



Parametric Rotor Innovation for Gravitational Vortex Turbines: Advancing Clean Energy through Integrated Computational Fluid Dynamics (CFD) Simulation and Experimentation to Support the Sustainable Development Goals (SDGs)

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ABSTRACT

This study aims to optimize the rotor design of gravitational vortex turbines through an integrated approach combining parametric modeling, computational fluid dynamics (CFD), and experimental validation. The rotor geometry was formulated using velocity triangle theory and refined using a CFD-based evaluation of flow behavior across multiple turbulence models. To ensure accurate comparisons, experimental tests were conducted under controlled flow conditions, and mechanical losses due to bearing friction were measured and accounted for. The simulation results demonstrated strong agreement with empirical data, validating the effectiveness of the proposed design. By enhancing energy conversion efficiency while minimizing environmental impact, this research supports the development of decentralized, renewable micro-hydropower systems. The findings contribute to the advancement of Sustainable Development Goals (SDGs).

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1. INTRODUCTION

The global population and economic growth have intensified attention toward renewable energy sources, particularly highlighting hydropower as a key resource aligned with the United Nations' Sustainable Development Goals (SDGs) (see <https://unstats.un.org/sdgs/report/2023/The-Sustainable-Development-Goals-Report-2023-Spanish.pdf>). However, current efforts remain insufficient to meet the 2030 targets, revealing an urgent need to implement innovative renewable technologies with reduced environmental and social impacts. Although hydropower is a renewable source that avoids combustion processes, traditional implementation through large-scale hydroelectric projects results in significant alterations to surrounding terrestrial and aquatic ecosystems [1], as well as social issues related to forced displacement and community conflicts caused by flooding [2]. In response to these challenges, there is a growing interest in small-scale hydropower technologies that allow for efficient, renewable, and socially inclusive energy generation with minimal impacts [3].

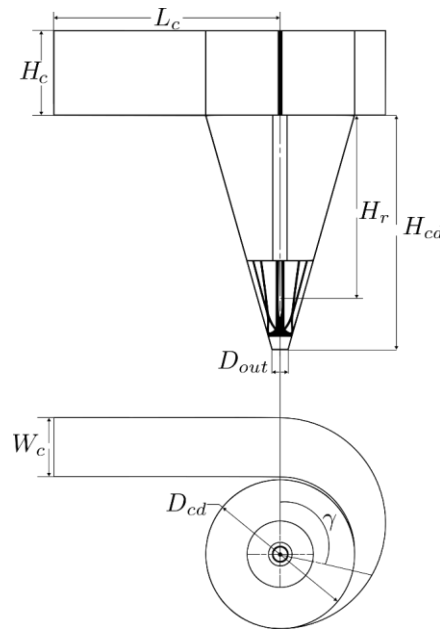
In this context, the gravitational vortex turbine (GVT), developed in 2004, has gained attention due to its constructive simplicity, low economic and environmental cost, and capability for small-scale energy generation (pico-generation) (see Zotlöterer patent in 2004 regarding Hydroelectric power plant (Patent No. WO2004061295A2)). The GVT harnesses the motion of a gravity-induced vortex in a rotation chamber to drive a rotor at its core, while also allowing the free passage of aquatic species without compromising their survival. Additionally, its construction can be carried out using accessible materials with low environmental impact [4]. GVT efficiency shows high sensitivity to changes in its geometric parameters, with different performances observed depending on the specific rotor configuration implemented [5,6]. Several studies have explored innovative geometric alternatives, including straight and curved blades, multiple rotors, and optimized configurations to harness vertical flows, highlighting a theoretical gap regarding vortex behavior in the presence of a rotor [7-11].

To properly analyze the behavior of these gravity-induced vortical flows in GVTs, numerical methods such as computational fluid dynamics (CFD) and advanced experimental techniques like Particle Image Velocimetry (PIV) are commonly employed [12-14]. However, few studies have integrated both methodologies to optimize rotor design based on a detailed analysis of the velocity triangle applied to swirling flows, a methodology that has proven effective in other types of hydraulic turbines [15,16]. In this research, a specific geometry of a GVT is analyzed, based on a conical chamber designed to induce and stabilize a gravitational vortex (**Figure 1**). This configuration includes a rectangular inlet channel, a main conical tank, and a lower discharge orifice that ensures stable vortex formation. The dimensions and specific geometric parameters used are summarized in **Table 1**.

The selection of this GVT geometry was based on various numerical and experimental studies available in the literature. These studies have demonstrated that a conical tank geometry optimizes turbine performance by improving the tangential velocity of the flow [5,17,18].

Table 1. Main geometric parameters for the turbine.

Parameter	Description	Value	Units
H_c	Height of the inlet channel	283	mm
L_c	Length of the inlet channel	759	mm
H_{cd}	Height of the discharge chamber	786	mm
H_r	Height of the rotor	659.5	mm
D_{out}	Outlet diameter	54	mm
W_c	Width of the inlet channel	198	mm
γ	Wrap-around angle	89.95	°
D_{cd}	Upper chamber diameter	500	mm

**Figure 1.** Main geometric parameters for the turbine.

2. METHODS

2.1. Rotor Design and Parametric Modeling

For the development of this study, a rotor design is proposed based on the fourth hydro-rotor [19]. This design is modifiable and allows the resizing of geometric parameters such as height, top and bottom radii, twist angle, and concavity. To this end, a parametric mathematical function was developed to describe the radius of a blade, using the next two principles to ensure that the rotor remains similar to its original model:

- (i) The variation of Z is not constant with the variation of θ ($\frac{d^2Z}{d\theta^2} \neq 0$).
- (ii) The twist angle of each blade equals 2π radians divided by the number of blades ($\gamma = \frac{2\pi}{n}$).

The parametric design used for this study is presented in Equations (1), (2), and (3).

$$z(t) = \frac{h_r\sqrt{2}}{2} \left(t + \sqrt{\left(t - \frac{\sqrt{2}}{2}\right)^2 + e^a} - \sqrt{\frac{1}{2} + e^a} \right) \quad (1)$$

$$\theta(t) = -\gamma\sqrt{2} \left(t - \sqrt{\left(t - \frac{\sqrt{2}}{2}\right)^2 + e^a} + \sqrt{\frac{1}{2} + e^a} \right) \quad (2)$$

$$r(t) = \frac{\sqrt{2}(r_{up} - r_{down})}{2} z(t) r_{down} \quad (3)$$

where z is the height parameter, θ is the sweep angle in cylindrical coordinates, r is the variable rotor radius, t is the parameter ranging from 0 to $\sqrt{2}$, and a is the parameter controlling the curvature of the blade. The parameter h_r is defined as 25 cm to maximize the use of the available 3D printing volume. The rotor selected for this study, shown in **Figure 2**, features a specific 4-blade configuration and an incidence angle (α) of -3 degrees. This design aims to maximize the torque generated by the rotor and consequently improve the overall hydraulic efficiency of the system. The specific blade geometry was defined using the aforementioned parametric equations, enabling precise control of blade curvature via a parameter.

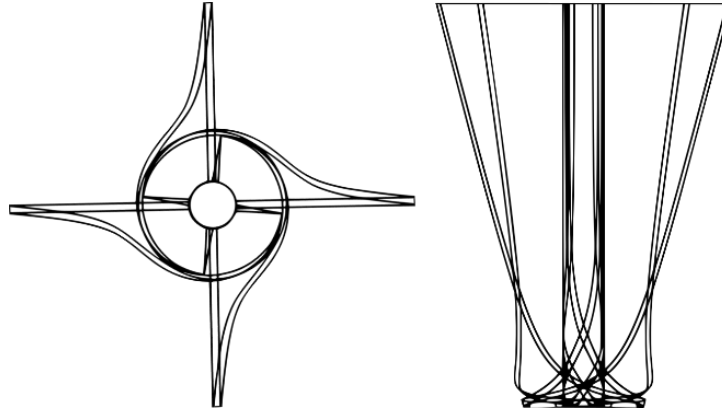


Figure 2. Design of the parametric runner.

For this analysis, the top surface (blue) is assumed to be the inlet to the control. For this study, efficiency is calculated using Equation (4), and for its maximization, it is necessary to maximize the torque generated in the rotor, which can be calculated using the angular momentum equation presented in Equation (5).

$$\eta = \frac{\tau\omega}{QgH\delta} \quad (4)$$

$$\tau = \oint \mathbf{r} \times \mathbf{V} \cdot (\mathbf{V} \cdot \hat{\mathbf{n}}) \delta dA \quad (5)$$

where η corresponds to the efficiency, τ represents total torque, ω represents angular velocity, Q corresponds to the total flow rate, g is the acceleration of gravity, H is the total head, and δ corresponds to the density of the water. \mathbf{r} represents the vector from the rotor axis to each point in the analyzed surface; \mathbf{V} is the velocity vector of the flow; $\hat{\mathbf{n}}$ is the normal vector perpendicular to the analyzed surface; and A is the area of each surface. The control surface selected for this analysis is shown in **Figure 3**.

Volume, and the inner surface (pink) as the outlet. The lateral surface (orange), which coincides with the chamber wall, does not contribute to torque generation in the rotor or domain (**Figure 3**). With the domain established, Equation (5) can be represented arithmetically as shown in Equation (6).

$$\begin{aligned} \tau &= \int_0^{2\pi} \int_0^{r_1} \mathbf{r} \cdot \mathbf{r} \times \mathbf{V}_1 \cdot (\mathbf{V}_1 \cdot \hat{\mathbf{n}}) \delta dr d\theta - \int_0^{2\pi} \int_0^{r_2} \mathbf{r} \cdot \mathbf{r} \times \mathbf{V}_2 \cdot (\mathbf{V}_2 \cdot \hat{\mathbf{n}}) \rho dr d\theta \\ &= \int_0^{2\pi} \int_0^r r^2 V_1^2 \cos(\theta_{\theta_1}) \cos(\theta_{z_1}) \rho dr d\theta - \int_0^{2\pi} \int_0^r r^2 V_3^2 \cos(\theta_{\theta_3}) \cos(\theta_{z_3}) \rho dr d\theta \\ &= \frac{2\delta V_1^2 r_1^3 \pi}{3} \cos(\theta_{z_1}) \cos(\theta_{\theta_1}) - \frac{2\delta V_3^2 r_3^3 \pi}{3} \cos(\theta_{z_3}) \cos(\theta_{\theta_3}) \end{aligned} \quad (6)$$

where $\cos(\theta_z)$ is the directional cosine of the velocity vector angle along the rotor axis and $\cos(\theta_\theta)$ is the tangential directional cosine. The total velocity of the water at the top of the rotor can be calculated using Torricelli's equation, shown in Equation (7).

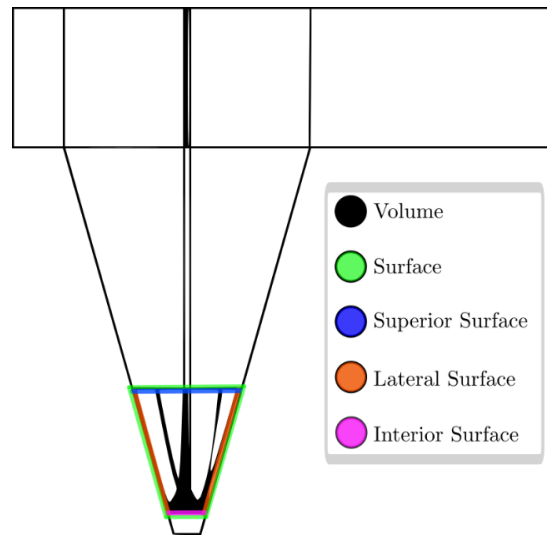


Figure 3. Main components of the GVT for the control volume details.

$$V = \sqrt{2gh} \quad (7)$$

Therefore, Equation (6) can be rewritten as shown in Equation (8).

$$\tau = \frac{4g\rho\pi}{3} (h_1 r_1^3 \cos(\theta_{z_1}) \cos(\theta_{\theta_1}) - h_3 r_3^3 \cos(\theta_{z_3}) \cos(\theta_{\theta_3})) \quad (8)$$

For this study, radial velocity is considered negligible in relation to tangential and axial velocities, so the directional cosines at the inlet are calculated using Equations (9) and (10):

$$\cos(\theta_z) = \frac{V_z}{\sqrt{V_z^2 + V_\theta^2}} = \sin(\theta_\theta) \quad (9)$$

$$\cos(\theta_\theta) = \frac{V_\theta}{\sqrt{V_z^2 + V_\theta^2}} = \sin(\theta_z) \quad (10)$$

where V_z and V_θ are the axial and tangential velocities, respectively. Since the analysis is performed under steady-state conditions, the average axial velocity can be calculated using the continuity principle and assuming constant density, while the tangential velocity is determined from the circulation (Γ) obtained through CFD simulations using the expression proposed for vortex characterization [20], as shown in Equation (11).

$$V_\theta = \frac{\Gamma}{2\pi r_m^2 + 2r^2} \quad (11)$$

where r_m is the radius at which the vortex reaches its maximum velocity, assumed to be $0.7r$ according to literature [20]. On the other hand, the terms associated with the lower surface (outlet surface 3) can be controlled via the velocity triangles of the rotor shown in **Figure 4**. This figure shows both the general 3D representation and the planar projection of the velocity triangles at the rotor inlet and outlet to facilitate analysis. This figure presents the velocity triangles at three key positions in the GVT rotor: the inlet (**Figure 4(a)**), the internal blade passage (**Figure 4(b)**), and the outlet (**Figure 4(c)**). At the inlet, the absolute velocity of the flow (V_1), the blade velocity (U_1), and the relative velocity (W_1) define the angle of incidence and the energy input into the rotor. Within the blade passage (**Figure 4(b)**), the direction and magnitude of these velocity components change as energy is transferred from the fluid to the rotor. At the outlet (**Figure 4(c)**), the remaining absolute (V_3) and relative (W_3) velocities, along with blade velocity (U_3), determine how much energy has been effectively converted into torque and how much is lost with the exiting flow. Understanding these velocity relationships is crucial for optimizing rotor design and accurately calculating hydraulic efficiency based on momentum theory.

Angle α_1 corresponds to the angle of θ_θ . Substituting Equations (8), (9), and (10) into Equation (4), and considering the velocity triangles, yields a hydraulic efficiency formulation for the machine, expressed in Equation (12).

$$\eta = \frac{4(W_1 h_1 A_1 \sin \alpha_1 \cos \alpha_1 - W_3 h_3 A_3 \sin \alpha_3 \cos \alpha_3)}{3Qh} \quad (12)$$

The relationships derived from the velocity triangles are presented in **Table 2**, and the correlation is in Equation (13).

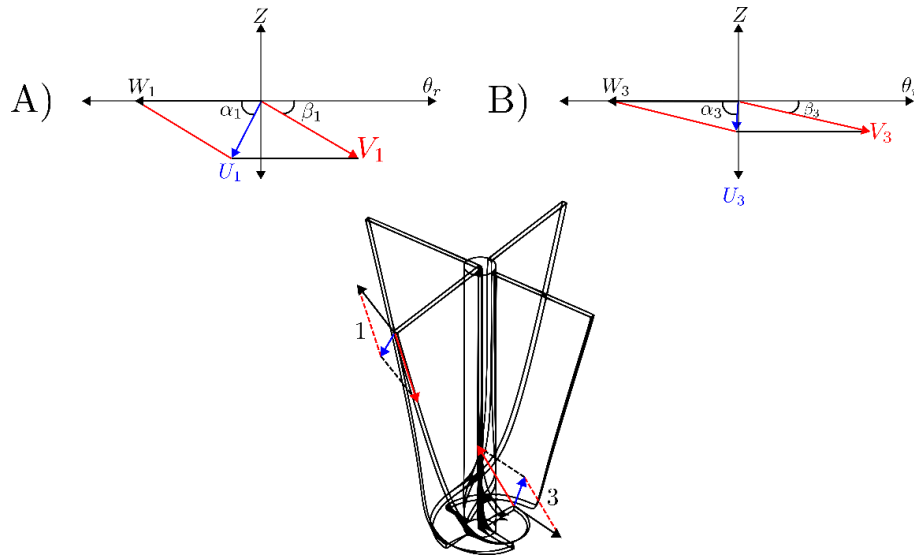


Figure 4. Velocity triangles at the inlet (a), inside the blade passage (b), and outlet (c) of the GVT rotor.

Table 2. Summary of velocity and flow equations.

Equation	Expression	Eq. No.
Velocity relationship	$U \cdot \cos \alpha + V \cdot \cos \beta = W$	(13)
Velocity components	$V \cdot \sin \beta = U \cdot \sin \alpha$	(14)
Flow rate equation	$Q = AU \cdot \sin \alpha$	(15)
Velocity magnitude	$V^2 = W^2 + U^2 - 2UW \cdot \cos \alpha$	(16)
Angular velocity	$W = \omega r$	(17)

$$H = \frac{4C(-h_3(\cos \alpha_3 + \cot \beta_3 \sin \alpha_3) \cos \alpha_3 + h_1(\cos \alpha_1 + \cot \beta_1 \sin \alpha_1) \cos \alpha_1)}{3H_1} \quad (13)$$

where i defines which height is used and can take the values of 1 or 3, referring to the distance from the free water surface at the chamber inlet to positions 1 or 3 as shown in **Figure 4**. The constant C is introduced to account for system losses. To reduce Equation (18) to independent terms only, it is necessary to express β_1 and β_3 in terms of α_1 and α_3 . This is done using Equations (14) and (15), which were calculated using the partial derivative of the parametric equation used for the description of the rotor.

$$\beta_1(a, \gamma) = \tan^{-1} \left(\frac{r\gamma}{h_r} \cdot \frac{\sqrt{\frac{4}{1+2e^a}-2}}{1+\sqrt{\frac{1}{1+2e^a}}} \right) \quad (14)$$

$$\beta_2(a, \gamma) = \tan^{-1} \left(\frac{r\gamma}{h_r} \cdot \frac{2+\sqrt{\frac{4}{1+2e^a}}}{\frac{1}{2}\sqrt{\frac{1}{1+2e^a}}-1} \right) \quad (15)$$

From the last mathematical formulation, an estimate for efficiency can be obtained by predicting the directional components of the water at the inlet of the rotor. This prediction can be developed using CFD simulation. Notwithstanding, unlike other turbines, the GVT

extracts energy from a flow that cannot be analyzed unidirectionally, making it necessary to predict vortex behavior in advance to estimate α_1 and, consequently, β_1 . Through CFD simulation, it is possible to predict the physical variables involved in the rotational flow within the chamber using finite element analysis and numerical approximations of the time-averaged Navier–Stokes equations (RANS) [21,22]. The tangential velocity of water inside the turbine is approximated by calculating the circulation (Γ) over the upper surface shown in **Figure 3** using Equation (16), developed in previous works [5,6].

$$\Gamma = \int \vec{\omega} \cdot d\vec{S} \quad (16)$$

2.2. Mesh Generation, Solver Configuration

To conduct a detailed numerical study, nine CFD simulations were carried out in ANSYS CFX, aiming to compare three different turbulence models (k - ε , SST, and BSL) using the GCI methodology [23]. The goal is to select the model that best fits the experimental conditions of the study. Both the geometry and the interfaces that define the basic simulation parameters (**Figure 5**) were kept constant, regardless of the turbulence model simulated [18,24]. Two servers from the Instituto Tecnológico Metropolitano (ITM) were used for this task; their technical specifications are shown in **Table 3**.

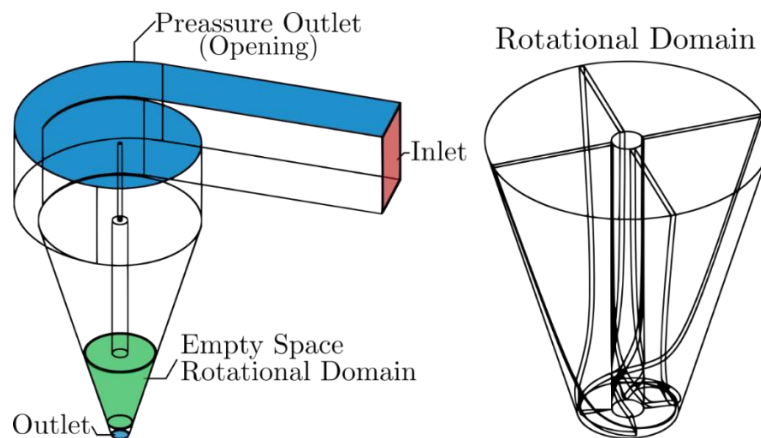


Figure 5. Domains for the CFD simulation of the GVT.

Table 3. Comparison of Computational Resources for CFD Simulations.

Specification	Server 1: CLUSTER	Server 2: NEFTIS
CPU / Processor		
Brand and Model	Intel Xeon E5-2690	Intel Xeon Silver 4216
Number of Cores	16	32
Clock Speed (GHz)	2.9	2.10
Number of Threads	32	64
RAM / Memory		
Total RAM Capacity (GB)	256	128
RAM Speed (MHz)	DDR3 1333	DDR4 1200
Operating System		
OS Name and Version	Windows 10	Windows 10
CFD Software / Version		
Software Name	ANSYS CFX	ANSYS CFX
Exact Version	ANSYS R2 2022	ANSYS R1 2024

The mesh used for the CFD simulations was generated using the ANSYS Meshing module, employing tetrahedral elements (**Figure 6**). To ensure the quality and reliability of the numerical results, different quality metrics were evaluated for the generated elements. **Table 4** presents the average values obtained for these metrics across the nine case studies [5].

Table 4. Mesh quality metrics for different levels of refinement.

Mesh Level	Min Orthogonal Quality	Avg Orthogonal Quality	Max Skewness	Avg Skewness	Max Aspect Ratio	Avg Aspect Ratio
Fine	0.09	0.75	0.9	0.24	14	1.85
Medium	0.11	0.82	0.9	0.28	15	1.83
Coarse	0.13	0.82	0.9	0.23	14.2	1.84

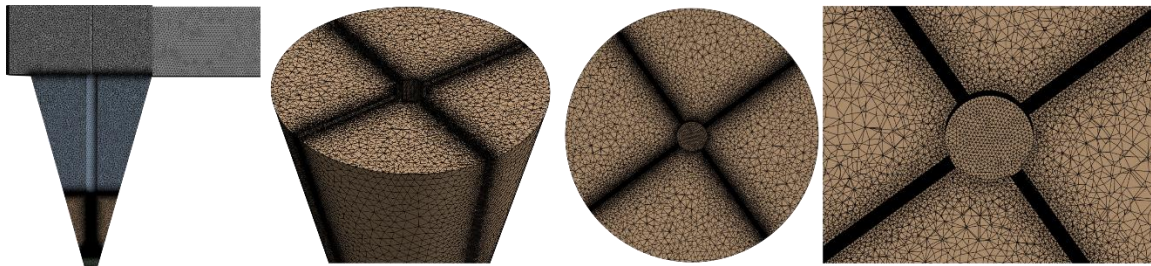


Figure 6. Mesh on the control volume.

The parameters used for configuring the CFD simulations in ANSYS CFX are summarized in **Table 5**. These parameters were carefully selected to accurately reflect the experimental conditions described in the previous section, thus ensuring proper numerical convergence in all analyzed cases. Additionally, the results obtained with the three evaluated turbulence models will be contrasted using the mesh independence methodology (GCI), ensuring the quality and reliability of the numerical study conducted [25].

Table 5. Parameters used in the CFD simulation configuration.

Parameter	Value
Inlet velocity [m/s]	0.0539
Angular velocity [rad/s]	11
Reference pressure [atm]	1
Outlet pressures [atm]	0
Time step	0.01
Density (ρ) [kg/m ³]	998.2
Gravitational acceleration (g) [m/s ²]	9.81
Average Courant number	12
Surface tension [N/m]	0.072
Rotor interface type	Frozen Rotor
Pitch angle	360

2.3. Experimental Methodology

2.3.1. Experimental setup and parameters

For the development of the experimental tests, a setup was assembled in the GEA laboratory using available components (**Figure 7**). The system is primarily composed of an inlet channel and a discharge chamber made of transparent acrylic to allow visualization of vortex formation, specific sensors for accurate measurement of flow rate (SITRANS F M MAG

5100W) and torque (FUTEK USB 520/530), and a specially designed runner manufactured through 3D printing [26]. It is worth noting that the rotor was printed using a Crealiti Ki printer with PLA filament, 1.75 mm in diameter, 0.2 mm layer height, and 100% infill. The printing parameters included an extruder temperature of 220 °C and a bed temperature of 45 °C, ensuring the dimensional accuracy and surface quality required for the experimental tests. During the tests, a constant water flow rate of 2.5 and 3 L/s was maintained. The rotor's dynamic variables, including speed and torque, were recorded over 30 seconds after the flow reached a steady state (**Figure 8**). The key components of the experimental setup are summarized in detail in **Table 6** [27].

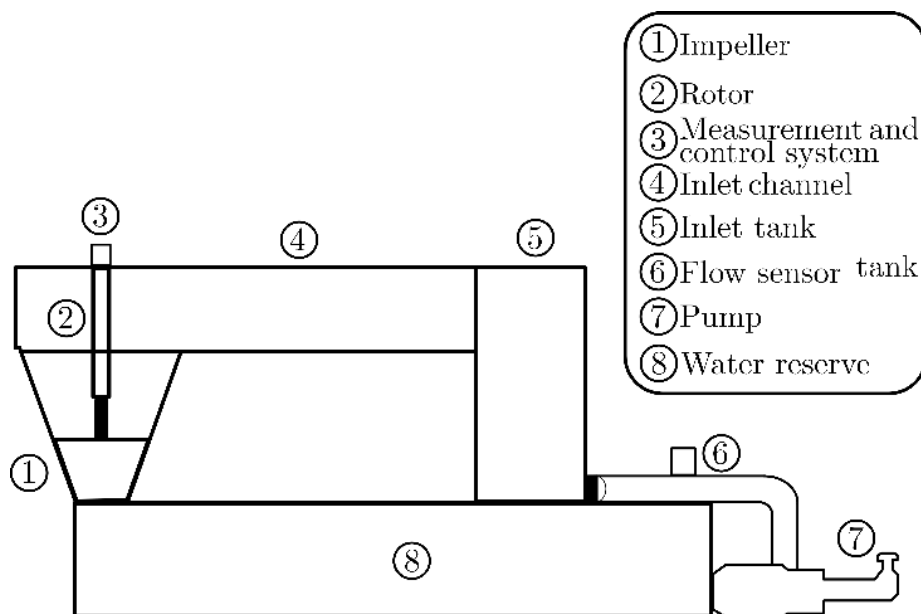


Figure 7. Elements of the experimental setup.



Figure 8. Experimental facilities.

Once the experimental setup was defined and validated, a comparison with the numerical study was carried out. The specific methodology and characteristics for this analysis and its results will be detailed in the following section.

Table 6. Components of the Measurement and Control System and Hydraulic Elements.

Element	Sub-Element	Description
Measurement and Control System	Flow Sensor	SITRANS F M MAG 5100W
	Torque Sensor	FUTEK USB 520/530
	CPU and Software	SENSIT software
	Dimensions	Structures for controlled water routing, made of 5 mm-thick transparent acrylic
Inlet Channel and Inlet Tank		Height: 250 mm Width: 250 mm Length: 1250
Water Reserve Tank		Capacity of 2 m ³
Pump		IHM 30A-15W – IE2, controlled with a variable frequency drive

2.3.2. Calculation of bearing losses

Due to the conditions of the experimental setup, specifically the state of the bearings coupled to the rotor shaft, it is necessary to consider friction losses when comparing the experimental results with the numerical data [28]. To quantify these losses, the experimental setup shown in **Figure 9** was used. In these tests, a constant speed of 100 RPM was maintained, equivalent to the simulation speed, and the rotor deceleration was recorded after interrupting the airflow. Thanks to the torque sensor, it was possible to determine the angular acceleration from the variation in angular velocity over time [29].

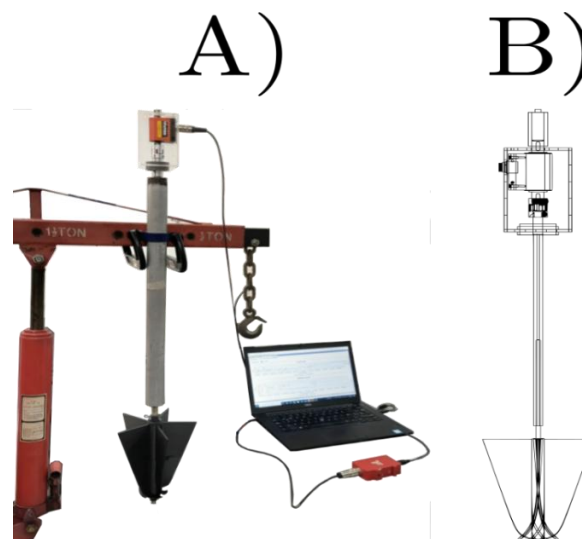


Figure 9. Experimental setup for determining bearing losses. (A) Physical assembly used for testing, and (B) Schematic diagram of the measurement system.

The angular acceleration is obtained from Equation (17).

$$\alpha = \frac{\Delta\omega}{\Delta t} \quad (17)$$

where $\Delta\omega$ is the change in angular velocity and Δt is the time interval between measurements. The moment of inertia of the shaft, modeled as a solid cylinder of mass m , length L , and radius R , is expressed as Equation (18).

$$I = \frac{1}{12}mL^2 + \frac{1}{4}mR^2 \quad (18)$$

The friction torque is calculated by multiplying this moment of inertia by the absolute value of the angular acceleration (Equation (19)).

$$T_{Fric} = I|\alpha| \quad (19)$$

Finally, the power lost due to friction at 100 RPM is determined by Equation (20).

$$P_{Fric} = T_{Fric}\omega_{100} \quad (20)$$

Table 7 summarizes all the experimental parameters needed to quantify friction losses in the rotor shaft bearings [30]. This corrected torque value will be used for comparison between experimental and numerical results. It is emphasized that, due to the state of the bearings, this value is essential for accurately describing the experimental results, as friction losses in the bearings account for approximately 11 % of the measured net torque. In this way, a corrected experimental torque of 0.664 Nm is obtained, which is essential for accurately accounting for the losses induced by the condition of the bearings.

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Table 7. Experimental parameters and calculated values to quantify friction losses.

Parameter	Symbol	Value	Units
Rotor shaft mass	m	0.895	kg
Moment of inertia	I	$6.042e - 02$	kgm^2
Average angular acceleration	α	-1.102	$\frac{rad}{s^2}$
Friction torque	T_{Fric}	0.066	Nm
Power loss	P_{Fric}	0.697	W
Net experimental torque	T_{ExpN}	0.597	Nm
Corrected experimental torque	T_{ExpC}	0.663	Nm

3. RESULTS AND DISCUSSION

This section presents both the numerical analyses and the efficiency evaluation. First, the numerical results obtained from simulations using three turbulence models ($k-\varepsilon$, SST, and BSL) are detailed, with emphasis on the prediction of rotor torque; this is achieved through the evaluation of convergence using the Richardson extrapolation method and the calculation of the GCI [31]. This analysis allows the identification of differences in stability and result dispersion among the various models. On the other hand, the efficiency of the turbine is addressed through a direct comparison between numerical and experimental values, describing the methods used to calculate the available power and the shaft power [5], which are essential to determine the system's performance, reaching a maximum experimental value close to 50.6%.

3.1. Numerical and GCI results

Once the configuration parameters for the numerical simulations were defined, the cases were executed for the three turbulence models considered (k - ϵ , SST, and BSL). The results obtained for rotor torque in the time interval between 34 and 37 seconds are summarized in **Table 8** and illustrated in **Figure 10**. In the finest mesh, composed of 9,974,357 elements, the BSL model yields the highest torque value at 0.73 Nm, while the k - ϵ model records the lowest value of 0.613 Nm. The SST model presents intermediate values with 0.661 Nm. This trend remains consistent across the different mesh configurations evaluated, considering in all cases a constant inlet velocity of 0.0539 m/s.

Table 8. Torque results for different mesh configurations and turbulence models.

Elements	K- ϵ	SST	BSL
9,974,357	0.613	0.661	0.73
3,610,398	0.604	0.632	0.62
1,295,088	0.598	0.577	0.612

The analysis using the Richardson extrapolation method enables the evaluation of numerical convergence and stability for each model in torque prediction, based on the calculation of the Grid Convergence Index (GCI) for fine and coarse meshes. In the case of the k - ϵ model, very close values are observed, with a GCI of 7.341% in the fine mesh and 7.450% in the coarse mesh, indicating stable and consistent convergence. In contrast, the SST model exhibits significantly higher indices, with 38.124% for the fine mesh and 39.873% for the coarse mesh, suggesting greater result dispersion. The BSL model presents the highest GCI values, reaching 48.493% on the fine mesh and 57.097% on the coarse mesh (**Figure 11**), indicating strong mesh dependence and potential convergence issues by this criterion [6].

However, when considering only the minimum relative error within the experimental uncertainty range, the BSL model shows the best performance, with an error of just 4.6%, followed by the SST model at 9.1%, and the k - ϵ model at 11.7%. These results suggest that, under ideal conditions within the uncertainty band, the BSL model approximates the corrected experimental torque value of 0.6025 Nm more closely than the others. Nonetheless, this agreement does not guarantee the model's reliability, as its high GCI indicates a strong sensitivity to mesh resolution, raising concerns about the robustness and generalizability of its predictions under varying simulation conditions (**Figure 12**).

Therefore, these findings underscore the importance of jointly considering both convergence indicators and direct comparison with experimental data. A seemingly accurate prediction may be misleading if it is not supported by a numerically stable and mesh-independent solution [32].

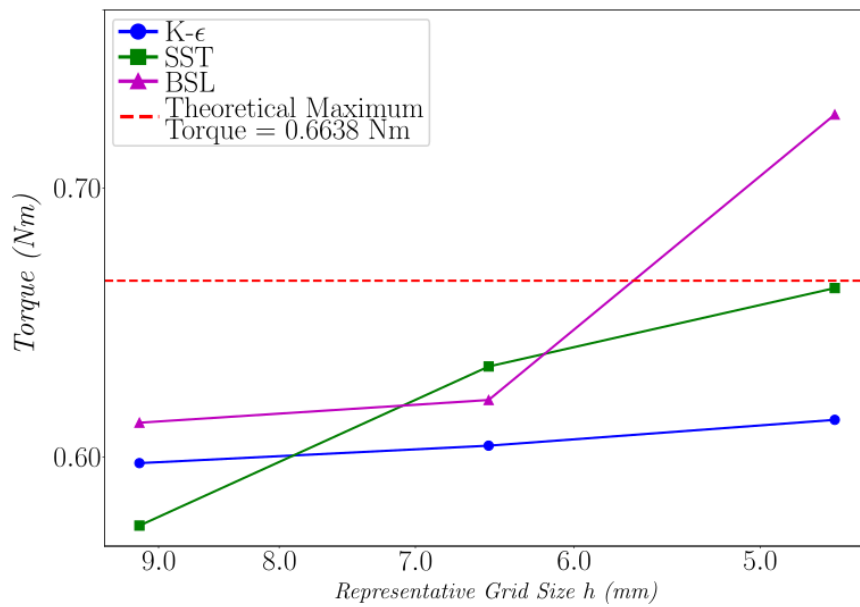


Figure 10. Numerical results by model.

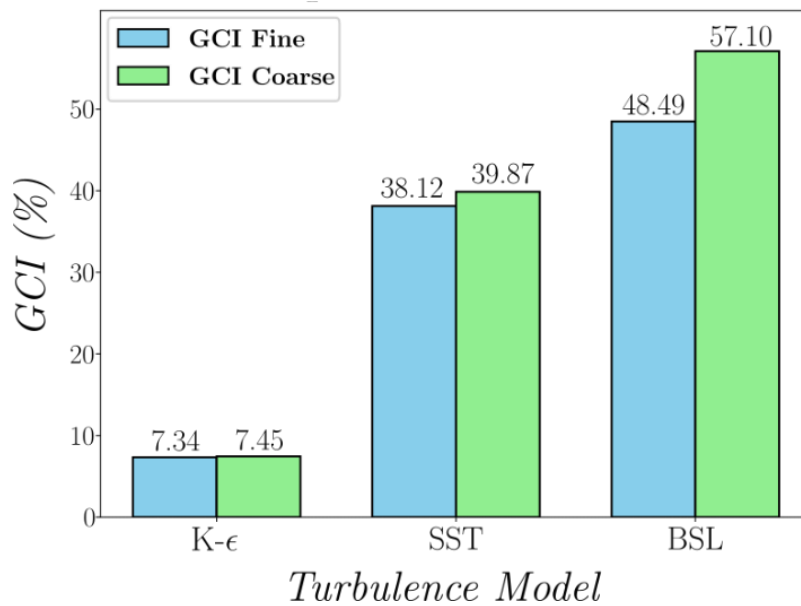


Figure 11. Richardson extrapolation method and Grid Convergence Index.

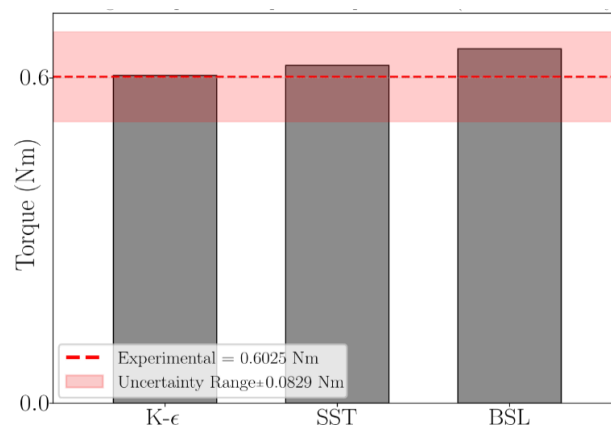


Figure 12. The average value of the numerical results of each model is relative to the experimental value.

3.2. Experimental Results

Table 9 summarizes the main findings from four experimental tests (two performed at a flow rate of 2.5 L/s and two at 3.0 L/s). Each row presents the maximum power measured during the experiment, the angular speed at which the maximum torque was observed, and the best efficiency predicted by the correlation equations derived from the experimental data. These correlation equations obtained by fitting the relationship among torque, power, angular speed, and flow rate are valid exclusively for the tested conditions. Their purpose is to relate the useful power (P_U) and the hydraulic power (P_H). Thus, the efficiency η can be predicted within the measured operating range (Equation (21)).

$$P_U = T\omega \quad (21)$$

where ω is the angular speed of the rotor and T is the torque (Equation (22)).

$$P_H = \rho g H_T Q \quad (22)$$

where the total head is $H_T = 0.488$ m, and all other relevant hydraulic and operational parameters such as fluid properties, flow conditions, and rotor speed are summarized in **Table 5**. The flow rate (Q) is derived from the inlet velocity and cross-sectional area, as detailed therein. A representative form of the power speed correlation used in these tests, for instance, can be written as Equation (23).

$$\tau = a + \frac{b}{\omega^c} \quad (23)$$

where τ is the torque, and ω is the angular speed. By substituting τ into (Equation (24)).

$$P = \omega\tau \quad (24)$$

We obtained Equation (25):

$$P(\omega) = \left(a + \frac{b}{\omega^c} \right) \omega \quad (25)$$

This expression shows how the power output depends on the rotor's angular speed. The constants a , b , and c were determined from experimental data and are valid only within the specific range of flow rates (2.5 - 3.0 L/s) and the geometrical parameters tested. The values obtained from the best-performing case are: $a = 0.6927$, $b = -0.6465$, and $c = -2.0833$. Test 3 in **Table 9** exhibits the best performance among the four tests, reaching the highest measured power of 13.262 W at a flow rate of 3.0 L/s. Moreover, for an angular velocity of 11 rpm, the measured torque in this test was 0.6025 Nm, with an associated uncertainty of ± 0.0829 Nm. This torque value should be taken as the representative experimental result under these operating conditions.

Table 9. Test results of experiments.

	Flow rate (L/s)	Maximum Measured Power (W)	Angular speed of max. measured torque (rad/s)	Best Efficiency (correlation)	Torque at max. power (N.m)	Torque at 11 rpm (N.m)
1	2.5	10.089	16.650	0.399	0.470	0.556
2	2.5	10.347	17.279	0.390	0.433	0.523
3	3	13.262	21.468	0.347	0.463	0.597
4	3	10.863	17.907	0.337	0.469	0.596

3.3. Comparative Discussion

To contrast the numerical results with the experimental ones, **Figure 13** shows the efficiency percentage (η) for the turbine design. The efficiency is calculated using Equation (26).

$$\eta = \frac{P_o}{P_H} 100\% \quad (26)$$

where P_o is the shaft power and P_H is the available power. With these parameters, the overall performance of the turbine is estimated, whose maximum measured value is approximately 50.6%. In contrast, the numerical simulations carried out showed efficiencies ranging around similar values (48-52%, depending on the turbulence model and mesh refinement). Although the *BSL model* presented a relatively high Grid Convergence Index (GCI), it ended up providing torque and efficiency results closer to the experimental data than other models (e.g., *k- ϵ*). This behavior can be attributed to the BSL model's better capability to represent the fluid dynamic behavior occurring in the vortex chamber, even when the dispersion according to the GCI is greater. The main discrepancies between numerical and experimental values can be explained by various causes, such as vortex and free surface flow, experimental limitations, or assumptions in the simulation setup. Nevertheless, the fact that the BSL model better reproduces the results, despite its higher GCI, highlights the relevance of comparing the physical accuracy of simulations with experimental data, in addition to analyzing mesh convergence. In other words, a favorable GCI does not necessarily guarantee the most accurate prediction, especially in flows with high circulation.

3.4. Discussion and Future Work

The GVT is reaffirmed in this study as a promising technology for small-scale renewable energy generation, particularly in geographically remote or underserved areas where conventional electrification infrastructure is limited or economically unfeasible [33,34]. Its design simplicity, low ecological footprint, and adaptability make it well-suited for decentralized energy applications. Within this context, optimizing hydrokinetic turbine components is essential for maximizing energy conversion efficiency from natural water flows. This research confirms that rotor geometry, specifically when derived from parametric curves and adjusted to circulation characteristics, plays a decisive role in determining overall turbine performance [35,36].

The proposed rotor configuration demonstrated improved hydraulic efficiency in both numerical simulations and physical testing, with performance results aligning well between experimental measurements and CFD predictions [37]. This alignment reinforces the effectiveness of integrating computational modeling with empirical validation. Moreover, the analysis revealed that physical constraints in the test setup (especially bearing friction) had a notable impact on performance, highlighting the importance of mechanical design considerations in real-world implementations.

From a broader perspective, these findings directly contribute to the United Nations SDGs. The advancement of clean and affordable energy technologies aligns with SDG 7, while the application of computational and experimental innovation supports SDG 9. Furthermore, by promoting the use of localized, non-polluting energy systems that reduce dependency on fossil fuels, this work contributes meaningfully to SDG 13 on climate action. Indeed, this study adds new information regarding support for sustainable energy and SDGs as reported elsewhere [38-55].

Future work should focus on holistic optimization of the entire GVT system. This includes redesigning additional components such as the inlet channel, the vortex chamber, and the

outlet to ensure seamless integration with the rotor's geometric characteristics. Implementing multi-variable optimization algorithms (based on circulation, vorticity, and turbulence modeling) could enable more precise control of internal flow behavior and further enhance energy recovery. In addition, real-world deployment studies in off-grid communities could help validate long-term performance, reliability, and social impact, further solidifying the role of GVTs as practical contributors to sustainable energy infrastructure.

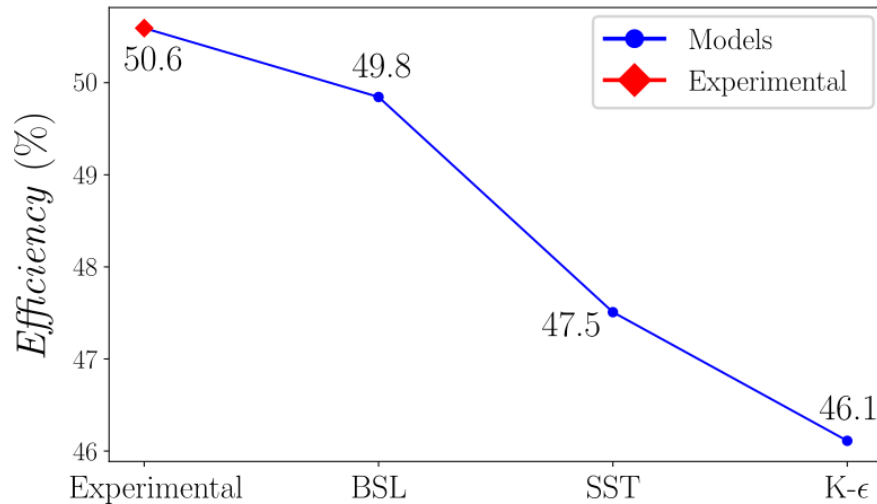


Figure 13. Comparison of experimental efficiency with numerical models.

4. Conclusion

This study proposed and validated a novel rotor design for a GVT using a combined approach of parametric modeling, CFD, and experimental testing. The proposed rotor, designed through mathematical parameterization to optimize blade curvature and flow interaction, achieved a maximum experimental efficiency of 50.6%, which is competitive with or superior to values reported in prior studies. Numerical simulations using three turbulence models (k - ϵ , SST, and BSL) were evaluated, with the BSL model demonstrating the closest agreement to experimental results despite exhibiting higher sensitivity to mesh resolution. These findings highlight the importance of not only ensuring numerical convergence but also validating model predictions against experimental data. The corrected torque and efficiency values accounted for bearing friction losses, ensuring accurate performance assessment. Overall, the results confirm that optimized rotor geometry significantly enhances energy conversion in GVT systems. Future work should focus on the integrated optimization of the rotor, inlet channel, and tank geometry, as well as the application of advanced optimization algorithms based on circulation and vorticity parameters. This study supports SDGs.

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6. AUTHORS' NOTE

The authors declare that there is no conflict of interest regarding the publication of this article. The authors confirmed that the paper was free of plagiarism.

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