

Article

Numerical Analysis of a Hybrid Turbine with Partial-Height Blades: Performance Gains Beyond Viscous Gap-Reduction

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Abstract

The Tesla turbine operates on viscous shear between parallel discs and, despite its mechanical simplicity, is typically characterized by low efficiency. In the present study, three-dimensional computational fluid dynamics (CFD) simulations performed using ANSYS Fluent are used to examine a hybrid Tesla turbine design in which 0.25 mm thick partial height blades are fitted on the disc faces, with 1 mm distance between them, thereby creating a 0.5 mm flow passage. Simulations employing the k- ω Shear Stress Transport (SST) turbulence model were performed for three blade counts (3, 6, and 9) and three blade geometries (curved, straight, and inverted curve) at rotational speeds from 1000 to 19,000 rpm and inlet pressures of 2 to 4 bar. Comparative analyses with standard 1 mm plane-disc rotors and reduced-gap 0.5 mm plane-disc rotors show that the hybrid arrangement consistently provides better torque and efficiency; this enhancement is not only due to the reduced gap but also to increased pressure-induced momentum and improved flow guidance provided by the blades. The curved blade was found to be the most favourable configuration, and the efficiency was positively related to the number of blades, with a maximum efficiency of 57.5% at 13,000 rpm using nine blades. The analyses sustain the conclusion that adding blades to the rotor discs positions the Tesla turbine model as a hybrid apparatus, combining viscous and pressure mechanisms to significantly enhance turbine performance.

Keywords: Tesla turbine; computational fluid dynamics; bladed-disc interaction; efficiency; power; renewable energy systems



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1. Introduction

Modern energy transition policies, such as the European Green Deal [1,2], increase the demand for new turbomachines that are not only efficient but also low-cost, robust, and flexible. Traditional axial and radial turbines are well developed; however, they are composed of highly complex elements, are expensive to manufacture, and have a restricted range of operation depending on the specific type of turbine [3]. This makes them less ideal for applications like microgeneration, low-temperature heat recovery, and adapting to the variable outputs of renewable energy sources.

The Tesla turbine, patented in 1913, features a simpler mechanism with fewer components compared to conventional turbines [4]. Its design, which utilizes a bladeless

rotor composed of parallel discs, extracts energy through viscous stress on the surfaces of the rotor discs [5]. While this makes it exceptionally robust, its performance is often limited; furthermore, a lack of investment and research has historically hindered its adoption compared to conventional pressure-driven turbines. Later designs employed steam as the working fluid, and reports indicate that one of Tesla's designs produced 22.3 kW at 20,000 rpm, although it could deliver a higher output. Tesla claimed to have built a turbine producing 149,000 watts (200 horsepower) with 25 discs of 0.457 m in diameter, rotating at 9000 rpm, using heated steam as the working fluid. The thermal efficiency of this machine was 19%, and its mechanical efficiency was 75%. Tesla posited that it would be possible to achieve even larger outputs by utilizing superheated steam and significantly higher pressures [4].

The absence of blades makes the turbine particularly appealing for low-enthalpy cycles, organic Rankine cycle (ORC) systems, and compressed-air microgeneration, where operating flexibility and sustainability are the most important factors [6–8]. Although Tesla turbines may be less efficient under conventional operating conditions than their blade-driven equivalents, recent studies reveal that significantly increased efficiency is possible through geometric optimization, proper selection of the number of discs and disc spacing, improved nozzle design, and control of operating conditions. The theoretical and experimental basis of the multi-disc turbine was laid by Rice et al. [9,10]. They showed that their analytical model described the velocity distribution and the flow behaviour inside the disc gaps, and in the optimum efficiency regime, the laminar-to-transient regime, viscous shear is dominant in the flow. They also emphasized the important effect of disc spacing and the number of discs on overall efficiency and pointed out that small changes in these parameters can result in big variations in energy conversion efficiency. Allen et al. [11] provided a comprehensive theoretical overview which incorporated equations for both momentum and energy balances, incorporating hydrodynamic and mechanical behaviours. The authors confirmed that the torque is inversely proportional to the square of the angular velocity.

As a result of advances in computational fluid dynamics (CFD), it is now possible to define the internal mechanisms that cause energy losses more accurately. A three-dimensional study of the flow inside the rotor was conducted by Ciappi et al. [12], which revealed recirculation zones and regions of non-uniform pressure gradients that contribute to viscous dissipation and the degradation of effective torque. Furthermore, Galindo et al. [13] provided quantitative measurements showing that between 64% and 88% of total energy dissipation stems from the inefficient conversion of pressure to kinetic energy in the rotor-stator system, thereby highlighting the importance of geometric and aerodynamic optimization. Rusin et al. [14] described an experimental and numerical study of a rotor composed of five 160 mm diameter discs spaced 0.75 mm apart, fed by compressed air with pressure ratios ranging from 1.4 to 1.88. The authors found a peak efficiency of approximately 10.3%, which was highly dependent on the mass flow rate and losses in the side clearances. Later, in 2021, Rusin et al. [15] used response surface optimization methods to achieve an efficiency increase of up to 100%, proving that substantial improvements are possible when structural and operational parameters are optimized simultaneously. Regarding small Tesla turbines and miniaturization, Qi et al. [16] showed that, all other factors being equal, a smaller turbine diameter leads to a higher isentropic efficiency. Another important consideration for overall performance is the injector design and angle. Lampart et al. [17] conducted CFD simulations to explore various tangential injector designs and determined that four inlets on 100 mm discs are the most efficient and stable in terms of flow. Neckel et al. [18] examined convergent-divergent nozzles and found that an outlet-to-throat ratio of approximately 2.1 minimizes shock wave formation, reduces pressure losses, and improves

energy conversion. Pahlavanzadeh et al. [19] assessed one-to-one and one-to-many nozzle configurations using Large Eddy Simulation (LES), featuring physical separation between each gap. They found that the former favours a uniform and stable flow, thereby leading to higher isentropic efficiency.

Beyond geometrical considerations, turbine design must also account for material properties such as surface finish and roughness. Given that its mode of operation relies on viscous friction between the working fluid and the encountering surfaces, surface roughness significantly influences momentum transfer. Rusin et al. [20] developed an extended analytical model that considers the velocity profile in microchannels with controlled roughness, showing that roughness on the order of 10^{-5} m could enhance the generated power and isentropic efficiency by up to 2.5 times. Pahlavanzadeh et al. [21] experimentally confirmed the porous layer model (PLM) by achieving an efficiency gain of about 4.6% at 22,000 rpm with a roughness of 60 μm . Nozzle geometry and seal enhancements were also experimentally studied by Mandal et al. [22] and Tiwari et al. [23], who discovered that nozzle seals of the type used by MTUs could reduce side losses by a factor of four and improve total efficiency by between 5% and 8%. Sanchez [24] and Klingl [25] demonstrated that further power and efficiency improvements can be achieved by combining optimal roughness and sealing with compressed air, making it a promising and sustainable alternative fluid for microgeneration. There is also growing interest in the use of Tesla turbines for thermal systems. Aghagoli et al. [26] substituted expansion valves with Tesla turbines in transcritical CO_2 cycles and found that they provided a 9.8% increase in the coefficient of performance (COP) and a 28.9% increase in exergetic efficiency. Talluri et al. [27,28] utilized the turbine in organic Rankine cycles (ORCs) with the fluid R1233zd(E) and reported an efficiency of over 60% at unit Mach conditions, whereas Sheikhnejad et al. [29] demonstrated its viability in refrigeration systems. Kachawong et al. [30] examined operations with saturated steam in ORCs, achieving an efficiency of 91% and demonstrating the effectiveness of the turbine under low enthalpy. Recent advances are also investigating the use of computational optimization and artificial intelligence [31]. Renuke et al. [32] proposed a zero-dimensional (0-D) prototype design utilizing non-dimensional parameters like the Reynolds and Ekman numbers, whereas Lisker et al. [33] compared various working fluids and found that steam yields the highest efficiency at 94.9%, followed by helium and air, both of which are suitable for domestic microgeneration.

Other recent research has focused on the use of blades in Tesla turbines, with the aim of enhancing momentum transfer and optimizing internal flow. The use of new materials and improvements in manufacturing accuracy have enabled the reinvention of the Tesla turbine. This includes the incorporation of modified surfaces and hybrid profiles, such as guide vanes implemented as blades at the disc extremities that span the entire gap width. This is illustrated by the research of Podergajs et al. [34], who cited a similar hybrid Tesla-Pelton design patented by Fuller in 2010. They discussed its potential for microgeneration, where small peripheral blades attached to the discs combine the viscous action of the Tesla design with the characteristics of impulse turbines, achieving higher torque and efficiency under variable flows. Similarly, Joshi et al. [35] presented a three-dimensional CFD numerical model of a Tesla turbine operating with compressed air. In their study, blades and internal channels added to the discs facilitated improved streamline organization, reduced recirculation regions, and substantially increased the conversion of pressure energy into kinetic energy, yielding almost twice the power generation. These developments support the use of such blades as a viable evolution of Tesla's original concept, as they can improve efficiency without compromising the turbine's inherent constructive simplicity, thereby broadening its applicability. However, in all cases where blades were previously studied, the blades

essentially acted as walls, blocking the inter-disc passages and forcing the flow to impinge directly upon them.

The main aim of this research is to optimize the performance of a Tesla turbine by inserting blades which direct the flow between the rotor discs based on previous investigations. This paper is designed to compare the geometric and aerodynamic characteristics of these blades with the fluid mechanics and performance of the turbine, including such variables as torque, efficiency, pressure distribution, entropy generation and internal flow. For this purpose, three-dimensional simulations will be performed in the software ANSYS Fluent 2025. The high resolution $k - \omega$ SST turbulence model will be used. This model is very useful in the simulation of complex flow phenomena such as flow separation and recirculation zones. Three types of blade design will be tested using different curves and angles of the blade. The goal is to find the design that helps to transfer momentum as much as possible without changing the basic principle that the Tesla turbine is based on, namely, viscous shear between parallel discs. Finally, the aim of the research is to provide guidelines which will allow higher turbine efficiencies and power output for compressed-air and other micro-generation systems, thereby helping the development of compact and high-performance turbomachinery.

2. Turbine Description and Boundary Conditions

The geometry analyzed in this study corresponds to a single-stage Tesla turbine composed of ten parallel rotor discs arranged within a cylindrical casing. A three-dimensional view of the turbine assembly is shown in Figure 1. The working fluid is injected into the system through a curved nozzle that directs the flow tangentially into the inlet chamber, thereby inducing rotor rotation via viscous forces acting along the disc surfaces.

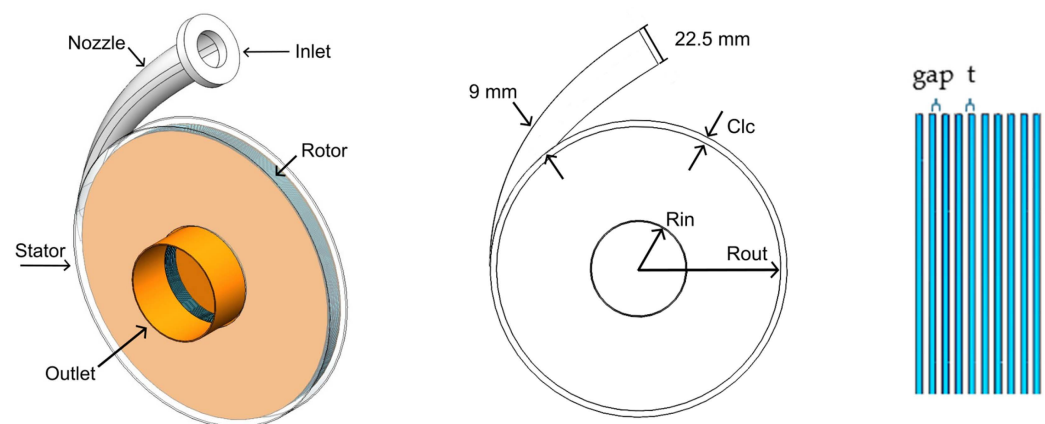


Figure 1. Schematic and dimensions of the Tesla turbine.

To reduce computational costs without compromising physical accuracy, a simplified single disc-gap model was adopted. Due to the geometric and fluid-dynamic symmetry between the inter-disc channels, the computational domain was restricted to a single representative disc. Symmetry boundary conditions were applied to the lateral faces to replicate the periodic flow behaviour of the full rotor. The boundary conditions were specified as follows: the turbine inlet and outlet were defined as a pressure inlet and a pressure outlet, respectively. Additionally, all solid surfaces were treated as adiabatic, no-slip walls. The specific parameter values are listed in Table 1.

The straight and curved-bladed discs are shown in Figure 2; the blade height was set to 0.25 mm and the blade thickness to 1 mm for all blade configurations.

The results obtained from the single-disc simulation were extrapolated to the full set of ten discs, allowing for the estimation of overall turbine performance, including torque, shaft

power, and efficiency. However, it should be noted that this approach does not account for interactions between multiple discs or casing and outlet losses; therefore, it represents an idealized rotor configuration. This simplification is widely adopted in other numerical studies of Tesla turbines [15,30], as it preserves the physical fidelity of the flow behaviour while significantly reducing computational effort.

Table 1. Dimensions of the Tesla turbine for the base case.

Parameter	Unit	Value
Inlet Pressure	bar	2, 3, 4
Inlet temperature	C	20
Rotational speed	rpm	1000–19,000
Stator diameter (St)	mm	276.15
Outer Disc Radius (Rout)	mm	135.0
Inner Disc Radius (Rin)	mm	45
Disc gap (gap)	mm	1
Nozzle exit (Ne)	mm	9
Disc Thickness (t)	mm	1
Number of discs (N)	-	10
Clearance (clc)	mm	3.15

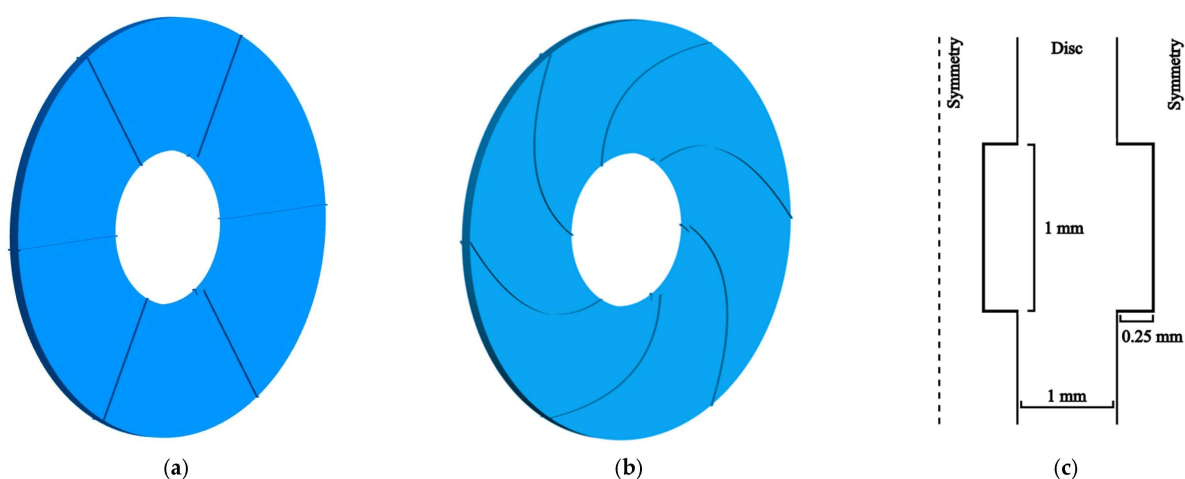


Figure 2. Schematic of the bladed disc: (a) straight blade; (b) curved blade; (c) blade configurations.

3. Numerical Model

In the present study, the Shear Stress Transport (SST) k - ω turbulence model was selected, as it provides reliable predictions of near-wall shear gradients, boundary-layer behaviour, and flow separation under strong adverse pressure gradients. The SST model combines the advantages of the k - ϵ formulation in the free-stream region with the robustness of the SST k - ω model for boundary layers through a set of blending functions. It is widely applied in turbomachinery simulations, where accurate prediction of shear-driven boundary layers, wall shear stresses, and flow separation phenomena is essential for reliable performance evaluation. Although the SST k - ω model combined with the MRF approach does not explicitly resolve transient rotor–stator interactions, it provides a computationally efficient framework for predicting the time-averaged flow behaviour and comparing the relative performance of different rotor geometries. Since the objective of the present study is to evaluate the mean performance characteristics and identify the dominant energy transfer mechanisms of the hybrid Tesla turbine, rather than analyzing transient flow phenomena, the steady-state approach provides an appropriate methodology for the present investigation.

Numerical simulations were performed in ANSYS Fluent utilizing the Moving Reference Frame (MRF) approach to model rotor motion [36]. The MRF method provides a steady-state representation of the rotating flow field, allowing efficient prediction of time-averaged turbine performance while maintaining reasonable computational cost. The working fluid was modelled as a Newtonian ideal gas, with its temperature-dependent thermophysical properties derived using NASA 9-coefficient thermodynamic polynomials; additionally, enthalpy and thermal conductivity were computed as proposed by McBride et al. [37]. The continuity equation, which expresses the conservation of mass, is given by

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathbf{u}_r) = 0, \quad (1)$$

where ρ represents the density, t represents time, and $\mathbf{u}_r$ represents the velocity vector in the rotating frame. The equation for linear momentum in the rotating frame of reference is given by

$$\frac{\partial(\rho \mathbf{u}_r)}{\partial t} + \nabla \cdot (\rho \mathbf{u}_r \mathbf{u}_r) = -\nabla p + \nabla \cdot \Gamma_{eff} - 2\rho(\mathbf{w} \times \mathbf{u}_r) - \rho(\mathbf{w} \times (\mathbf{w} \times \mathbf{r})), \quad (2)$$

where p is the static pressure, $\mathbf{w}$ is the angular velocity vector, and $\mathbf{r}$ is the position vector. The last two terms represent the Coriolis and centrifugal forces, respectively [24].

The effective stress tensor is $\Gamma_{eff} = \Gamma_l + \Gamma_t$.

For Newtonian fluids, the laminar viscous stress tensor is defined by

$$\Gamma_l = \mu \left(\frac{\partial u_{r,j}}{\partial x_i} + \frac{\partial u_{r,i}}{\partial x_j} \right) - \left(\frac{2}{3} \right) \mu (\nabla \cdot \mathbf{u}_r) \delta_{ij}, \quad (3)$$

where μ is the molecular viscosity and δ_{ij} is the Kronecker delta.

The Reynolds stresses are modelled using the Boussinesq hypothesis for the turbulent component:

$$\Gamma_t = -\overline{\rho u'_{r,i} u'_{r,j}} = \mu_t \left(\frac{\partial u_{r,i}}{\partial x_j} + \frac{\partial u_{r,j}}{\partial x_i} \right) - \left(\frac{2}{3} \right) \rho k \delta_{ij}. \quad (4)$$

The equation for conservation of total energy is written as follows [14]:

$$\frac{\partial(\rho E)}{\partial t} + \nabla \cdot [\mathbf{u}_r(\rho E + p)] = \nabla \cdot (k_e \nabla T) + \nabla \cdot (\Gamma_{eff} \cdot \mathbf{u}_r), \quad (5)$$

where E is the total specific energy defined by $(E = e + \frac{1}{2} |\mathbf{u}_r|^2)$, T is the temperature, k_e is the effective thermal conductivity.

The Shear Stress Transport (SST) k - ω model was selected, the transport equations for turbulent kinetic energy k and the specific dissipation rate ω are given by

$$\frac{\partial(\rho k)}{\partial t} + \nabla \cdot [\rho \mathbf{u}_r k] = \left[\nabla \cdot \left(\mu + \frac{\mu_t}{\sigma_k} \right) \nabla k \right] + P_k - \beta^* \rho k \omega, \quad (6)$$

$$\frac{\partial(\rho \omega)}{\partial t} + \nabla \cdot [\rho \mathbf{u}_r \omega] = \left[\nabla \cdot \left(\mu + \frac{\mu_t}{\sigma_k} \right) \nabla k \right] + \alpha \frac{k}{\omega} P_k - \beta \rho \omega^2 + CD_{k\omega}, \quad (7)$$

where P_k is the turbulent production term, $CD_{k\omega}$ is the cross-diffusion correction, and the empirically calibrated constants are σ_ω , c , β and β^* . The turbulent viscosity is obtained by

$$\mu_t = \rho \frac{c_1 k}{\max(c_1 \omega, SF_2)}, \quad (8)$$

where S is the shear rate and F_2 the characteristic mixing function of the SST formulation.

Compressible modelling enables the accurate capture of density variations, pressure gradients, and viscous dissipation, crucial phenomena in the injection throat regions and between the discs, where the Mach number can approach or exceed unity. This approach aligns with the recommendations of Versteeg and Malalasekera [38].

The mechanical power is expressed as

$$P_{exit} = T_q \cdot \omega, \quad (9)$$

where T_q is the total torque on the rotor and ω is the angular velocity.

The total-to-static efficiency is determined by

$$eff = \frac{P_{exit}}{\dot{m} C_p T_i [1 - (p_o/p_i)^{(\gamma-1/\gamma)}]}, \quad (10)$$

where $\dot{m}$ denotes the inlet mass flow rate, C_p the specific heat, T_i the inlet temperature, and p_i total inlet pressure and p_o the outlet static pressures, and γ the ratio of specific heats. This set of equations allows for a precise description of the combined effects of compressibility, viscous shear, and rotation, all of which are fundamental to understanding and predicting the aerodynamic behaviour of the Tesla turbine.

The working fluid is modelled as an ideal compressible gas obeying the equation of state:

$$p = \rho RT, \quad (11)$$

where R is the specific gas constant.

Thermodynamic properties were evaluated using the NASA polynomial formulation implemented in ANSYS Fluent. Accordingly,

$$C_p = C_p(T), \quad h = h(T), \quad e = h(T) - RT, \quad (12)$$

where C_p , h , and e are the specific heat, specific enthalpy, and specific internal energy, respectively.

To quantify the individual contributions of viscous shear and pressure forces to the total power output, the total torque (T_{total}) was decoupled into its viscous (T_{visc}) and pressure (T_{pres}) components with respect to the axis of rotation ($\hat{k}$). The viscous torque is generated by the tangential shear stress (τ_w) acting on the fluid-solid interfaces of the parallel discs and is calculated by integrating the shear stress over the disc surface area (A_{disc}):

$$T_{visc} = \int_{A_{disc}} (\vec{r} \times \vec{\tau}_w) \cdot \hat{k} dA. \quad (13)$$

The pressure-induced torque is generated by the pressure difference acting across the pressure and suction sides of the partial-height blades, and is obtained by integrating the pressure over the normal the disc surface area (A_{disc}):

$$T_{pres} = \int_{A_{disc}} (\vec{r} \times p\hat{n}) \cdot \hat{k} dA. \quad (14)$$

In the solver, the viscous and pressure-induced torque components were obtained from surface moment reports by selecting the corresponding wall boundaries and defining the rotor axis as the moment axis. The associated power contributions were calculated by multiplying their respective torque components by the angular velocity (Ω): $P_{visc} = T_{visc} \cdot \omega$ and $P_{pres} = T_{pres} \cdot \omega$.

To quantify these aerodynamic losses, the entropy generation (S_{gen}) is analyzed. The entropy generation from viscous dissipation, denoted as S_v , have from spatial gradients within the time-averaged velocity field, whereas the turbulent dissipation, denoted as S_t , is driven by the inhomogeneity of the fluctuating velocity components [39]. Their mathematical formulations are expressed as follows:

$$S_v = \frac{\mu}{T_s} \left\{ 2 \left[\left(\frac{\partial \bar{u}_x}{\partial x_x} \right)^2 + \left(\frac{\partial \bar{u}_y}{\partial x_y} \right)^2 + \left(\frac{\partial \bar{u}_z}{\partial x_z} \right)^2 \right] + \left(\frac{\partial \bar{u}_x}{\partial x_y} + \frac{\partial \bar{u}_y}{\partial x_x} \right)^2 + \left(\frac{\partial \bar{u}_z}{\partial x_x} + \frac{\partial \bar{u}_x}{\partial x_z} \right)^2 + \left(\frac{\partial \bar{u}_y}{\partial x_z} + \frac{\partial \bar{u}_z}{\partial x_y} \right)^2 \right\}, \quad (15)$$

$$S_t = \frac{\mu}{T_s} \left\{ 2 \left[\overline{\left(\frac{\partial u'_x}{\partial x_x} \right)^2} + \overline{\left(\frac{\partial u'_y}{\partial x_y} \right)^2} + \overline{\left(\frac{\partial u'_z}{\partial x_z} \right)^2} \right] + \overline{\left(\frac{\partial u'_x}{\partial x_y} + \frac{\partial u'_y}{\partial x_x} \right)^2} + \overline{\left(\frac{\partial u'_z}{\partial x_x} + \frac{\partial u'_x}{\partial x_z} \right)^2} + \overline{\left(\frac{\partial u'_y}{\partial x_z} + \frac{\partial u'_z}{\partial x_y} \right)^2} \right\}, \quad (16)$$

where $\bar{u}_x$, $\bar{u}_y$, and $\bar{u}_z$ represent the time-averaged velocity components; u'_x , u'_y , and u'_z are the fluctuating velocity components. Because the exact fluctuating velocity field is not resolved in standard RANS simulations, the aerodynamic cost of the pressure-driven torque can be quantified by $S_t = \frac{\beta \rho \omega k}{T}$, using the parameters of the $k - \omega$ SST model [40].

4. Numerical Procedure

ANSYS Fluent was used to carry out the RANS simulation and solve the Navier–Stokes equations in three-dimensional, compressible flow, using the finite volume method (FVM). The coupling between pressure and velocity was achieved, the COUPLED algorithm was selected with second order spatial discretization. The same method is commonly applied to research problems involving high speed flows, as it demonstrates good stability with a reduced dispersion error [36]. A second-order upwind scheme was adopted to discretize the convective transport, which provides an optimal balance between accuracy and robustness. The solution was deemed successful when the residuals were reduced to less than 10^{-4} and the average torque and pressure did not change by more than 1% for over 300 iterations.

The computational mesh considered for the simplified model consists of a single rotor disc and two half inter-disc gaps, which together represent one complete inter-disc flow channel between two consecutive rotor discs. The domain was entirely meshed with structured hexahedral elements, ensuring high orthogonality and numerical accuracy. Near the discs and the nozzle walls, prismatic layers with a first-cell height of 0.002 mm were generated to maintain the dimensionless wall distance (y^+) close to 1. This condition is essential for the $k - \omega$ SST turbulence model to adequately capture boundary layer behaviour and shear stresses along the solid surfaces. The domain was divided into two main computational regions, as shown in Figure 3.

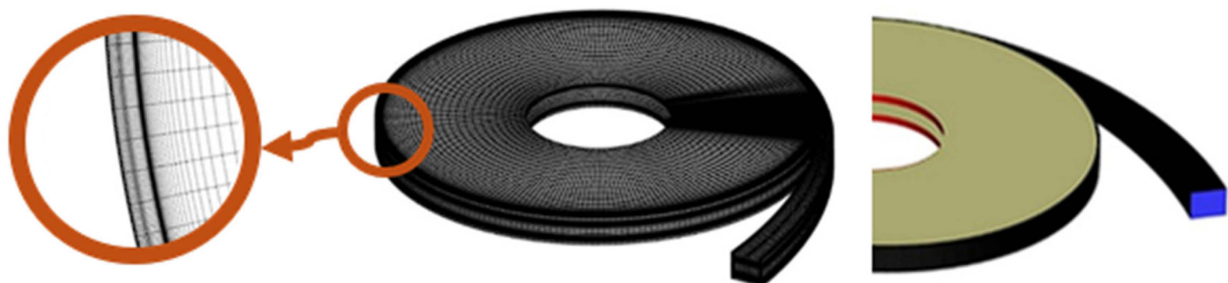


Figure 3. Computational grid of the simplified model, showing the rotating zone (brown), the stationary zone (black), representing the stator and the nozzle regions, the inlet boundary (blue), and the outlet boundary (red).

In the injection nozzle region, the computational mesh was locally refined to accurately resolve the inlet jet flow and its interaction with the rotating discs. This refinement ensures

a faithful representation of the velocity profiles and pressure variations at the tangential interface between the nozzle and the rotor. The mesh refinement adopted for the bladed-disc cases is illustrated in Figure 4, showing the mesh distribution around both the curved and straight blades as well as detailed views near the blade surfaces and blade tip.

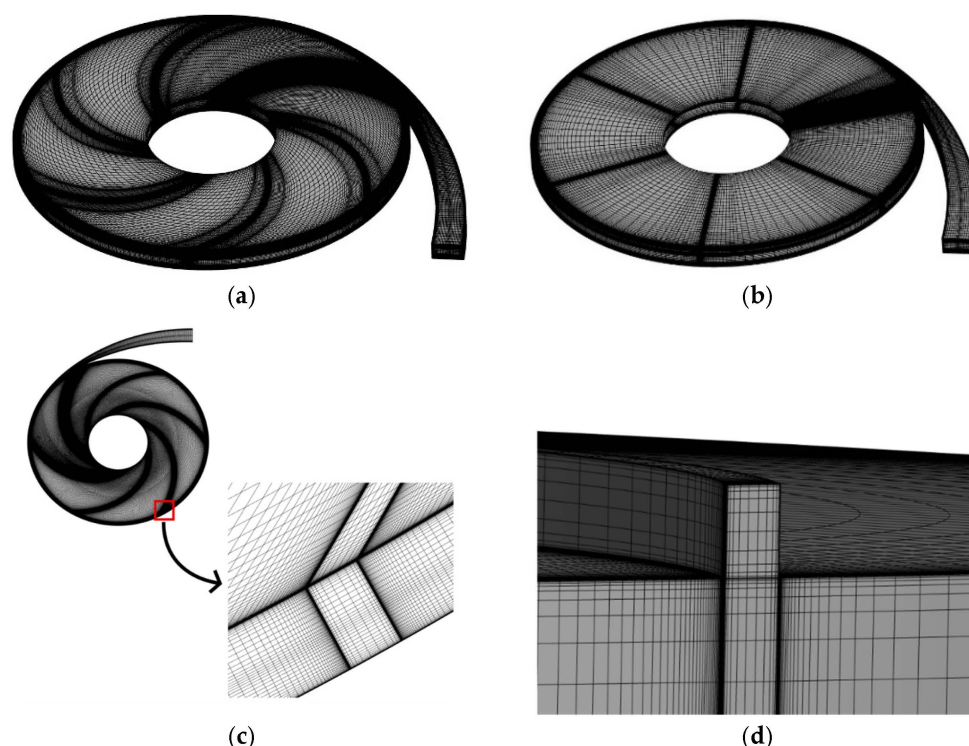


Figure 4. Mesh for the bladed-disc cases, showing in detail the refinement in blade-wall direction for the curved blade (a); the straight blade (b); zoom in the distribution on blade in the plane normal to symmetry (c) and distribution in z direction for the blade tip (d).

5. Mesh-Independence Study

A mesh-independence analysis was performed by comparing the predicted power output and mass flow rate obtained using coarse, medium, and refined meshes. The results showed that, beyond a certain mesh resolution, the relative variations between successive refinements were below 1%, indicating that the solution was effectively independent of further grid refinement. The final mesh was therefore selected by considering the balance between numerical accuracy and computational cost.

To further quantify the discretization uncertainty, Richardson's extrapolation method was applied. This method estimates the asymptotic solution by evaluating the variation between three systematically refined meshes as the characteristic cell size decreases [41]. The obtained results confirmed that the selected mesh provides sufficient accuracy for the numerical analysis.

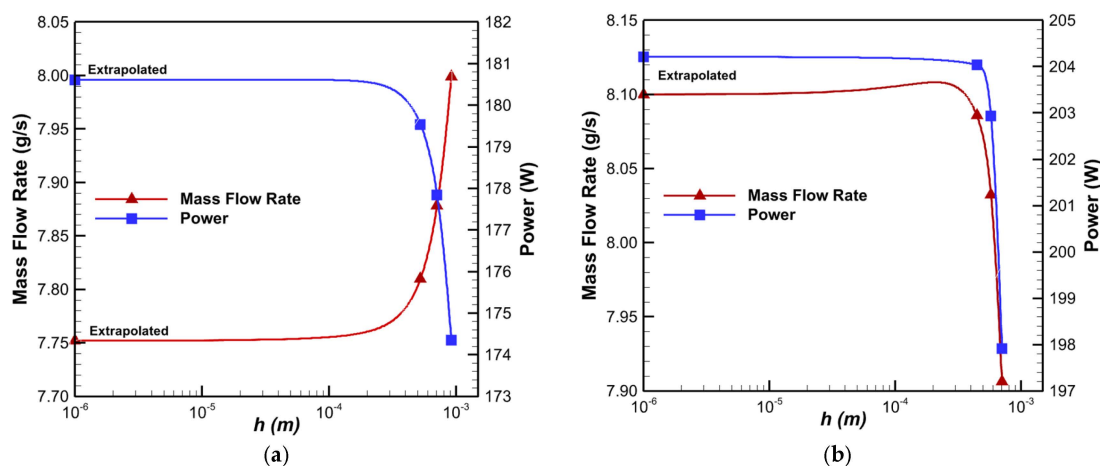
Three grid levels were evaluated for both the plane and bladed-disc cases, as summarized in Table 2. This yielded refinement ratios of $r_{21} = 1.35$ and $r_{32} = 1.30$ for the plane disc case, and $r_{21} = 1.30$ and $r_{32} = 1.22$ for the bladed-disc case, which aligns closely with the values recommended by Roache [41]. The Grid Convergence Index (GCI) values presented in Table 3 confirm that the discretization error is minimal, demonstrating high numerical precision and solution stability, as also illustrated in Figure 5. Consequently, Grid 2 was adopted for the primary simulations, representing the best balance between computational cost and result reliability, as the relative error between Grid 1 and Grid 2 was less than 2.5% in all cases.

Table 2. Number of cells and average representative length for each grid (Grid 1 the most refined and Grid 3 the coarsest) for the plane-disc and the bladed-disc cases.

	Plane		Bladed Disc	
	N° Cells	h	N° Cells	h
Grid 3	1,640,052	9.21×10^{-4}	3,095,388	7.08×10^{-4}
Grid 2	3,314,892	7.11×10^{-4}	7,103,498	5.79×10^{-4}
Grid 1	6,715,271	5.22×10^{-4}	14,320,664	4.46×10^{-4}

Table 3. Results for the Plane Disc and Bladed Disc GCI and relative error for the extrapolated result.

Parameter		Grid 3	Grid 2	Grid 1	P	Err extrapolated [%]	GCI 21 [%]	GCI 32 [%]
Plane Disc	Power (W)	174.35	177.84	179.54	3.14	0.59	0.74	1.94
	Mass flow (g/s)	7.99	7.88	7.81	2.58	0.74	0.92	2.00
Bladed Disc	Power (W)	197.92	202.93	204.05	7.91	0.079	0.099	0.784
	Mass flow (g/s)	7.90	8.03	8.08	4.92	0.25	0.319	1.15

**Figure 5.** Grid convergence study using the Richardson extrapolation method for the plane disc configuration, showing results for mass flow rate and power: (a) plane disc; (b) curved bladed disc.

6. Model Validation

The consistency and reliability of the numerical model were evaluated using the experimental results reported by Zhao et al. [42], as no experimental data are currently available for the specific Tesla turbine configuration considered in this work. In the reference study, the turbine consisted of five discs with an outer diameter of 80 mm, an inner diameter of 30 mm, a disc thickness of 0.7 mm, a disc spacing of 0.8 mm, and a nozzle width of 4 mm. Although the geometric and operating conditions differ from those of the current study, the comparison provides a valuable basis for assessing the accuracy of the adopted numerical methodology, as shown in Figure 6.

The maximum deviation observed between the simulated and experimental results was less than 4%, demonstrating the high consistency and reliability of the employed numerical approach. Differences in this magnitude are common and considered acceptable in CFD analyses of turbomachinery, as they may arise from experimental uncertainties, geometric simplifications, or the intrinsic limitations of turbulence models. Recent studies [17,21,29] have reported deviations between 2% and 6% in similar comparisons, highlighting that errors below 5% indicate an excellent correlation between experimental and numerical data. Thus, the validation stage confirms that the proposed model is physically consistent and

numerically robust, ensuring it can be confidently used to analyze the various scenarios simulated in this study.

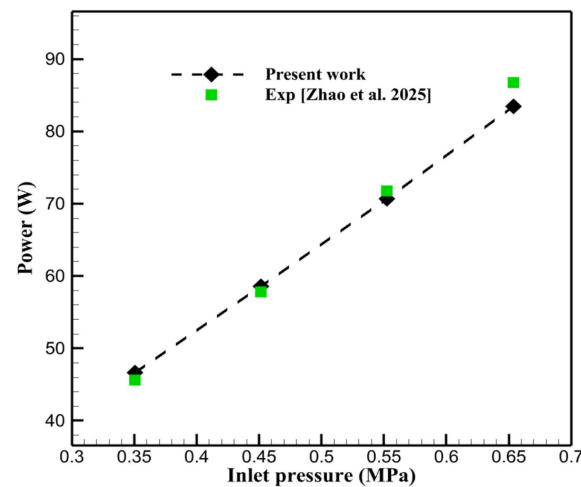


Figure 6. Comparison of simulated and experimental power output at various inlet pressures, based on Zhao et al. [42].

7. Results and Analysis

To evaluate the performance of the Tesla turbine and the influence of incorporating partial-height blades, a comprehensive series of numerical simulations was conducted. The simulations covered rotational speeds ranging from 1000 to 19,000 rpm and inlet pressures from 2 to 4 bar for the conventional plane-disc configuration. A comparative analysis with the bladed-disc configurations was then performed to evaluate the influence of blade geometry and provide a deeper understanding of the associated flow mechanisms.

The specific energy (J/kg), representing the mechanical energy extracted per unit mass of working fluid, was evaluated to compare the energy transfer capability of the rotor under different operating conditions. Since the mass flow rate varies with inlet pressure, this parameter provides additional insight into the effectiveness of the energy-extraction process. As illustrated in Figure 7, the specific energy initially increases with rotational speed for all investigated inlet pressures. The results show that the optimal operating point strongly depends on the inlet pressure. Increasing the inlet pressure from 2 bar to 4 bar increases the maximum achievable specific energy and shifts the optimal rotational speed toward higher values, from approximately 11,000 rpm at 2 bar to nearly 15,000 rpm at 4 bar.

Figure 7 highlights the balance between performance improvement and aerodynamic limitations. Although higher inlet pressure increases the available flow energy and shifts the performance curves upward, stronger compressibility effects at elevated rotational speeds introduce additional losses and reduce the effective operating range. Therefore, among the investigated cases, the 2-bar condition provides a wider operating range before significant aerodynamic losses occur.

The evolution of the flow is presented in the tangential velocity fields in Figure 8 for the inlet pressure of 2 bar at 4000 rpm, 9000 rpm and 14,000 rpm, respectively. At 4000 rpm, the flow is still weak and non-uniform, and there are large low-velocity regions near the inner radius. The high-velocity jet from the nozzle penetrates deeper into the inter-disc gap at high rotational speed (9000 rpm), which decreases the size of the low-velocity zones. At 14,000 rpm, the tangential velocity exhibits a uniform distribution, consistent with the increase in specific energy observed in Figure 7. This behaviour confirms that the optimal operating conditions occur near or above 10,000 rpm for most pressure levels.

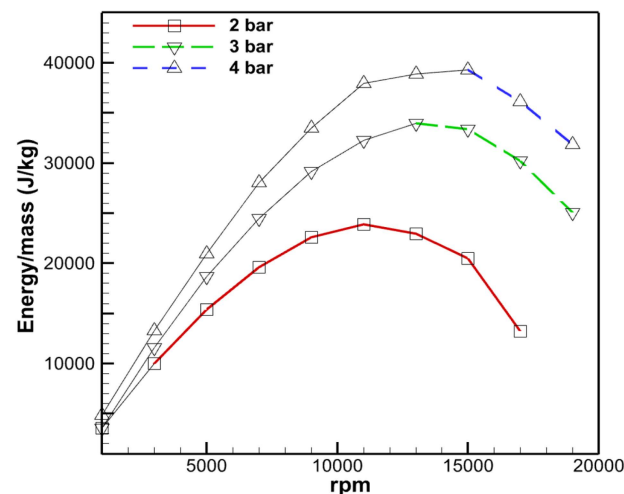


Figure 7. Specific energy (energy per unit mass) as a function of rotational speed for different inlet pressures, with black segments indicating supersonic ($Ma > 1$) internal-flow conditions.

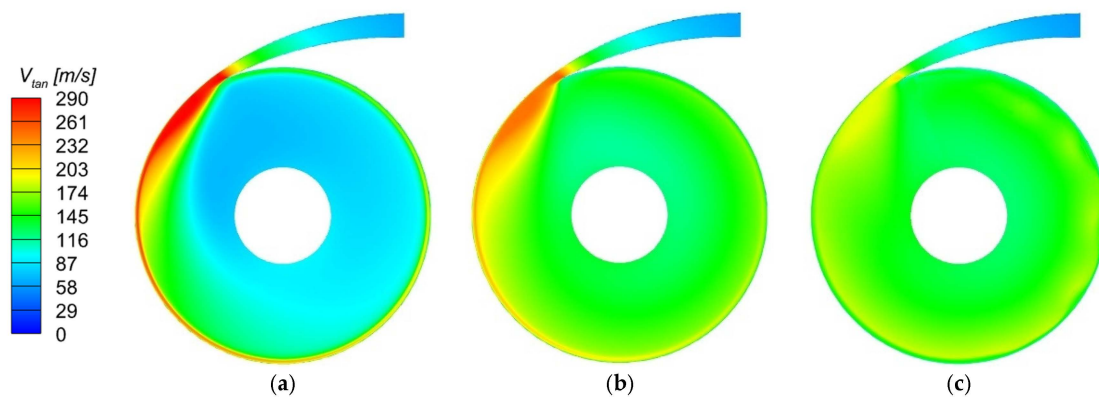


Figure 8. Tangential velocity contours in the Tesla Turbine with plane discs at 2 bar for different rotational speeds: (a) 4000 rpm; (b) 9000 rpm; (c) 14,000 rpm.

In the plane-disc configuration, momentum extraction was exclusively based on viscous shear, which becomes continuously less predominant at high rotational speeds due to the boundary layer becoming thinner and increased flow separation. To check if the partial-height blades can reduce these losses inherent to the rotor, bladed and planar rotor designs were compared to each other under the same set of boundary conditions. The following results indicate the effect of blade insertion on power production, efficiency, pressure distribution, and development of tangential velocities.

The power output of the curved-bladed disc configuration is shown in Figure 9a for pressures at the inlet of 2, 3 and 4 bar. There is a strong dependence of maximum power on the inlet pressure. As shown in Figure 9, the increase in mechanical power associated with higher inlet pressures is fundamentally driven by enhanced momentum transfer from the fluid to the rotor. For a fixed outlet condition, elevating the inlet total pressure increases the overall pressure gradient across the turbine. This thermodynamic potential translates into a higher mass flow rate and increased flow velocities, particularly the tangential velocity component at the rotor inlet. Consequently, the higher kinetic energy of the fluid intensifies both the viscous shear stresses on the parallel disc surfaces and the pressure differentials acting across the individual partial-height blades. This combined amplification of shear-driven and pressure-driven torque directly yields the higher power output observed. As the speed of rotation increases, the power output increases up to a certain point, and then the power output decreases. This drop starts at around 11,000 rpm in the 2 bar cases, 15,000 rpm in the 3 bar cases and 17,000 rpm in the 4 bar cases. This transition indicates

that increased inlet pressures will need to be met with increased rotational speeds before the losses due to aerodynamics begin to take precedence over the losses due to the fluid energy. The curved-bladed rotor consistently generates more power, up to 30% higher at 14,000 rpm at 2 bar inlet pressure than the plane-disc case, as illustrated in Figure 9b. The fluid-dynamic mechanisms that contribute to this macroscopic enhancement, particularly the enhanced flow guidance, the tangential acceleration, and the momentum imparted from the pressure distribution, are described later by analyzing the distributions of surface shear stress and pressure.

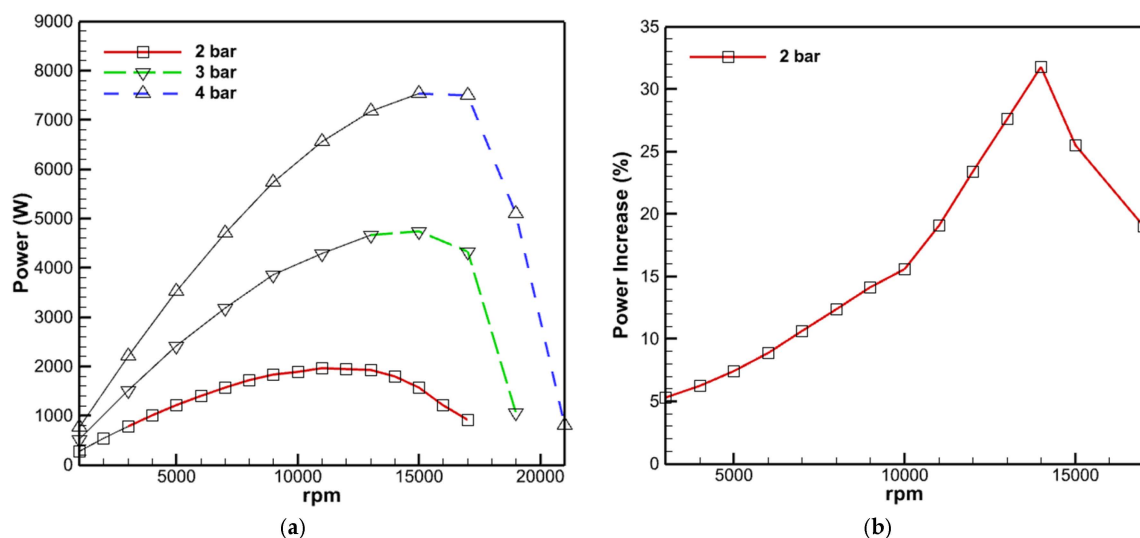


Figure 9. (a) Power output as a function of rotational speed for different inlet pressures for the curved bladed disc; (b) power increase between the plane-disc and curved bladed-disc configuration at 2 bar.

The contours in Figure 10a,b present the effect of the partial-height blades on the skin-friction distribution. The baseline plane-disc (Figure 10a) exhibits a uniformly distributed and visually higher overall level of skin friction across the entire disc surface. In contrast, the bladed configuration (Figure 10b), while presenting localized peaks of elevated shear stress directly along the blade paths and near the inlet nozzle, visually demonstrates lower average wall shear stress over the broader inter-blade regions. This indicates an overall reduction in the purely viscous momentum transfer on the disc walls.

However, this reduction in global viscous efforts is compensated by a secondary mechanism, as revealed by the static pressure fields extracted at the blade mid-span plane. While the plane-disc (Figure 10c) shows a simple, near concentric radial pressure drop, the bladed disc (Figure 10d) disrupts this symmetry. Distinct pressure gradients are observable across the individual blade faces, with high-pressure stagnation zones on the advancing faces and lower-pressure regions on the trailing faces. This cross-blade pressure differential introduces a new, pressure-induced torque production that is entirely absent in the conventional plane-disc configuration. The introduction of this new pressure-driven torque overcomes the reduction in uniform wall friction, driving the macroscopic power gains characteristic of this hybrid design.

Corroborating the visual evidence from the skin-friction coefficient contours discussed previously, Figure 11 quantitatively demonstrates the individual torque contributions as a function of rotational speed. As expected from the previous analysis, the purely viscous torque generated by the bladed turbine is consistently lower than that of the conventional plane-disc rotor across the entire operating range. For instance, at 14,000 rpm, the viscous torque decreases from approximately 0.103 Nm in the plane-disc configuration to 0.094 Nm in the bladed configuration, corresponding to a reduction of nearly 9%.

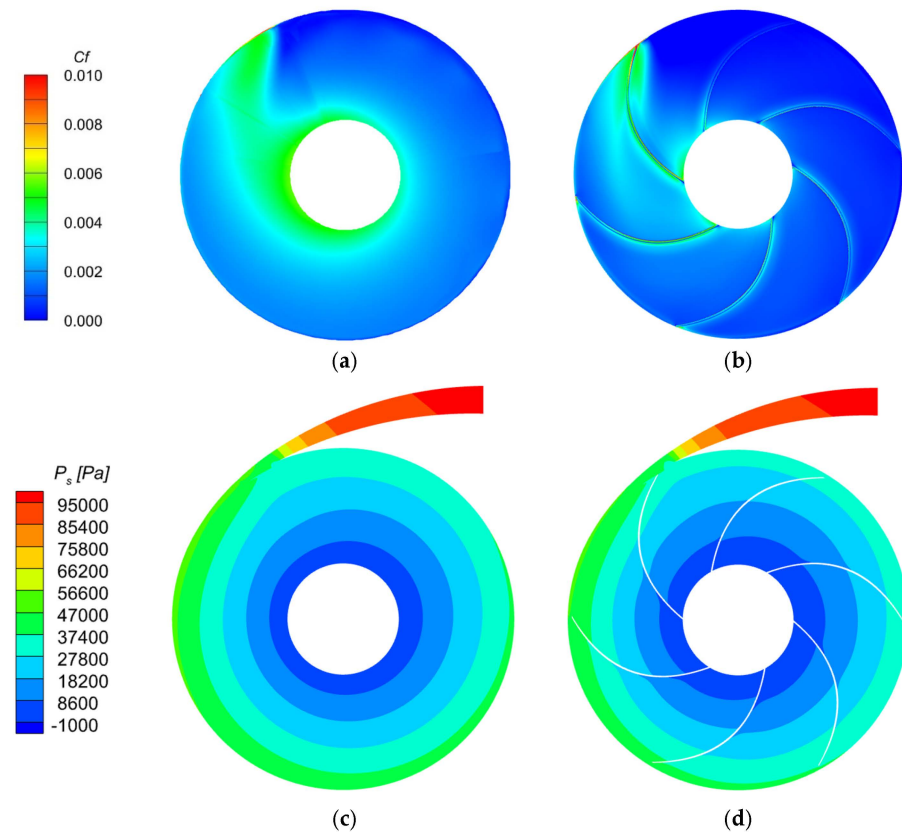


Figure 10. Skin-friction and static pressure contours for the plane-disc (left) and bladed-disc (right) configurations at 2 bar and 11,000 rpm: (a) plane-disc skin friction; (b) bladed-disc skin friction; (c) plane-disc static pressure; and (d) bladed-disc static pressure.

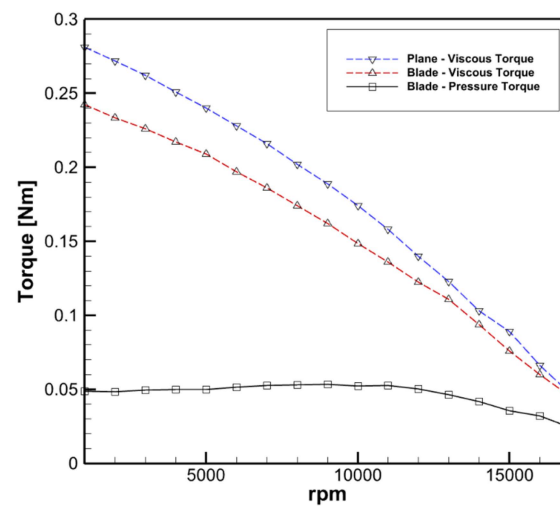


Figure 11. Viscous and pressure-induced torque components for the plane-disc model and bladed model as a function of rotational speed for one disc at 2 bar.

However, the graph clearly illustrates the fundamental advantage of the proposed hybrid mechanism. At the same rotational speed, the pressure-induced torque component, generated by the localized form drag across the partial-height blades, contributes an additional 0.042 Nm. When combined with its corresponding viscous contribution, the total torque delivered to the shaft reaches 0.136 Nm, representing an overall enhancement of approximately 31.5% compared with the plane-disc baseline.

This quantitative breakdown confirms that the pressure contribution compensates for the reduction in viscous torque and represents the main factor responsible for the overall performance improvement.

As shown in Figure 12, increasing the inlet pressure significantly increases the turbine power output, from 1954 W at 2 bar to 6558.6 W at 4 bar (at 11,000 rpm). However, this power improvement is accompanied by additional aerodynamic limitations. At 4 bar, the flow velocity downstream of the nozzle reaches Mach 1.21, resulting in localized supersonic regions. These compressibility effects can promote the formation of shock waves, leading to increased stagnation pressure losses, viscous dissipation, and possible boundary-layer separation.

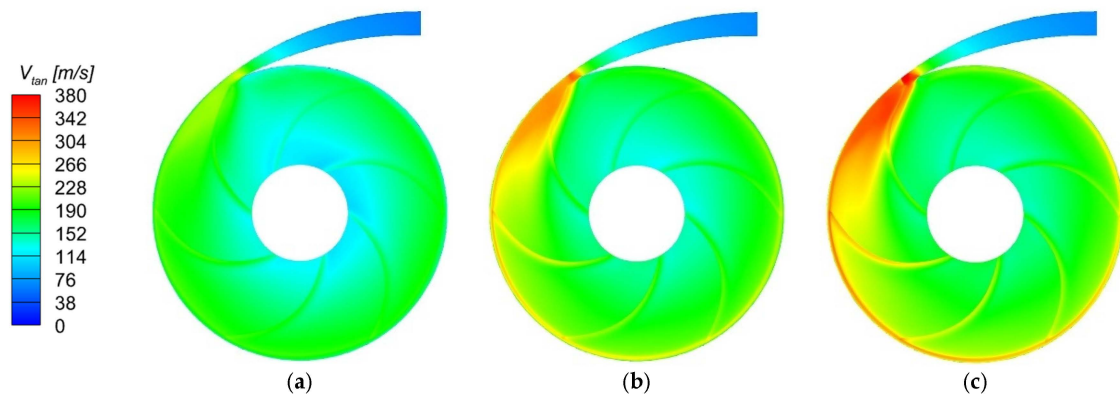


Figure 12. Tangential velocity contours for different inlet pressures at 11,000 rpm: (a) 2 bar, (b) 3 bar, and (c) 4 bar.

Therefore, although higher inlet pressures allow greater power extraction, they also introduce additional aerodynamic losses and more complex internal flow behaviour. Consequently, operation at 4 bar may exceed the optimal operating range of the current subsonic turbine design.

By comparing the spatial distribution of S_{gen} with the static pressure field, regions of high turbulent dissipation and their relationship with the pressure-driven energy-extraction mechanism can be identified. This combined analysis provides a clearer understanding of how the proposed blade geometry enhances pressure-induced work extraction while controlling the associated turbulent energy losses.

This mechanism is illustrated by the volumetric entropy generation contours. As shown in Figure 13a, the plane-disc configuration exhibits a large region of high dissipation near the inlet, resulting from unguided expansion and local flow stagnation. In contrast, the introduction of partial-height blades (Figure 13b) modifies the irreversibility distribution. The entropy generation becomes more organized and is mainly concentrated near the leading edges and suction surfaces of the blades due to wake formation and local flow disturbances. By improving the guidance of the working fluid, the blades enable additional energy extraction through pressure differences across their surfaces instead of relying exclusively on boundary-layer viscous shear. This transition toward a hybrid energy-extraction mechanism increases the total torque production, compensating for the localized increase in turbulent losses.

To analyze whether the performance improvement originates only from the reduction in the inter-disc gap caused by the blade tips, an additional plane-disc configuration was simulated with a gap equivalent to the minimum flow passage created by the blades. Since the blades have a height of 0.25 mm on each disc surface, an equivalent plane-disc simulation with a 0.5 mm gap was performed. The performance results shown in Figure 13c indicate that the reduced-gap (0.5 mm) configuration provides similar performance to the

original 1.0 mm gap configuration at lower rotational speeds. However, above 11,000 rpm, the 0.5 mm configuration achieves higher efficiency and power output than the 1.0 mm gap turbine. Nevertheless, the bladed configuration achieves the highest efficiency for most operating conditions, reaching a maximum value of approximately 53.3% at 13,000 rpm compared with 46.6% for the 0.5 mm gap configuration. At high rotational speeds, all configurations experience a reduction in efficiency; however, the bladed rotor maintains its performance advantage. This demonstrates that the improvement is not solely caused by gap reduction but also results from enhanced flow guidance and additional pressure-induced energy extraction provided by the partial-height blades.

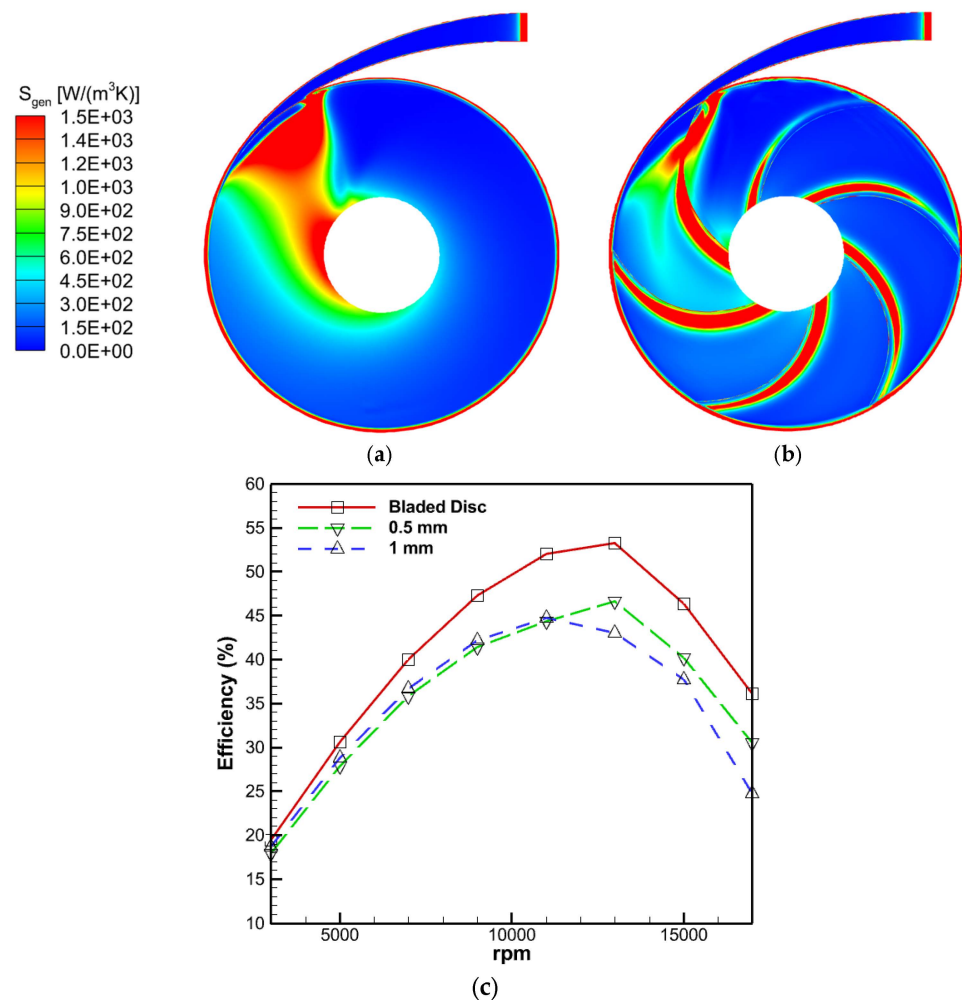


Figure 13. Volumetric entropy generation (S_{gen}) contours for (a) the plane-disc turbine with a 0.5 mm gap and (b) the bladed-disc configuration at an inlet pressure of 2 bar. (c) Turbine efficiency as a function of rotational speed for the 1.0 mm gap, 0.5 mm gap, and bladed-disc configurations.

The influence of three different blade geometries, namely curved, straight, and inverted curved blades, on the tangential and radial velocity fields extracted at the blade mid-span plane for the six-blade configuration is shown in Figure 14. As illustrated in Figure 14a,d, the curved blade provides a well-distributed velocity field along the disc. The peak tangential velocities are aligned with the natural flow path created by the injector, while the radial velocity contours demonstrate a smooth, progressive inward flow. The curvature of the blades effectively assists the desired spiralling motion of the Tesla turbine, enhancing jet directionality and promoting continuous flow development across the disc channel. This aerodynamic guidance is explicitly corroborated by the radial velocity contours, which reveal a distinct zone of negative radial velocity immediately downstream

of the blades. This negative velocity mathematically indicates an inward-directed flow towards the central exhaustion, in contrast to the third geometry, which presents positive values after the blades.

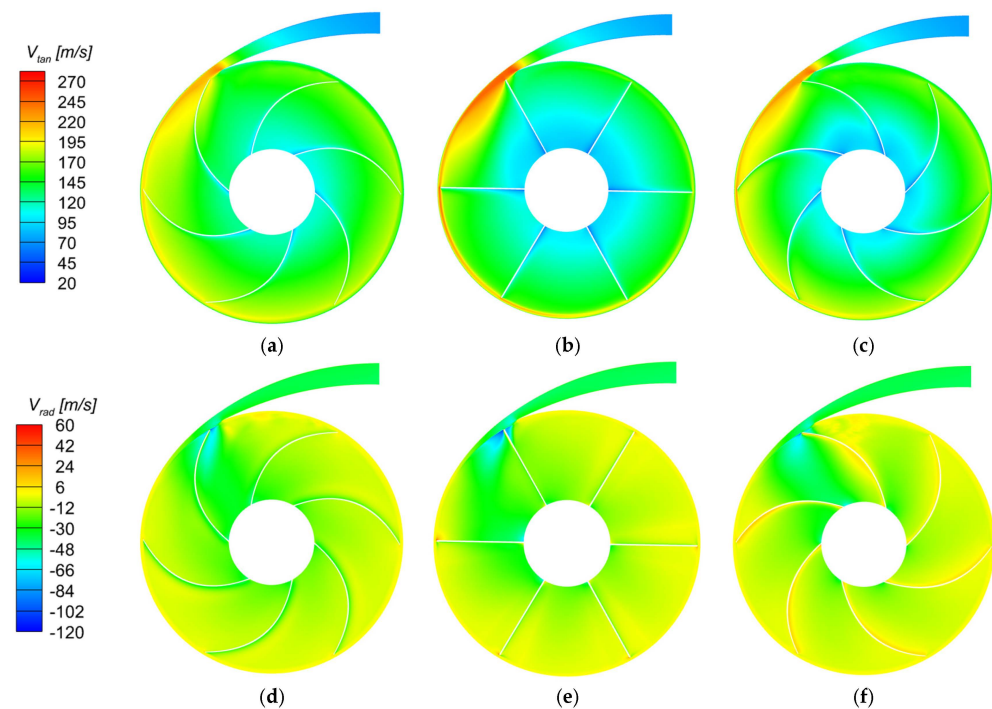


Figure 14. Contours of tangential velocity (top row) and radial velocity (bottom row) evaluated at the blade mid-span plane for different blade geometries at 2 bar and 11,000 rpm: (a,d) curved blade, (b,e) straight blade, and (c,f) inverted curved blade.

The straight blade, in contrast, presents a much more abrupt interaction with the flow. Although it demonstrates higher localized tangential velocities and a radial velocity field (Figure 14e) that reveals a similar, inward-directed behaviour compared to the curved model. The tangential velocity peaks are primarily concentrated on the clearance, failing to effectively penetrate the inter-disc spacing. This peripheral concentration, coupled with significantly lower tangential velocities towards the centre of the turbine, is strongly indicative of severe aerodynamic losses. These characteristics demonstrate that the geometric mismatch forces a high incidence angle, which acts as a partial flow blockage at the inlet, restricting the fluid from smoothly entering the disc channel and resulting in a highly inefficient flow redirection.

The inverted curved blade, shown in Figure 14c,f, exhibits the least favourable flow behaviour. Although its tangential velocity distribution within the inter-disc spacing is slightly improved compared with the straight-blade configuration, it still presents relatively low velocity values near the central region of the turbine. By comparing the region immediately downstream of the injector with the optimized curved-blade configuration, it can be observed that the inverted geometry redirects part of the flow toward the outer region of the discs. This unfavourable flow behaviour is confirmed by the radial velocity contours, which show positive radial velocity regions immediately downstream of the blades. Unlike the curved-blade configuration, these positive radial velocities indicate a tendency for the flow to move outward, opposing the desired inward radial transport toward the outlet. This reduces the effective flow-guidance capability and negatively affects the energy-extraction process.

Figure 15 presents a comprehensive comparison of the different blade shapes in terms of torque generation mechanisms, mass flow rate, and specific energy production at an

inlet pressure of 2 bar across different rotational speeds. As shown in Figure 15a, the total torque generated by one disc decreases monotonically with increasing rotational speed for all bladed configurations. However, an important distinction can be observed regarding the contribution of each torque generation mechanism. For both the straight and inverted curved blades, the percentage of pressure-induced torque increases significantly with rotational speed, eventually becoming the dominant contribution to power extraction (reaching approximately 60% and 55% at 17,000 rpm, respectively). This increased dependence on form drag and pressure forces shifts the operating mechanism away from the traditional viscous-driven principle of the Tesla turbine. In contrast, the curved blade design effectively mitigates this issue, maintaining a pressure-induced torque fraction well below 40% (mostly hovering between 20% and 35%) across the entire operational range. This proves that the curved geometry enhances overall performance while preserving the fundamental boundary-layer shear mechanism. This physical behaviour is further contextualized by the mass flow rate for one slice trends shown in Figure 15b. As the rotational speed increases, stronger centrifugal effects generate greater resistance to the inward radial flow, resulting in a progressive reduction in mass flow rate across all geometries. While the bladed configurations provide slightly higher mass flow rates than the plane-disc model at lower rotational speeds, all configurations approach a limiting flow condition at high rotational speeds, decreasing to approximately 0.0064–0.0068 kg/s at 17,000 rpm. This limitation restricts the maximum mechanical power that can be extracted from the turbine.

In Figure 15c, the straight blade exhibits moderate performance with higher energy values than the plane-disc configuration but lower values than the curved blade. It can produce a small increase in power; however, with a slightly higher mass flow, reaching a peak of 1986 W at 11,000 rpm, compared with 1954 W shown by the curved blade. This behaviour can be linked to non-optimal guidance, which contributes to the mass flow and limits effective energy transfer. The straight blade presents a very high local acceleration in Figure 14b but also exhibits a clear performance penalty compared to the normal curved geometry, although it still performs better than the plane-disc configuration. An additional insight is provided in Figure 15b, which demonstrates the mass flow through turbine slices. The curved blade reveals a steady mass-flow rate, whereas the straight and inverted curve blades reveal a decreasing mass-flow rate as rotational velocity increases, similar to the original disc case. This decrease in mass flow can be interpreted as approaching a choking limit, which limits the mechanical power that can be generated.

It has been demonstrated that incorporating blades into Tesla turbine discs improves torque generation and overall performance compared with the conventional plane-disc configuration. However, the blade number plays a critical role in controlling the balance between viscous coupling, pressure-induced torque generation, and aerodynamic blockage. Therefore, three configurations (3, 6, and 9 blades), uniformly distributed as shown in Figure 16, are investigated to evaluate the influence of blade number on the internal flow behaviour and mechanical performance. Increasing the blade number reduces the angular spacing between adjacent blades, modifying the available flow passages. The three-blade configuration provides wider channels with lower flow restriction, while the six-blade configuration represents an intermediate case. The nine-blade arrangement provides stronger flow guidance due to the narrower passages, although with a potential increase in aerodynamic losses. The following results analyze these effects through tangential velocity, skin-friction distribution, and power generation.

Figure 17 shows the contours of the tangential velocity, illustrating how these geometrical differences translate into fluid behaviour. With the minimum number of blades, the flow is accelerated to generate a relatively smooth but less powerful tangential field. The six-blade design exhibits higher acceleration and better alignment with the direction of

rotation, leading to a more effective transfer of momentum. The maximum magnitudes of tangential velocity are found near the inlet when nine blades are employed, though with steeper gradients that indicate the initiation of stronger shear layers. A similar trend can be observed in the skin-friction coefficient contours: the three-blade layout produces lower friction values, which are related to lower levels of shear stress; the six-blade layout promotes greater flow attachment, resulting in the formation of a stronger skin-friction profile; and the nine-blade layout presents the highest values of skin friction, which is associated with the strengthened interaction between the fluid and the blade surfaces.

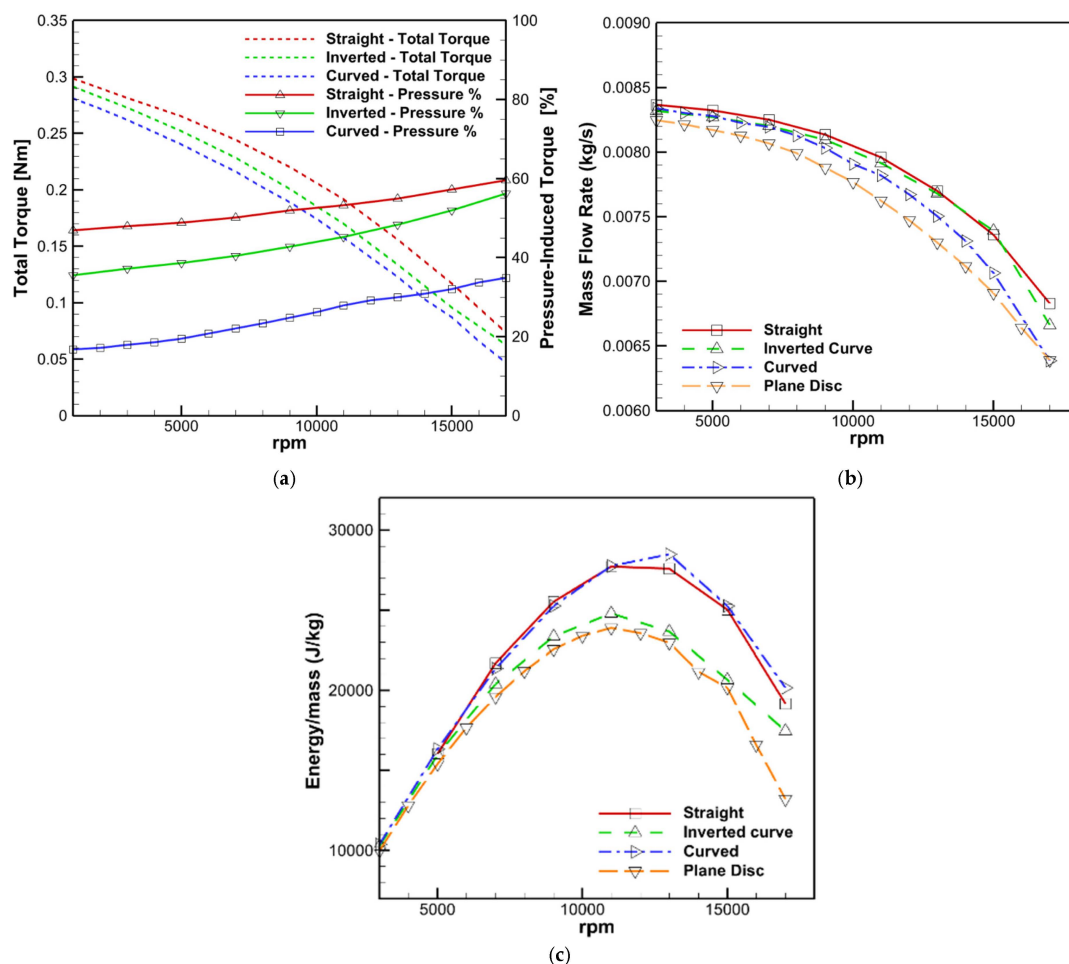


Figure 15. Comparison of different blade shapes in terms of (a) total torque and percentage contribution of pressure-induced torque, (b) mass flow rate per gap, and (c) specific energy production at an inlet pressure of 2 bar for different rotational speeds.

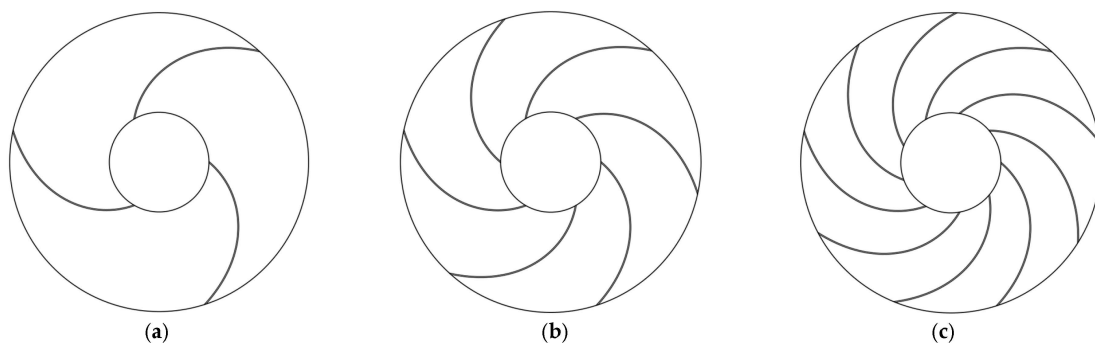


Figure 16. Schematic of blade distribution for three (a), six (b), and nine (c) blades in the curved blade configuration.

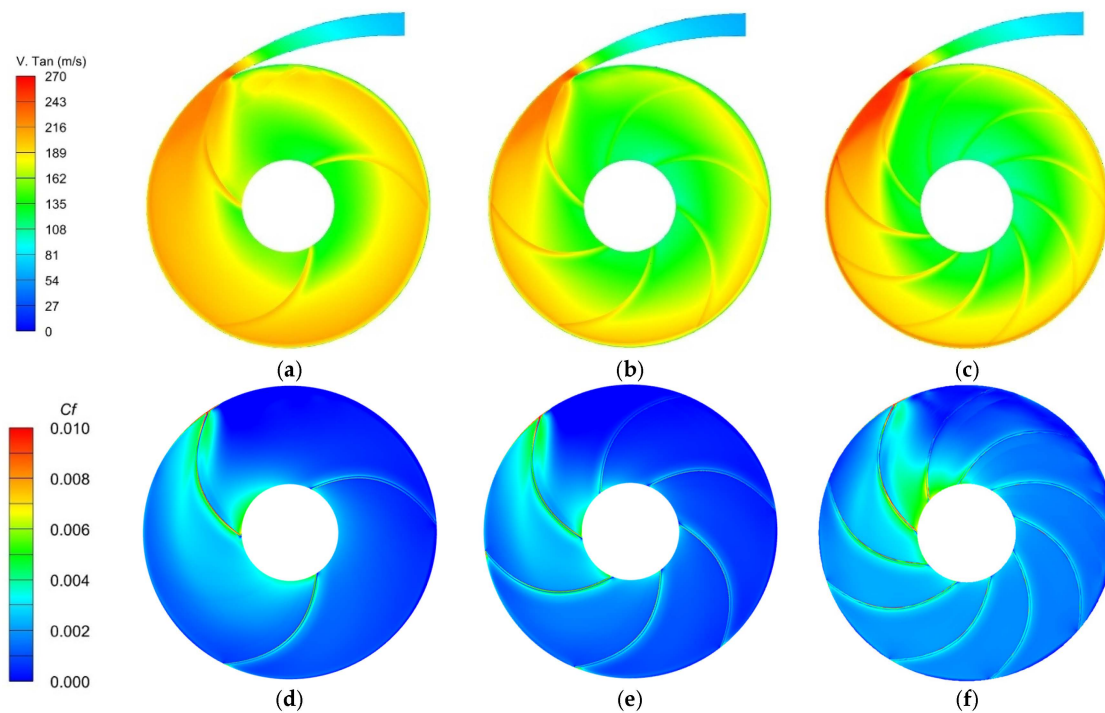


Figure 17. Tangential velocity and skin-friction coefficient contours for three, six, and nine-blade disc configurations at 2 bar and 11,000 rpm: (a,d) three blades; (b,e) six blades; (c,f) nine blades.

The impact of blade count on the internal flow gives a preliminary idea of the effect that these geometric changes will have on different fluid-dynamic behaviours. As seen in the tangential velocity contours, a few blades produce a less energetic, but more uniform tangential velocity field. The six-blade arrangement results in a higher level of acceleration and a greater degree of rotational alignment, demonstrating a more efficient transfer of torque. The nine-blade configuration has maximum tangential-velocity magnitudes, especially near the nozzle, with larger velocity gradients, which imply the presence of higher shear stresses in the system (see Figure 17c,f). The six-blade configuration exhibits the lowest friction distributions, and it can be explained by the reduced shear interaction with the fluid caused by the blockage generated by the blades, which disrupts the flow, whereas the three-bladed configuration has increased friction distributions with a more uniform friction zone. In the nine-blade configuration, the highest friction distributions can be explained by increased interaction with the fluid along the disc surfaces. These observations are followed by the more detailed quantitative investigation that follows, which examines the effects of blade count on radial profile and overall power production in detail.

The tangential velocity contours increase from the three-blade to the nine-blade configuration, and the three-blade case has the highest velocity along the radial direction. The same does not apply to the skin-friction contours, which exhibit different trends. Atypically, more blades mean higher velocity gradients and, for the nine-bladed model, higher viscous work in the disc surfaces. Analysis of these results shows that the performance of the turbine, and power in particular, is generally improved in blade-modified discs.

Figure 18 presents the power output and each specific contribution for different blade counts at 2 bar and 11,000 rpm, along with their percentage contribution to the total power compared to the plane-disc model. As shown in Figure 18a, increasing the number of blades increases the total power output and enhances the contribution of pressure-induced effects generated by the blades. Figure 18b shows that although power output increases with blade count, the rate of improvement gradually decreases, indicating the

existence of an optimal blade number. However, it is important to highlight that viscous shear remains the dominant driving mechanism across all investigated configurations. Even for the optimal nine-blade design, the pressure contribution, while providing a significant performance improvement, does not exceed the viscous contribution. These partial-height blades act as localized obstacles in the primary flow path. As the fluid impinges upon the blade profiles, local deceleration occurs, generating a high-pressure zone on the pressure side and a corresponding lower-pressure region on the suction side. This pressure difference creates a form-drag contribution acting on the blade surfaces, generating an additional pressure-induced torque component similar to the mechanism observed in impulse-type turbomachinery.

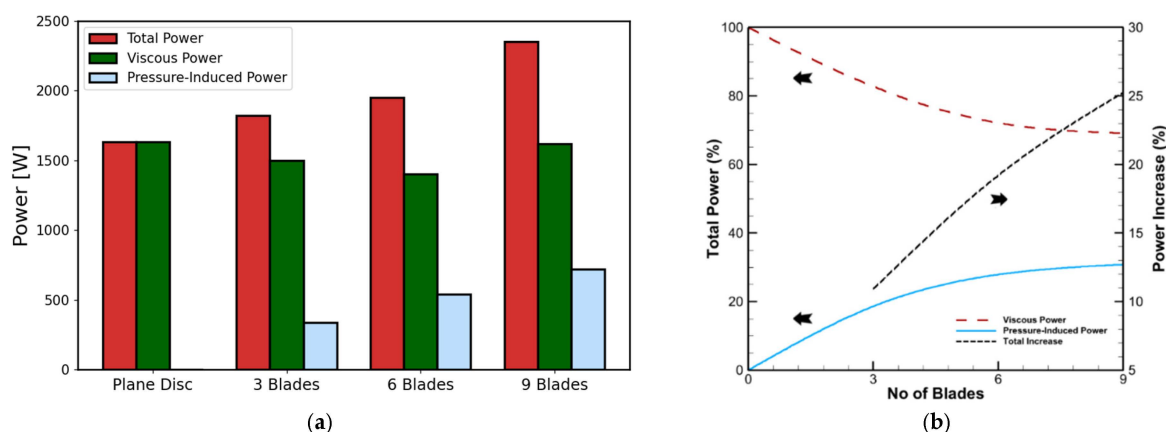


Figure 18. (a) Total power, viscous power, and pressure-induced power for different numbers of blades at 2 bar and 11,000 rpm; (b) percentage contribution to total power and relative power increase in the viscous and pressure-induced components with respect to the plane-disc configuration.

As shown in Table 4, increasing the number of blades leads to a gradual improvement in both power output and total-to-static efficiency at the best operating point, with the nine-blade model achieving a maximum efficiency of 57.5% at 13,000 rpm. This performance enhancement is governed by two main physical mechanisms.

Table 4. Results of mass flow rate, power and efficiency for 2 bar inlet pressure in discs with 3, 6 and 9 blades on each side.

No of Blades	Measurement	Rotational Velocity (rpm)						
		5000	7000	9000	11,000	13,000	15,000	17,000
3	Mass Flow [g/s]	8.25	8.15	7.97	7.75	7.43	7.00	6.53
	Power [W]	1180.4	1505.8	1727.0	1818.7	1745.6	1439.1	685.1
	Efficiency [%]	29.5	38.1	44.7	48.5	48.4	42.5	25.2
6	Mass Flow [g/s]	8.28	8.19	8.03	7.82	7.50	7.06	6.39
	Power [W]	1217.0	1574.5	1826.4	1954.0	1924.9	1573.9	1110.0
	Efficiency [%]	30.6	40.0	47.3	52.0	53.3	46.3	36.1
9	Mass Flow [g/s]	8.30	8.22	8.08	7.84	7.60	7.13	6.29
	Power [W]	1256.5	1638.2	1920.3	2053.1	2101.6	1834.3	1358.6
	Efficiency [%]	31.5	41.4	49.4	54.4	57.5	53.5	44.9

First, increasing the blade count provides more surfaces for the generation of pressure-induced torque through the pressure difference formed across the blades. Second, higher blade numbers create narrower inter-blade channels, improving flow guidance and promoting a more effective transfer of fluid momentum to the rotor.

This behaviour is supported by the flow contours shown in Figure 17, which show that increasing the number of blades produces higher tangential velocities and modifies the skin-friction distribution. Although the three-blade configuration generates higher skin friction than the six-blade model due to wider flow passages, it does not provide the same pressure-induced contribution as the nine-blade design. In the nine-blade configuration, improved flow guidance increases tangential acceleration and enhances the overall torque generation mechanism.

As demonstrated in Figure 18, despite the additional torque generated by blade-induced pressure differences, viscous shear remains the dominant power-extraction mechanism. Therefore, the modified rotor maintains the fundamental operating principle of the Tesla turbine while benefiting from an additional pressure-induced contribution. These results demonstrate that blade number represents an important design parameter, requiring a balance between improved flow guidance, pressure-induced torque generation, and potential aerodynamic blockage effects to maximize turbine performance.

Table 5 presents the maximum velocity (V_{max}) and the corresponding Mach number in the nozzle region for different rotational speeds and blade counts. The results show that the presence of blades has only a modest influence on peak velocity for most operating conditions, with the nine-blade configuration consistently exhibiting the highest values. For example, at 9000 rpm, V_{max} increases from 262.2 m/s for the plane-disc case to 272.7 m/s for the nine-blade configuration, with the Mach number changing from 0.81 to 0.85. At low rotational speeds (1000–3000 rpm), all configurations operate close to sonic conditions ($Ma = 1.00$ – 1.02), regardless of blade count, due to the strong geometric contraction imposed by the nozzle, which dominates over rotor-induced effects. As rotational speed increases, the Mach number decreases, falling into the subsonic range of 0.54–0.79 at 17,000 rpm across all bladed configurations. Overall, the results indicate that the local acceleration within the nozzle remains within stable limits and does not lead to premature choking or the formation of internal shock waves.

Table 5. Maximum velocity (m/s) in the nozzle region and corresponding Mach number (Ma) at the same reference location for different blade counts and rotational.

Rotational Velocity (rpm)	Number of Blades							
	0		3		6		9	
	V_{max}	Ma	V_{max}	Ma	V_{max}	Ma	V_{max}	Ma
1000	314.6	1.02	318.37	1.02	319.1	1.02	319.4	1.03
3000	302.7	0.96	309.15	0.98	311.0	0.98	312.3	1.0
5000	287.6	0.90	296.9	0.93	300.1	0.95	297.2	0.94
7000	279.3	0.87	281.0	0.87	284.3	0.89	286.8	0.90
9000	262.19	0.81	263.8	0.81	268.2	0.83	272.7	0.85
11,000	243.9	0.75	246.6	0.76	251.1	0.77	275.2	0.79
13,000	225.4	0.72	234.2	0.70	236.6	0.72	261.0	0.77
15,000	205.6	0.62	209.3	0.63	210.5	0.64	213.4	0.65
17,000	182.3	0.56	185.4	0.57	182.3	0.55	179.7	0.54

8. Conclusions

This study presents a comprehensive analysis of the geometric and operating parameters of a Tesla turbine with rotor modifications, with a particular focus on the introduction of partial-height blades and the effects of blade number and shape. All simulations were performed using ANSYS Fluent. Throughout the evaluation of power production, tangential and radial velocity distributions, skin-friction distributions, pressure gradients, and mass-flow behaviour, a clear and coherent physical interpretation of the blade-induced

modifications in the internal flow has been obtained. Based on these results, several important conclusions can be drawn:

- The fundamental premise of the ‘hybrid’ performance lies in the synergistic coupling of two distinct momentum transfer mechanisms within the same fluid domain. In a conventional Tesla turbine, energy extraction relies exclusively on the viscous shear stress developed along the boundary layers of the smooth parallel discs. The presence of blades produces a significant increase in momentum transfer within the admissible operating regime. While mechanically simple, this purely viscous mechanism, the integration of partial-height blades, alters this dynamic by introducing a pressure-driven mechanism; designs that introduce blades exhibit higher tangential acceleration, steeper pressure gradients, and higher skin-friction coefficients along the disc surface compared to the plane-disc design, which directly contributes to the enhancement of torque generation in the subsonic regime. As a result, performance improvements are physically realized whenever operation is restricted to regimes characterized by Mach numbers less than 1. This behaviour is a result of a trade-off between flow guidance, viscous dissipation, and operational stability;
- Regarding the geometric parameters, increasing the number of blades positively impacted the energy conversion process. The configuration with nine curved blades emerged as the optimal design among the tested geometries, delivering the highest isentropic efficiency (57.5% at 13,000 rpm and 2 bar) and torque by maximizing the pressure-induced forces operating under a subsonic regime ($Ma < 1$) at 2 bar.

Overall, the results demonstrate that a Tesla turbine equipped with optimized partial-height blades can significantly outperform the conventional plane-disc configuration. Under identical operating conditions in the subsonic regime, the modified turbine achieves a 27% increase in power output and an efficiency improvement of approximately 10%. The incorporation of partial-height blades enhances the torque-generation process by modifying the pressure distribution and guiding the flow more effectively toward the outlet, thereby improving momentum transfer, while viscous shear remains the dominant energy-transfer mechanism. This confirms the hybrid operating behaviour of the proposed turbine concept. However, these effects are inherently non-linear, and both blade number and blade curvature require careful optimization to avoid excessive losses associated with viscous dissipation, aerodynamic blockage, and compressibility effects. Among all investigated operating conditions and geometrical configurations, the nine curved partial-height blade configuration achieves the highest power output and efficiency improvement.

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