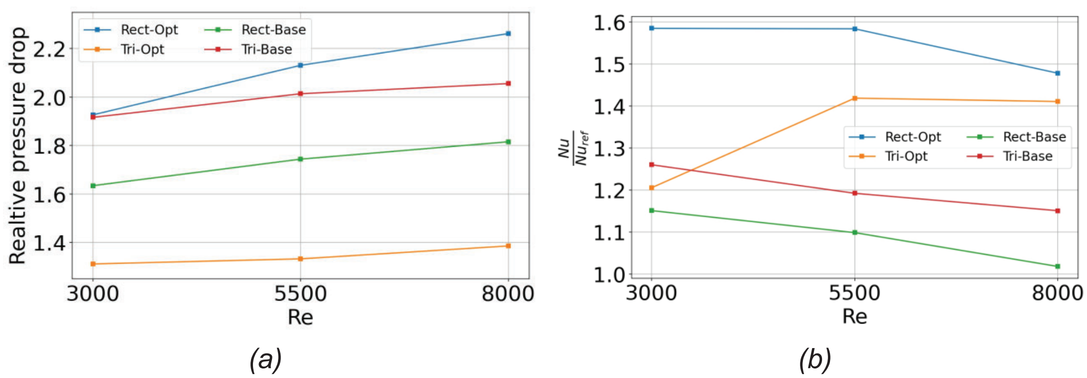

Figure 5. Results for optimized models: a) pressure drop and b) Nu for different values of Re (Rect-Base – base pipe with rectangular turbulators, Rect-Opt – optimized pipe with rectangular turbulators, Tri-Base – base pipe with rectangular triangular, Tri-Opt – optimized pipe with rectangular triangular).

The optimized pipes have much better convective heat transfer performances. The increase in the Nu for the optimized pipe with rectangular turbulators is 47-58%, while for the pipe with triangular turbulators is 21-42%. For non-optimized pipes (base geometries), it was 2-15% and 15-26%, respectively. The average increments of the Nu for optimized pipes relative to the smooth ones are several times greater in both cases. The situation is different in the case of pressure drops, and the optimized model with rectangular turbulators has higher pressure drops. Optimized triangular turbulators are much better in this regard. For the pipe with triangular turbulators, a lower pressure drop was recorded for the considered Re range than for the base pipe, i.e., the relative pressure drop varied in the 31-39% range. For the optimized pipes with rectangular turbulators, the increase in pressure drop is 92-126%.

It should be noted that in these simulations, the Re was in the same range for all cases. This resulted in significantly higher glycol mass flow rates in modified pipes than the smooth ones, as modified pipes had lower cross-sectional area and lower hydraulic dimeters than the smooth pipe. For optimized pipes, the mass flow rates were higher, c.a. 32 and 13% for rectangular and triangular vortex promoters, respectively. Applying the same mass flow rates of the working medium in the smooth and modified pipes will result in a much lower pressure drop increase than shown in this work, while Nu will still be significantly higher. This shows that the practical implementation of pipes with turbulence promoters in ground heat exchangers may be profitable.

6. Conclusions

This paper presents numerical optimization of the geometry of turbulence promoters in polymeric-made pipes in terms of heat transfer and fluid flow performances. Two rib shapes, i.e., rectangular and triangular, were

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selected based on the previous studies [12] as the most promising. Two geometrical parameters, i.e., vortex generators' height and pitch of the helix along which the ribs were extruded, were selected as variable parameters.

In the first step, a series of simulations were conducted for four ribs heights and four helix pitches. The results of these simulations allowed for finding the pressure drop and Nu characteristics for each pipe with internal surface modifications for Re in the range of 3000-8000. The results were analyzed, and optimal turbulators parameters were proposed based on that. For both rectangular and triangular vortex generators, their optimal height was selected to be 1 and 0.6 mm, respectively, while the pitch of the helix in both cases was chosen to be 150 mm.

In the second step, optimal turbulators' performances were compared to the initial (base) geometries. The optimized pipes have much better convective heat transfer performances. The increase in the Nu for the optimized pipe with rectangular vortex generators was 47-58% regarding the smooth pipe, while for the pipe with triangular modifications, it was 21-42%. These values, on average, are several times higher than for the initial turbulence promoters' geometries. However, for rectangular modifications, the pressure drops were higher than for base pipes and were in the range of 92-126%. Optimized triangular turbulators are much better in this regard. The triangular vortex promoters attained pressure drops in the 31-39% range, which was lower than for the base case.

In the performed analyses, the Re was in the same range for all simulated cases. This resulted in significantly higher glycol mass flow rates in modified pipes than the smooth ones. Applying the same mass flow rates of the working medium in the smooth and modified pipes will result in a much lower pressure drop increase than shown in this work for the fixed Re range, while Nu will still be significantly higher. This shows that the practical implementation of pipes with turbulence promoters in ground heat exchangers might be profitable.

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