Evaluation of in-pipe turbine performance for turbo solenoid valve system

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ABSTRACT
This study is related to the newly developed and patented turbo solenoid turbine system. In this context, a numerical study was conducted on turbines with Darrieus, Gorlov and Lucid® configurations. As a result of the tests made on turbines with a NACA0018 aerofoil profile, the Lucid® turbine produced the highest moment and also highest pressure loss. Although a 10° twisted aerofoil Gorlov turbine produced 31.3% less moment than the Lucid® turbine, it also caused 51.4% less pressure loss. Also, it was observed that a Darrieus turbine with NACA0021 profile produced 1.5% less moment and 3.3% more pressure loss than a NACA0018 profile and turbine with NACA0015 profile, which produced 9% more moment and 1.4% more pressure loss than NACA0018.

On the performance impact studies of the aerofoil twist angle, 20° twisted Gorlov turbines produced 19.1% more moment and 5.5% less pressure loss than a 10° twisted aerofoil turbine. Finally, a 20° twisted Gorlov turbine with NACA0015 aerofoil profile was analyzed. Following the studies, it was observed that the 20° twisted Gorlov turbine with NACA0015 aerofoil profile gave the best result by creating a 49.3% less pressure loss, in spite of only producing 4.7% less moment than the Lucid® turbine.

1. Introduction
Fluid control technology is a field with numerous applications in the industry, the service industry and for domestic uses. In 2017, the worldwide valve sector is expected to reach $80 billion. This value is estimated to be $98.5 billion by 2019 (Freedonia Group, 2017).

On the other hand, with the proliferation of concepts such as the industry 4.0 industrial revolution in the world and the internet of things, the interest and need for smart systems is increasing day by day. Two issues have gained importance in this respect, particularly by the routing of current user trends: energy efficiency and self-sufficiency.

Especially in regions where electrical power is difficult to deliver, it is very important to provide the energy of the flow automation equipment that will be used for flow control in remote areas. Cable networks established for the energy supply and control of flow automation equipment in installations, such as international crude oil and natural gas transmission lines, large power plants, petrochemical plants and petroleum refineries bring serious installation, repair and maintenance costs to the enterprises. The failure of fluid control equipment to operate properly in such facilities may sometimes lead to adverse situations. Because of this reason, it is ensured that these systems work continuously in a healthy way with the repairs and maintenance that are perpetually made in the enterprises. In areas of explosion risk, in particular, these costs are increasing even more, due to the dedicated equipment that needs to be used, and any malfunction can lead to greater catastrophes.

The numerical and experimental study presented here is concerned with the establishment of a locally self-sustaining system for the control of the fluid control equipment, producing energy with the aid of the fluid to be controlled. Thanks to this system, automation processes will become much easier, and it will be possible to provide only wireless power and flow control with wireless communication, especially in big pipe areas without any cabling system.

In the Turbo Solenoid Valve System, first of all, the rotation is obtained from the running fluid by means of a turbine; in this way, a special electrical generator is operated and the solenoid valve is energized by the electric power obtained. A battery system is used to keep the required electrical energy at an appropriate level and to ensure that the control system operates in all situations.

The operation of the system is briefly as follows: the fluid is driving the turbine, the turbine drives the
generator and the alternative current power obtained from the generator is converted by a rectifier and a regulator to a constant voltage, which charges the battery and powers the control system. In case of both the first operations and insufficient power generation, the battery will provide the necessary energy. It will also feed the control system in standby mode, which requires a certain amount of energy, especially for wireless data communication.

As a result of the experimental measurements made, it has been observed that a 4W power source is sufficient for a latching solenoid valve that opens or closes in 12 seconds. In the experiments performed, the Wi-Fi system’s power consumption and the system's incapability of generating power when the valve is closed have been considered.

As is well known, the prime equipment used to generate electric power from the flow energy is turbines. The turbines can be divided into two main groups according to the position of the turbine's axis of rotation in the flow: (1) axial flow turbines, (2) cross-flow turbines. In axial flow turbines, the turbine's axis of rotation is parallel to the flow direction. In order to obtain a good efficiency in such turbines, the direction of the rotor must always be parallel to the flow. In cross-flow turbines, the flow direction is perpendicular to the turbine's axis of rotation. In such turbines, the axial direction of the turbine does not need to be continuously changed as in the axial flow turbines as well as the flow axis. Therefore, they are advantageous over the axial flow turbines in this respect. However, it is very difficult to predict the design rules and the hydrodynamic behavior of such turbines (Zanette, Imbault, & Tourabi, 2010). On the other hand, the flow structure in cross-flow turbines introduces varying loads on the aerofoils. In these turbines, the turbine is exposed to variable loads at each rotation, which causes material fatigue. The severity of these mechanical loads determines the turbine lifetime. Therefore, material fatigue analysis is very important in these types of turbines.

In the last decades, by the advancements of computer technology, CFD softwares are used widely for the study of different turbine designs. Tchon and Paraschivoiu (1994) use a noninertial stream function–vorticity formulation of the two-dimensional unsteady Navier–Stokes equations to study the dynamic stall phenomenon on a Darrieus wind turbine and simulate the incompressible flow field around a moving aerofoil. They simulate the flow around a NACA0015 aerofoil in Darrieus turbine and a Reynolds number of 6700 was used which was one tenth of real conditions. The results were compared with experimental observations. Horiuchi, Ushiyama, and Seki (2005) analyzed the flow velocity around the lift base of a vertical axis wind turbine. In their study, both numerical and experimental methods were employed to confirm the precision of the analysis. They used Detached Eddy Simulation and compared the measured and calculated wind velocities. It was shown that for cross-flow turbines numerical simulation is applicable. It was also noticed that flow was not able to recover its freestream condition up to the distance of 5 times of turbine diameter from turbine center which is important for the installing of wind turbines in large quantities. Guerri, Sakout, and Bouhadef (2007) use moving meshing technique and time accurate Reynolds Averaged Navier Stokes (RANS) solver for simulation of fluid flow around a small rotating vertical axis wind turbine. In all simulations, 2D incompressible fully turbulent flow was assumed. Turbulence was modeled by the SST k/ω model of Menter. By integrating the pressure and shear stress over the blade surface, the blade forces and torques were calculated. They compare the power coefficient results calculated by RANS solver and double multiple stream tube model (DMSM). It was shown that RANS model gave a better prediction result than DMSM. Ferreira, Kuijk, Bussel, and Scarano (2009) conducted research about the aerodynamic behavior of a vertical axis wind turbine. In their study, PIV (Particle Image Velocimetry) was used to find the development of a dynamic stall at different tip speed ratios (TSRs). At low TSRs, dynamic stall had a detrimental effect on the operation of VAW turbines effecting both loads and power ratio. They try to understand this issue by visualizing and quantifying it and create a database for model validation. Ferreira, Van Bussel, and Van Kuijk (2007) also try to evaluate the differences between the different commonly used turbulence models (Laminar, Spalart–Allmaras and k–ε) and verify them with PIV experimental data. They found that Spalart–Allmaras and k–ε have same average normal and tangential forces over the upwind semi rotation. Hamada, Smith, Durrani, Qin, and Howell (2008) also study the VAWT with straight blades (H type) by using the sliding mesh vertical axis wind turbine technique. They try to compare 2D and 3D simulations and highlight the 3D effects. Also, they indicated that an appropriate time step needs to be used to obtain an accurate solution. Hwang, Lee, and Kim (2009) try to find the optimum characteristic values by changing of various characteristics of a turbine like a number of blades, chord length, TSRs, aerofoil shapes, and pitch and phase angles. Through the optimum values, they can approximately achieve a 70% better performance than a fixed pitch turbine. They also try to find different values for a controllable pitch angle to reach the maximum rotor performance at different operating conditions. This changeable pitch angle causes 25% improvement compared to the
constant value turbines. Paraschivoiu, Trifu, and Saeed (2009) try to maximize the torque generation of H type Darrieus wind turbine by the change of the blades pitch angle. A DMSM code was utilized to determine the performance of straight blade VAWT at the operating wind velocity. Takao et al. (2009) study the effect of guide vane geometry on the performance of a straight-bladed vertical axis wind turbine (S-VAWT). They try to optimize the effects of setting angle and gap between the rotor blade and guide vane on the power coefficient of three straight blade rotor with NACA0018 aerofoils. Mohamed (2012) studied the efficiency of Darrieus turbine with 20 different symmetric and non-symmetric aerofoils. An unsteady two-dimensional Computational Fluid Dynamics method by realizable $k-\varepsilon$ turbulence model was used in this study. By this new design, he can obtain 10.9% increase on absolute efficiency. Also, he mentioned that dynamic stall assumes a significant role in determining the cross-flow turbine performance, especially at low TSRs. By using CFD and also experimentally he investigates the capability of Darrieus turbine and some techniques to improve certain drawbacks. He also studied the effect of the turbine solidity and the usage of hybrid system between drag and lift types (Mohamed, 2013).

Chen and Lian (2015) study the vortex dynamics of a two-dimensional cross-flow turbine. The impact of dynamic vortices on the force field was determined using the RNG $k-\varepsilon$ turbulence model. Their result concludes that the torque coefficient of a thicker aerofoil is larger than a thinner one.

When designing the turbine in cross-flow turbines, factors such as limiting the fatigue effects of the material and decreasing the maximum and mean load difference need to be considered.

Generally speaking, cross-flow turbines have two important advantages over axial flow turbines: (1) they can be connected directly to the generator (without using intermediate gear mechanism), (2) they can be easily installed. On the other hand, the hydrodynamics of cross-flow turbines are very difficult due to three different conditions that cause uncertainty, complex turbulence and flow separation. (1) ever differing angle of attack, (2) influence of aerofoil turbulences on each other, (3) influence of connecting rods on flow (Marsh, Ranmuthugala, Penesis, & Thomas, 2015).

In cross-flow turbines, at low rotational speed, changes in the angle of attack very effectively influences turbine performance and causes static and dynamic stalls to occur (Paraschivoiu, 2002). On the other hand, the performance of the turbine at high speeds is influenced by the turbulence of the flow running into the turbine, the vortices created by the aerofoils and the interactions they have on each other (Scheurich, Fletcher, & Brown, 2011). In addition, as the rotational speed increases, the drag effect of the connecting rods increases and the total generated moment decreases (Marsh, Ranmuthugala, Penesis, & Thomas, 2013).

In this study, it is planned to use a cross-flow turbine in design because it is more suitable for the system considered, it is simpler and longer lasting for mass production in the future, the system does not need a very high moment generation and low pressure loss is more important in this system.

In this study, three different types of cross-flow turbines were numerically compared to each other and the most suitable type of turbine was selected for the turbo solenoid system. (1) Darrieus Turbine (U.S. Patent No. 1.835.018, 1931), (2) Gorlov Turbine (U.S. Patent No. 5.451.137, 1997) 3) Lucid® Turbine (U.S. Patent No. 7959411, 2011) (Figure 1). The numerical study was based on Sliding Mesh approach where the interaction of eddies shed by blades with consequent blades was taken into account.

**Figure 1.** Analyzed cross-flow turbines.
2. Performance criteria in cross-flow turbines

As can be seen in Figure 2, the flow rate in a cross-flow turbine varies with the angle of the turbine.

Using the velocity triangles, the relation of the relative velocity \( W \) with respect to the absolute velocity \( U_\infty \) and the tangential velocity \( V \)

\[
W = \sqrt{(U_\infty \cos \theta + \vec{V})^2 + (U_\infty \sin \theta)^2}
\]

(1)

\[
W = \sqrt{U_\infty^2 \cos^2 \theta + V^2 + 2U_\infty \vec{V} \cdot \cos \theta + U_\infty^2 \sin^2 \theta}
\]

(2)

\[
W = \sqrt{U_\infty^2 + V^2 + 2 \vec{U}_\infty \vec{V} \cdot \cos \theta}
\]

(3)

On the other hand, it is possible to write the tangential or chord velocity as a function of the angular velocity as follows:

\[|\vec{V}| = r \cdot \omega\]

(4)

Also, in the literature, the ratio of tangential velocity to absolute velocity is called the tip speed ratio.

\[
\lambda = \frac{|\vec{V}|}{|U_\infty|} = \frac{r \cdot \omega}{U_\infty}
\]

(5)

By substituting tip speed ratio into the Equation 3:

\[
W = U_\infty \sqrt{1 + \lambda^2 + 2 \lambda \cdot \cos \theta}
\]

(6)

\( U_\infty \) velocity varies along the diameter of the pipe and it is a function of the height \( h \) of the turbine.

\[
\bar{W} = \frac{1}{2\pi} \int_0^{2\pi} W d\theta
\]

(7)

On the other hand, the angle of attack is a function of angle \( \theta \):

\[
\alpha = \arctan \left( \frac{U_\infty \sin \theta}{U_\infty \cos \theta + V} \right)
\]

(8)

\[
\alpha = \arctan \left( \frac{\sin \theta}{\cos \theta + \lambda} \right)
\]

(9)

If there is an angle of attack (\( \beta \)) of the aerofoil at \( \theta = 0 \), then equation 9 is written in the following form:

\[
\alpha = \arctan \left( \frac{\sin \theta}{\cos \theta + \lambda} \right) - \beta
\]

(10)

Figure 3 shows drag and lift force in a cross-flow turbine aerosfoil. Of these two forces, the component vertical to the aerosfoil is called the normal force and the tangential component as the tangential force.

Another term used in turbine performance calculations is the tangential force coefficient \( (C_t) \). The tangential force coefficient is the difference between the tangential component of the lift and drag force. Another term used in turbine performance calculations is the normal force coefficient \( (C_n) \). The normal force coefficient is derived from the difference of the normal component of the lift and drag force.

\[
C_t = C_l \sin \alpha - C_d \cos \alpha
\]

(11)

\[
C_n = C_l \cos \alpha + C_d \sin \alpha
\]

(12)

In this case, the net tangential and normal force can be defined as follows:

\[
F_t = C_t \frac{1}{2} \rho \ c \ h \ W^2
\]

(13)

\[
F_n = C_n \frac{1}{2} \rho \ c \ h \ W^2
\]

(14)

The above two equations show the Chord Length \( c \), fluid density and turbine height \( h \).

\( F_t \) varies depending on the rotation angle \( (\theta) \). The average tangential force for an aerosfoil can be obtained

\[
\bar{F}_t = \frac{1}{2\pi} \int_0^{2\pi} F_t d\theta
\]
from the following equation.

\[ F_t = \frac{1}{2\pi} \int_{0}^{2\pi} F_t(\theta) d\theta \]  

(15)

In this case, the total moment for an N-aerofoil turbine is as follows:

\[ M = N F_t r \]  

(16)

From here the total power can be obtained as follows:

\[ P = M \cdot \omega \]  

(17)

Tip speed ratio (equation 5) depends mostly on the rotor rotational speed. The tip speed ratio is the ratio of the tangential velocity of the tip flow to the main flow velocity.

As the tip speed ratio decreases, the friction forces become effective. As the tip speed ratio increases, greater loads are exerted on aerofoils. The high tip speed ratio causes a high noise level in the wind turbines. As the tip speed ratio increases, stronger aerofoils need to be used due to the increase in centrifugal force.

One of the most important performance criteria for turbines is the power to weight ratio. Using the Buckingham \( \pi \) theorem, the non-dimensional power to weight ratio can be obtained as shown in equation 18.

\[ C_p = \frac{P}{\rho \frac{U_\infty^3}{2} D h} \]  

(18)

Power to weight ratio is the ratio of the power obtained in its expression to the power present in the flow. This equation is like efficiency, therefore, it usually takes a value between 0 and 1.

3. Numerical approach

The characteristics of the turbines to be subjected to the CFD analysis are given in Table 1. In this design study, the FLUENT software program was used to solve the flow equations.

| Characteristic | Darrieus | Gorlov | Lucid® |
|----------------|----------|--------|--------|
| Number of Aerofoil | 3        | 3      | 3      |
| Attack Angle | 0°        | 0°      | 0°      |
| Turbine Height (h) | 32 mm    | 32 mm  | 37 mm  |
| Turbine Diameter (D) | 30 mm    | 30 mm  | 45 mm* |
| Aerofoil Profile | NACA0018 | NACA0018 | NACA0018 |
| Main Axis Diameter | 10 mm    | 10 mm  | 10 mm  |
| Support Bars Diameter | 7 mm     | 7 mm   | 7 mm   |
| Chord Length | 15 mm     | 15 mm  | 15 mm  |
| Twist Angle | 0°        | 10°/20°| 120°   |

* diameter of the sphere.

3.1. Mesh sensitivity analysis

To establish appropriate numerical models and obtain accurate computed results, many factors should be taken into consideration, such as computational domain, mesh generation, boundary conditions, solver setting, and residual control (Mou, He, Zhao, & Chau, 2017). As a first step, good mesh production is the most important phase of the CFD analysis. In this study, the mesh independence of the results for a design was examined before beginning the general analysis. In this step, three different mesh areas as coarse, medium and fine was applied. As a result, it’s seen that there is 1.5% difference between coarse and fine mesh but there is less than 1% difference between medium and coarse mesh so medium mesh area with 8 million girds was produced (Figure 4).

3.2. Solving method

In this study, FLUENT’s Multiple Rotating Reference Frame (MRF) analysis method was used to analyze the rotating turbine (Figure 5).

A more sophisticated analysis can be achieved by creating a higher quality mesh in the framed section (Figure 6). The separation of this high-quality mesh segment from the other segments is identified by interface boundary condition.

![Figure 4. Torque value of different grid density around aerofoil.](image-url)

![Figure 5. MRF analysis method.](image-url)
During the analysis, a sliding mesh method has been used. For this method which is a time dependent method, time step of $10^{-4}$ has been selected. The results have been evaluated after 6 tours returning of the turbine.

One of the most important factors in the CFD analysis is the turbulence solution. In this design study, given that all the flows analyzed are pipe flow and well over the 2300 Reynolds number, the flow is fully turbulent and the standard $k-\epsilon$ model was chosen as the analysis model.

As the near wall approach in turbulence analysis is more sensitive, enhanced wall treatment solution, which performs the solutions using the surface modeling approach, is preferred in the analysis. In the analysis performed in this context, it was ensured that $y^+ < 5$ (Figure 7).

Four different boundary conditions were used in this study. The ‘velocity inlet’ condition at the inlet surface of the fluid to the pipe, the ‘outflow’ condition for the outlet surface of the fluid from the channel, the ‘interface’ condition to the surface between the rotating cylinder network and the running fluid network and the ‘wall’ condition for the pipe and turbine surfaces were used (Figure 8).

In the analyses, the inlet velocity was determined to be 3 m/s and the turbine rotational speed to be 157 rad/s.

4. CFD results

In this study, three different factors were examined: (1) Turbine type effect, (2) Aerofoil profile effect, (3) Aerofoil twist angle effect.
In order to examine the turbine type effect, as already mentioned, analyses were performed for Darrieus, 10° twisted Gorlov and Lucid® turbines, having a profile of NACA0018. Analyses of the NACA0015 and NACA0021 Darrieus turbines were conducted to investigate the effect of the aerofoil profile. Finally, an analysis of a 20° twisted Gorlov turbine with NACA0018 profile was carried out to investigate the twist angle effect.

It is very important that the moment generated by the rotating turbine at different positions and the pressure loss it creates are not too variable. The excess moment variation at different angles when the turbine is rotating will lead to variations of the load on the bearing area and material fatigue. Furthermore, the variation in pressure loss caused by the turbine at different angles will cause an increase in hydrodynamic instability. In this context, by comparing the turbines with one another, we will consider the moment produced by the turbines and the pressure loss they create, as well as the standard deviation of these two values at various angles. Tables 2 and 3 present the average moment values produced by different types of turbines at different angles and the average pressure loss values they produce in descending order along with their standard deviation.

By examining the results, it can be seen that the highest moment production is in the Lucid® turbine with the lowest standard deviation. Due to the structure of the Lucid® turbine, low average moment deviation in these types of turbines is an expected result; however, it seems that a very high pressure loss occurs in this turbine type. Since the objective in the turbo solenoid system is to generate the power required by the system with the lowest pressure loss, it seems that the Lucid® turbine is not very suitable for this application because it causes a pressure loss of about 100% more compared with other turbine types.

When the other turbine types are examined, it is observed that the 20° twisted NACA0018 profile Gorlov turbine is the second highest in moment generation and the lowest in terms of pressure loss.

In addition, in order to examine the effect of the aerofoil profile, the analysis of the Darrieus turbine showed that the NACA0015 profile gave the best result in terms of moment generation. The Darrieus turbine with the NACA0015 profile had a pressure increase of 1.4% against a 9% moment increase. As a result of this analysis, it was concluded that the 20° twisted Gorlov turbine could give the best result with the NACA0015 profile and the analysis of this turbine was carried out under similar conditions. For an easy comparison, the average moment generated by all the turbines analyzed in Tables 4 and 5

Table 2. Average moment values produced.

| Turbine type   | Mean torque [N.m] | Standard deviation |
|---------------|------------------|-------------------|
| Lucid® NACA0018 | 0.043856         | 0.010011          |
| Gorlov20 – NACA0018 | 0.035866         | 0.014829          |
| Darrieus-NACA0015 | 0.035221534      | 0.018102          |
| Darrieus-NACA0018 | 0.0323035         | 0.019721          |
| Darrieus-NACA0021 | 0.031807847      | 0.015979          |
| Gorlov10-NACA0018 | 0.030112992      | 0.018246          |

Table 3. Average created pressure loss values.

| Turbine type   | Pressure loss [Pa] | Standard deviation |
|---------------|-------------------|-------------------|
| Lucid® NACA0018 | 16650             | 1001.1            |
| Darrieus-NACA0021 | 7651             | 641               |
| Darrieus-NACA0015 | 9485             | 825.7             |
| Darrieus-NACA0018 | 9332             | 783.4             |
| Gorlov10-NACA0018 | 9659             | 828.4             |
| Gorlov20-NACA0018 | 8097             | 678.3             |

Table 4. Proportioning of average moment values.

| Turbine type   | Lucid® NACA0018 | Darrieus-NACA0018 | Darrieus-NACA00015 | Darrieus-NACA0021 | Gorlov10-NACA0018 | Gorlov20-NACA0018 | Gorlov20-NACA0015 |
|---------------|----------------|------------------|-------------------|------------------|------------------|------------------|------------------|
| Lucid® NACA0018 | *              | 35.8%            | 24.5%             | 37.9%            | 45.6%            | 22.3%            | 5%               |
| Darrieus-NACA0018 | −26.3%          | *                | −8.3%             | 1.6%             | 7.3%             | −9.9%            | −22.7%           |
| Darrieus-NACA00015 | −19.7%          | 9%               | *                 | 10.7%            | 17%              | −1.8%            | −15.7%           |
| Darrieus-NACA0021 | −27.5%          | −1.5%            | −9.7%             | *                | 5.6%             | −11.3%           | −23.9%           |
| Gorlov10-NACA0018 | −31.3%          | −6.8%            | −14.5%            | −5.3%            | *                | −16%             | −27.9%           |
| Gorlov20-NACA0018 | −18.2%          | 11%              | 1.8%              | 12.8%            | 19.1%            | *                | −14.2%           |
| Gorlov 20-NACA0015 | −4.7%           | 29.4%            | 18.6%             | 31.4%            | 38.8%            | 16.5%            | *                |

Table 5. Proportioning of average pressure values.

| Turbine type   | Lucid® NACA0018 | Darrieus-NACA0018 | Darrieus-NACA00015 | Darrieus-NACA0021 | Gorlov10-NACA0018 | Gorlov20-NACA0018 | Gorlov20-NACA0015 |
|---------------|----------------|------------------|-------------------|------------------|------------------|------------------|------------------|
| Lucid® NACA0018 | *              | 78%              | 75.5%             | 72.4%            | 105.6            | 117.6%           | 97%              |
| Darrieus-NACA0018 | −43.8%          | *                | −1.4%             | −3.2%            | 15.5%            | 22.2%            | 10.7%            |
| Darrieus-NACA00015 | −43%            | 1.4%             | *                 | −1.8%            | 17.1%            | 24%              | 12.3%            |
| Darrieus-NACA0021 | −42%            | 3.3%             | 1.8%              | *                | 19.3%            | 26.2%            | 14.3%            |
| Gorlov10-NACA0018 | −51.4%          | −13.4%           | −14.6%            | −16.2%           | *                | 5.8%             | −4.2%            |
| Gorlov20-NACA0018 | −54%            | −18.2%           | −19.3%            | −20.8%           | −5.5%            | *                | −9.4%            |
| Gorlov 20-NACA0015 | −49.3%          | −9.7%            | −10.9%            | −12.5%           | 4.4%             | 10.4%            | *                |
and the average pressure loss they created are compared in a crosswise fashion. In these two tables, by proportioning the turbines identified in the vertical turbine types column to the turbines identified in the horizontal turbine types column, how much more or less moment they produce or create pressure loss in percentage was determined.

As shown in Table 4, the Gorlov turbine with a 20° twisted profile turbine is the second best in terms of moment generation. This turbine generates only 4.7% less moment than the Lucid® turbine with a NACA0018 profile. However, as shown in Table 5, it causes 49.3% less pressure loss.

As shown in Table 5, the Gorlov turbine with a 20° twisted NACA0015 profile is ranked 5th in terms of the pressure loss it creates. The pressure loss produced by this turbine is 4.4% higher than that of Gorlov turbine with a 10° twisted NACA0018 profiled turbine. However, the corresponding moment produced is 38.8% higher. Similarly, the pressure loss produced by the Gorlov turbine with a 20° twisted NACA0015 profile is 10.4% greater than the Gorlov turbine with a 20° twisted NACA0018 profile. However, it produces 16.5% more moment.

Figures 9 and 10 show the moment generated at different angles and the pressure loss created by all the turbines analyzed.

5. Experimental study

For the experimental studies, a 20° twisted Gorlov turbine with a NACA0015 aerofoil profile was produced. In addition, a turbine housing was built for a special pipe made from a DN50 size plexiglass material (Figure 11).

For the experimental studies, a special test setup was used in the SMS-TORK mechanical laboratory. This test setup is a specially designed test rig to measure the performance of solenoid valves. In this test setup, one frequency modulated pump, flowmeter and pressure transmitter are used to calculate the flow coefficient of the valves in the size range from 1/8” to 2”.

The pump frequency was set above 42.2 Hz to obtain a flow speed of 3 m/s in the CFD analysis. In this case, the flow meter reads 21.2 m³/h flow rate. This value means an average flow speed of 3 m/s in a DN50 pipe.

In the first stage, the turbine is rotating at 141 rad/s with no generator connection and the turbine is neutral. Since the CFD analyses are performed at 157 rad/s, which is greater than 141 rad/s, this time the CFD analyses were repeated with 78.5 rad/s for the Gorlov turbine with a 20° twisted NACA0015 wing profile in order to make an accurate comparison between experimental and numerical analyses. In this case, the average moment value is 0.0176 Nm and the mean pressure loss is calculated as 6.3 kPa. After this step, a special test setup for moment measurement was developed. This system consists of a braking system, a torque meter and a tachometer (Figure 12).

When the turbine rotates under the effect of 3 m/s flow, the rotation speed is fixed at 78.5 rad/s by means of the brake provided in the system. In these conditions, the moment value is measured as 0.016 Nm. In this case, a difference of 10% is seen between the numerical and
experimental result, which is caused by the friction in the bearing and sealing elements.

On the other hand, a pressure loss of 28.3 kPa was measured in experimental studies with a calibration uncertainty of ±3.5% and equipment accuracy of 0.2%. This value was predicted as 6.3 kPa in the CFD studies. This difference is due to local losses in pipes which was not taken into account in CFD runs. When the turbine is taken off the line, the pressure loss of the pipe is measured as 20.1 kPa. In this case, the pressure loss due to the turbine is calculated as 8.2 kPa. This difference between the CFD and the experimental studies is thought to be caused by surface and windage friction on the turbine.

6. Conclusions

For the operation of the turbo solenoid system, a power of 4W is predicted. The average power output for the turbine types analyzed using Equation 17 in Table 6 is presented in descending order.

One of the most important performance criteria of turbines is the power to weight ratio. The average power to weight ratio for the turbine types analyzed in Table 7 are presented in descending order.

As a result of the comparisons made in Tables 6 and 7, the Gorlov turbine with a 20° twisted NACA0015 profile is considered as the best turbine in terms of moment production and low pressure loss. Conversely, when Tables 6 and 7 are examined, the Gorlov turbine with a 20° twisted NACA0015 profile shows the best performance after the Lucid turbine.

In contrast, when the CFD and the experimental results are compared, it is observed that these results are close to each other and that the difference is stemmed from the losses in the bearing and sealing components.

In the continuation of these studies, it is deemed useful to analyze 30° and 40° twisted aerofoil Gorlov turbines and compare them with other results but because of pipe internal diameter structurally it is impossible to replace these turbines in a 50 mm diameter pipe. Since the NACA0012 profile's aerofoil cross-section is very thin, it is not easy to manufacture it in the studied dimensions and it is considered that it would not be very useful to conduct an analysis in this respect. However, for larger pipe diameters, the NACA0012 profile can also be considered.

As a result of these studies, two important conclusions may be withdrawn: Firstly, the numerical approach chosen should be Sliding Mesh approach since otherwise the interaction of shed eddies with consequent blades may not be seen. Secondly, it can be concluded that the turbo solenoid system will be useful for large pipe diameters. By using similar considerations, it is thought that the turbo solenoid system can be used to control the solenoid of the pneumatic actuator’s directional valve in the pneumatic valve actuators.

Disclosure statement

No potential conflict of interest was reported by the author(s).
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