Karadeniz Fen Bilimleri Dergisi, 12(1), 204-228, 2022. DOI: 10.31466/kfbd.1014483

KFBD

Karadeniz Fen Bilimleri Dergisi
The Black Sea Journal of Sciences
ISSN (Online): 2564-7377

Araştırma Makalesi / Research Article

Theorical Analysis of New Propeller Type Turbulator Design in Parallel Flow Double Tube Heat Exchanger

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Abstract

In this study, the usage purposes of turbulators are mentioned. The advantages and disadvantages of two different turbulator models are examined. In addition, the working principle of the turbulator was examined to create the analysis conditions. Parts used in turbulators and their alternatives are introduced. The necessary engineering calculations for the analysis were made, the flow analysis to be applied to the turbulator was made and the results were interpreted. Flow simulation, one of the simulation plug-ins containing the SolidWorks commercial program, was used in the analysis. Also turbulator analysis; it was made to determine fluid pressure, fluid velocity, fluid temperature, turbulent energy, turbulent viscosity. The heat exchanger is approximately 1000 mm long. For the turbulators, copper material was chosen as its thermal conductivity is better than aluminum. In the study conducted with two types of turbulators; numerical simulation results were calculated as 4,771 Mpa, 20,15 ^{0}C, 211,168 J/kg, 14,81 Pa.s for the spring turbulator, while these values were calculated as 7,811 Mpa, 20,32 ^{0}C, 165,518 J/kg, 18,62 Pa.s for the propeller turbulator. In addition, it has been observed that the fluid velocities are 61.5 m/s for the spring turbulator and about 75 m/s for the propeller turbulator. Due to the continuous increase in turbulent energy and turbulent viscosity over time, it has been recommended to use a finned turbulator instead of a spring tube. When all these results are examined, it will open new ideas and fields of study to further improve the overall thermal-aerodynamic performance of turbulators. In this study, a design that will be an alternative to turbulator designs has been realized.

Keywords: Turbulator, Turbulence, Heat Exchanger, Fluid Analysis.

Paralel Akışlı Çift Borulu Isı Eşanjöründe Yeni Pervaneli Türbülatör Tasarımının Teorik Analizi

Öz

Bu çalışmada türbülatörlerin kullanım amaçlarından bahsedilmiştir. İki farklı türbülatör modelinin avantaj ve dezavantajları incelenmiştir. Ayrıca analiz koşullarını oluşturmak için türbülatörün çalışma prensibi incelenmiştir. Türbülatörlerde kullanılan parçalar ve alternatifleri tanıtılmıştır. Analiz için gerekli mühendislik hesapları yapılmış, türbülatöre uygulanacak akış analizi yapılmış ve sonuçlar yorumlanmıştır. Analizde SolidWorks ticari programını içeren simülasyon eklentilerinden biri olan akış simülasyonu kullanılmıştır. Ayrıca türbülatör analizi; akışkan basıncını, akışkan hızını, akışkan sıcaklığını, türbülans enerjisini, türbülans viskozitesini belirlemek için yapılmıştır. Isı eşanjörü yaklaşık 1000 mm uzunluğundadır. Türbülatörler için, termal iletkenliği alüminyumdan daha iyi olduğu için bakır malzeme seçilmiştir. İki tip türbülatör ile yapılan çalışmada; sayısal simülasyon sonuçları yaylı türbülatör için 4,771 Mpa, 20,15 ^{0}C, 211,168 J/kg, 14,81 Pa.s iken, pervaneli türbülatör için bu değerler 7,811 Mpa, 20,32 ^{0}C, 165,518 J/kg, 18,62 Pa.s olarak hesaplanmıştır. Ayrıca akışkan hızlarının yaylı türbülatör için 61,5 m/s, pervaneli türbülatör için yaklaşık 75 m/s olduğu gözlemlenmiştir. Zaman içinde türbülans enerjisi ve türbülans viskozitesindeki sürekli artış nedeniyle, yaylı boru yerine kanatlı türbülatör kullanılması tavsiye edilmiştir. Tüm bu sonuçlar incelendiğinde, türbülatörlerin genel termal-aerodinamik performansını daha da iyileştirmek için yeni fikirler ve çalışma alanları açacaktır. Bu çalışmada türbülatör tasarımlarına alternatif olacak bir tasarım gerçekleştirilmiştir.

Anahtar Kelimeler: Türbülatör, Türbülans, Isı Eşanjörü, Akış Analizi

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1. Introduction

Today, due to the rapid consumption of energy sources, it is important to save on the use of depletable energy while searching for different energy sources and increasing the use of infinite energy sources. While the search for different energy sources continues, there are various methods of saving energy resources and energy efficiency used today. One of these methods is turbulators. In order to increase the formation of turbulence and improve heat transfer, the elements put in the pipe are turbulators. In addition, it has a very important role in saving the energy resources used today.

In one study, the nozzle was used as a turbulator and experiments were carried out. Experimental results show that the heat transfer efficiency increased by 270% (Promvonge et al., 2006).

Conical spring turbulators for heat transfer and pressure loss were investigated in the range of 10,000-34,000 Reynolds at 30, 45 and 60 degree angles. Based on the data, heat transfer, pressure loss and exergy analysis were performed. The best results are 30 degrees, 45 degrees and 60 degrees, respectively. The pressure loss increased with the increase of heat transfer. (Karakaya and Durmus 2013).

Guo investigated the two-phase flow characteristics, density change and pressure drop during heat transfer in the pipe in their experimental and theoretical study. It has been found that the pressure drop is directly proportional to the increase in thermal power and is higher in the long heat transfer line than in the short heat transfer line. The increase in pressure drop was at smaller inclination angle and it was observed that the steam quality was lower (Guo et al., 2014).

According to Yeşilyurt, in their experimental study, the effects of conical coiled elements consisting of different spring steps as an in-pipe heat transfer improvement element in a two-phase flow system with constant pressure at constant pressure, constant inlet and forced convection boiling in a horizontal straight pipe, temperature, constant thermal power and output limiting. conditions were examined. As a result, with the increase of the minimum point thermal power, the characteristic curve shifted to the right, but it was observed that the pressure drop increased in direct proportion to the thermal power at the given mass flow rate. The system with the highest and lowest pressure drop was determined for four different experimental studies carried out in the two-phase region (Yesilyurt 2015).

In another study, the effect of continuous helical geometry turbulent on flow and heat transfer in air-to-water double-pipe heat exchanger was experimentally examined. Experimental analysis was performed for different values such as open area ratio (0-0,0625), Reynolds number (6,000-12,000) and pitch ratio (1,83-5,83). According to experimental data, Nusselt number, friction factor and correlations for thermal performance are presented as functions of variable parameters. The results show that the friction factor and Nusselt number decreased with the increase in open space ratio and inclination (Ganji et al., 2016).

The two-phase (liquid and gas) flow distribution in the head of a plate heat exchanger was investigated experimentally. In this study, small mass fractions of the liquid phase are taken into account in the liquefaction process of natural gas. The liquid and gas flow rate was measured using optical methods such as PIV / PTV / LIF. The flow distribution of liquid and gas flows was quantified and discussed. The problem of installing a porous baffle or modifying the inlet nozzle configuration was investigated. The results showed that an impeller containing a diffuser, eddy-generating liquid and gas phases increased the distribution of liquid flow (Xin-Cheng Tu et al., 2018).

Steel examined the effects of turbulators on heat transfer numerically on hermetic water heaters with fluent program. First, examinations were carried out without the use of turbulators, after which four turbulators were added in the form of ellipses and accordingly, examinations were carried out. As a result of the analyses, although the yield was not high, increases were seen (Steel et al., 2019).

In a study focusing on turbulent flow in a heat exchanger with a new type of vane turbulator, Reynolds number values were between 5,000 and 20,000 and elevation ratios between 0.6 and ∞. The new type of blade turbulators have a square length. In addition, it is aimed to eliminate the awake eddies in the middle with the existing wing-shaped turbulators. (Chen et al., 2020).

In this study, the effect of helix angle and pitch ratio of helical surface disc turbulators on the thermal performance of the double tube heat exchanger is discussed. The tests were performed with air as the working fluid for different step rates, multiple helix angles, and various Reynolds numbers. Water and air were allowed to flow through the inner tube and the ring, respectively. In all cases, it is seen that the thermal strengthening factor is higher than the unit value. The smallest pitch rate and highest helix angle gave the highest thermo-hydraulic performance with 1.39 per Re = 3,500. Correlations for various parameters were developed (Yadav et al., 2020).

In a master's study, the effects of ring steps as in-pipe heat transfer improvement element in a forced convection boiling two-phase flow system in a horizontal straight pipe under constant pressure constant inlet temperature, constant thermal power and output restrictive conditions were investigated. As a result, with the increase of the minimum point thermal power, the characteristic curve shifted to the right, however, it was observed that the pressure drop increased in direct proportion to the thermal power at a given mass flow rate (Göcücü, 2020).

In another study, numerical simulation was performed on a double-pipe heat exchanger. Effects of conical and fusiform turbulators have been reported. 21 configurations are simulated, including conical and fusiform turbulator. Heat transport and turbulent flow models are shown. The maximum convective heat coefficient is obtained for the circular inner pipe. Optimum k configuration for 12 mm fusiform turbulators has been achieved (Xiong et al., 2021).

2. Materials and Methods

This section consists of two stages. In the first stage, general equations are given to calculate the heat transfer in the heat exchanger. In the second stage, the geometry of the turbulent to be used in the heat exchanger is explained.

The momentum differential equation of the mass in the boundary layer, continuity and energy conservation equations can be solved and the heat transmission generated in the heat exchanger can be calculated. Another name given to these equations is borderline equality. The boundary conditions of the current boundary layer equations should be adjusted according to the problem. Units that are resistant to analysis of conservation equations are units such as mass, energy and momentum. The temperature distribution of the solid area inside the heat exchanger can be solved by a three-dimensional heat conduction equation. The continuity, momentum and energy equations of uncompressed flow are given below (Baysal, 2009).

2.1. Energy Conservation

The three-dimensional energy equation for a cylindrical geometry can be written as follows.

$$\rho c_p \left(\frac{\partial T}{\partial t} + u_r \frac{\partial T}{\partial r} + \frac{u_\theta}{r}\frac{\partial T}{\partial \theta} + u_z \frac{\partial T}{\partial z}\right) = k\left(\frac{1}{r}\frac{\partial}{\partial_r}\left(r\frac{\partial T}{\partial r}\right)\frac{1}{r^2}\frac{\partial^2 T}{\partial \theta^2} + \frac{\partial^2 T}{\partial z^2}\right) + \mu q \qquad (1)$$

2.2. Continuity Equation

The other name of this equation can also be called the preservation of mass. In the region where there is a three-dimensional fluid, the equality that will occur if the law of mass protection is referenced according to the control volume element is continuity equality. The speed components of the fluid r, θ, z can be expressed in u_r, u_θ, u_z format and as given in Equation 3.2. (Cengel and et al., 2008).

$$\frac{1}{r}\frac{\partial (r u_r)}{\partial r} + \frac{1}{r}\frac{\partial (u_\theta)}{\partial \theta} + \frac{\partial (u_z)}{\partial z} \qquad (2)$$

2.3. General Description of the Geometric Model

In heat exchangers with turbulators, heat transfer is increased by increasing the heat transfer coefficient instead of increasing the surface area. The thickness of the thermal boundary layer that appears between the inner surface of the pipe and the fluid is important for the heat transfer coefficient. Since the layer thickness in laminary flow is thicker than in turbulent flow, heat transfer efficiency is higher in turbulent flow. Turbulent emulators in turbulent flow are used to further reduce the thickness of the laminary boundary layer near the thin surface. Using turbulators, the flow lines that appear in the pipeline will be deformed and can be improved by passive methods. Using turbulent, fluid can be affected, such as the breakage of the boundary layer of the fluid, and the fact that the fluid is a physical object brings additional turbulence to the environment in which the fluid is located, and secondary flow occurs in the environment. The liquid flows and the difference between the wing slope and the wing angle of the turbulent and the expansion of the liquid flow distance are ensured.

In the analyses to be carried out, comparisons of two different types of geometry turbulators were made. In the analysis to be carried out, the material of the wall is copper in order to increase the heat transfer of the wall. The inner diameter of the wall is 60 mm and the outer diameter is 62 mm. The thickness of the meat is 1 mm. The total length of the heat exchanger is 1000 mm. Since the internal diameter of the heat exchanger is 30 mm, the material of turbulators for different steps that can be positioned in this heat exchanger is also preferred as copper. This copper turbulator is placed inside the heat exchanger for each analysis stage.

The outer diameter of the spring turbulent is 54 mm and the inner diameter is 45 mm. The length of the turbulent is selected as 900 mm. Meat thickness is 1 mm. The spring turbulent is drawn in the Solidworks commercial drawing program (Figure 1).

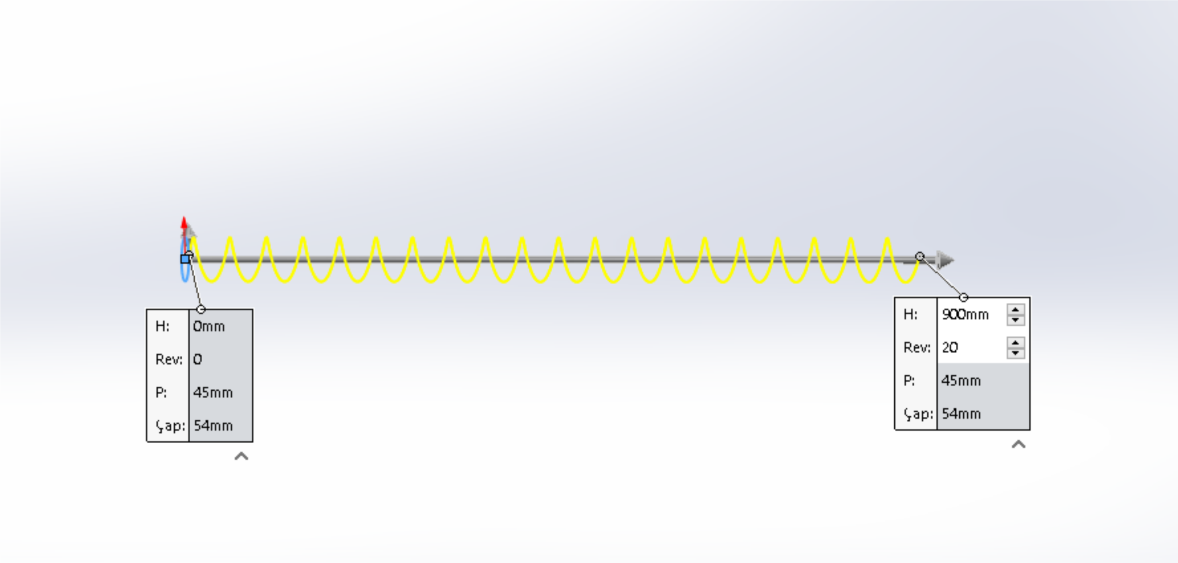

Figure 1. Mathematical model of the turbulent to be used in the analysis

Physical properties of the first turbulent heat exchanger to be created in Table 1 is provided.

Table 1. Heat exchanger physical properties

Parameter	Dimension
Length of heat exchanger (L)	1000 mm
Pipe inner diameter (d)	60 mm
Pipe outer diameter (D)	62 mm
Pipe meat thickness (t)	1 mm
Turbulator length (L)	900 mm
Turbulator spring outer diameter(D)	54 mm
Turbulator spring inner diameter (d)	45 mm
Turbulator meat thickness (s)	1 mm
Arc revolutions (n)	20

The model of the spring turbulator to be used in the analysis is given in Figure 2.

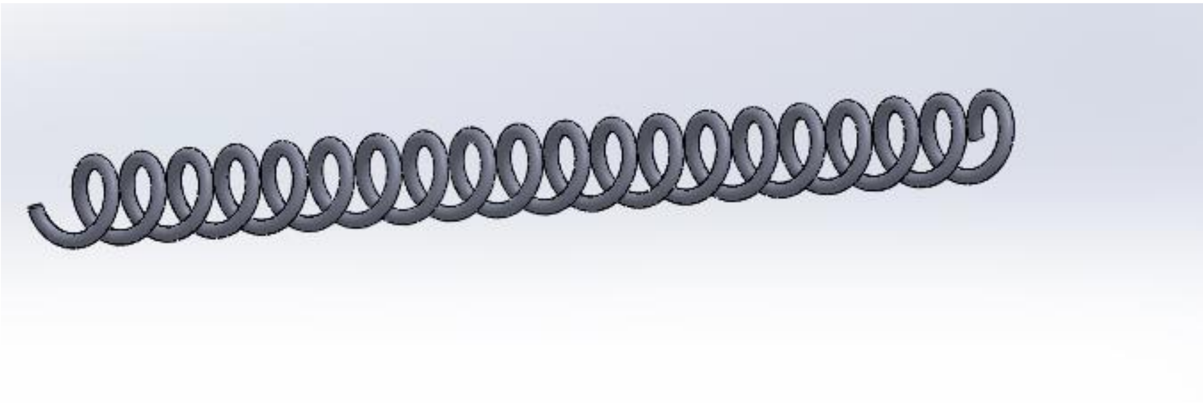

Figure 2. Spring turbulator to be used in analysis

The physical properties of the other turbulent to be made are given in Table 2. In this turbulator, the general mode is manually drawn on the Solidworks commercial program. Since the propeller type turbulator drawing is considered, a normal pipe is drawn primarily to the drawing. This will be the main part of the turbulent. The total length of this pipe is 900 mm. The diameter of the pipe was selected as and a solid model was created. The following illustrated solid model created in Figure 3 is shown.

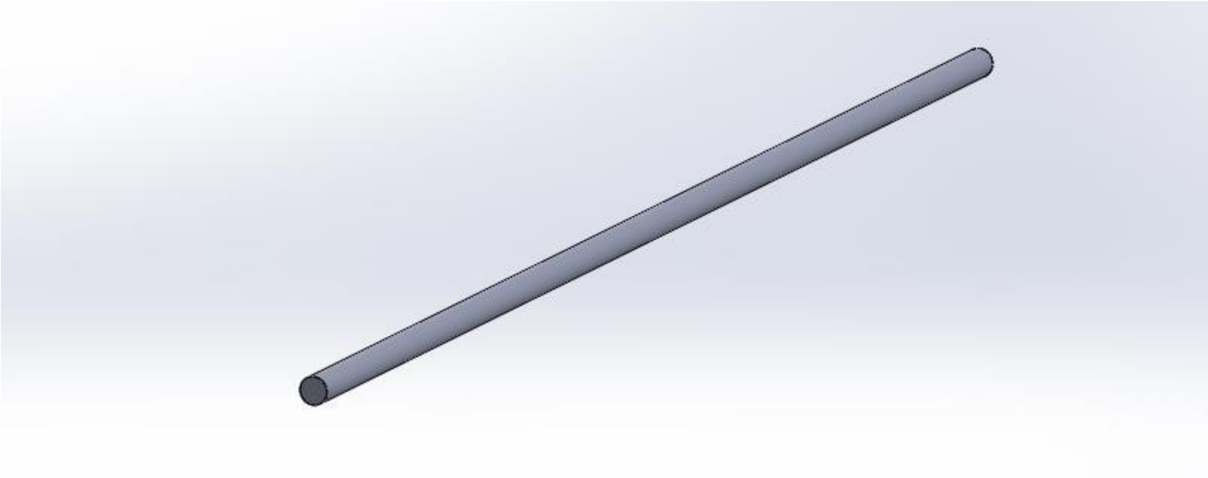

Figure 3. Pipe part of the turbulator

After the pipe part is drawn, the propeller model is drawn. In the propeller model, the inner pipe diameter is 8 mm and the outer pipe diameter is 16 mm. The meat thickness of the main part of the propeller is preferred as 5 mm. The total length of length was approximately 64 mm. The angle between the two propellers is 60 degrees. A total of 6 protrusions have been created. The solid model of the propeller is shown in Figure 4.

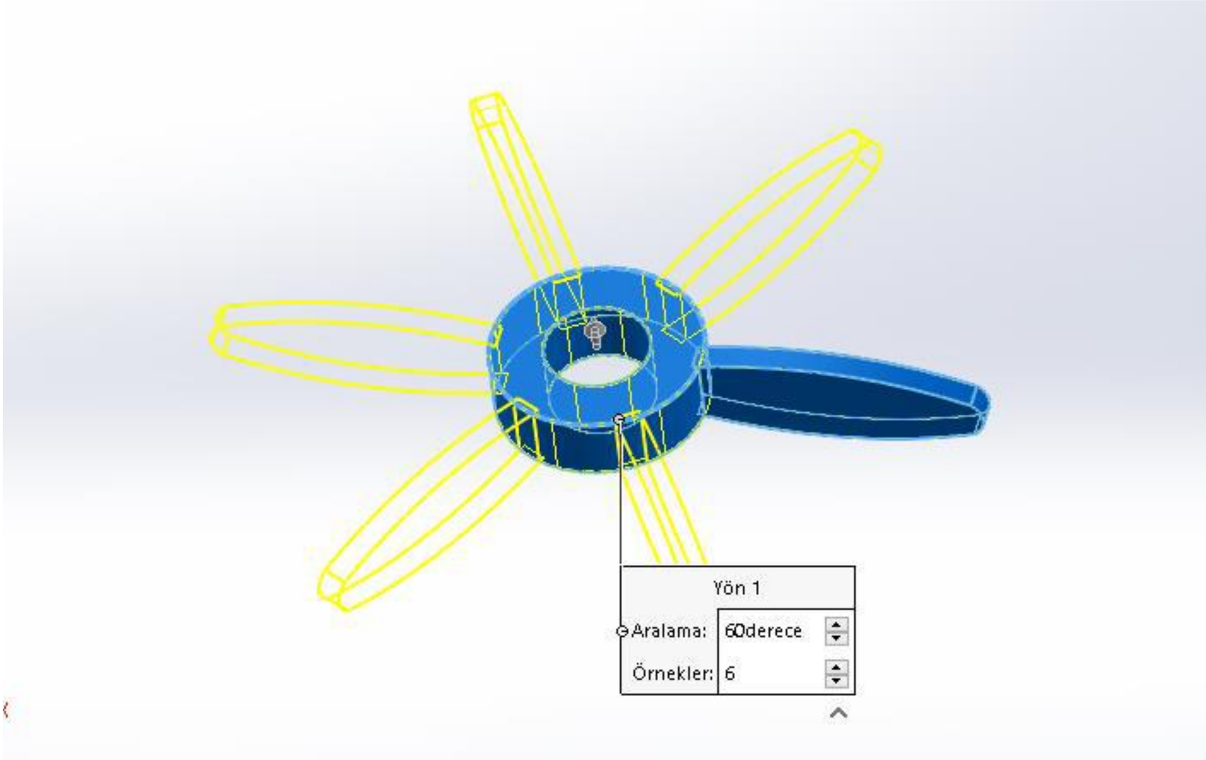

Figure 4. Propeller solid model

As the last part, the installation of the pipe and propeller was carried out. These transactions were carried out in solidworks commercial program. The solid model of the second type of turbulent is shown in Figure 5.

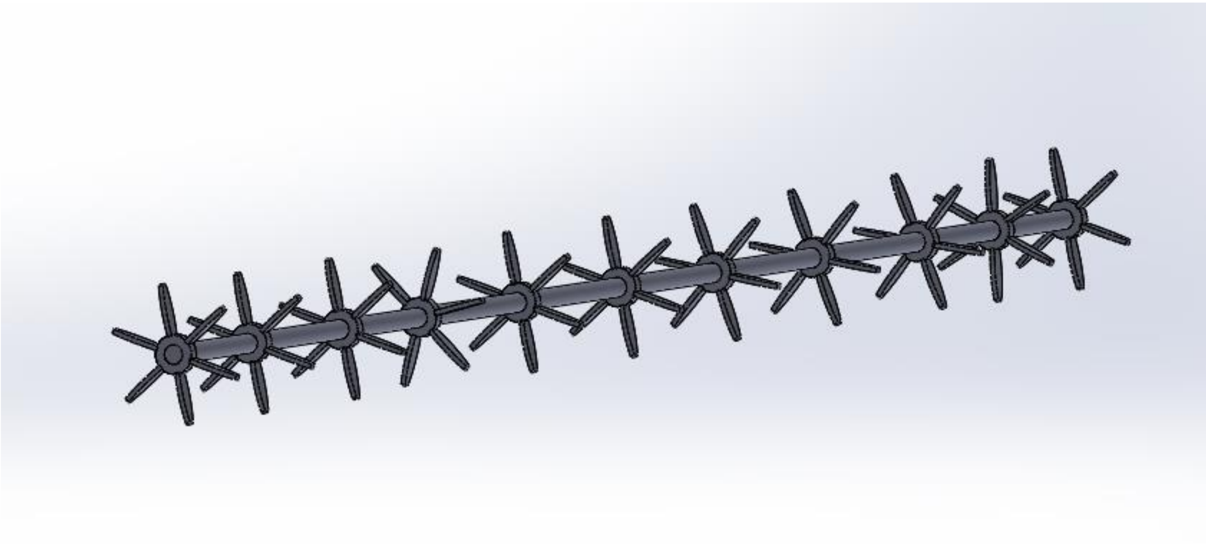

Figure 5. Solid model of propeller type turbulator

The physical properties of the propeller type turbulator are shown in Table 2.

Table 2. Physical parameters of the propeller type turbulator

Parameter	Dimension
Heat exchanger length	1000 mm
Turbulator length	900 mm
Pipe inner diameter	8 mm
Pipe outer diameter	16 mm
Total length of propeller	64 mm
Angle between the propeller	60 degrees
Pipe meat thickness	5 mm
Propeller meat thickness	2 mm

In the next section, comparisons of these two turbulators will be made. In this turbulator, as in the spring turbulator, copper pipe in the heat exchanger and the material to be used in the turbulent will be preferred as steel.

3. Findings and Discussion

In this section, 3D models of two turbulators are made in solidworks from certain parameters. The models were then examined and analyzed in the flow simulation. Water was used as a fluid and analyzed in a specific speed data. The 3D model of turbulators is given in Figure 6 and Figure 7.

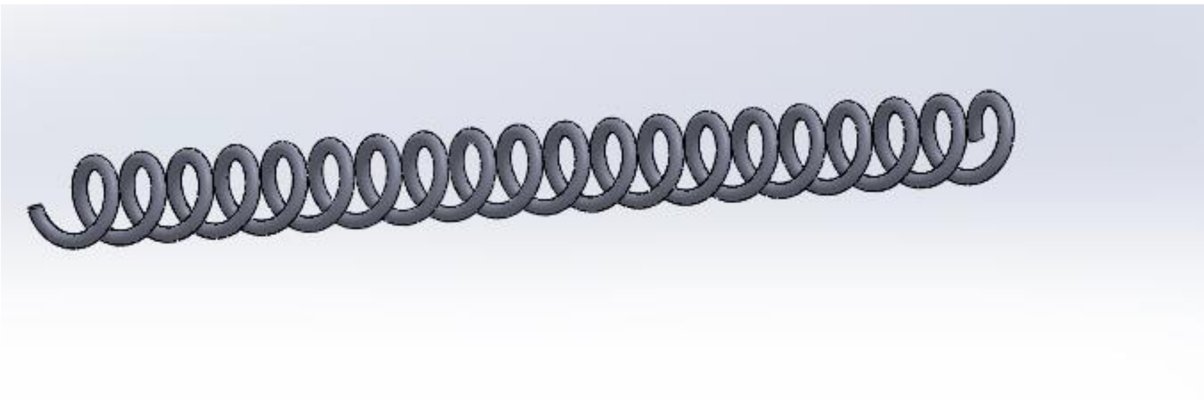

Figure 6. 3D model of the classic spring turbulator to be used in analysis

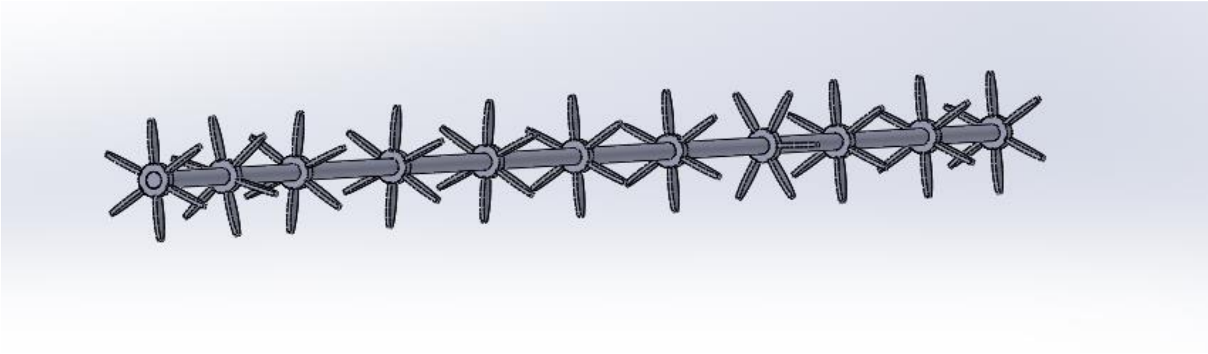

Figure 7. 3D model of propeller type turbulator to be used in analysis

Comparisons of these two types of turbulent, pressure, temperature, turbulence length, turbulence energy, fluid speed, etc. will be compared individually. This counter-heating will be done through the flow-simulation plug-in tool located in solid works commercial program.

First, a speed was determined for flow analysis and two turbulators were compared at the same speeds. Before the analysis of the turbulators, the installation process was carried out on 1000 mm pipe, which is considered as a heat exchanger. The border conditions are given in Table 3.

Table 3. Boundary requirements of turbulators

Turbulator material	copper
Heat production rate	20
Heat transfer coefficient	25 W/m^2K
Number of Reynolds	10,000
Starting solid temperature	20 ^oC
External liquid temperature	25 ^oC

After the boundary requirements and acceptances are made, a speed is determined to perform flow analysis. An input flow rate of 25 m/s was determined and pressure and other parameters were found as output parameters for the heat exchanger. First, the velocity analysis in the heat exchanger is compared (Figure 8 and Figure 9).

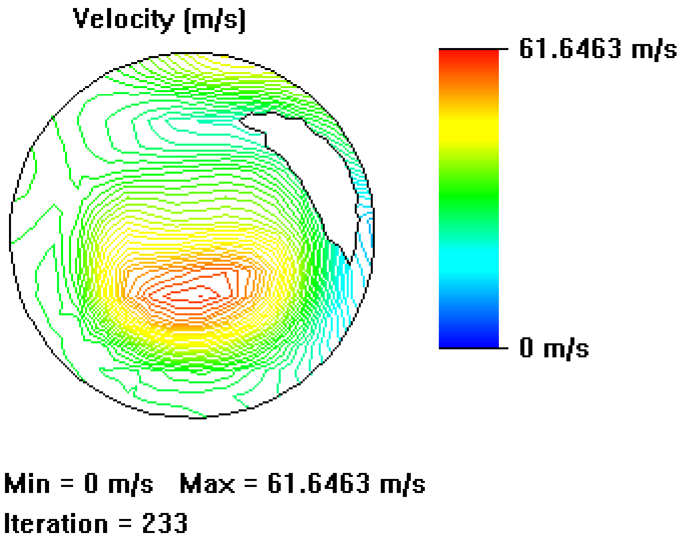

Figure 8. Spring turbulator speed change

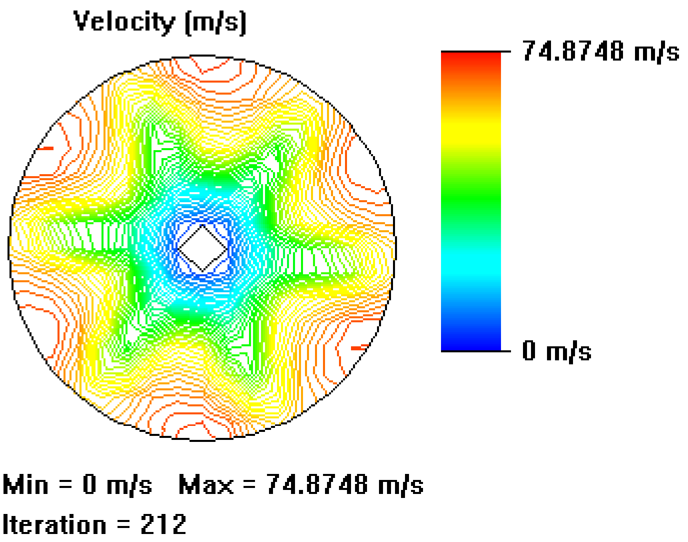

Figure 9. Propeller turbulator speed change

When we compared the two turbulators in the speed analysis, it was observed that the spring turbulent reached a maximum speed of approximately 61,5 m/s. In the propeller turbulator, the speed reached a speed of approximately 75 m/s. The maximum speed of these two turbulators varies from the different geometries. Due to the geometry of the spring turbulator, the speed occurs in the middle of the maximum, while the maximum speed in the propeller turbulent occurs in the edges. In the propeller turbulator, it was observed that the speed was also high in the intermediate parts of the fins. Graphical representation of the velocity changes of the spring and propeller turbulators over time are given in Figure 10 and Figure 11. The maximum velocity variation was greater in the propeller turbulator.

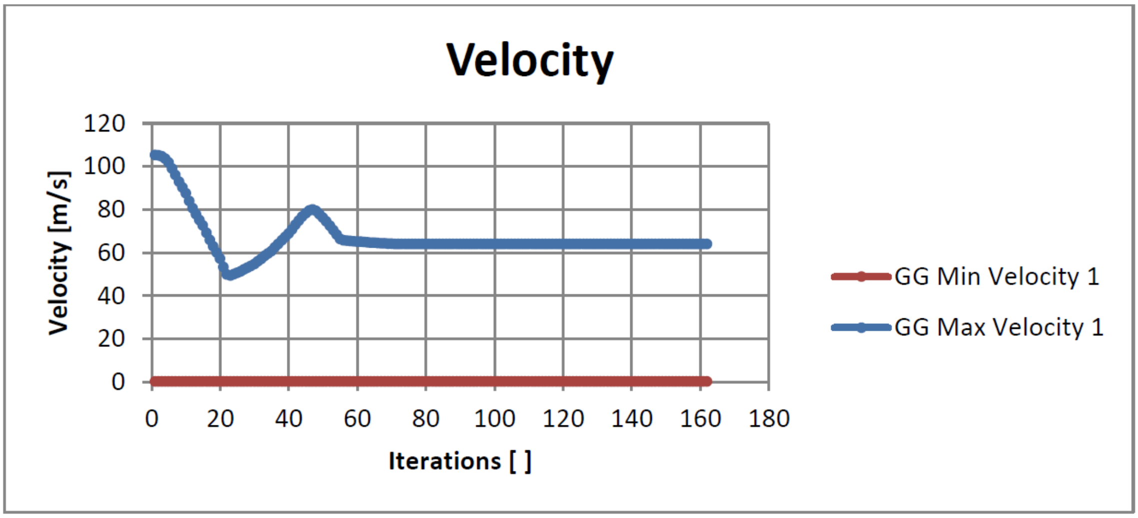

Figure 10. Graphical representation of the speed change of the spring turbulator over time

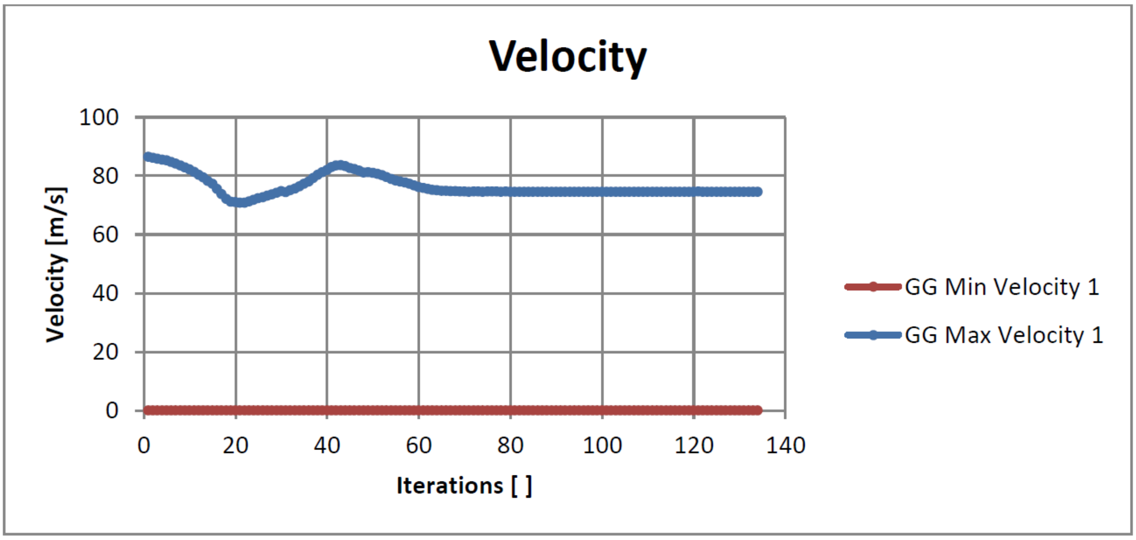

Figure 11. Graphical representation of the speed change of the propeller turbulator over time

After the speed analysis is realized, the analysis of the pressure change is given in Figure 12 and Figure 13.

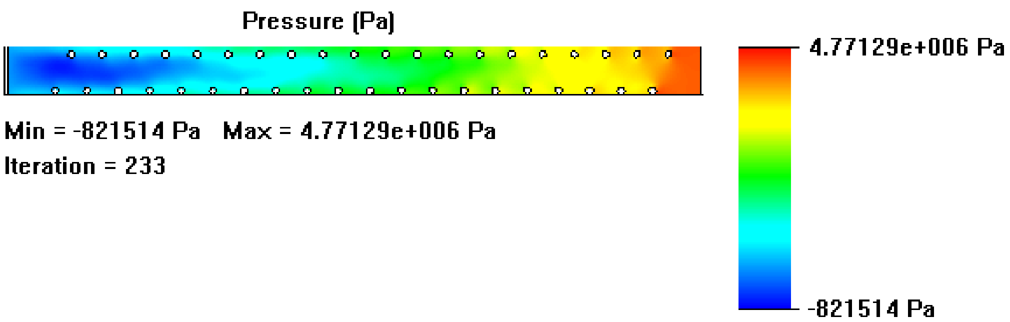

Figure 12. Spring turbulator pressure change

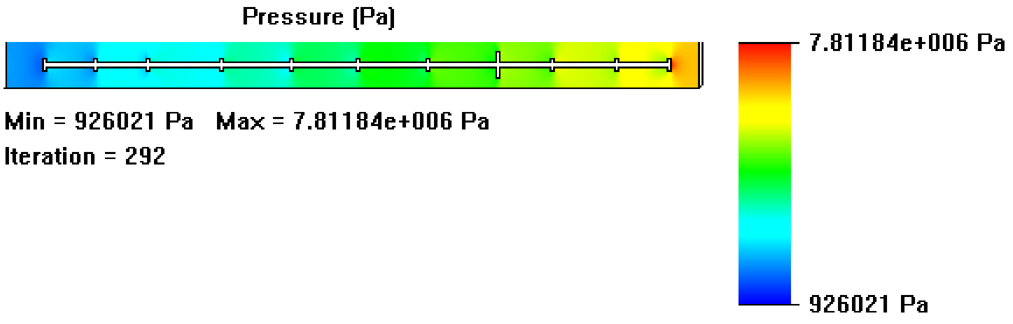

Figure 13. Propeller turbulator pressure change

If the two turbulators are compared; The pressure of the propeller turbulator is approximately 7,8 MPa. The pressure change of the spring turbulator is approximately 4,8 MPa. Similarities occurred in the parts of the two turbulators where the pressure was the most and the least. Pressure change of propeller and of spring turbulators over time are given in Figure 14 and Figure 15. Pressure variation remained the same in spring and propeller types.

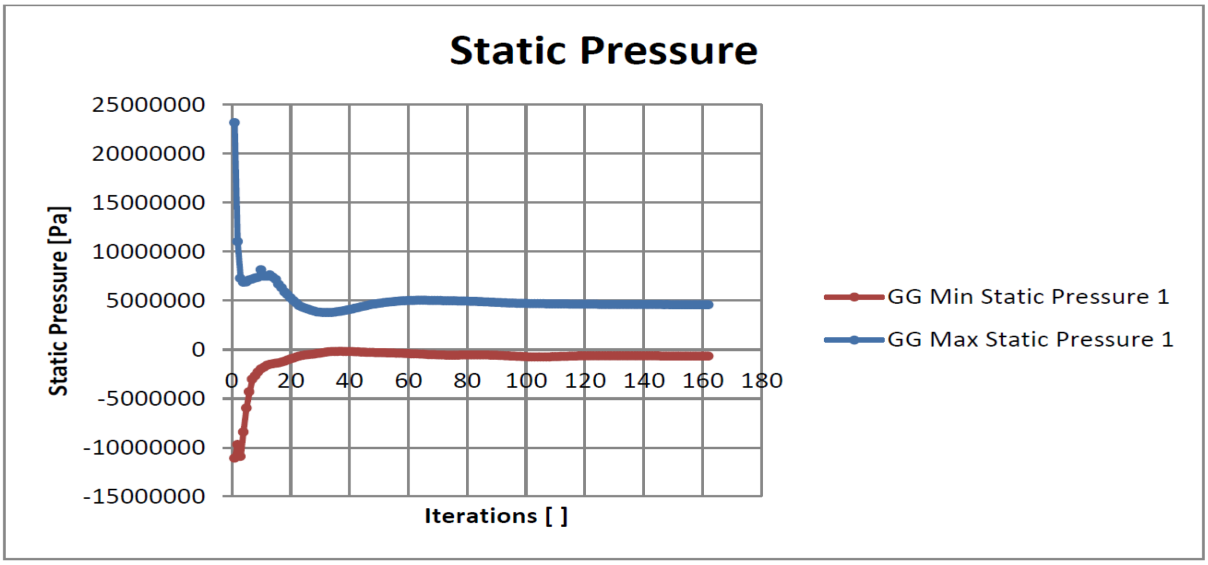

Figure 14. Spring turbulator pressure change over time

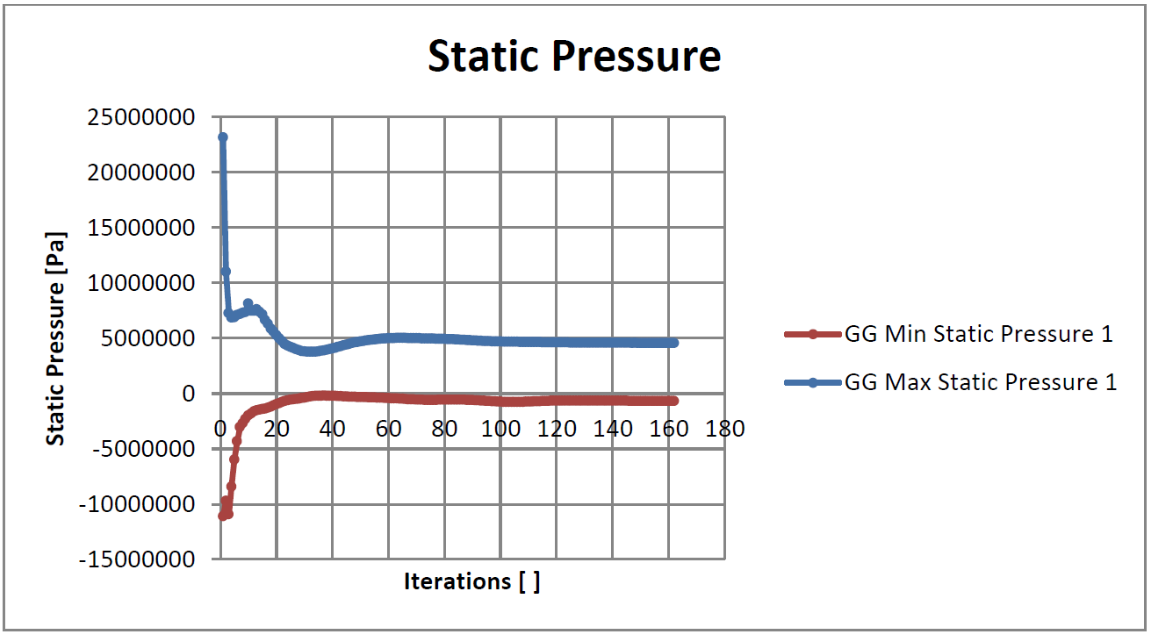

Figure 15. Pressure change of propeller turbulator over time

After pressure analysis, temperature change analyses are given (Figure 16 and Figure 17).

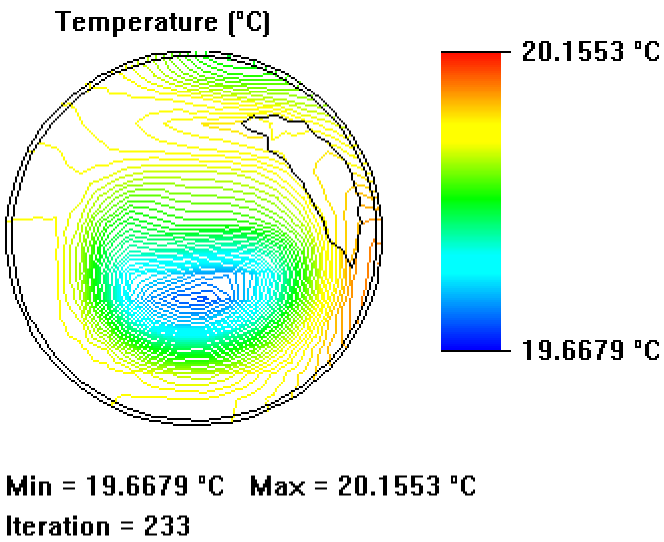

Figure 16. Spring turbulator temperature change analysis

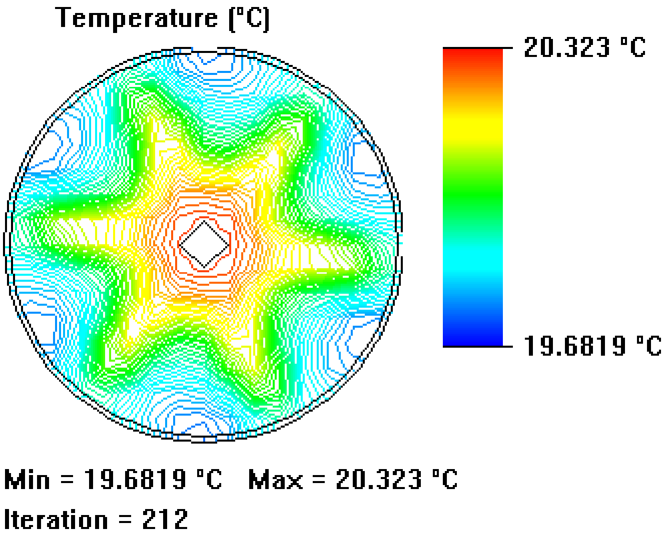

Figure 17. Propeller turbulator temperature change analysis

Temperature values were observed to be close to each other in the two turbulators. However, changes have been observed where the temperature is maximum. In the propeller turbulator, the temperature was seen to be the highest in the middle part and the temperature decreased as we went to the outer edges. In spring turbulator, the opposite is determined. Temperature change of propeller and of spring turbulators over time are given in Figure 18 and Figure 19. The temperature variation is greater in the propeller turbulator.

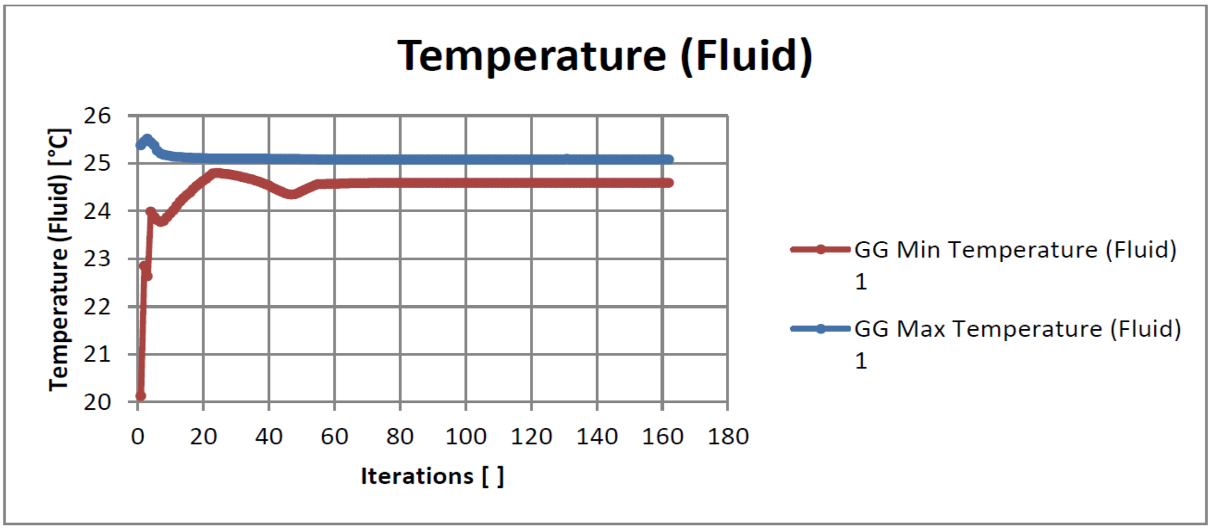

Figure 18. Spring turbulator temperature change over time

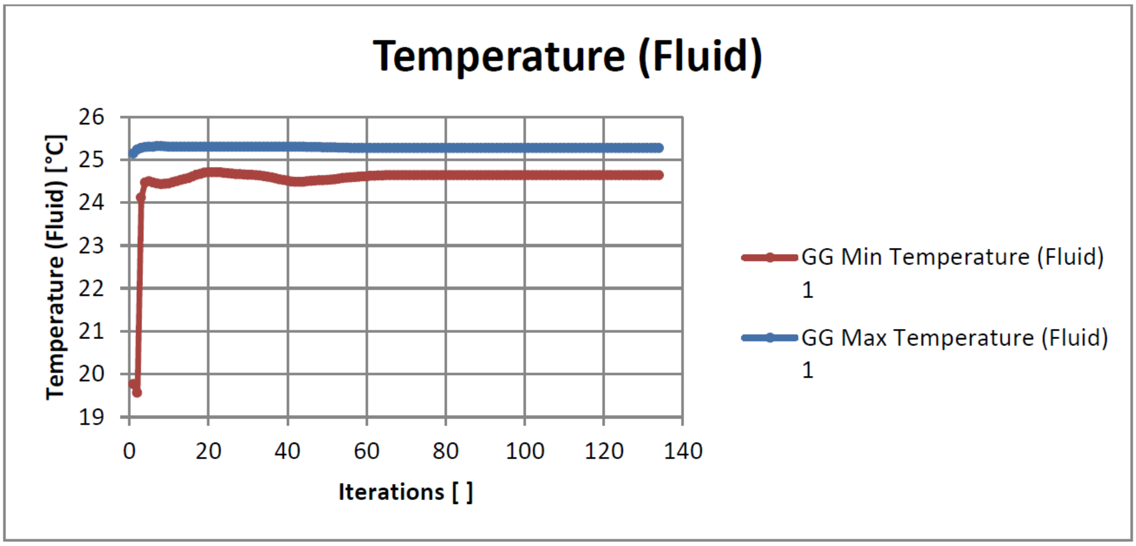

Figure 19. Propeller turbulator temperature change over time

After the temperature, turbulator energy change was examined (Figure 20 and Figure 21).

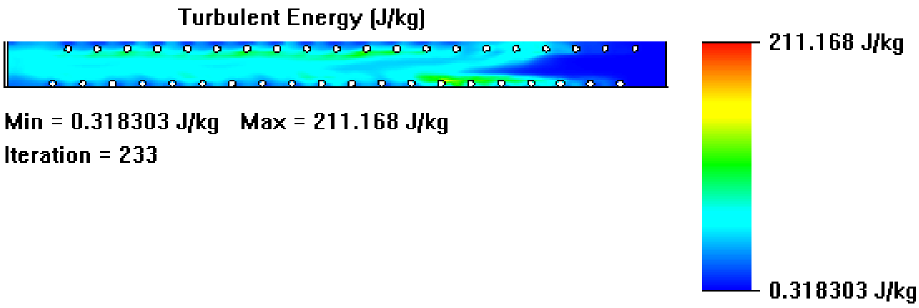

Figure 20. Spring turbulator turbulence energy change

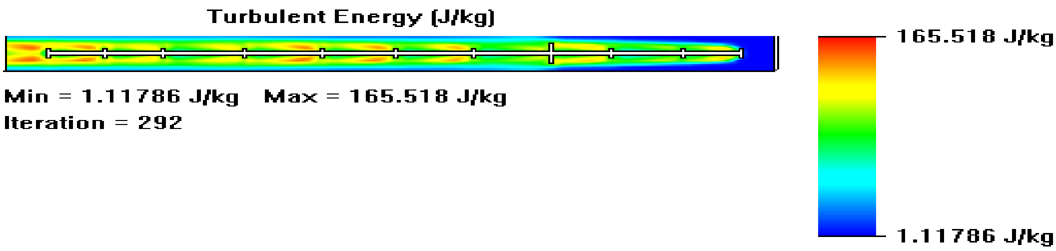

Figure 21. Propeller turbulator turbulence energy exchange

Turbulence in the spring turbulator in energy has been observed to have a higher energy density. It was observed that the spring turbulent was approximately 211 J/kg. In propeller turbulator, it is approximately 165,5 J/kg. Turbulence energy change of propeller and of spring turbulators are given in Figure 22 and Figure 23. Turbulent energy change is higher in propeller turbulator.

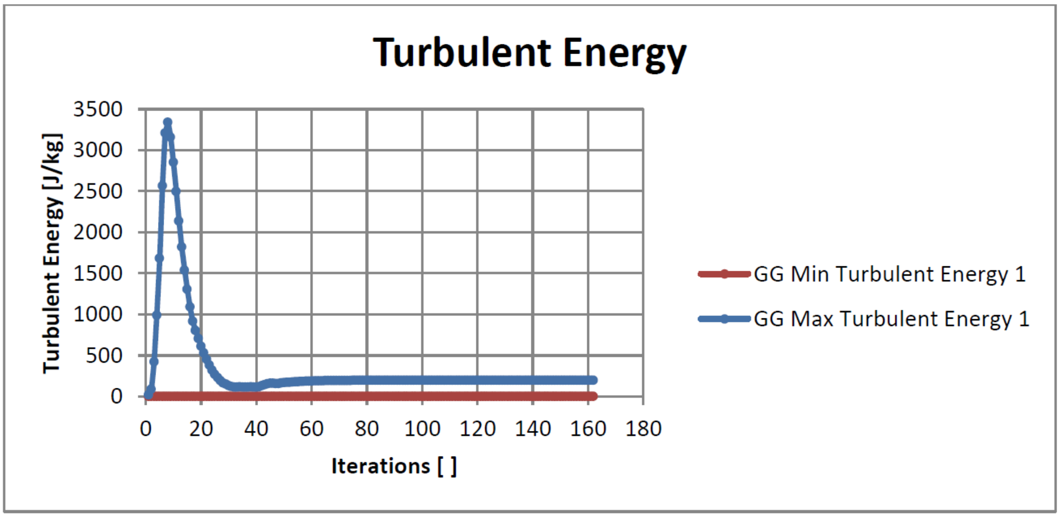

Figure 22. Spring turbulator turbulence energy change

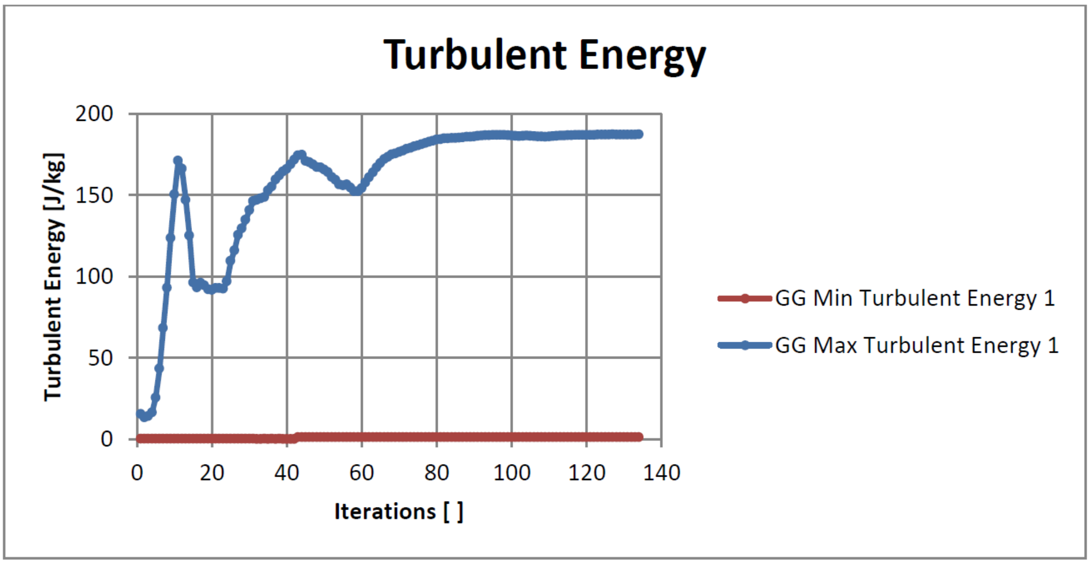

Figure 23. Propeller turbulator turbulence energy exchange

Turbulent viscosity changes were compared in Figure 24 and Figure 25.

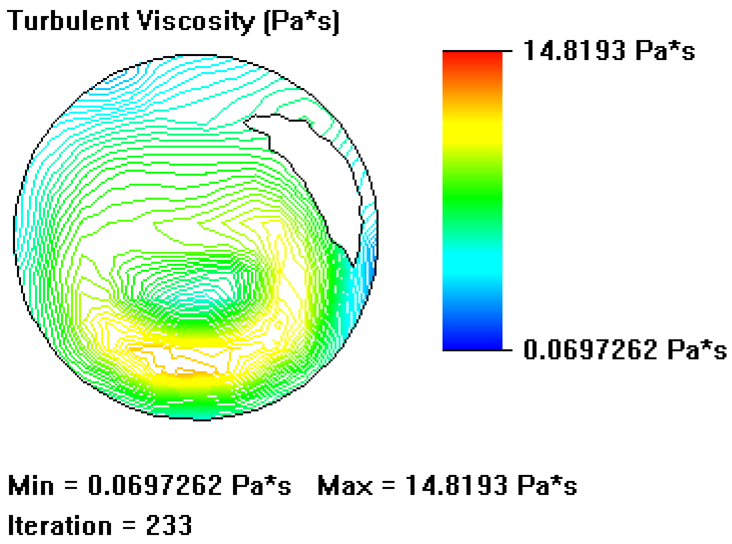

Figure 24. Spring turbulator turbulence viscosity change analysis

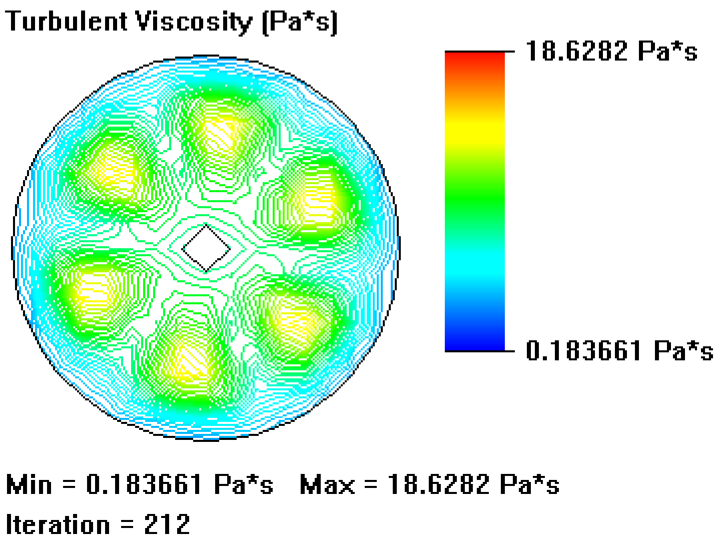

Figure 25. Propeller turbulator turbulence viscosity change analysis

When the turbulence viscosity exchange comparisons of these two types of turbulator were realized, it was understood that the turbulence viscosity value of the propeller turbulator was higher. The turbulence viscosity value of the propeller turbulator was approximately 18,7 Pa.s. The spring turbulator was found to be approximately 14,9 Pa.s. When the places where turbulence viscosity is most common, the publication occurs in the middle due to its own geometry. The maximum viscosity value was detected in the fluid passing through the fins located in the propeller parts of the propeller turbulator. Turbulence viscosity change of propeller and of spring turbulators are given in Figure 26 and Figure 27. The turbulence viscosity variation was larger in the propeller turbulator.

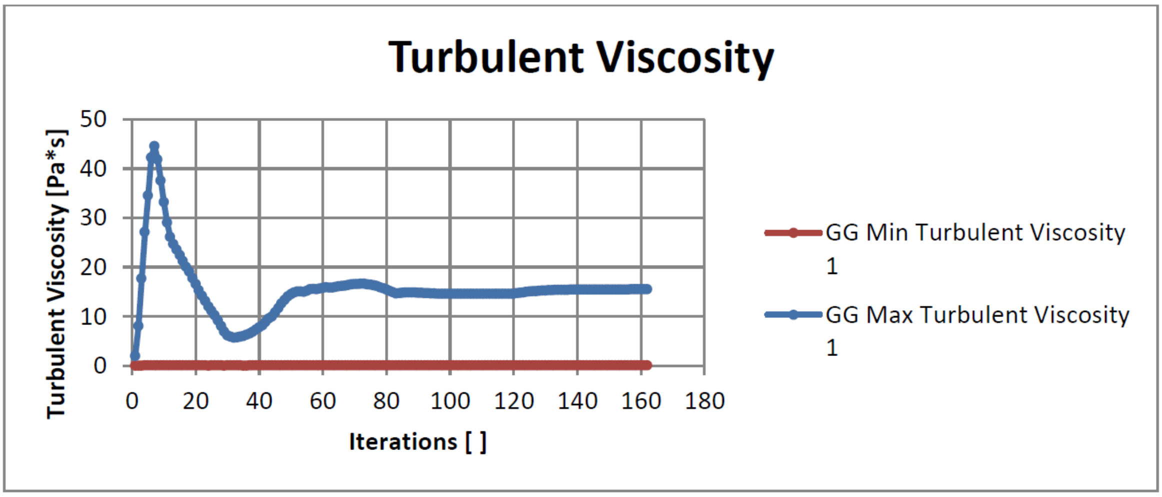

Figure 26. Spring turbulator turbulence viscosity change graph

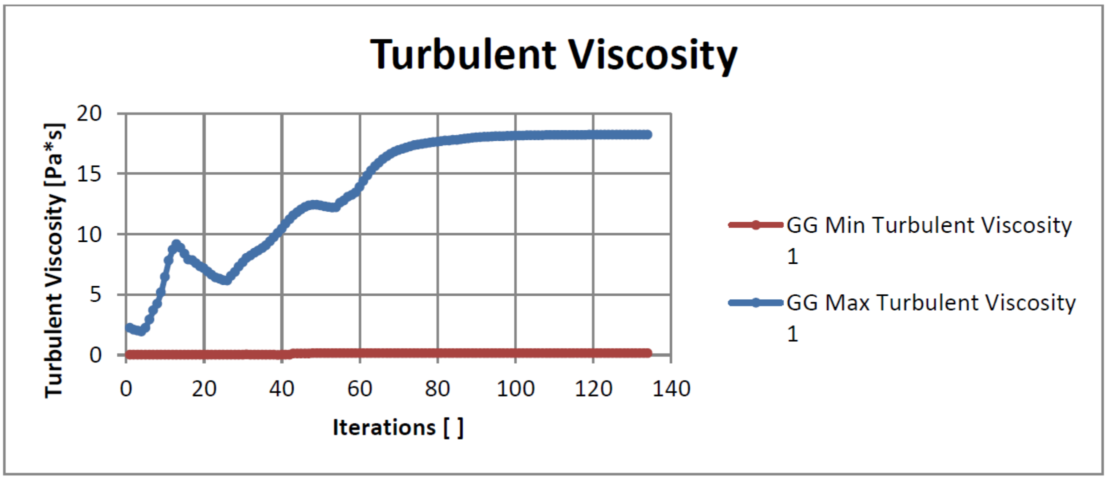

Figure 27. Propeller turbulator turbulence viscosity change graph

After viscosity change analysis, turbulent length variation analysis was performed (Figure 28 and Figure 29).

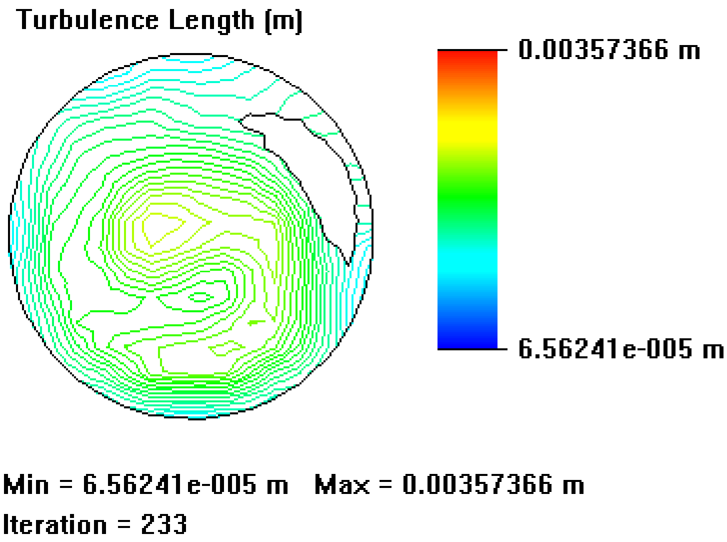

Figure 28. Spring turbulator turbulence length change analysis

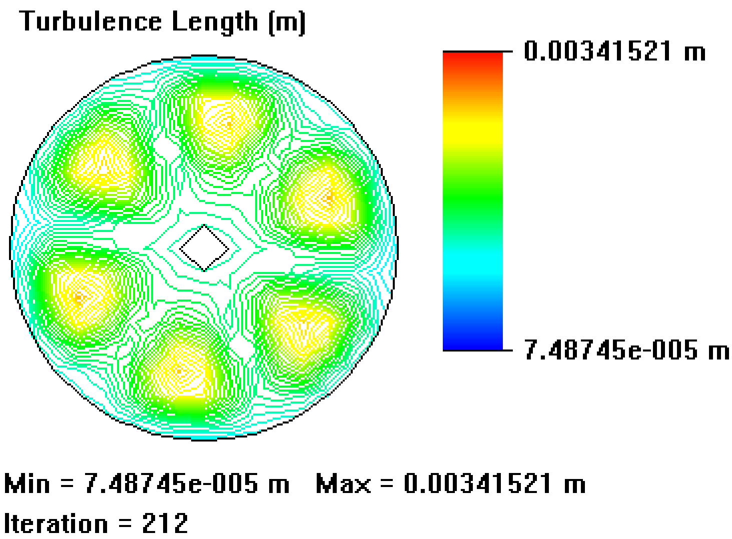

Figure 29. Propeller turbulator turbulence length change analysis

Turbulence length changes compared; The propeller turbulator was analyzed as a maximum of 0,00341521 m. In the spring turbulator, the maximum length was 0,0037366 m. As in turbulence viscosity change analysis, turbulence length change analysis showed the maximum lengths between fins in the propeller turbulator and occurred in the middle part of the spring in the form of half a moon due to its geometry. Turbulence length change of propeller and of spring turbulents are given in Figure 30 and Figure 31.

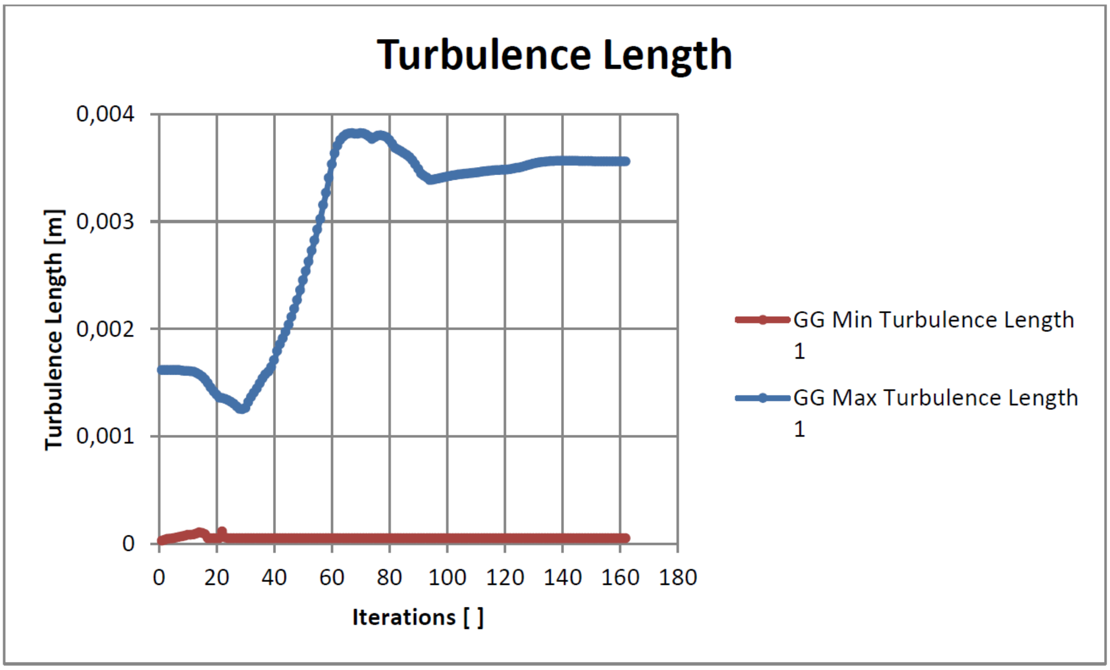

Figure 30. Spring turbulator turbulence length change graph

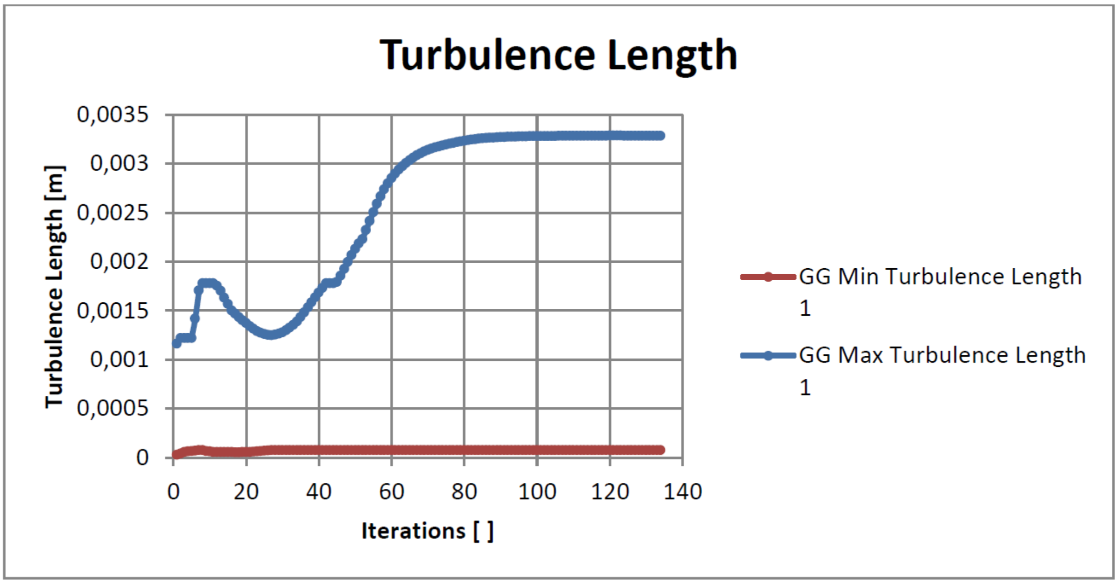

Figure 31. Propeller turbulator turbulence length change graph

Finally, turbulence time was analyzed (Figure 32 and Figure 33).

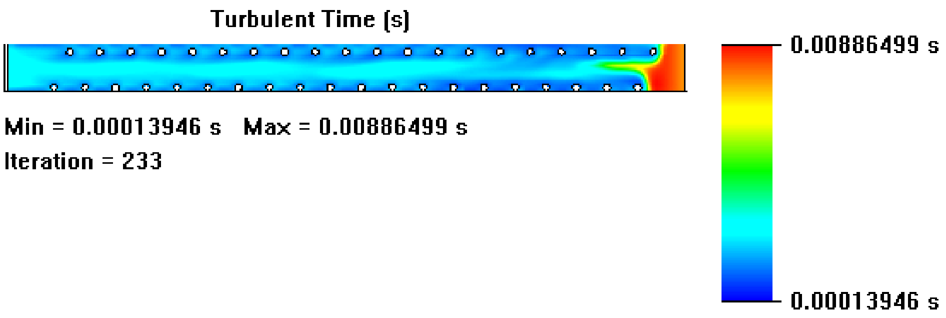

Figure 32. Spring turbulator turbulence time analysis

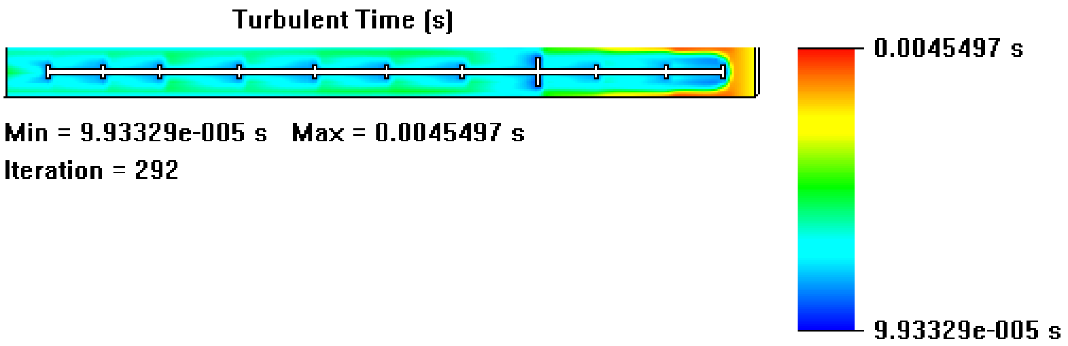

Figure 33. Propeller turbulator turbulence time analysis

When turbulent times are compared, it is understood that the spring turbulator has longer turbulence in terms of duration. The longest turbulence time in the spring turbulator was 0,00886499 s. In the propeller turbulator, the longest turbulence time was analyzed as 0,0045497 s. Turbulence time of propeller and of spring turbulators are given in Figure 34 and Figure 35.

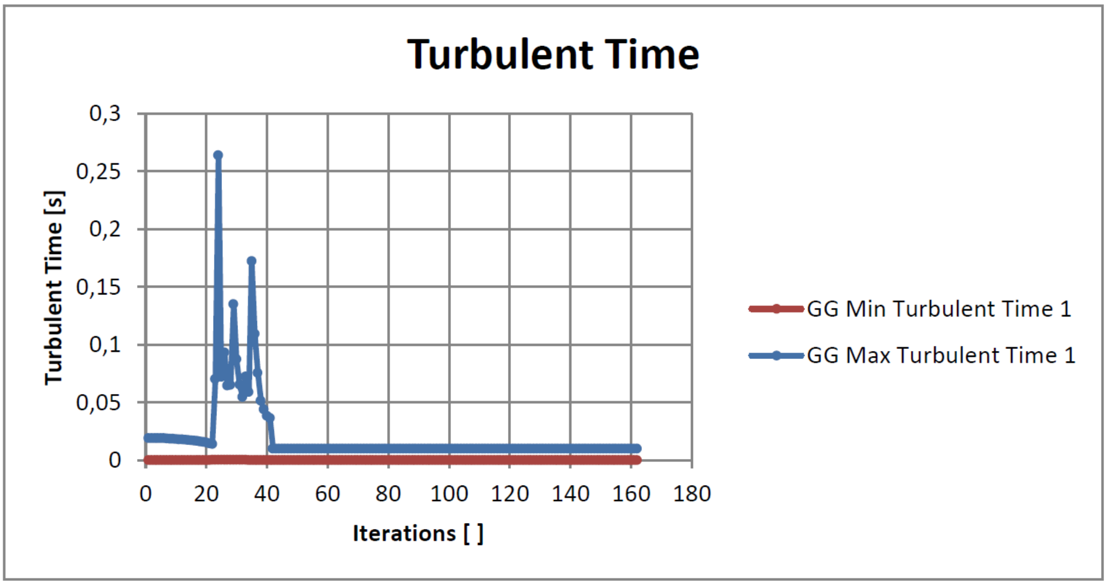

Figure 34. Spring turbulator turbulence timeline

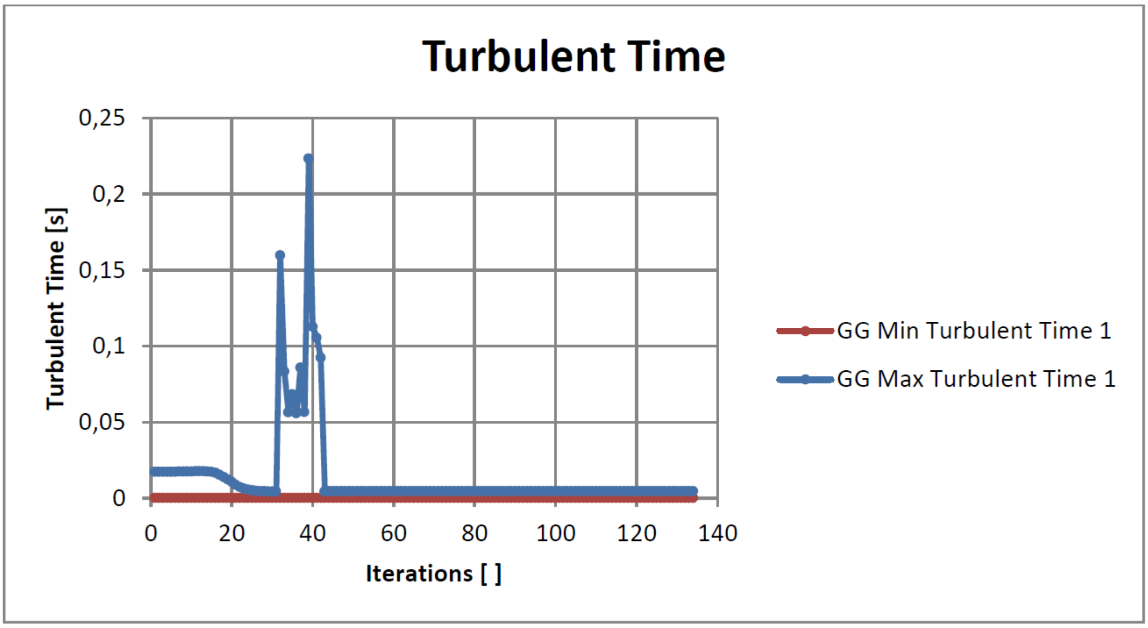

Figure 35. Propeller turbulator turbulence timeline

4. Conclusions and Recommendations

Turbulators have various geometries and can be placed in circular section pipes. The manufacture of turbulators is inexpensive and easy. It can also be easily placed in piped heat exchangers. As a result, turbulators are optimal solutions to improve the performance of the previously manufactured heat exchanger. In addition, the pipes can be easily cleaned, as they are easy to disassemble and install. In the use of new equipment, it allows for smaller heat exchanger sizes.

In this study, heat transfer and flow simulation of the spring type turbulator placed in parallel flow double tube heat exchanger, as well as analysis of the propeller type turbulator, which is a different production, were carried out. Turbulator parts were drawn in the Solidworks commercial program and the analyses were carried out through the flow-simulation plugin. When analyzing the turbulators to be compared, reynolds number, flow rate, flow temperature, heat transfer coefficients, etc. were performed in the same way. Copper is preferred as a material in two turbulators. Both light and thermal conductivity of copper is preferred because it is known. Pure water was selected as fluid in the analysis.

The evaluation of the results of the resulting analysis can be summarized as follows.

- In the Solid works commercial program, the spring type and propeller type turbulator heat exchanger model has been successfully applied.
- From the analysis results obtained in this study, each parameter was compared with a maximum and minimum.
- When the resulting speed graphs were examined, it was observed that the fluid speed of the propeller type turbulator was higher.
- When the resulting turbulence energy graphs are compared, the turbulence curve of the propeller type turbulator is constantly increasing and it has been determined that there is a decrease in the spring type.
- Turbulence viscosity curve propeller turbulator is found to increase continuously and spring turbulent decreases.

The results of the analysis obtained are shown to the maximum in Table 4.

Table 4. Approximate maximum values of the analysis results

Turbulator Types	Spring Turbulent	Propeller Turbulent
Speed (m/s)	61,6463	74,8748
Static Pressure (MPa)	4,771	7,811
Temperature (^{o}C)	20,15	20,32
Turbulence Energy (J/kg)	211,168	165,518
Turbulence Viscosity(Pa.s)	14,81	18,62
Turbulence Length (m)	0,00357366	0,00341521
Turbulence Time (s)	0,00886499	0,0045497

When all these results were examined, it showed that in the turbulator, known as the turbulence generator, the propeller-type turbulator is better because the turbulence energy and turbulence viscosity increase steadily over time. As seen in the analysis, a design has been realized that will be an alternative to other turbulator designs.

Statement of Conflicts of Interest

There is no conflict of interest between the authors.

Statement of Research and Publication Ethics

The author declares that this study complies with Research and Publication Ethics.

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